Journal of Mechanical Engineering *Vol SI 1 (1), 31-42, 2017*

Compensated Hole-Entry Hybrid Journal Bearing by CFV Restrictor under Micropolar Lubricants

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ABSTRACT

In this work, the performance of hole-entry hybrid journal bearing compensated by constant flow valve (CFV) restrictor under micropolar lubricants has been numerically simulated. Reynolds equation for micropolar fluid lubricated bearing has been solved with finite element technique. Performance of bearing has been evaluated as a function of coupling number N^2. *The simulated characteristics of bearing under micropolar lubricants have been compared with similar bearing under Newtonian lubricant. The results are presented for the selected micropolar parameters* N^2 *and* l_m. *Simulated results indicate that the bearing under micropolar lubricants exhibits the increased values of minimum fluid film thickness, stiffness and damping coefficients than similar bearing under Newtonian lubricant. Further, the coefficient of friction decreases for a bearing when it is operating under micropolar lubricant than Newtonian lubricant.*

Keywords: *Journal Bearings, Micropolar Fluids, FEM, Restrictors*

ISSN 1823-5514, eISSN 2550-164X

Received for review: 2016-09-01
Accepted for publication: 2016-11-17
Published: 2017-01-19

Introduction

The estimation of flow behavior of lubricant plays an important role to predict the behavior of bearings accurately. The idea of micropolar fluid theory was given by Eringen [1] which accounts for the internal structures of fluids. This theory of fluids exhibits microrotational effect and microrotational inertia [2]. Migun [3] introduced a technique for determining the parameters, characterizing the microstructure of lubricant experimentally. Later on, Kolpaschikov et al. [4] determined the material micro-polar fluid constants experimentally. Several researchers studied the operation of journal bearings under micropolar lubrication [5-11]. Tipei [5] presented the characteristic parameters of short journal bearing under micropolar lubrication. Singh and Sinha [6] studied the rotor dynamic coefficients of a short journal bearing under sinusoidal load. Das et al. [7] evaluated stability of journal bearings under micropolar fluids using finite difference technique. They compared the results of linear and non-linear analyses of bearing under micropolar lubricant. Later on, Wang and Zhu [8] presented the influence of micropolar lubricant on hydrodynamic journal bearing. Elord's cavitation algorithm was used to solve the Reynolds equation. They reported the increased pressure distribution of bearing under micropolar lubrication. Krasowski [9, 10] reported the hydrodynamic pressure distribution and load carrying capacity for slider journal bearing with micropolar lubricant. Later on, Nathi and Sharma [11] investigated the effect of micropolar lubricant on hole-entry hybrid bearing considering orifice compensation. Many studies about the performance of hole-entry journal bearings have been reported [12-18]. EI Kayaret al. [12] analyzed the hole-entry orifice compensated journal bearings using finite difference technique. They reported the static and dynamic performance of bearing against eccentricity ratio and speed parameter. The performance of hole-entry journal bearing was reported by Rowe et al [13]. Later on, Rowe [14] reported the comprehensive review for the development of hydrostatic and hybrid journal bearing technology. Cheng and Rowe [15] proposed a selection model which is concerned with the selection of bearing type and configuration, the fluid feeding device and bearing material. Nathi and Sharma [16] worked on micropolar lubrication's influence on asymmetric slot-entry hybrid bearing. Recently, Nathi [17, 18] analyzed symmetric/asymmetric hole-entry hybrid bearing compensated by capillary restrictor under micropolar lubrication and symmetric hybrid bearing under turbulent regime.

The compensated hole-entry hybrid bearing by CFV under micropolar fluid as shown in Figure 1 has been studied analytically. Reynolds equation, modified for micropolar fluid lubricated bearing has been solved by the finite element method. Considering the various values of micropolar parameters, the performance hole-entry bearing have been presented.

Analysis

The Reynolds equation, modified for a bearing hole-entry journal bearing [Figure 1] under micropolar lubrication is expressed as [8, 11]

$$\frac{\partial}{\partial\alpha}\{\frac{\bar{h}^3}{12\bar{\mu}}\bar{\phi}\frac{\partial\bar{p}}{\partial\alpha}\}+\frac{\partial}{\partial\beta}\{\frac{\bar{h}^3}{12\bar{\mu}}\bar{\phi}\frac{\partial\bar{p}}{\partial\beta}\}=\frac{\Omega\partial\bar{h}}{2\partial\alpha}+\frac{\partial\bar{h}}{\partial\bar{t}} \qquad (1)$$

Where,

$$\bar{\phi}=1+\frac{12}{\bar{h}^2 l_m^2}-\frac{6N}{\bar{h}l_m}\coth(\frac{N\bar{h}l_m}{2}),\ N=(\frac{k}{2\mu+k})^{1/2},\ \ l=(\frac{\gamma}{4\mu})^{1/2}$$

The coupling number N and characteristics length of lubricant l are the parameters which characterize the micropolar lubricant and make it different from the Newtonian lubricant. When l_m approaches to infinity and N^2 tends to zero, the lubricant behaves like a Newtonian lubricant.

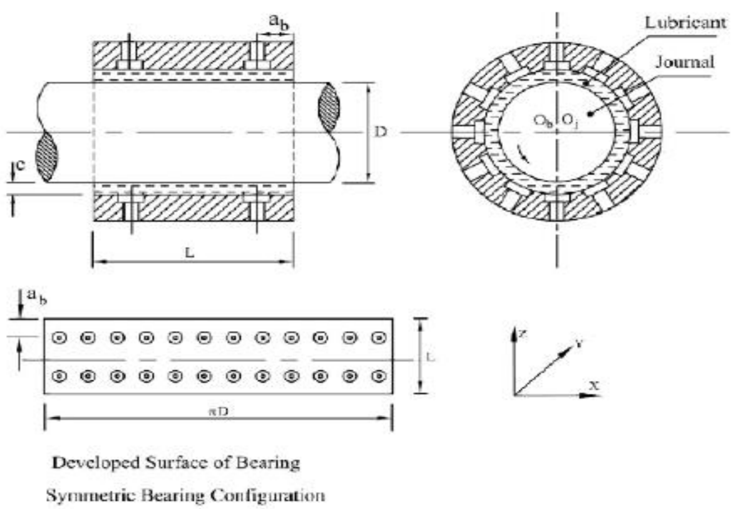

Figure 1: Hole-entry journal bearing

Fluid Film Thickness

The fluid film thickness $\bar{h}$ for hole-entry bearing in non-dimensional form is given as [11]

$$\bar{h} = 1 - \bar{X}_j \cos\alpha - \bar{Z}_j \sin\alpha \tag{2}$$

Restrictor Flow Equation

The flow of lubricant through a constant flow valve restrictor is expressed in non-dimensional form as

$$\bar{Q}_R = \bar{Q}_c \tag{3}$$

Where, $\bar{Q}_c$ is specified flow rate for CFV restrictor.

Finite Element Formulation

The isoparametric four noded quadrilateral elements are used for discretized flow domain. After using the Galerkin's technique for solving the Reynolds equation (1), the equation is given [11] in the matrix form as:

$$[\bar{F}]^e\{\bar{p}\} = \{\bar{Q}\}^e + \Omega\{\bar{R}_H\}^e + \dot{\bar{X}}_j\{\bar{R}_{xj}\}^e + \dot{\bar{Z}}_j\{\bar{R}_{zj}\}^e \tag{4}$$

Where,

$$\bar{F}_{ij}^e = \int_{A^e}\int [\frac{\bar{h}^3}{12\bar{\mu}}\bar{\phi}\frac{\partial N_i}{\partial\alpha}\frac{\partial N_j}{\partial\alpha} + \frac{\bar{h}^3}{12\bar{\mu}}\bar{\phi}\frac{\partial N_i}{\partial\beta}\frac{\partial N_j}{\partial\beta}]d\alpha d\beta \tag{4a}$$

$$\bar{Q}_j^e = \int_{\Gamma^e}\{[(\frac{\bar{h}^3}{12\bar{\mu}}\bar{\phi}\frac{\partial\bar{p}}{\partial\alpha}) - \frac{\Omega}{2}\bar{h}] + (\frac{\bar{h}^3}{12\bar{\mu}}\bar{\phi}\frac{\partial\bar{p}}{\partial\beta})m\}N_i d\Gamma \tag{4b}$$

$$\bar{R}_{Hi}^e = \int_{A^e}\int \frac{\bar{h}}{2}\frac{\partial N_i}{\partial\alpha}d\alpha d\beta \tag{4c}$$

$$\bar{R}_{xji}^e = \int_{A^e}\int N_i \cos\alpha \times d\alpha d\beta \tag{4d}$$

$$\bar{R}_{zji}^e = \int_{A^e}\int N_i \sin\alpha \times d\alpha d\beta \tag{4e}$$

Where, l and m are direction cosines.

Boundary Conditions

The boundary conditions used for solving Reynolds equation (1) for micropolar fluid lubricated hole-entry bearing are:

1. The nodes situated on external boundary of bearing have zero relative pressure with respect to atmospheric pressure i.e. $\bar{p}\,|_{\beta=\pm1.0} = 0.0$.
2. Nodes situated on holes have equal pressure.

Solution Procedure

The results of constant flow valve compensated bearing have been obtained after solving the Reynolds equation using numerical procedure [11]. After feeding the values of $\bar{X}_j$ and $\bar{Z}_j$ for external load $\bar{W}_o$ initially, the fluid film thickness is determined from eqn. (2). Then, the system eqn. (4) and constant flow valve restrictor eqn. (3) has been solved after applying boundary conditions to calculate the pressure field. When the program gets converged, the performance of hole-entry symmetric bearings has been computed using the expressions as reported in [11].

Result and Discussion

A computer program has been developed according analytical model. For checking the genuineness of developed model, the computed results have been verified with given results in Ref. [8] as shown in Figure 2. The obtained results are found very close to the given in Ref. [8]. The characteristics of hole-entry bearing are presented for the range of micropolar parameters $N^2 = 0.1 - 1.0$, $l_m = 5 - 20$ and Newtonian lubricant through Figs.3 to 10 for bearing parameters such as Land width ratio $(\bar{a}_b) = 0.25$; Bearing aspect ratio $(\lambda) = 1.0$; No. of rows of holes = 2; No. of holes per row = 12; Speed parameter $(\Omega) = 1$; External load $(\bar{W}_o) = 1.25$;concentric design pressure ratio $(\beta^*) = 0.5$

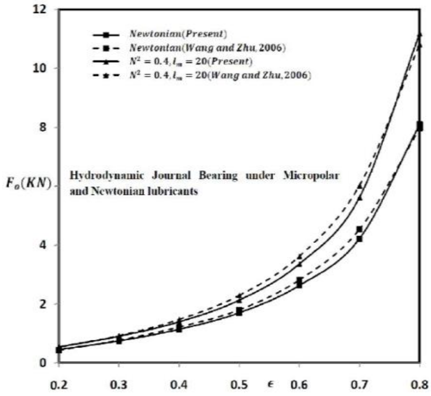

Figure 2: Variation of F_o with ε

The variation in maximum pressure is shown in Figure 3. It may be observed that the maximum fluid film pressure $\bar{p}_{\max}$ increases significantly when the bearing is operating with increased value of coupling number N^2 at fixed characteristic length of micropolar lubricant l_m. The enhancement in the maximum fluid film pressure $\bar{p}_{\max}$ is 46.58% at l_m = 5 for N^2 = 0.9 in comparison to Newtonian fluid.

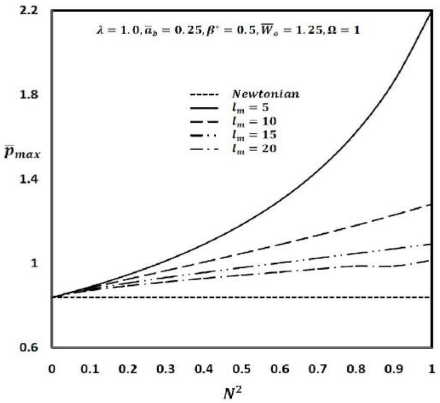

Figure 3: Variation of $\bar{p}_{\max}$ with N^2

Figure 4 shows the variation of minimum fluid film thickness $\bar{h}_{\min}$ with coupling number N^2. It may be noticed that the value of $\bar{h}_{\min}$ increases with decrease in characteristic length of micropolar lubricant l_m at constant coupling number N^2 for a bearing operating under micropolar lubricants than Newtonian lubricant. However, an enhancement in the film thickness $\bar{h}_{\min}$ at N^2 = 0.9 and l_m = 5 is found of the order of 10.26% than the bearing under Newtonian lubricant.

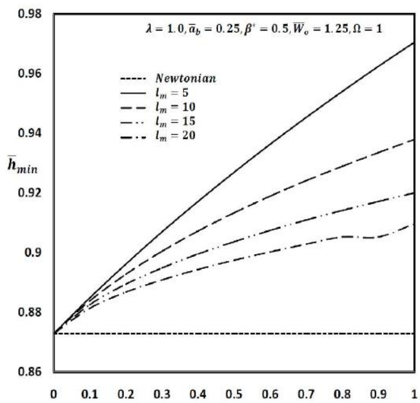

Figure 4: Variation of $\bar{h}_{\min}$ with N^2

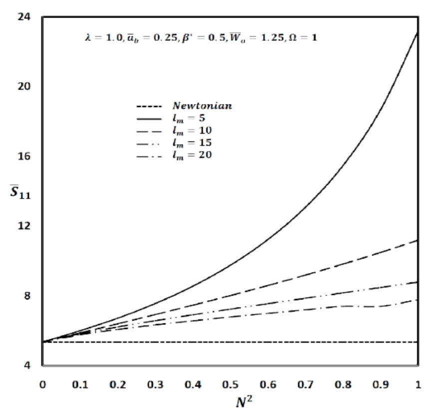

Figure 5: Variation of $\bar{S}_{11}$ with N^2

The values of direct stiffness coefficients ($\bar{S}_{11}, \bar{S}_{22}$) are presented in Figure 5 and Figure 6. The values of stiffness coefficients are found to be increased when bearing operates with lower value of the characteristic length l_m for the coupling number N^2 than Newtonian lubricant. An enhancement of 96.04% in stiffness coefficient ($\bar{S}_{11}$) and 94.31% in stiffness coefficient ($\bar{S}_{22}$) respectively is found for micropolar parameters N^2 = 0.9 and l_m = 5 as compared to similar bearing under Newtonian lubricant.

Figure 7 and Figure 8 indicate the increasing trend for direct fluid film damping coefficients ($\bar{C}_{11}, \bar{C}_{22}$) as a function of coupling number for a bearing operates under either Newtonian or micropolar lubricant. Further, the values of damping coefficients increase with an increase in coupling number N^2 and the percentage of increase in the values of damping coefficient $\bar{C}_{22}$ is 76.89% and for damping coefficient $\bar{C}_{22}$ is found 79.22% respectively corresponding to micropolar parameters N^2 = 0.9 and l_m = 10 for a bearing under micropolar lubricant than the same bearing with Newtonian lubricant.

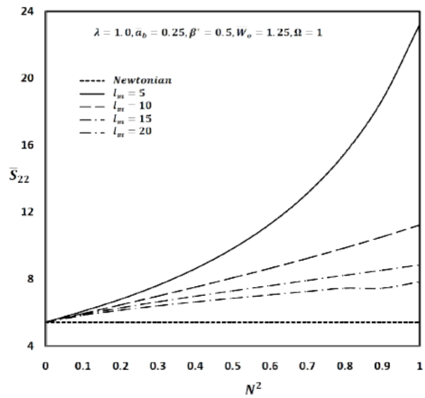

Figure 6: Variation of $\bar{S}_{22}$ with N^2

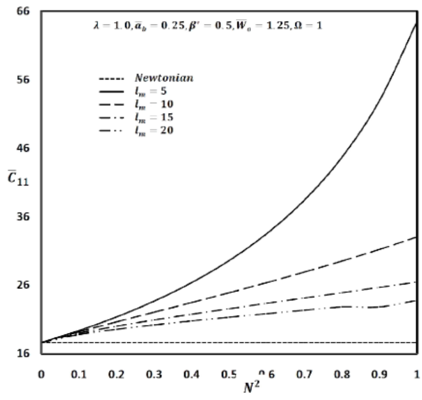

Figure 7: Variation of $\overline{C}_{11}$ with N^2

The influence of micropolar parameters N^2 and l_m on stability threshold speed margin $\overline{\omega}_{th}$ is presented in Figure 9. An increase of 85.21%, 39.11%, 25.31% and17.15% in stability threshold speed is found corresponding to the coupling number N^2 = 0.9 for the values of l_m at 5, 10, 15 and 20 respectively as compared to the bearing under Newtonian lubricant.

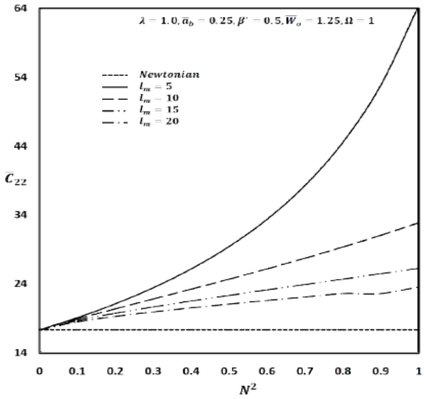

Figure 8: Variation of $\overline{C}_{22}$ with N^2

The influence of N^2 and l_m on friction coefficient ($\bar{c}_f$) is presented in Figure 10. It may be observed that the value of $\bar{c}_f$ reduces with increasing the value of coupling number N^2 when the bearing is operating under micropolar lubricant. However, the maximum reduction in the value of $\bar{c}_f$ is 37.88% at l_m = 5 and 19.55% at l_m = 10 respectively for N^2 = 0.9 for bearing operates under micropolar lubricant vis-a-vis Newtonian lubricant.

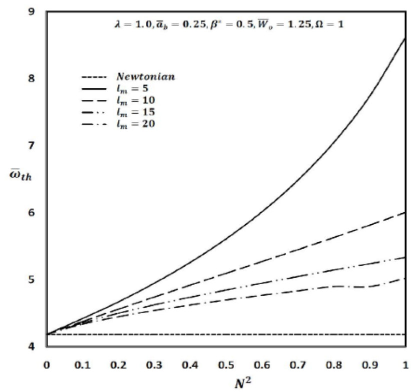

Figure 9: Variation of $\bar{\omega}_{th}$ with N^2

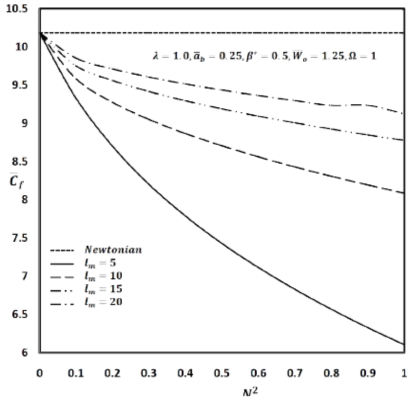

Figure 10: Variation of $\bar{c}_f$ with N^2

Conclusion

The conclusions have been drawn from numerically simulated results in this work, are:

1. A significant increase in maximum fluid film pressure $\bar{p}_{max}$ is observed for bearing under coupling number $N^2 = 0.9$ at $l_m = 5$ than similar bearing under Newtonian lubricant.
2. The minimum fluid film thickness $\bar{h}_{min}$ enhances for a bearing operating with increasing coupling number and decreasing characteristics length of micropolar lubricant than Newtonian lubricant.
3. The bearing stiffness and damping coefficients get improved significantly by using micropolar lubricant than Newtonian lubricant
4. An increase of 39.11% in stability threshold speed margin $\bar{\omega}_{th}$ is found at $N^2 = 0.9$ and $l_m = 10$ for bearing under micropolar lubricant than bearing under Newtonian lubricant.
5. The coefficient of friction reduces by 37.88% corresponding to micropolar parameters at $N^2 = 0.9$ and $l_m = 10$ than Newtonian lubricant.

Acknowledgement

The authors are grateful to Indira Gandhi Delhi Technical University for Women, Delhi, India for providing the financial assistance.

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