Analysis of Performance Characteristics of Journal Bearing with Micropolar Fluid and Comparison to Newtonian Fluid

Sachin Kumar

***Abstract:** The present work deals with the dynamic behaviour of a Plane Journal Bearing working in condition of Micropolar lubrication. As from the characteristics of Micropolar fluid the Plane Journal Bearing is being observed under the increase in fluid film pressure and fluid film thickness but a decrease in the side flow as compared to Newtonian fluids. On the basis of the theory of micropolar fluids, the modified Reynolds's equation for dynamic loads is derived. Results from the numerical analysis indicated that the effects of micropolar fluids on the performance of a dynamically loaded journal bearing are evaluated. Applying the half sommerfeld's boundary conditions, the pressure distribution in journal bearing is obtained and the dynamic characteristics in terms of the components of stiffness and damping coefficients, friction drag and side flow obtained with respect to the micropolar property for varying eccentricity ratios. The results show that micropolar fluid exhibits better stability in comparison with Newtonian fluid. The finite element analysis of journal bearing has done by using programming software package MATLAB*

***Index Terms:** Finite Element Analysis, Newtonian Fluid, Reynolds's Equation, Micropolar Fluid*

I. INTRODUCTION

Journal bearings are very common engineering components and are used in almost all types of machinery. Combustion engines and turbines virtually depend on journal bearings to obtain high efficiency and reliability. A journal bearing consists of a shaft rotating within a stationary bush. The hydrodynamic film which supports the load is generated between the moving surfaces of the shaft and the bush.

The sliding action is acting along the circumference of a circle or an arc of a circular and carrying radial loads are known as journal or sleeve bearing. In these bearings, the forms a sleeve around a shaft. The part of the shaft which rotates in the bearing is called "journal". The journal bearing are classified as:

1. FULL BEARING: In these bearings, the bearing completely envelops the journal. This type of bearing is commonly used in Industrial machinery to accommodate bearing loads in any direction.
2. PARTIAL BEARING: The enveloping angle is not 360^o but is 120^o .The full bearing and partial bearings are also known as clearance bearings. The friction in a partial bearing will be less than in Full bearing, but its applications are limited to only those situations where the load is always in one direction.
3. FITTED BEARING: This is a special case of partial bearing in which the sizes of the journal and bearing are equal and hence there is no clearance.

Eringen was the first scholar to propose the theory of micro polar fluids to simplify the traditional analysis of micro polar fluids. In 1972, Eringen also proposed the theory of thermo-micro fluids, and it was also succeeded to interpret the behavior of some kind on Non-Newtonian fluids. In 1975, Shukla and Isa presented the generalized Reynolds's equation for micropolar fluid with the applications of one- dimensional slider bearings. Meanwhile, Prakash and Sinha analysed the infinitely long journal bearing with the lubrication of micropolar fluid. In 1978, Zaheeruddin and Isa learned the steady- state characteristics for one-dimensional journal bearings and took both the infinitely short and long bearing under micropolar lubrication into consideration. In 1989, Khonsari and Brewe also presented the analysis of the performance of finite journal bearing lubricated with micropolar fluid. In 1996, Lin performed the analysis of lubricants and three dimensional irregularities. In 2002, Das et al. studied the performance of the steady-state characteristics of Hydrodynamic journal bearings with the consideration of misalignment. In 2004, Wang and Zhu presented a study of the lubricating effect of micropolar fluids in a dynamically loaded journal bearing. It was found that the application of micropolar fluid lubricants can increase the fluid film pressure and fluid film thickness, and decrease the side leakage flow. In 2005, Das et al proposed the linear stability analysis for the Hydrodynamic journal bearing lubricated with Newtonian fluids. In November 2010 by N.B. Naduvinamani, S.Santosh, the effect of micropolar fluid on the static and dynamic characteristics of squeeze film lubrication in finite porous journal bearings was studied. The finite modified Reynolds equation was solved numerically using the finite difference technique and the squeeze film characteristics are obtained. According to the results obtained, the micropolar fluid effect significantly increases the squeeze film pressure and the load-carrying capacity as compared to the corresponding Newtonian case. Under cyclic load, the effect of micropolar fluid is to reduce the velocity of the journal centre. Effect of porous-matrix is to reduce the film pressure, load-carrying capacity and to increase the journal centre velocity. In May 2011 by Xiao-Li Wang n, Jun-Yan Zhang, HuiDong, Numerical computation of the dynamically loaded journal bearings lubricated with micropolar fluids

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Was undertaken based on the improved Elord cavitation algorithm and over-relaxation method. The results show that the average inflow and average outflow based on the mass conservation boundary conditions are almost equal, which is in accordance with the fact. However, under the Reynolds boundary conditions, large difference is shown between the average inflow and outflow. It is also demonstrated that with the micropolar fluids lubrication, the minimum film thickness, bearing capacity and friction power loss are increased while the maximum film pressure is decreased.

A. Problem Formulation

In the present work, the hydrodynamic analysis of Plain circular Journal bearing with micropolar lubrication is studied. For this purpose a plain circular journal bearing is taken. The behavior of the system is analyzed using the Reynolds's Equation of plain circular bearing lubrication. For that the Reynolds's equation in the non-dimensionalised form has been solved using the CAE tool software package MATLAB.

In the problem we had taken a plain journal bearing of length L^2=12, width to diameter ratio B/D ratio 1.0, $\pi/10$, eccentricity ratio $\epsilon = 0.5$, dimensionless parameter N^2= 1/3. As the journal is analyzed under the micropolar lubrication of viscosity 0.04 and attitude angle 40^o. The questions to analyze with the model are:

- What amount of pressure distribution in model subjected to micropolar lubrication?
- The calculation of various characteristics for the journal bearing i.e. stiffness, load capacity , damping coefficient etc.

To analyze the pressure distribution, the comparison of micropolar lubrication with Newtonian lubrication has been studied. The Reynolds's equation in dimensionless form has been evaluated for both lubrications i.e. for micropolar:

$$\frac{\partial}{\partial\theta}\left\{\bar{h}^3\,\bar{g}\,\frac{\partial\bar{p}}{\partial\theta}\right\}+\left(\frac{D}{B}\right)^2\frac{\partial}{\partial\xi}\left\{\bar{h}^3\,\bar{g}\,\frac{\partial\bar{p}}{\partial\xi}\right\}=\frac{-1}{2}\,\epsilon\sin\theta$$

Where

$$\bar{g}(\bar{N}, L, \bar{h}) = \frac{1}{12}+\left(\frac{1}{L\bar{h}}\right)^2-\frac{N}{2L\bar{h}}\coth\left(\frac{NL\bar{h}}{2}\right)$$

Subjected to boundary conditions

$$\bar{p}(0,\xi)=\bar{p}(2\pi,\xi)=0$$

$$\bar{p}(\theta,1)=\bar{p}(\theta,-1)=0$$

$$\bar{p}(\theta,\xi)=\bar{p}(\theta,-\xi)$$

$$p(\theta,\xi)\geqslant 0$$

Using the half Summerfield boundary conditions for the half section of the bearing model has been used i.e.

P=0 at $\theta = 0$ and π

For the various characteristics like Load intensity , Stiffness and Damping coefficient the different conditions are analyzed. For the load intensity the two load components in X-direction and in Y-direction are taken into account i.e.

$$\bar{W}_{\pi/2} = \bar{W}\sin\phi = \int_0^{\pi} P\,\mathrm{din}\,\theta\,d\theta$$

And

$$\bar{W}_0 = \bar{W}\cos\phi = \int_0^{\pi} P\cos\theta\,d\theta$$

While the resultants load intensity is:

$$\bar{W} = (\bar{W}_0^{\,2} + \bar{W}_{\pi/2}^2)^{1/2}$$

To calculate the stiffness the condition of displacement/deflection of journal with respect to bearing has been analyzed that when the journal is displaced from original position to slightly in X-direction of an amount Δx , so that the stiffness components are obtained such as K_{xx} , K_{yx} . K_{xx} is the stiffness component in X-direction due to load intensity in X-direction, whereas K_{yx} is the stiffness component in X-direction due to load component in Y-direction. i.e.

$$K_{xx} = (F_x - F_x`)/\Delta x$$

$$K_{yx} = (F_y - F_y`)/\Delta x.$$

Similarly when the journal get displaced in Y-direction there are also two stiffness components due to two load vectors .when the journal get displaced in Y-direction of an amount Δy , so in resultant the stiffness components are be K_{yy} and $K_{xy.}$ i.e. the K_{yy} is the stiffness components due to load vector in Y-direction and displacement is also in same direction and K_{xy} is the stiffness component in Y-direction due to load vector in X-direction i.e.

$$K_{yy} = (F_y - F_y`)/\Delta y .$$

$$K_{xy} = (F_x - F_x`)/\Delta y.$$

Similarly the damping coefficients are evaluated in same manner as in case of stiffness. The damping effect are analysed when the journal are having rotational motion and the behavior of load components are taken into account i.e.

$$D_{xx} = (F_x - F_x')/ \Delta x' .$$

Represent the damping coefficient in X-directional motion due to load component in X-direction.

$$D_{yx} = (F_y – F_y')/ \Delta x' .$$

Represent the damping coefficient in X-directional motion due to load component in Y-direction.

$$D_{yy} = (F_y – F_y')/ \Delta y' .$$

Represent the damping coefficient in Y-directional motion due to load component in Y-direction.

$$D_{xy} = (F_x - F_x')/ \Delta y' .$$

Represent the damping coefficient in Y-directional motion due to load component in X-direction.

II. METHODOLOGY

Reynolds' Equation:

Under the usual assumptions made for the lubrication film, the generalized Reynolds equation is derived as for eqn.

(53) in ref. 7. Particularly, for the special conditions with Uiz = Uzi = Uz2 = 0, eqn. (53) of ref. 7 will provide the Reynolds equation for the problem considered in this study.

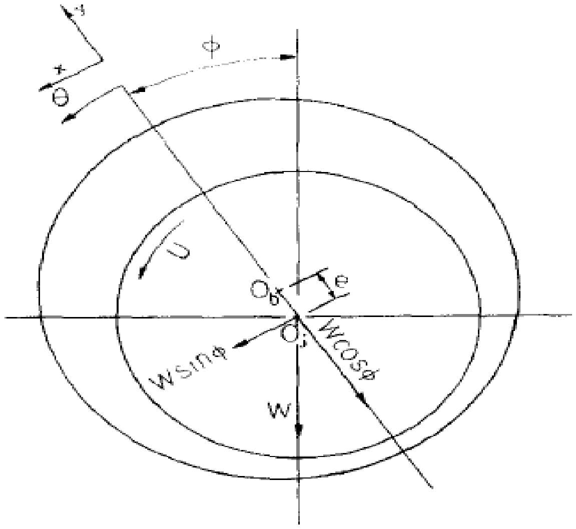


Fig 1:- Journal Bearing Configuration

Consequently, it is also equivalent to eqn. (7) of ref. 4. The Reynolds equation necessary for this problem is then written in the following form:

$$\frac{\partial}{\partial x}\left\{\frac{g(N,\ell,h)}{12\mu}\frac{\partial p}{\partial x}\right\}+\frac{\partial}{\partial z}\left\{\frac{g(N,\ell,h)}{12\mu}\frac{\partial p}{\partial z}\right\}=\frac{1}{2}\frac{\partial}{\partial x}(Uh)$$

(1)

Where

$$g(N,\ell,h)=h^3+12\ell^2h-6N\ell h^2\coth\left(\frac{Nh}{2\ell}\right) \qquad (2)$$

$$\mu=\mu_v+\frac{1}{2}\kappa_v$$

$$\ell=\left(\frac{\gamma}{4\mu}\right)^{1/2} \qquad (3)$$

$$N=\left(\frac{\kappa_v}{2\mu+\kappa_v}\right)^{1/2}$$

and in which μ_v is the viscosity coefficient for a newtonian fluid, K_v, is the spin viscosity, γ is the material coefficient, p is the hydrodynamic pressure and the film thickness h is given by

$$h=C(1+\epsilon\cos\theta) \qquad (4)$$

where ϵ is the eccentricity ratio.

Introducing the following dimensionless quantities:

$$\bar{h}=\frac{h}{c} \quad \bar{g}=\frac{g}{12h^3} \quad L=\frac{C}{\ell} \quad \theta=\frac{x}{R} \quad \xi=\frac{z}{B/2} \quad \bar{p}=\frac{pC^2}{\mu UR} \qquad (5)$$

then the dimensionless form of the Reynolds equation is

$$\frac{\partial}{\partial\theta}\left\{\bar{h}^3\bar{g}\frac{\partial\bar{p}}{\partial\theta}\right\}+\left(\frac{D}{B}\right)^2\frac{\partial}{\partial\xi}\left\{\bar{h}^3\bar{g}\frac{\partial\bar{p}}{\partial\xi}\right\}=\frac{-1}{2}\epsilon\sin\theta \qquad (6)$$

Where

$$\bar{g}(\bar{N},L,\bar{h})=\frac{1}{12}+\left(\frac{1}{L\bar{h}}\right)^2-\frac{N}{2L\bar{h}}\coth\left(\frac{NL\bar{h}}{2}\right) \qquad (7)$$

The pressure boundary conditions in dimensionless form are :

$$\bar{p}(0,\xi)=\bar{p}(2\pi,\xi)=0$$
$$\bar{p}(\theta,1)=\bar{p}(\theta,-1)=0$$
$$\bar{p}(\theta,\xi)=\bar{p}(\theta,-\xi)$$
$$p(\theta,\xi)\geqslant 0$$

(8)

The last condition yields the Reynolds boundary condition $\bar{p}=\frac{\partial\bar{p}}{\partial\theta}=0$ at the trailing edge of the film, when coupled with the correct numerical method.

Pressure Distribution:

we have

$$\frac{\partial P}{\partial\theta}=\frac{1}{2.H^2.f(N,L,H)}-\frac{\bar{Q}}{H^3.f(N,L,H)}$$

Where

$$f(N,L,H)=\left[\frac{1}{12}+\left(\frac{1}{LH}\right)^2-\frac{N}{2LH}\frac{(1+\cosh NLH)}{\sinh NLH}\right].$$

Integrating equation $\partial P/\partial\theta$ using half Summer fold boundary condition

P=0 at $\theta=0$

P=0 at $\theta=\pi$

The non-dimensional pressure distribution is obtained as

$$P(\theta)=\frac{1}{2}F_1(\theta)-\bar{Q}F_2(\theta)$$

Where

$$\bar{Q}=\frac{1}{2}\frac{F_1(\pi)}{F_2(\pi)}$$

$$F_1(\theta)=\int_0^\theta\frac{d\theta}{H^2.f(N,L,H)}$$

$$F_2(\theta)=\int_0^\theta\frac{d\theta}{H^3.f(N,L,H)}.$$

In accordance with the half Somerfield boundary conditions, the pressure is assumed to be zero for $\theta >= \pi$.

Load Capacity:

The load components per unit length along and perpendicular to the line of centers are obtained in the usual way by integrating pressure over the bearing surface. Normal to the line of centers, the non-dimensional load component is

$$\bar{W}_{\pi/2} = \bar{W} \sin\phi = \int_0^{\pi} P \text{ din } \theta \, d\theta$$

Substituting for P and integrating by parts

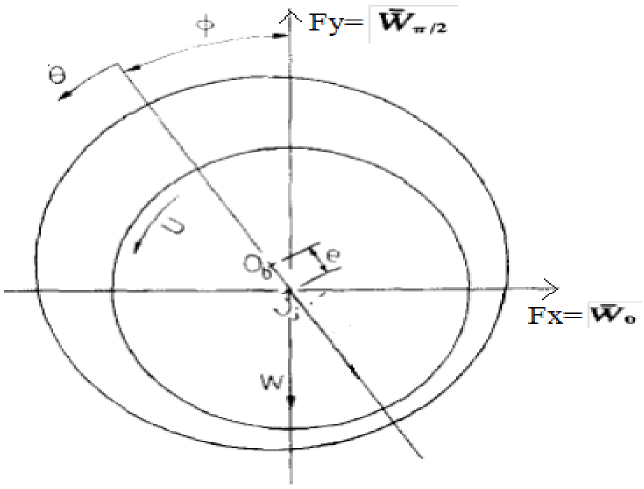


Fig 2: Load Vectors in Journal Bearing

$$\bar{W}_{\pi/2} = \frac{1}{2}\int_{\theta}^{\pi} \frac{(1+\cos\theta)\,d\theta}{H^2 . f(N,L,H)} - \bar{Q}\int_0^{\pi} \frac{(1+\cos\theta)d\theta}{H^3 . f(N,L,H)}$$

Similarly the non-dimensional load component along the line of centers $\bar{W}_0$ is

$$\bar{W}_0 = \bar{W}\cos\phi = \int_0^{\pi} P \cos\theta d\theta$$

Substituting for P and integrating by parts

$$\bar{W}_0 = \frac{1}{2}\int_0^{\pi} \frac{\sin\theta d\theta}{H^2 . f(N,L,H)} - \bar{Q}\int_0^{\pi} \frac{\sin\theta d\theta}{H^3 . f(N,L,H)}$$

Where

$$\bar{W} = \frac{WC^2}{\mu U R^2}.$$

Resultant non-dimensional load is given by

$$\bar{W} = (\bar{W}_0^{\,2} + \bar{W}_{\pi/2}^{2})^{1/2}$$

Stiffness: As from the basic concept the stiffness of the journal will be equal to the load per unit displacement or deflection. Whenever there is sudden disturbance given to the journal in the X-direction there is significant change in eccentricity and respective resultant pressure angle. The pressure angle at initial position of journal bearing at ϕ and after displacement the change in pressure angle is of amount ψ and similarly there is initial eccentricity ϵ and after having displaced in x-direction the eccentricity also changed to ϵ', So in resultant there will be a change in the load/force component appeared in respective direction.

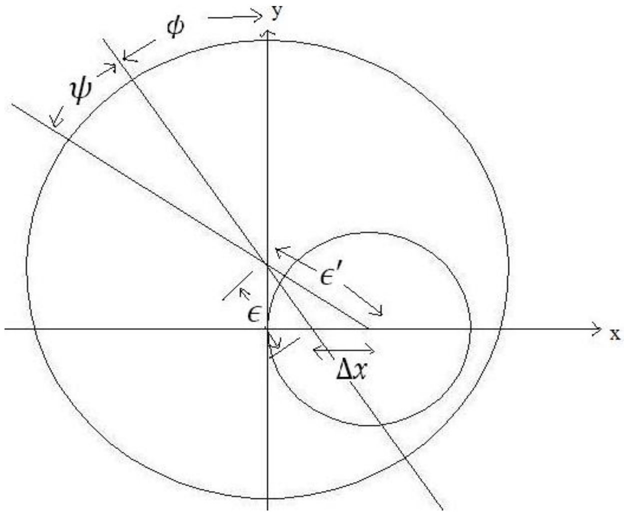


Fig. 3: Position of Journal Subjected To Displacement in X-Direction

Let the journal is displaced from its original position to a position Δx in X-direction, so due to that the change in force component in X-direction will be of amount F_x`.

so the stiffness in X-direction due to force component in X-direction is:

$$K_{xx} = (Fx - F_x`)/\Delta x .$$

Similarly when the journal is displaced or deflected with an amount Δy in Y-direction from its original position, there is significant change in the eccentricity and also the F_y has been changed to F_y`. So the stiffness of the journal for having displacement in Y-direction and also due to Force component in Y-direction will be:

$$K_{yy} = (F_y - F_y`)/\Delta y .$$

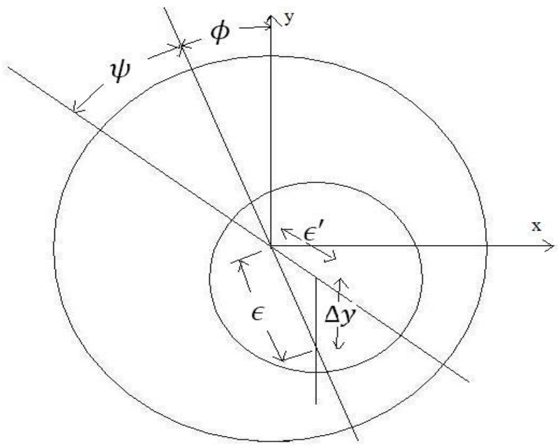


Fig 4: Position of Journal Subjected to Displacement in Y-Direction

Similarly when the position of journal has been changed in X-direction there will be change in load vector in Y-direction also. So when Δx amount of displacement has been given to journal and effect of change of load vector in Y-direction has been observed which provide the new stiffness K_{yx} i.e. the stiffness in X-direction due to load component in Y-direction will be:

$$K_{yx} = (F_y - F_y`)/\Delta x.$$

And when the position of journal is displaced in Y-direction of an amount Δy there will be change in load vector in X-direction that provide the new stiffness value K_{xy} i.e. the stiffness in Y-direction due to load component in X-direction will be :

$$Kxy = (Fx - Fx`)/\Delta y.$$

Damping Co-Efficient:

Damping Coefficients in journal bearing are concluded by same process as in case of stiffness. As the journal has been provided velocity in X-direction and the change in load component are taken into account which prove the value of damping coefficient in X-direction. Let the journal get velocity $\Delta x'$ in X-direction and the force/load component changes from F_x to F_x' so in resultant the damping coefficient in X-direction due to load component in X-direction is:

$$Dxx = (Fx - Fx')/\Delta x'.$$

Similarly when velocity has been given in X-direction and load component in Y-direction are observed then the damping coefficient obtained will be:

$$Dyx = (Fy – Fy')/\Delta x'.$$

When journal has provided $\Delta y'$ amount of velocity in Y-direction and the stiffness in Y-direction due to load components in Y-direction obtained as:

$$Dyy = (Fy – Fy')/\Delta y'.$$

Similarly when velocity has been provided in Y-direction and effect of load component in X-direction observed that will give the damping coefficient in Y-direction due to load component in x-direction i.e.:

$$Dxy = (Fx - Fx')/\Delta y'.$$

Critical Mass Parameter

The critical mass parameter of journal bearing can be calculated as:

M_{crit}=min (M_I or M_{II}) when kxx+kyy<0

=min (M_I) when kxx+kyy>0

$$M_I=(\bar{D}_{XX}\bar{D}_{YY}-\bar{D}_{XY}\bar{D}_{YX})\Bigg/\left[\frac{(\bar{D}_{XX}+\bar{D}_{YY})(\bar{K}_{XX}\bar{K}_{YY}-\bar{K}_{XY}\bar{K}_{YX})}{\bar{K}_{XX}\bar{D}_{YY}+\bar{K}_{YY}\bar{D}_{XX}-\bar{K}_{YX}\bar{D}_{XY}-\bar{K}_{XY}\bar{D}_{YX}}\right.$$

$$\left.-\frac{\bar{K}_{YX}\bar{D}_{XY}+\bar{K}_{YY}\bar{D}_{YY}+\bar{K}_{XX}\bar{D}_{XX}+\bar{K}_{XY}\bar{D}_{YX}}{\bar{D}_{XX}+\bar{D}_{YY}}\right]$$

And

$$M_{II}=\frac{(\bar{D}_{XX}\bar{D}_{YY}-\bar{D}_{XY}\bar{D}_{YX})}{\bar{K}_{XX}+\bar{K}_{YY}}$$

III. RESULT AND DISCUSSION

Two non-dimensional parameters **N** and ***l*** characterize the fluid micro-polarity and distinguish a micropolar fluid from a Newtonian fluid. The coupling number **N**, which characterizes the coupling between the Newtonian and micro-rotational viscosities, is defined in eqn. (3) and the parameter ***l*** is defined in eqn. (5). Since the parameter ***l*** has the dimensions of length, and may be considered as a fluid property depending on the size of the microstructure, the parameter ***l*** characterizes the interaction between the geometry and micropolar fluids.

In the following, we shall analyse the characteristics of micropolar fluids with specific values of N and ***l*** and make a comparison with Newtonian fluids. Following results shows the pressure distributions at z = 0 for ϵ = 0.5, B/D = π/10 and B/D = 1.0 in the cases of Newtonian and micropolar fluids.

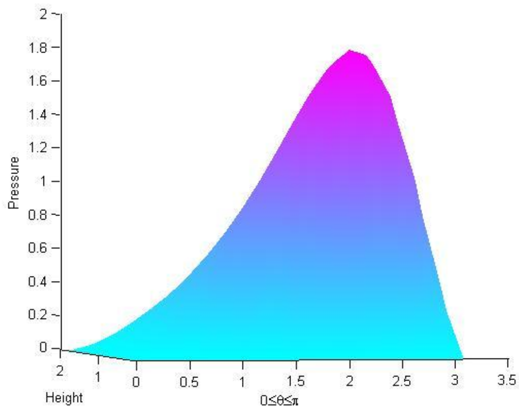
Results -

1 Pressure Distributions

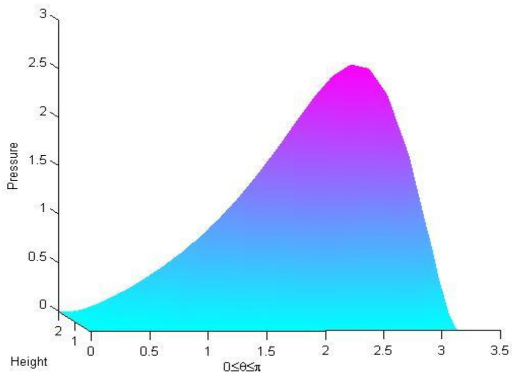
The pressure distributions for Newtonian fluids and micropolar fluids are evaluated for different parameters i.e. based upon B/D ratio, eccentricity, N2, l etc.

It has seen that the pressure distribution along the circumference of the journal in case of micropolar fluid have large value as compared to Newtonian's fluid.

When ϵ=0.5 ,B/D =1, N^2 =.57 ,l =3.5 :-



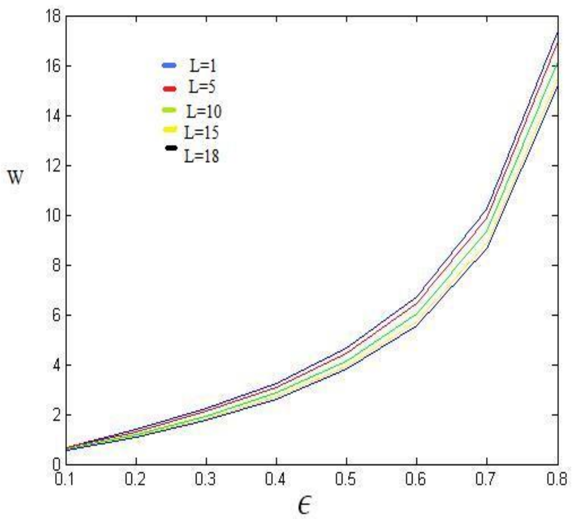
Newtonian's Fluid



Micropolar Fluid

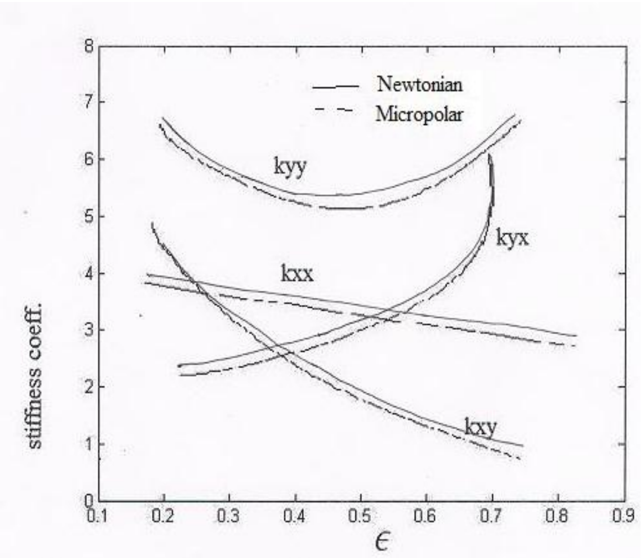
2 Load intensity in journal bearing with micropolar fluid based upon 'L'

When B/D = 1, N =0.577 :-



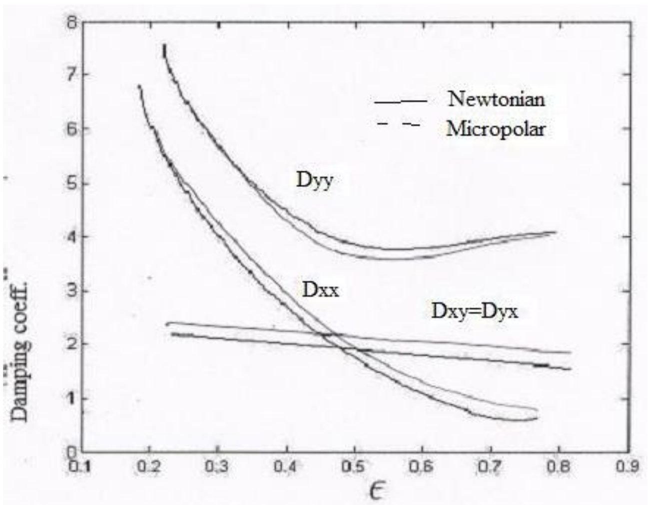
3. Stiffness coefficient in journal bearing with micropolar fluid and newtonian fluid:

When B/D =1, N^2 =1/3 ,l =3.5 :-



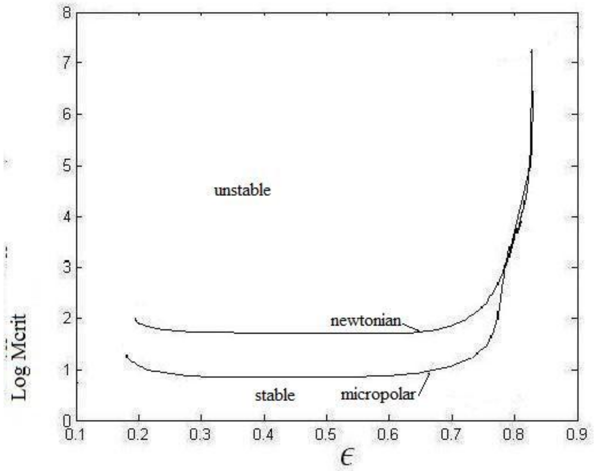
4 Damping coefficients in journal bearing with micropolar fluid and Newtonian fluid:

When B/D =1, N^2 =1/3 ,l =3.5 :-



5 Critical mass Vs eccentricity in micropolar and Newtonian fluid

When ϵ=0.5 ,B/D =1, N =0.577,l =3.5



IV. CONCLUSION

Hydrodynamic analysis of the Journal bearing has been done using CAE tool MATLAB. From the results obtained from hydrodynamic analysis, some discussions have been made. On the basis of the current work and comparing the result of Newtonian and Micropolar lubricant, it is concluded that the numerical study of plain journal bearing gives out some following results.

- The modified equation derived on the basis of the micropolar theory in this paper is more general than the classical Reynolds equation for dynamic loads.
- With the same dynamic loads applied, the micropolar fluids yield a higher oil film pressure and oil film thickness then Newtonian fluid.
- The feature of increasing load capacity for micropolar fluid is more evident at higher eccentricity ratio and lower width-to-diameter ratio then Newtonian fluid.

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23. Partial differentiation equation tool box –MATLAB

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