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PERFORMANCE CHARACTERISTICS OF A FINITE WIDTH CIRCULAR BEARING USING MICROPOLAR FLUID

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ABSTRACT

To study and compare stability phenomena using Newtonian and non-Newtonian fluid for circular journal bearing. Modified Reynolds equation is obtained for micropolar fluid. Stiffness characteristics, damping characteristics, Critical mass, whirl frequency ratio and threshold speed are evaluated for different eccentricity. The results show that micropolar fluid exhibits better stability in comparison with Newtonian fluid.

Keywords: Critical mass, dynamic characteristics, eccentricity, modified Reynolds equation, stability parameter

I. INTRODUCTION

With the development of modern machine elements the increasing use of complex fluids as lubricants has been emphasized. The classical theory of hydrodynamic lubrication assumes that the lubricant behaves as a Newtonian viscous fluid. However, the characteristics of the lubricants are often controlled using additives, in order to meet the specific requirements of many engineering applications. The viscosities of these lubricants follow non-linear relationships between shear stress and shear strain rates. As the contaminated sub-structures can translate, rotate or even deform independently, the classical (Newtonian) theory becomes limited to predict the accurate flow behaviors and thus the micro-continuum theories have been developed. The study of the flow behaviors using the theory of micropolar lubrication was initiated with the problem of a two-dimensional slider bearing. For such micropolar lubrication the non-dimensional material length was found to have considerable influence on the lubricating properties.

Micropolar lubrication also showed better lubricating effectiveness in terms of higher load carrying capacity, higher misalignment moment and lower friction parameter for a finite length misaligned journal bearing. However, though the dynamic characteristics of finite width journal bearings were analyzed using micropolar lubricants along with the stiffness, damping coefficients and the critical stability parameters of the journal by the linear stability theory.

The linear stability analysis has been carried out in the present paper using perturbation method around the steady-state equilibrium motion. Moreover, linear stability analysis becomes important for arriving at an estimate of critical mass parameter to be used as a guess value for nonlinear transient analysis to obtain the limit cycle of the journal centre trajectory.

II. MATHEMATICAL ANALYSIS

2.1 Modified Reynolds's equation[1]

The basic assumptions in micropolar lubrication to a journal bearing include the usual lubrication assumptions in deriving Reynold's equation and the assumptions to generalize the micropolar effects.

i. The Flow is Incompressible and steady, i.e. ρ= constant and $\frac{\partial \rho}{\partial t} = 0$.

ii. The flow is laminar i.e. free of vortices and turbulence.

iii. Body forces and body couples are negligible, i.e. FB=0 and CB=0.

iv. The variation of pressure across the film $\partial p/\partial y$ is negligibly small.

v. 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 translator velocities.

vi. No slip occurs at the bearing surfaces.

vii. Bearing surface 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.

viii. No fluid flow exists across the fluid film i.e. the lubrication characteristics are independent of y-direction.

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

$$\left.\begin{aligned} V &= [V_x(x,y,z), V_y(x,y,z), V_z(x,y,z)] \\ \upsilon &= [\upsilon_1(x,y,z), \upsilon_2(x,y,z), \upsilon_3(x,y,z)] \\ p &= p(x,y,z) \end{aligned}\right\}$$

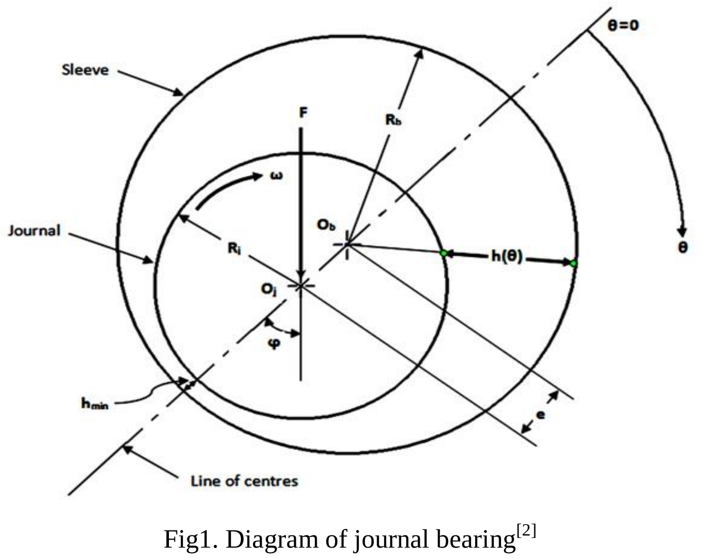


Fig1. Diagram of journal bearing[2]

The non- dimensional form of modified Reynold's equation for an incompressible fluid flow in micropolar lubrication is given by:

$$\frac{\delta}{\delta\theta}\left(\Phi\left(N,\overline{l_m},\bar{h}\right)\frac{\delta\bar{p}}{\delta\theta}\right)+\frac{\delta}{\delta\bar{z}}\left(\Phi\left(N,\overline{l_m},\bar{h}\right)\frac{\delta\bar{p}}{\delta\bar{z}}\right)=06\frac{\delta\bar{h}}{\delta\theta}+12\frac{\delta\bar{h}}{\delta\bar{t}}$$

For steady flow, $\frac{\delta\bar{h}}{\delta\bar{t}}=0$

$$\frac{\delta}{\delta\theta}\left(\Phi\left(N,\overline{l_m},\bar{h}\right)\frac{\delta\bar{p}}{\delta\theta}\right)+\frac{\delta}{\delta\bar{z}}\left(\Phi\left(N,\overline{l_m},\bar{h}\right)\frac{\delta\bar{p}}{\delta\bar{z}}\right)=6\frac{\delta\bar{h}}{\delta\theta}$$

Where, $\Phi(N,\overline{l_m},\bar{h})=\bar{h}^3+\frac{12\bar{h}}{\overline{l_m}^2}-\frac{6N\bar{h}^2}{\overline{l_m}}coth\left(\frac{N\bar{h}\overline{l_m}}{2}\right)$

2.2 Load carrying capacity

The hydrodynamic dimensionless forces developed in the bearing can be evaluated by integrating the dimensionless fluid pressure over the entire bearing area:

$$\overline{F_x}=\int_0^{\frac{L}{R}}\int_0^{\theta_2}\bar{p}\cos(\theta+\phi)\,d\theta\,d\bar{z} \qquad \overline{F_z}=\int_0^{L/R}\int_0^{\theta_2}\bar{p}\sin(\theta+\phi)\,d\theta\,d\bar{z}$$

2.3 Stiffness and Damping coefficients

With the dynamic pressure fields known, the non-dimensional components of stiffness and damping coefficients are obtained as follows:

$$\begin{bmatrix}\overline{K_{xx}} & \overline{K_{xz}}\\ \overline{K_{zx}} & \overline{K_{zz}}\end{bmatrix}=\begin{bmatrix}-\frac{\delta\overline{F}_x}{\delta\bar{x}_J} & -\frac{\delta\overline{F}_x}{\delta\bar{z}_J}\\ -\frac{\delta\overline{F}_z}{\delta\bar{x}_J} & -\frac{\delta\overline{F}_z}{\delta\bar{z}_J}\end{bmatrix}$$

$$\begin{bmatrix}\overline{C_{xx}} & \overline{C_{xz}}\\ \overline{C_{zx}} & \overline{C_{zz}}\end{bmatrix}=\begin{bmatrix}-\frac{\delta\overline{F}_x}{\delta\dot{\bar{x}}_J} & -\frac{\delta\overline{F}_x}{\delta\dot{\bar{z}}_J}\\ -\frac{\delta\overline{F}_z}{\delta\dot{\bar{x}}_J} & -\frac{\delta\overline{F}_z}{\delta\dot{\bar{z}}_J}\end{bmatrix}$$

2.4 Stability parameters

The linearized equation of motion can be rewritten as:

$$\begin{bmatrix}\overline{M}_J & 0\\ 0 & \overline{M}_J\end{bmatrix}\begin{bmatrix}\ddot{\bar{x}}\\ \ddot{\bar{z}}\end{bmatrix}+\begin{bmatrix}\overline{C_{xx}} & \overline{C_{xz}}\\ \overline{C_{zx}} & \overline{C_{zz}}\end{bmatrix}\begin{bmatrix}\dot{\bar{x}}\\ \dot{\bar{z}}\end{bmatrix}+\begin{bmatrix}\overline{K_{xx}} & \overline{K_{xz}}\\ \overline{K_{zx}} & \overline{K_{zz}}\end{bmatrix}\begin{bmatrix}\bar{x}\\ \bar{z}\end{bmatrix}=\begin{bmatrix}0\\ 0\end{bmatrix}$$

For a non-trivial solution, determinant of the above equations is given as:

$$\begin{vmatrix}\overline{M}_J\lambda^2+\lambda\overline{C_{xx}}+\overline{K_{xx}} & \lambda\overline{C_{xz}}+\overline{K_{xz}}\\ \lambda\overline{C_{zx}}+\overline{K_{zx}} & \overline{M}_J\lambda^2+\lambda\overline{C_{zz}}+\overline{K_{zz}}\end{vmatrix}=0$$

2.4.1 Critical mass

The critical mass is evaluated by using the stiffness and damping coefficients incorporated with Routh's-Hurwitz stability criteria. The non-dimensional critical mass $\overline{M}_c$ of the journal is expressed as:

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

Where,

$$a_0 = \overline{C_{xx}} * \overline{C_{zz}} - \overline{C_{xz}} * \overline{C_{zx}}$$

$$b_0 = \frac{(\overline{C_{xx}} + \overline{C_{zz}}) * (\overline{K_{xx}} * \overline{K_{zz}} - \overline{K_{zx}} * \overline{K_{xz}})}{(\overline{K_{xx}} * \overline{C_{zz}} + \overline{C_{xx}} * \overline{K_{zz}} - \overline{K_{zx}} * \overline{C_{xz}} - \overline{K_{xz}} * \overline{C_{zx}})}$$

$$c_0 = \frac{\overline{K_{xx}} * \overline{C_{zz}} + \overline{C_{xx}} * + \overline{K_{zx}} * \overline{C_{xz}} + \overline{K_{xz}} * \overline{C_{zx}}}{\overline{C_{xx}} + \overline{C_{zz}}}$$

2.4.2 Whirl frequency ratio

It is the ratio of the rotor whirl or processionals frequency to the rotor onset speed at instability. If the equivalent stiffness is less than or equal to zero, reflects that the whirl is absent.

$$\overline{\omega} = \sqrt{\frac{(\overline{K_{xx}} - \overline{K_{eq}}) * (\overline{K_{zz}} - \overline{K_{eq}}) - \overline{K_{zx}} * \overline{K_{xz}}}{\overline{C_{xx}} * \overline{C_{zz}} - \overline{C_{zx}} * \overline{C_{xz}}}}$$

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

2.4.3 Threshold speed

Threshold speed is the utmost value of speed at which the bearing remains stable. A further increase in rotational speed will induce sudden and violent whirling at a sub synchronous frequency that is equal to a natural frequency of the rotor-bearing system. Threshold speed is obtained using the relation given below:

$$\overline{\Omega_s} = \sqrt{\frac{\overline{K_{eq}}}{\overline{K_{th}}}}$$

Where,

$$\overline{K_{th}} = \left[\frac{(\overline{K_{eq}} - \overline{K_{xx}}) * (\overline{K_{eq}} - \overline{K_{zz}}) - (\overline{K_{xz}} * \overline{K_{zx}})}{\overline{C_{xx}} + \overline{C_{zz}}}\right]$$

III. SOLUTION PROCEDURE

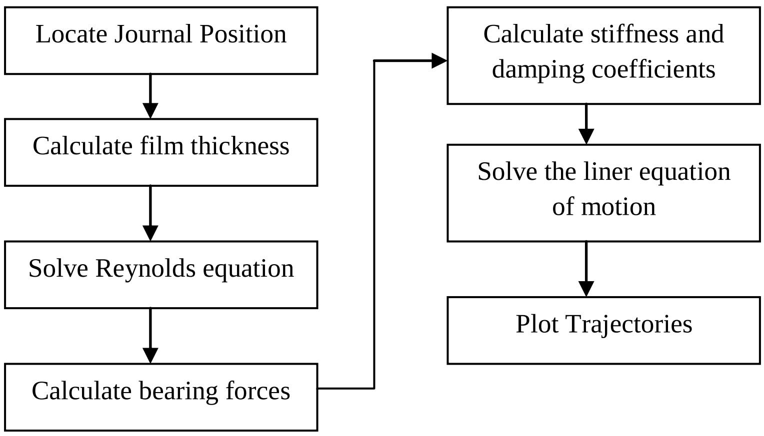
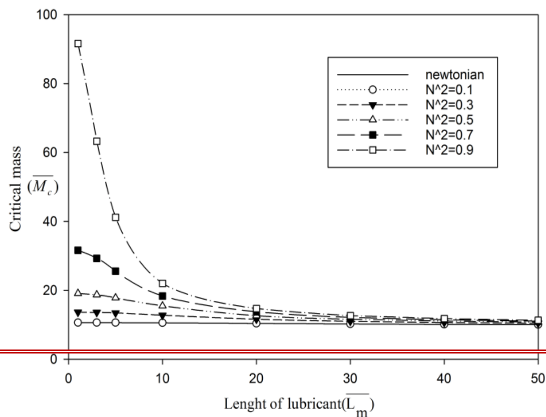


Fig2. Solution Process



IV. RESULT AND DISCUSSION

Since the components of stiffness and damping coefficients and the critical mass parameter representing the stability of the journal are

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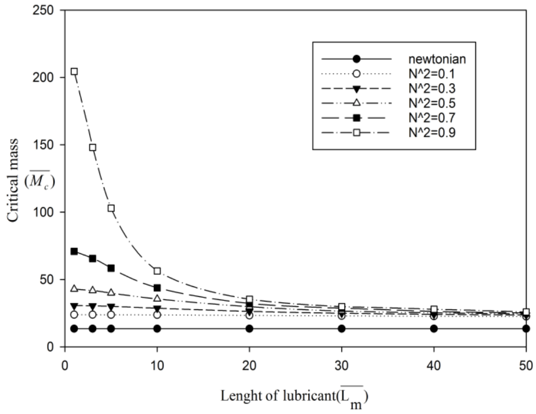
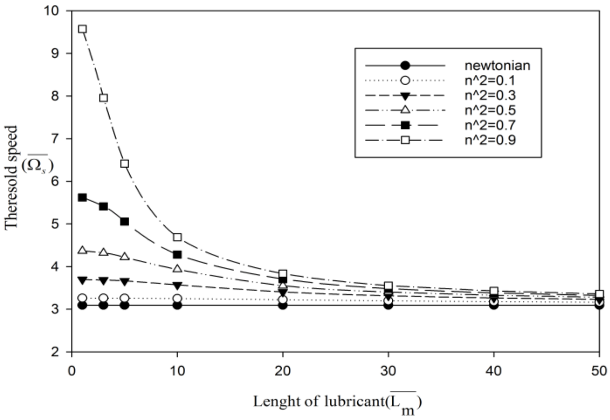
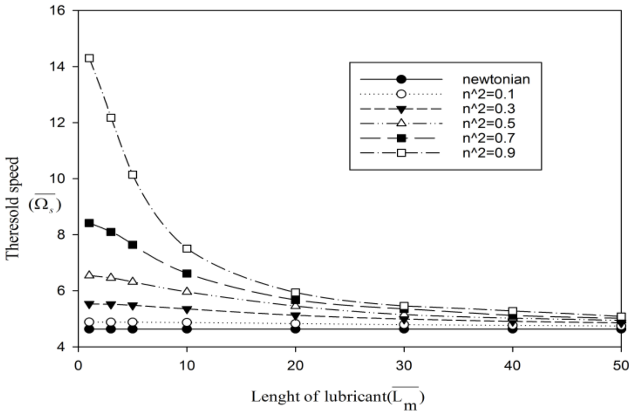
dependent on the steady state and perturbed film pressures depend upon the micropolar parameters length of lubricant($\overline{L_m}$), coupling number(N). The parametric studies are done to show their effects on the critical mass parameter and threshold-speed.



Comparison of critical mass and length of lubricants

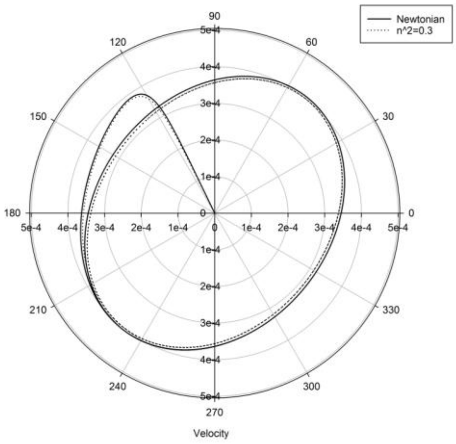
Fig3. For eccentricity 0.3 Fig4.For eccentricity 0.5





Comparison of Threshold speed and length of lubricants

Fig5. Foe eccentricity 0.3 Fig6. For eccentricity 0.5



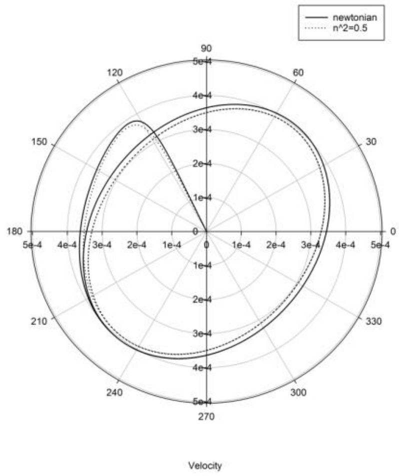


Fig7. For eccentricity 0.3 and N^2=0.3 Fig8. For eccentricity 0.3 and N^2=0.5

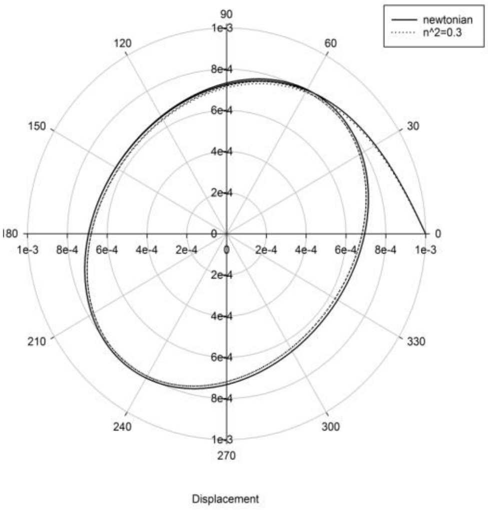


Fig9. For eccentricity 0.3 and N^2=0.3

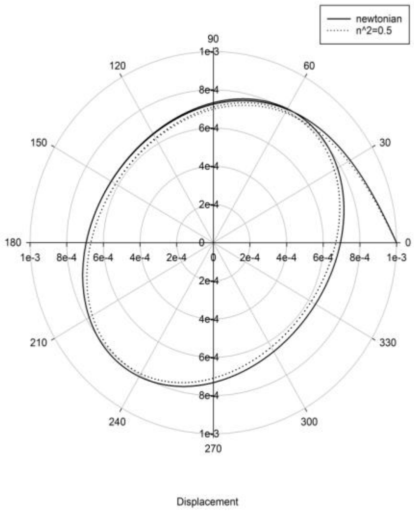


Fig10. For eccentricity 0.3 and N^2=0.5

Fig 3, 4, 5, 6 gives the comparison of critical mass and threshold speed parameters in case of Newtonian and non-Newtonian lubricants for different eccentricities (e=0.3,0.5).

For eccentricity 0.3 the value of critical mass in case of Newtonian fluid is approximate 9.5 and for eccentricity 0.5 values is approximate 21.4 which is lower value then non-Newtonian fluid. In case of Newtonian fluid value of critical mass is not effect by length of lubricants. The comparison made for different length of lubricant. For higher value of coupling number value of critical mass parameter is higher than the lower coupling number as well as Newtonian fluid. This comparison shows that for lower length of lubricant and higher coupling number parameter critical mass value is increasing.

In case of threshold speed values for eccentricity 0.3 and 0.5 are respectively approximately 3 and 4.6 which is lower than non-Newtonian fluid. Result is compare with different value of length of lubricants. Value of threshold speed in case of Newtonian is not varying with the length of lubricants. With the increasing value of coupling number the values of threshold speed is increase. Also value is increasing for shorter characteristic length of lubricated bearing.

Fig 7, 8, 9, 10 gives the comparison of velocity and displacement trajectory for both Newtonian and non-Newtonian lubricants for different eccentricities (e=0.3, 0.5).

Displacement and velocity profile of center of journal are given for eccentricity 0.3 in case of Newtonian and non-Newtonian (n^2=0.3, 0.5).For different time value of velocity and displacement is find with reference of center of bearing. This comparison shows that non-Newtonian lubricated circular bearing give more stable displacement and velocity compare to Newtonian lubricated circular bearing. In case of 0.5 coupling number both velocity and displacement trajectories are more closed from the center of shaft compare to 0.3 coupling number trajectories. So with increasing the coupling number stability of bearing increases.

Comparing the result with previous published result by R. Sinhasan and K. C. Goyal[4] it gives approximate same result in case of eccentricity 0.3.

V. CONCLUSION

- The values of critical mass and threshold speed for non-Newtonian lubricated bearings are higher than Newtonian lubricated bearing. So bearings lubricated form non-Newtonian fluid is more stable compare to Newtonian fluid. For shorter characteristic length ($\overline{L_m}$) bearings the values of critical mass and threshold speed are higher than longer characteristic length ($\overline{L_m}$) bearings.
- In case of non-Newtonian lubricated bearings displacement and velocity trajectories are closer from the center of shaft compare to Newtonian lubricated bearing so stability is more in non-Newtonian fluid compare to Newtonian fluid.
- With increasing coupling number trajectories are more closed from center of shaft so stability is increasing with increasing coupling number.

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