Analysis of Plain Circular Journal Bearing lubricated with Micropolar Fluid

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Abstract— The research area of hydrodynamic lubrication is widely diversified. The literature includes the analysis and various optimized technique of determination of static and dynamic characteristics of plain circular bearing. However these studies were unable to describe the simultaneous effect of both micropolar fluid parameters and eccentricity ratio on bearing characteristics. In the present study combined effect of micropolar parameters and eccentricity ratios are considered with wide range of variation on the bearing performance characteristics. For linear studies critical mass of journal is determined by Routh-Hurwitz stability criteria. Comparative analysis on critical values of journal mass required for both linear and nonlinear studies is carried out. Also the stability of these bearing are analyzed at various operating conditions.

Keywords— Journal Bearing, Hydrodynamic Lubrication, static characteristics, Dynamic characteristics, eccentricity, eccentricity ratio, comparative analysis.

INTRODUCTION

As per the history a theory of hydrodynamics in lubrication was introduced by Reynolds in 1886. Many numbers of researcher performed analytical and practical studies over hydrodynamic lubrication and put very important concepts and theories. Analysis of literature showed that the research area of hydrodynamic lubrication is widely diversified. The literature includes the analysis and various optimized technique of determination of static and dynamic characteristics of plain circular bearing.

The bearings have been in use for centuries but the recent developments in science and technology demands critical designs of bearings with high precision and optimized performance for the most adverse working conditions. The rapid developments in the fields of rocketry and missile technology, cryogenics, aeronautics and space engineering, nuclear engineering, electronics, computer sciences and technologies, bio-medical engineering and a lot more fields in science and technology make the aspects of designing bearings more and more challenging and innovative. Moreover, the mode, time and place of operations demand exploration of new materials, lubricants and even lubrication theories and technologies.

I. MATHEMATICAL ANALYSIS

BASIC ASSUMPTIONS

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 $\partial p/\partial t = 0$.

(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:
$V= [Vx\ (x,y,z\ ),\ Vy\ (x,y,z\ ), Vz\ (x,y,z)]$
$\upsilon= [\upsilon 1\ (x,y,z)\ ,\upsilon 2\ (x,y,z\ ),\upsilon 3\ (x,y,z)]$
$p=p\ (x,y,z)$

Principle of Conservation of Mass

$\partial\rho/\partial t+\nabla\ (\rho V) =0$ (1)

Principle of Conservation of Linear Momentum

$(\lambda+2\mu)\nabla(\nabla.V)-[(2\mu+\mathcal{X})/2]\nabla*(\nabla*V)+\mathcal{X}\nabla*\upsilon-\nabla.\pi+FB=\rho*DV/Dt$ (2)

Principle of Conservation of Angular Momentum $(\alpha+\beta+\gamma)\nabla(\nabla.\upsilon)-\gamma\nabla*(\nabla*\upsilon)+\mathcal{X}\nabla*V-2\mathcal{X}\upsilon+CB=\rho jD\upsilon/Dt$ (3)

Where, ρ is the mass density, V is the velocity vector, υ is the micro-rotational velocity vector. π is the thermodynamic pressure and is to be replaced by the hydrodynamic film pressure, p, since, $\pi=-\ [\partial E/\partial\rho^{-1}\ ] = p$. Where E is the internal energy and p is to be determined by the boundary conditions. λ and μ are the familiar viscosity coefficients of the classical fluid mechanics, while α,β and γ are the new viscosity coefficients derived as the combinational effects of the gyroviscosities for the micropolar fluid as defined by ERINGEN**.** $\mathcal{X}$ is also a new viscosity coefficient for micropolar fluid, termed as spin viscosity, which establishes the link between velocity vector and the microrotational velocity vector. *FB* is the body force per unit mass, *CB* is the body couple per unit mass and j is the microinertia constant. D/Dt represents the material derivative. The constitutive equations of micropolar are

$tk1= (-\pi+\lambda Vr,)1+ (\mu-1/2*\mathcal{X})\ (Vk,1+V1,k)\ +\mathcal{X}(V1,k+\eta k1r*\upsilon r)$ (4)
$mk1 = \alpha\upsilon r,1+\beta\upsilon k,1+\gamma\upsilon 1,k$ (5)
Where, $tk1$ and $mk1$ are the stress tensor and the couple stress tensor respectively. $\eta k1r$ is an permutation tensor. $\delta k1$ is Kronekar delta. The index following a prime represents the partial derivative to spatial variable $\mathcal{X}k$.

Note that for $\alpha=\beta=\gamma=\mathcal{X}=0$ and for negligible body couple per unit mass equation (3) yields $\upsilon=0$ and so, equation (2) reduces to the classical Navier-Stokes equation. For $\mathcal{X}=0$ the velocity vector and the microrotational velocity vector are uncoupled and the global motion of the fluid becomes free of the microrotation and their effects.

The theoretical prediction of hydrodynamic pressures in the bearing is obtained by the solution of modified Reynolds equation satisfying the appropriate boundary conditions. The steady state and dynamic pressure profile is obtained by finite difference technique.

II. SOLUTION PROCEDURE

Journal bearing systems are analyzed by linear and nonlinear study with the help of MATLAB software incorporation of PDE toolbox. The analysis is carried out by linear and nonlinear study of journal bearing system. These studies are carried out for circular bearing geometries.

Operating conditions of journal bearing system can be varied by combination of characteristic length of the micropolar lubricant ($\overline{l_m}$), Coupling number (N), and eccentricity ratio. Hence with the help of these programs one can obtain results over wide range. Hence it becomes necessary to execute a program at each operating condition separately.

Solution procedure for linear analysis of a plain circular journal bearing with micropolar fluid

1. Acquire input parameters such as attitude angle (ϕ) = 60o, the initial guess for fluid film extent are specified by considering circular coordinate axis X originating from line of centers and Y axis along bearing width. Hence For present finite width bearing Xmax=180o, Xmin=0o, Ymax= 2 (since $\beta = 2$) and Ymin= 0. Characteristic length of the micropolar lubricant. ($\overline{l_m}$), Coupling number (N) and eccentricity ratio ($\bar{\varepsilon}$) specifies the various operating conditions and it acts as variable.
2. Journal centres are located as ($\overline{X_j}$, $\overline{Y_j}$) using Cartesian co-ordinate system originated at geometric centre of bearing.
3. In order to get the solution of PDE finite difference method is employed in practice hence fluid film domain is discretized into optimum mesh size.
4. Fluid film thickness is determined at centroid of each elemental area by using thickness equation.
5. Modified Reynolds equation is solved by using PDE toolbox.
6. The pressure distribution over fluid film thickness is calculated.
7. On the basis of pressure distribution pressure gradient $\left(\frac{dp}{dx}\right)$ is determined along X axis between a mid-node on trailing edge and nearest node. If the pressure gradient $\left(\frac{dp}{dx}\right)$ becomes 'negative' i.e., termination of positive fluid pressure zone and negative pressure start buildings onwards.
8. Bearing load components along horizontal and vertical directions are calculated.
9. In the present case bearing is subjected pure radial this condition is satisfied when bearing load ratio (fx/fy) tends to zero. Here bearing load ratio less than 0.001 is considered as bearing subjected to pure radial load. This load ratio can be reduced to desired value by adjusting the attitude angle in iterations. In each iteration attitude angle is modified by 10^{o}.
10. Once the attitude angle and trailing edge determined the equilibrium position of journal is located.
11. Bearing load calculated in step (8) are considered as static equilibrium forces.
12. For new displaced position of journal again instantaneous bearing load components are calculated followed by steps (4) to (6).
13. On the basis of Routh- Hurwitz stability criteria critical mass is calculated using equation.

Other dynamic characteristics such as whirl frequency and threshold speed ratios are calculated

IV ANALYSIS RESULTS

1 STATIC CHARACTERISTIC

1.1. LOAD CARRYING CAPACITY

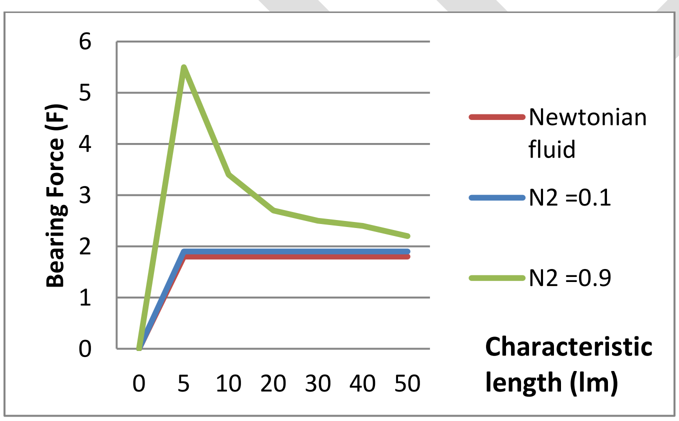

Fig.1 Variation of Bearing force as a function of non-dimensional Characteristic length (lm) at ϵ=0.3

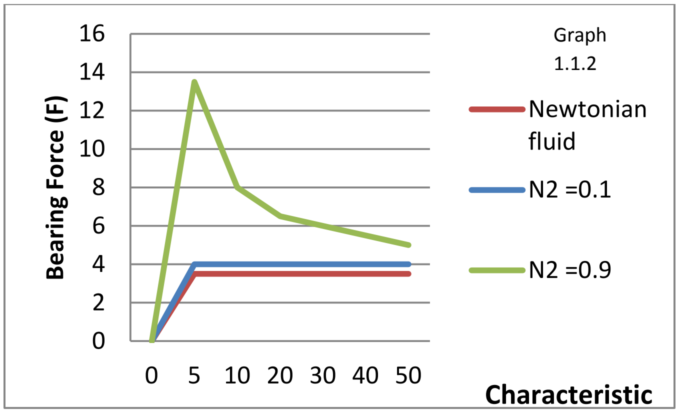

Fig.2 Variation of Bearing force as a function of non-dimensional Characteristic length (lm) at ϵ=0.5

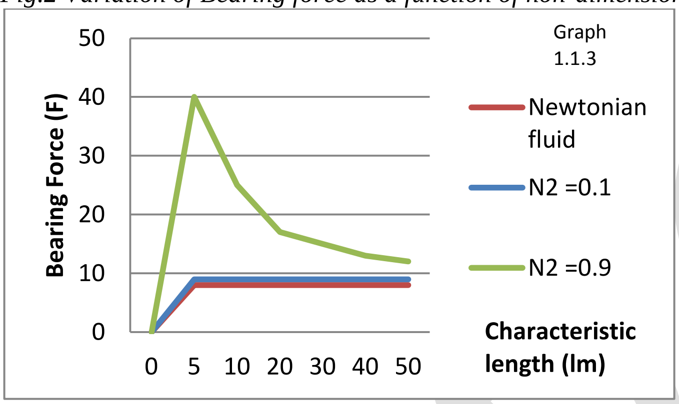

Fig.3 Variation of Bearing force as a function of non-dimensional Characteristic length (lm) at ϵ=0.7

The load carrying capacity reduces as coupling number reduces and approaches to Newtonian fluid for $N2 \rightarrow 0$. It shows that for high coupling number and low characteristic length a plain circular journal bearing lubricated with micropolar fluid provides maximum load carrying capacity as compared to the Newtonian fluid.

It can be seen that for a particular value of *lm* and *N2*, the load carrying capacity increases for micropolar fluid as well as for Newtonian fluid with increase in eccentricity ratio. It is also been observed that the load carrying capacity is higher i.e. 4-5 times as the eccentricity ratio changes from 0.3 to 0.7 for a distinct value of *lm* and *N2***.** The figures depict that the load capacity at any eccentricity ratio is much higher at lower value of *lm* and approaches to Newtonian fluid as $lm \rightarrow \infty$.or $N2 \rightarrow 0$

1.2 ATTITUDE ANGLE (Φ)

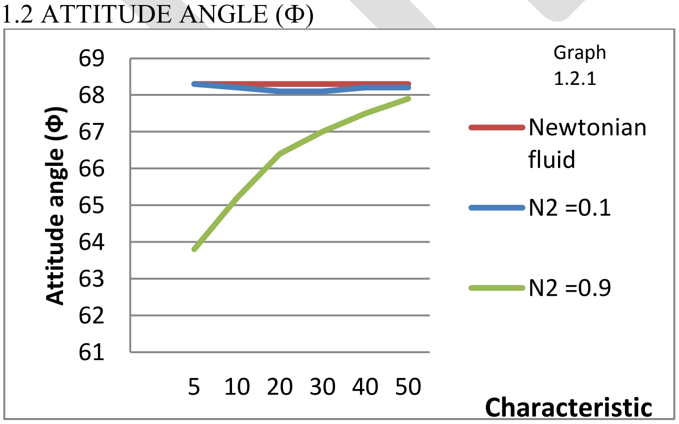

Fig.4 Variation of Attitude angle (ϕ) as a function of non-dimensional Characteristic length (lm) at ϵ=0.3

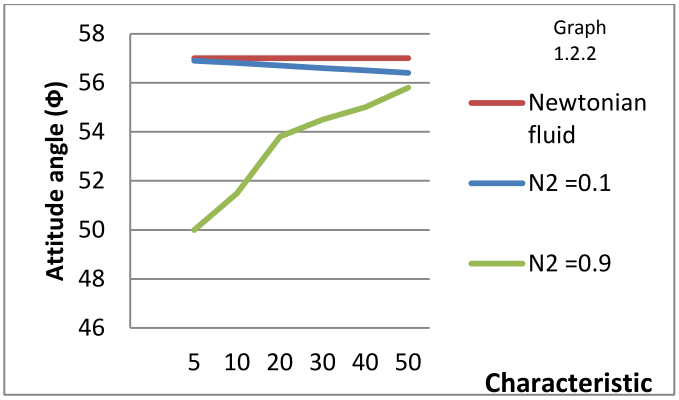

Fig.5 Variation of Attitude angle (ϕ) as a function of non-dimensional Characteristic length (lm) at ϵ=0.5

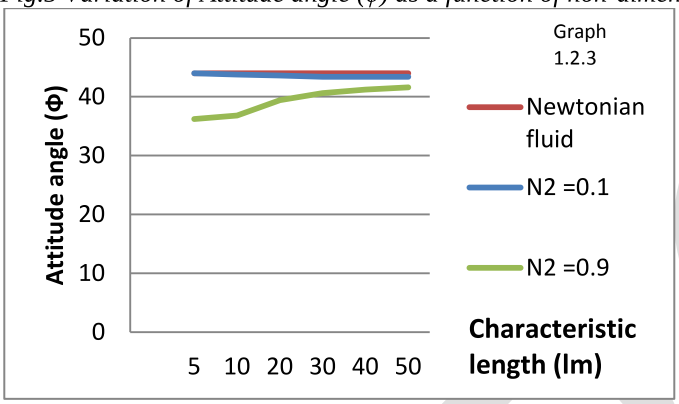

Fig.6 Variation of Attitude angle (ϕ) as a function of non-dimensional Characteristic length (lm) at ϵ=0.7

The attitude angle reduces as coupling number increases and approaches to Newtonian fluid for $N2 \rightarrow 0$. It shows that for high coupling number and low characteristic length a plain circular journal bearing lubricated with micropolar fluid provide low attitude angle as compared to the Newtonian fluid.

It can be seen that for a particular value of lm and N2, the attitude angle decreases for micropolar fluid as well as for Newtonian fluid with increase in eccentricity

2. DYNAMIC CHARACTERISTICS

2.1 CRITICAL MASS PARAMETER (*Mc*)

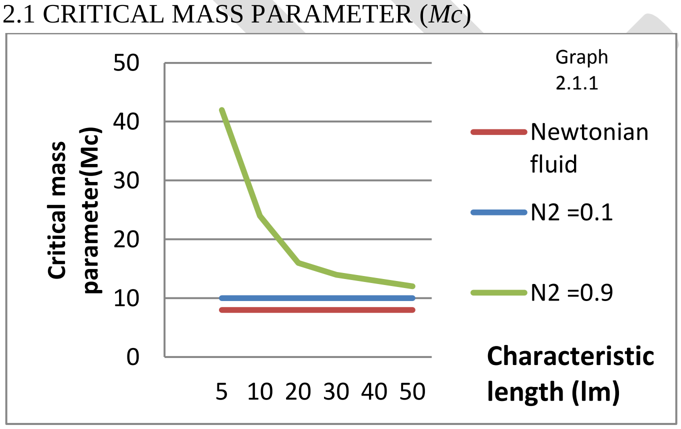

Fig. 7 Variation of Critical Mass (Mc) as a function of non-dimensional Characteristic length (lm) at ϵ=0.3

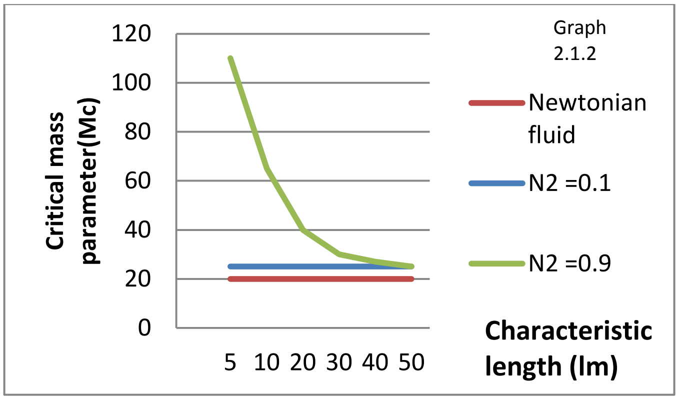

Fig. 8 Variation of Critical Mass (Mc) as a function of non-dimensional Characteristic length (lm) at ϵ=0.5

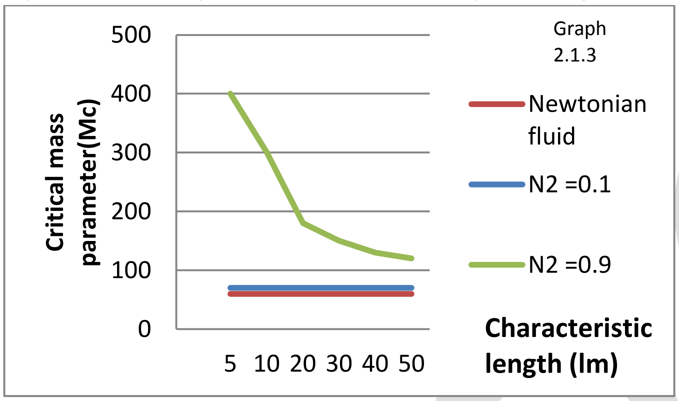

Fig. 9 Variation of Critical Mass (Mc) as a function of non-dimensional Characteristic length (lm) at ϵ=0.7

The critical mass parameter increases as N is increased. It also has been found that when $N2 \rightarrow 0$ for any value of lm, micropolar fluid approaches to Newtonian fluid. It can be observed from figure that critical mass decreases as increasing lm and approaches to Newtonian fluid as lm grows indefinitely

2.2 WHIRL FREQUENCY RATIO

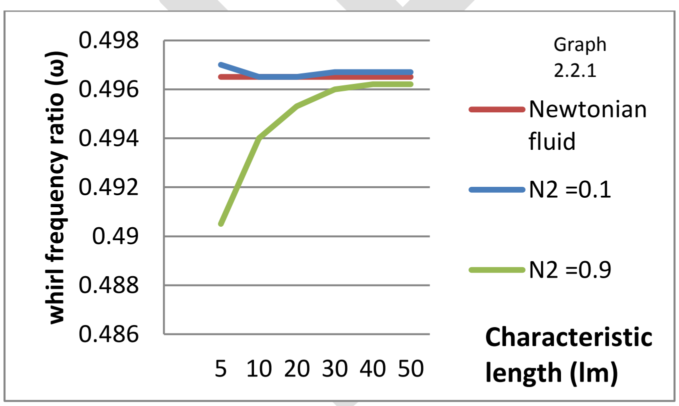

Fig.10 Variation of Whirl frequency ratio (ω) as a function of non-dimensional Characteristic length (lm) at ϵ=0.3

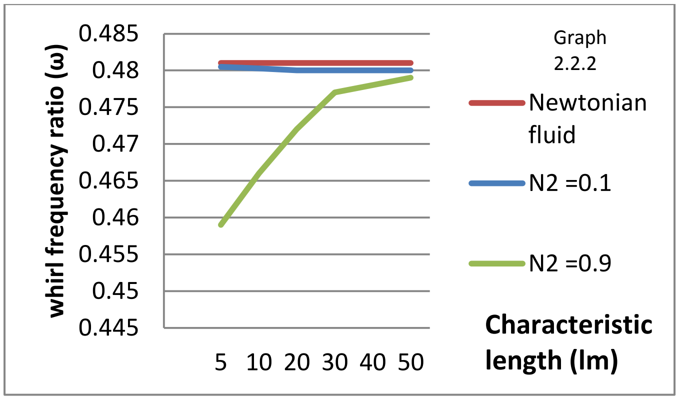

Fig. 11 Variation of Whirl frequency ratio (ω) as a function of non-dimensional Characteristic length (lm) at ϵ=0.5

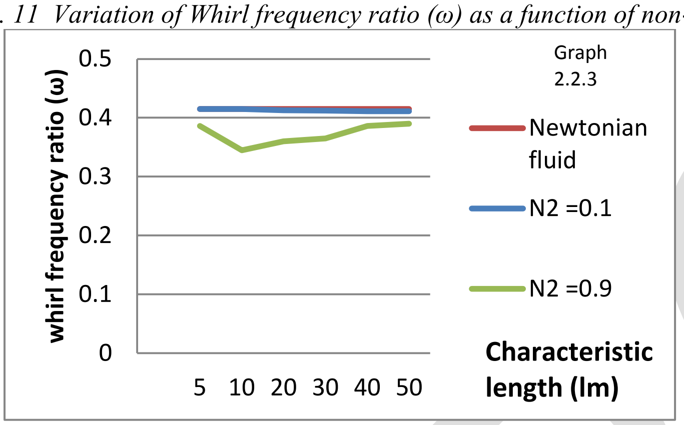

Fig. 12 Variation of Whirl frequency ratio (ω) as a function of non-dimensional Characteristic length (lm) at ϵ=0.7

It observed that for a particular value of coupling number, whirl frequency ratio firstly decreases than increases with increase in *lm*. It clearly shows that there is a decrement in the whirl frequency ratio at small values of *lm*.

It can be seen that for a particular value of *lm* and *N2*, whirl frequency ratio decreases for micropolar fluid as well as for Newtonian fluid with increase in eccentricity ratio.

2.3 THREESHOLD SPEED

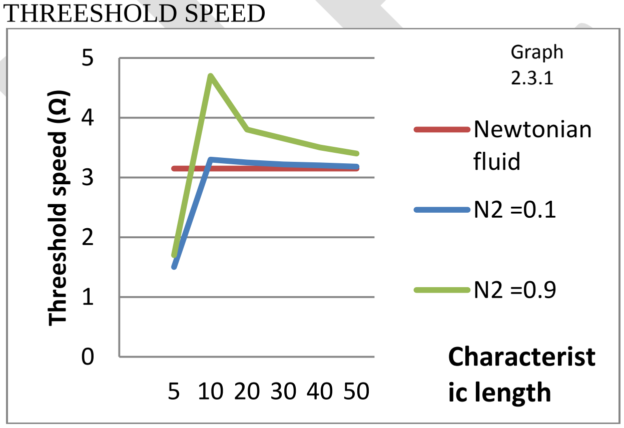

Fig. 13 Variation of Threshold Speed (Ω) as a function of non-dimensional Characteristic length (lm) at ϵ=0.3

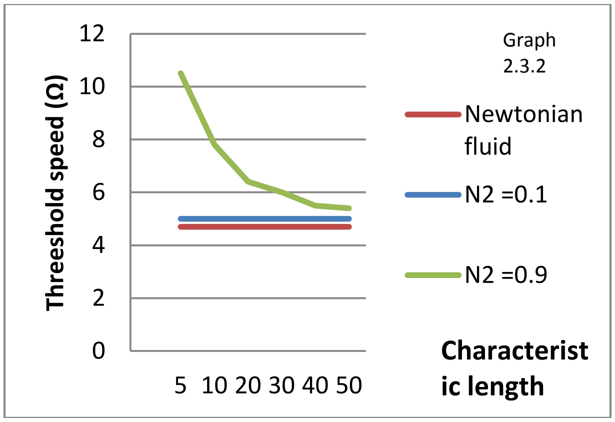

Fig. 14 Variation of Threshold Speed (Ω) as a function of non-dimensional Characteristic length (lm) at ϵ=0.5

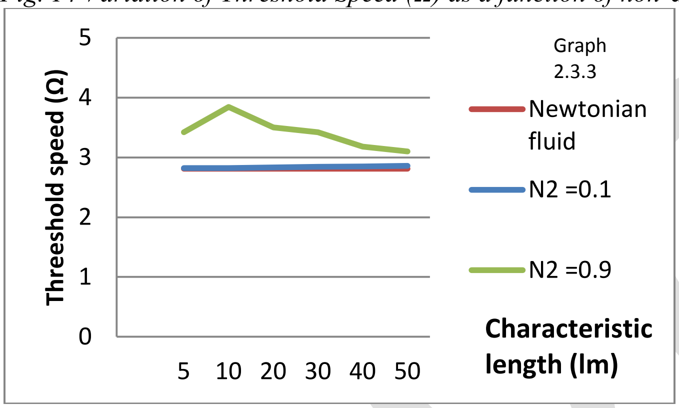

Fig. 15 Variation of Threshold Speed (Ω) as a function of non-dimensional Characteristic length (lm) at ϵ=0.7

we can see that as the coupling number decreases threshold speed decreases. It observed that for a particular value of coupling number, threshold speed increases with increase in *lm*. It clearly shows that there is an augmentation in the threshold speed at small values of lm. Curves illustrate that when lm increases or N2 decreases, the value of threshold speed approaches to that of the Newtonian value.

2.3 STIFFNESS COEFFICIENT

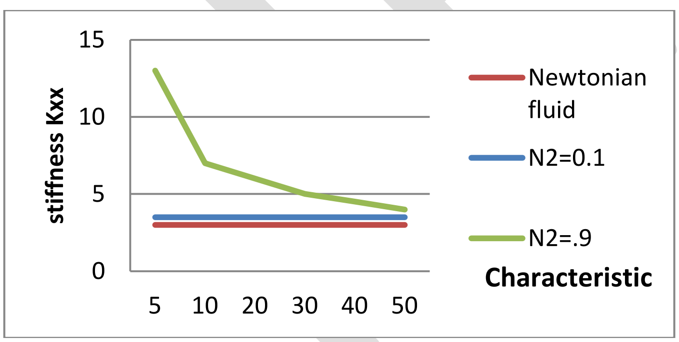

Fig.16 Variation of Stiffness (Kxx) as a function of non-dimensional Characteristic length (lm) at ϵ =0.3

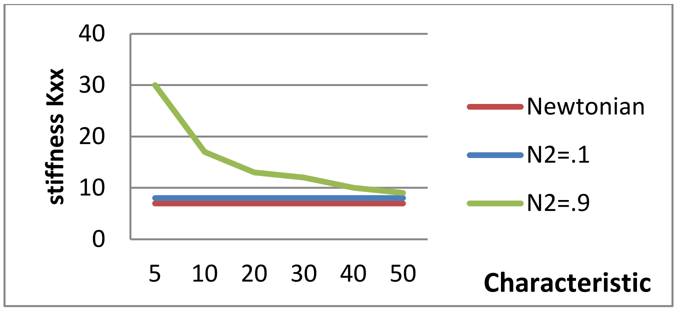

Fig.17 Variation of Stiffness (Kxx) as a function of non-dimensional Characteristic length (lm) at ϵ =0.5

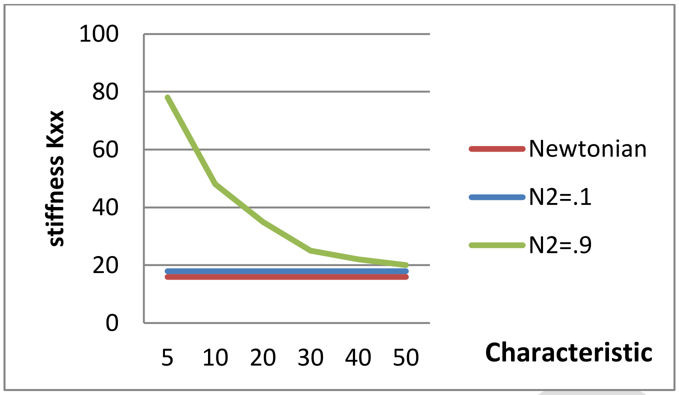

Fig.18 Variation of Stiffness (Kxx) as a function of non-dimensional Characteristic length (lm) at ϵ =0.7

Figs. shows the variation of the non- dimensional components of stiffness coefficients as function of lm for any coupling number, keeping eccentricity ratio and L/R constant at 0.3 and 2 respectively. It can be observed from the figures that at any value of lm, the direct stiffness coefficient Kxx increases with increase in coupling number. For any value of coupling number, Kxx decreases with lm and approaches to Newtonian fluid as lm becomes infinitely large.

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CONCLUSION

Important conclusion can be made about the use of micropolar fluid in plain circular and elliptical bearing are as follows:

1. It has been observe that load carrying capacity of both the journal bearing with micropolar lubricant at a particular eccentricity ratio increases when compared with that of bearing with Newtonian fluid.

2. The micropolar fluid approaches to Newtonian fluid as characteristic length of the micropolar fluid grows indefinitely or coupling number tends to zero.

3. The critical mass and threshold speed for bearings under micropolar fluid is increases for high coupling number and decreases when characteristic length decreases. Hence, stability of a bearing increases at high coupling number and low characteristic length.

4. The analysis predicts a lower value of critical mass for Newtonian than micropolar fluid.

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