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Research Article

A Study on the Performance of a Magnetic-Fluid-Based Hydrodynamic Short Journal Bearing

N. S. Patel,[1] D. P. Vakharia,[2] and G. M. Deheri[3]

[1] *Department of Mechanical Engineering, Faculty of Technology, Dharmsinh Desai University, Nadiad 387001, Gujarat, India*
[2] *Department of Mechanical Engineering, Sardar Vallabhbhai National Institute of Technology, Surat 395007, Gujarat, India*
[3] *Department of Mathematics, Sardar Patel University, Vallabh Vidyanagar, Anand 388120, Gujarat, India*

Correspondence should be addressed to N. S. Patel, nimeshsp@yahoo.co.in

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Efforts have been made to study and analyze the performance of a hydrodynamic short journal bearing under the presence of a magnetic fluid lubricant. With the usual assumptions of hydrodynamic lubrication, the associated Reynolds equation for the fluid pressure is solved with appropriate boundary conditions. In turn, this is then used to calculate the load-carrying capacity which results in the calculation of friction. The computed results presented in graphical form suggest that the bearing system registers an improved performance owing to the magnetic fluid as compared to the conventional lubricant. It is clearly observed that the load-carrying capacity increases nominally while the coefficient of friction decreases significantly. Besides, it is seen that the bearing can support a load even when there is no flow of lubricant. In addition, this type of study may offer an additional degree of freedom from design point of view in terms of the forms of the magnitude of the magnetic fluid.

1. Introduction

Oliver [1] made a comparison between the lubricating performance of Newtonian and highly elastic liquid. It was seen that the elastic liquid induced load enhancement ratio and reduction in the coefficient of friction. Lin [2] dealt with the theoretical study of squeeze film behavior for a finite journal bearing lubricated with couple stress fluid. It was evaluated that the couple stress effects increased the load-carrying capacity significantly and lengthened the response time of the squeeze film.

Kuzma [3] presented an analysis of an infinitely long journal bearing for the case of an electrically conducting fluid in the presence of a magnetic field. It was found that the bearing performance got improved due to the magnetization as compared to the case of conventional lubricant-based bearing. Chang et al. [4] considered two types of four-pad step-pocket journal bearing, lubricated with a ferromagnetic fluid. It was observed that the ferrofluid lubrication yielded higher overall bearing performance. Besides, the side leakage of the ferrofluid at both ends was found to be avoided.

Nada et al. [5] derived the modified Reynolds equation based on the momentum and continuity equation for a ferrofluid under an applied magnetic field, in order to analyze the effect of using current carrying wire model in the design of a hydrodynamic journal bearing lubricated with a ferrofluid. The results concluded that the magnetic lubrication provided higher load-carrying capacity and reduced friction coefficient as compared to a conventional fluid-based bearing.

Naduvinamani et al. [6] made an investigation on the rheological effects of the couple stress fluids on the static and dynamic behavior of squeeze films in a short porous journal bearing. It was revealed that as compared to the Newtonian lubricants, the lubricant which sustains the couple stresses yielded an increase in the load-carrying capacity. Shah and Bhat [7] discussed the squeeze film behavior in an infinitely long journal bearing using the ferrofluid flow

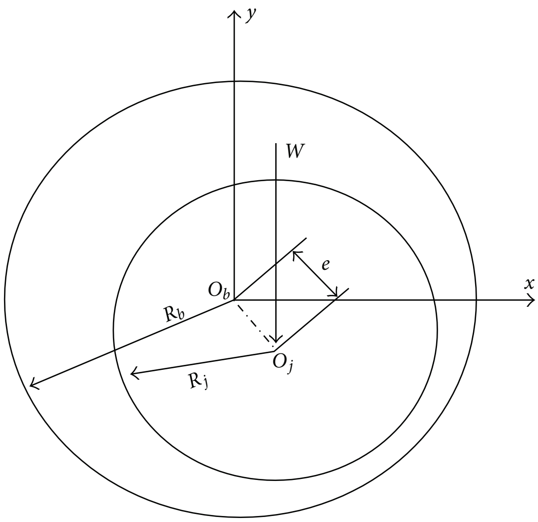

FIGURE 1: Configuration of the problem.

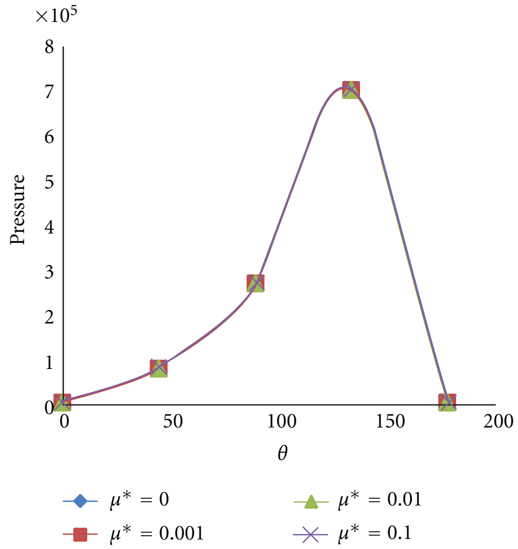

FIGURE 2: Nondimensional pressure distribution p versus θ for different values of magnetic parameter μ^*.

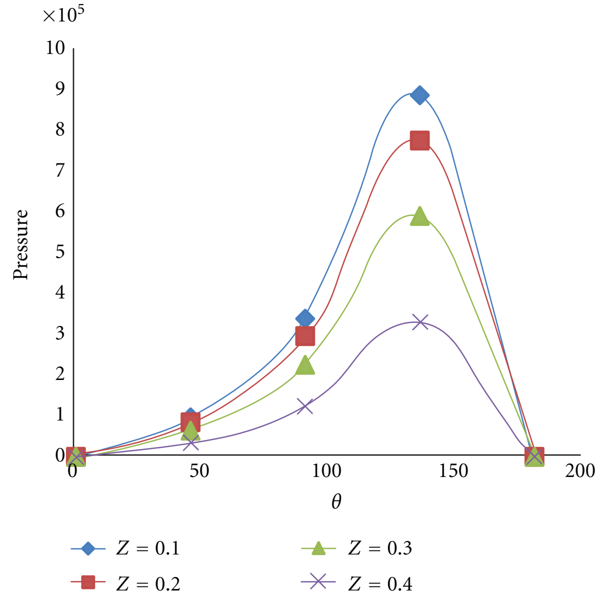

FIGURE 3: Nondimensional pressure distribution p versus θ for different values of Z.

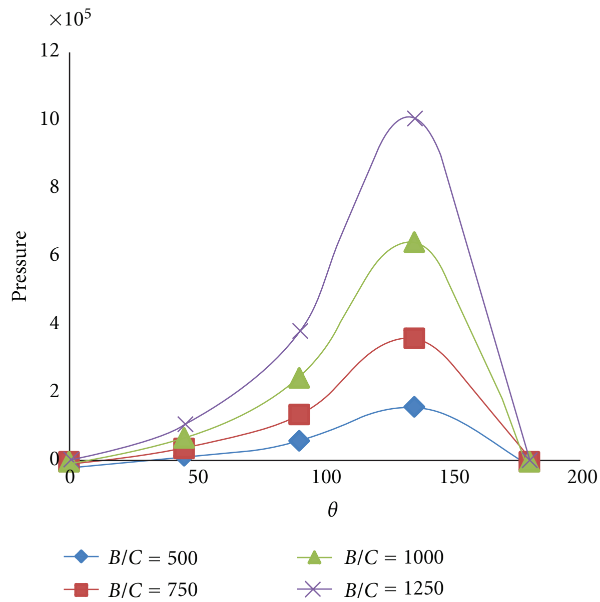

FIGURE 4: Nondimensional pressure distribution p versus θ for different values B/C.

models of Neuringer-Rosensweig and Jenkins and Shliomis models for uniform and nonuniform magnetic fields. It was established that a uniform magnetic field failed to produce magnetic pressure in the Neuringer-Rosensweig model, on the other hand, it could affect the bearing performance characteristics in the Shliomis model owing to the rotational viscosity parameter. Further, the load-carrying capacity and squeeze time were more in the case of nonuniform magnetic field than in the case of a uniform magnetic field. Nada and Osman [8] investigated the problem of lubrication of a finite hydrodynamic journal bearing lubricated with magnetic fluids considering the couple stress effects. The results indicated that the influence of couple stresses and magnetic effects on the bearing performance characteristics were significantly apparent. It was concluded that fluids with couple stresses were better than Newtonian fluids and the magnetic effect enhanced the performance characteristics. Urreta et al. [9] conducted a theoretical analysis with numerical solutions of the associated Reynolds equation for the pressure distribution in a hydrodynamic journal bearing, based on viscosity modulation for ferrofluid and Bingham model for MR fluid. It was demonstrated that the magnetic fluid could be used to manufacture active journal bearings. Gertzos et al. [10] carried out CFD analysis of hydrodynamic journal bearing lubricated by a Bingham lubricant.

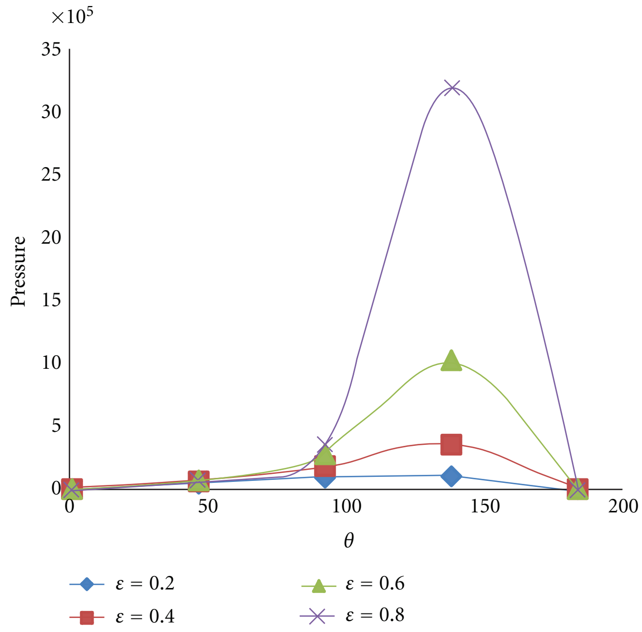

Figure 5: Nondimensional pressure distribution p versus θ for different values of ε.

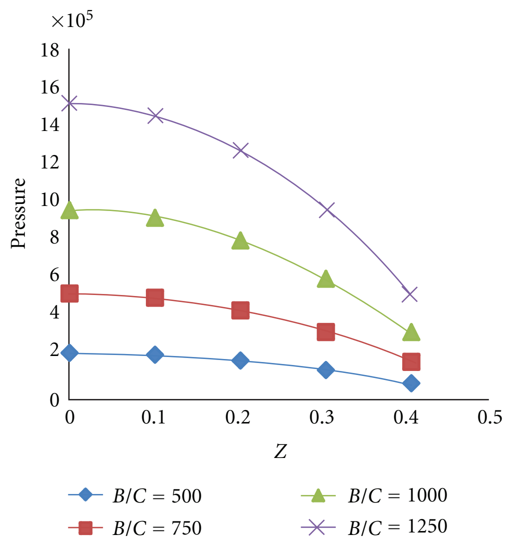

Figure 6: Nondimensional pressure distribution p versus Z for different values of B/C.

The charts presented here could be used by the designer to design smart journal bearings. Further, the results obtained from the developed 3D CFD model were found to be in very good agreement with experimental data from previous investigations on Bingham fluids.

Here, it has been sought to study and analyze the performance of a short journal bearing under the presence of a magnetic fluid lubricant based on Neuringer and Rosensweig model.

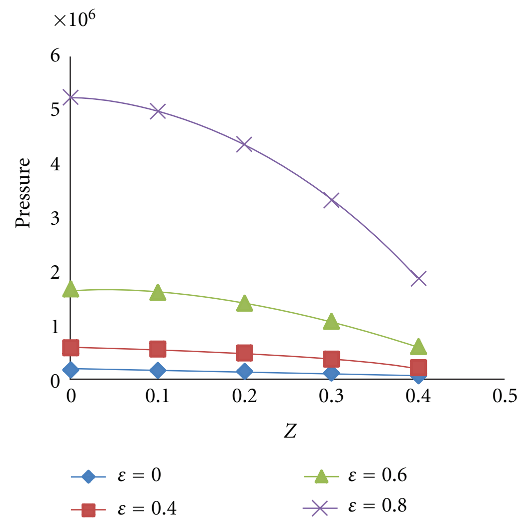

Figure 7: Nondimensional pressure distribution p versus Z for different values of ε.

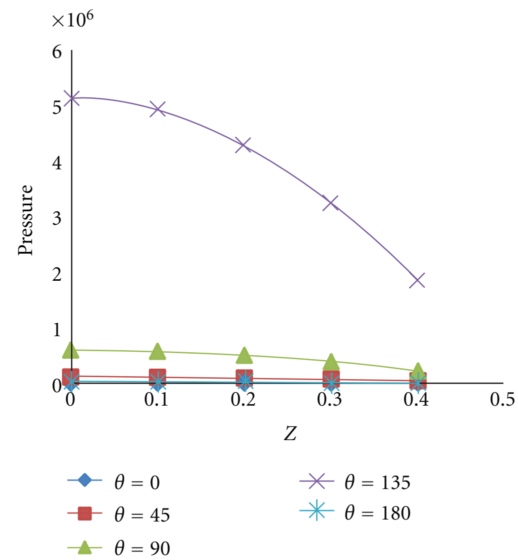

Figure 8: Nondimensional pressure distribution p versus Z for different values of θ.

2. Analysis

The configuration of the bearing which is infinitely short in Z-direction is presented in Figure 1. The journal has radius R_j rotating inside a bearing, and the space between journal and bearing is filled with a magnetic fluid. If the journal is infinitely short, the pressure gradient $\partial p/\partial z$ is much larger than the pressure gradient $\partial p/\partial x$, as a result of which the latter can be neglected. The magnetic field is oblique to the stator as in Agrawal [11] and its magnitude is given by

$$H^2 = k\left(z - \frac{B}{2}\right)\left(z + \frac{B}{2}\right), \tag{1}$$

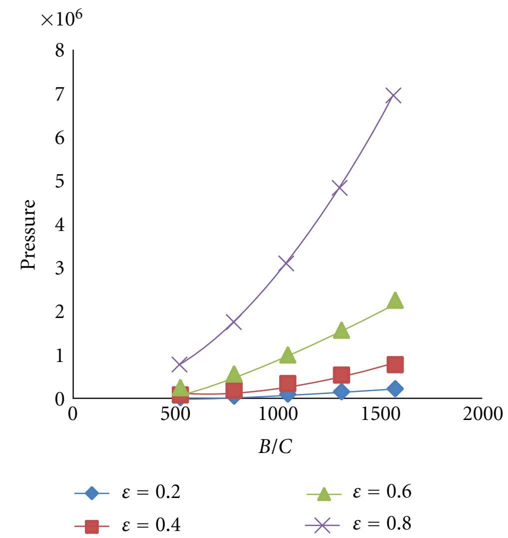

Figure 9: Nondimensional pressure distribution p versus B/C for different values of ε.

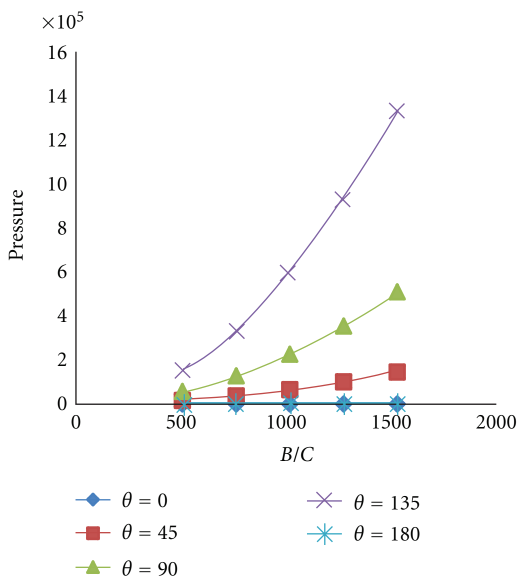

Figure 10: Nondimensional pressure distribution p versus B/C for different values of θ.

where k is a constant to suit the dimensions and the strength of the magnetic field [10].

In 1964, Neuringer and Rosensweig developed a simple model to study the steady flow of magnetic fluids in the presence of slowly changing external magnetic fields. The model consisted of the following equations:

$$\rho(\overline{q}.\nabla)\overline{q} = -\nabla p + \eta\nabla^2\overline{q} + \mu_o\left(\overline{M}.\nabla\right)\overline{H}, \tag{2}$$

$$\nabla.\overline{q} = 0, \tag{3}$$

$$\nabla \times \overline{H} = 0, \tag{4}$$

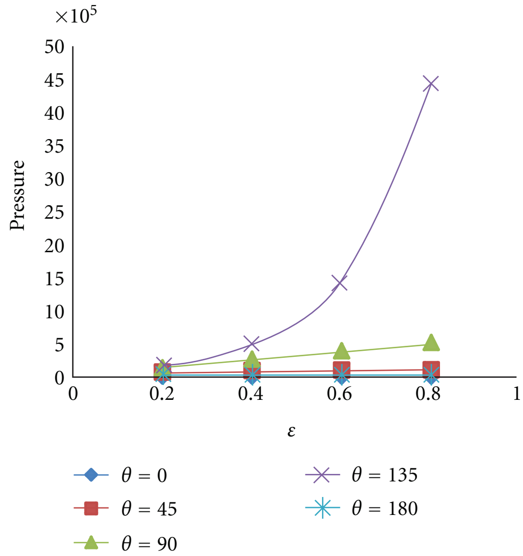

Figure 11: Nondimensional pressure distribution p versus θ for different value of ε.

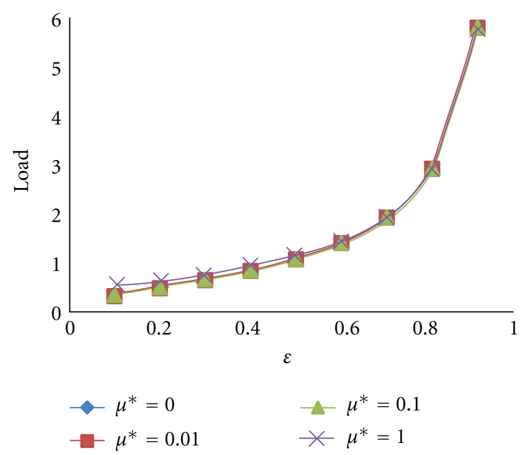

Figure 12: Nondimensional load-carrying capacity versus ε for different value of magnetic parameter μ^*.

$$\overline{M} = \overline{\mu}\overline{H}, \tag{5}$$

$$\nabla.\left(\overline{H} + \overline{M}\right) = 0, \tag{6}$$

where ρ is the fluid density, $\overline{q} = (u, v, w)$ is the fluid velocity in film region, p is the film pressure, η is the fluid viscosity, μ_o is the permeability of free space, $\overline{M}$ is the magnetization vector, $\overline{H}$ is the external magnetic field, and $\overline{\mu}$ is the magnetic susceptibility of the magnetic particles.

Using Equations (4) and (5), Equation (2) becomes

$$\rho(\overline{q}.\nabla)\overline{q} = -\nabla\left(p - \frac{\mu_0\overline{\mu}H^2}{2}\right) + \eta\nabla^2\overline{q}. \tag{7}$$

This shows that an extra pressure $\mu_0\overline{\mu}H^2/2$ is introduced into the Navier-Stokes equation when magnetic fluid is

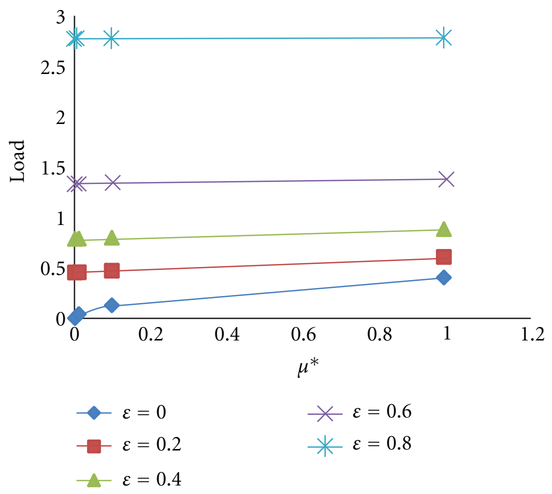

FIGURE 13: Nondimensional load-carrying capacity versus μ^* for different value of ε.

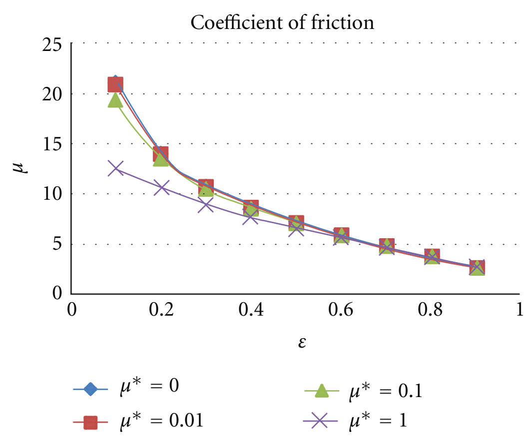

FIGURE 14: Coefficient of friction versus ε for different value of magnetic parameter μ^*.

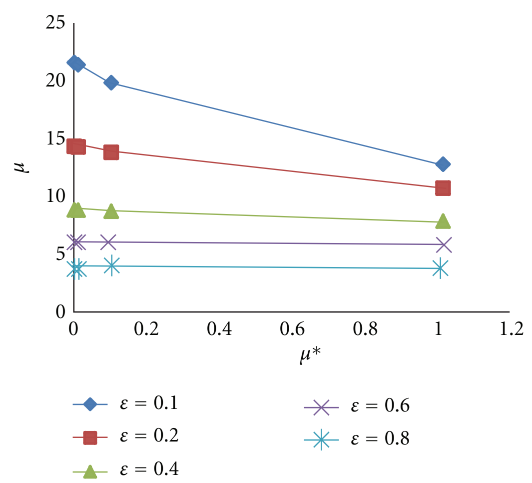

FIGURE 15: Coefficient of friction versus μ^* for different value of magnetic parameter ε.

used as a lubricant. Thus, the modified Reynolds equation for magnetohydrodynamic short journal bearing under the usual assumptions, the lubricant film is taken to be isoviscous, incompressible, and the flow is laminar (Bhat and Deheri [12], Agrawal [11], Bhat [13]), turns out to be

$$\frac{d^2}{dz^2}\left(p - \frac{\mu_o\,\bar{\mu}\,H^2}{2}\right) = \frac{6\eta u}{h^3 R} \cdot \frac{dh}{d\theta}. \tag{8}$$

The associated boundary conditions are

$$p = 0 \quad \text{at } z = +\frac{B}{2} \text{ and } -\frac{B}{2}, \tag{9}$$

$$\frac{dp}{dz} = 0 \quad \text{at } z = 0. \tag{10}$$

Introduction of the dimensionless quantities,

$$Z = \frac{z}{B}, \qquad P = \frac{R}{\eta u}\,p, \qquad \mu^* = -\frac{kB^2 R\mu_o\bar{\mu}}{\eta u}, \tag{11}$$

leads to the expression for the pressure distribution in dimensionless form:

$$P = \left[\frac{\mu^*}{2} + 3\left(\frac{B}{C}\right)^2\left(\frac{\varepsilon\sin\theta}{(1+\varepsilon\cos\theta)^3}\right)\right]\left[\frac{1}{4} - Z^2\right]. \tag{12}$$

The load-carrying capacity in x direction is given by

$$w_X = -2\int_0^{\pi}\int_0^{B/2} p\ \cos\theta\ R\,d\theta\,dz. \tag{13}$$

Thus, the dimensionless load-carrying capacity in x direction is obtained from

$$\begin{aligned} W_x &= \frac{c^2}{\eta u B^3}\,w_x \\ &= \frac{\varepsilon^2}{(1-\varepsilon^2)^2}. \end{aligned} \tag{14}$$

The load-carrying capacity in z direction is given by

$$w_Z = 2\int_0^{\pi}\int_0^{B/2} p\ \sin\theta\ R\,d\theta\,dz. \tag{15}$$

Then, the nondimensionless load-carrying capacity in z direction is obtained from:

$$\begin{aligned} W_z &= \frac{c^2}{\eta u B^3}\,w_z \\ &= \frac{\mu^*}{6} + \frac{1}{4}\frac{\pi\varepsilon}{(1-\varepsilon^2)^{3/2}}. \end{aligned} \tag{16}$$

Therefore, the resultant load-carrying capacity is given by

$$\begin{aligned} W &= \sqrt{W_x^2 + W_z^2}, \\ W &= \left[\frac{\varepsilon^2}{(1-\varepsilon^2)^2} + \frac{\mu^*}{6} + \frac{1}{4}\frac{\pi\varepsilon}{(1-\varepsilon^2)^{3/2}}\right]^{1/2}. \end{aligned} \tag{17}$$

The friction force is determined by

$$f = \int_0^{2\pi} \eta \frac{u}{h} \, LR \, d\theta, \tag{18}$$

which gives the nondimensional friction force as

$$\begin{aligned} F &= f \frac{C}{\eta U L B} \\ &= \frac{2\pi}{(1-\varepsilon^2)^{1/2}}. \end{aligned} \tag{19}$$

The coefficient of friction is given by

$$\begin{aligned} \mu &= \frac{F}{W} \\ &= \frac{2\pi}{(1-\varepsilon^2)^{1/2}} \frac{1}{\left[\dfrac{\varepsilon^2}{(1-\varepsilon^2)^2} + \dfrac{\mu^*}{6} + \dfrac{1}{4}\dfrac{\pi\varepsilon}{(1-\varepsilon^2)^{3/2}}\right]^{1/2}}. \end{aligned} \tag{20}$$

3. Results and Discussion

The variation of pressure distribution, load-carrying capacity, friction force, and coefficient of friction are presented in Equations (12), (16), (19), and (20), respectively. A theoretical comparison with the conventional lubricants suggests that the nondimensional pressure and load-carrying capacity get increased while friction force and coefficient of friction get decreased.

From Figure 2, it is clear that the pressure increases marginally with respect to the magnetization parameter. This is due to the fact that the magnetic pressure adds generated by the magnetic force developed due to the magnetic particles suspended in the lubricant. Figures 3, 4, and 5 indicate that the pressure increases substantially with respect to Z, B/C, and eccentricity ratio, respectively. If B/C is less, then more fluid passes through the gap between journal and bearing and, therefore, more pressure develops. Increase in eccentricity ratio increases the convergent region of fluid film between the bearing and the journal which again increases the pressure. Also, Figures 6 and 7 show pressure distribution with respect to Z for different values of B/C and eccentricity ratio. Figure 8 states that the pressure decreases significantly with increase in Z. Figure 9 represents the pressure distribution versus ε which makes clear that increase in pressure is significant at higher value of eccentricity ratio. Figures 10 and 11 represent pressure distribution versus B/C and ε for different values of and θ, respectively. It makes clear that the pressure increases substantially at $\theta = 135^\circ$. Besides, it is found from Figure 12 that the load-carrying capacity increases as the magnetic parameter increases. It is clear from Figure 13 that the effect of magnetic parameter decreases with increase in eccentricity ratio. In addition to this, it is found from Figure 14 that the coefficient of friction decreases with respect to the increasing values of the magnetic parameter. This probably due to the fact that the magnetic force increases the load-carrying capacity and decrease the friction. Figure 15 indicates that decrease in coefficient of friction is considerable at higher value of eccentricity ratio.

This article reveals that a proper selection of the eccentricity ratio and the magnetic parameter may result in a better performance. The present investigation not only presents the method for enhancing the performance of bearing system but also provides sufficient scopes for extending the life period of the bearing system.

4. Conclusions

This investigation reveals the following.

(1) The magnetization has an overall positive effect on the performance of the bearing system.

(2) The bearing can support a load even when there no motion due to magnetostatic force generated by magnetic particles.

(3) The wear between bearing and the journal is decreases as the initial contact during starting period between them is less.

(4) The evaluation of wear is necessary from bearing life period point of view.

(5) The study establishes the crucial role of eccentricity ratio, even if there is a suitable strength of the magnetic field.

Nomenclature

B: Breadth of bearing (mm)
M^2: Magnetic field
p: Lubricant pressure (N/mm^2)
P: Dimensionless pressure
w: Load-carrying capacity (N)
F: Dimensionless friction force
μ: Coefficient of friction
W: Dimensionless load carrying capacity
μ^*: Dimensionless magnetization parameter
η: Lubricant viscosity (N.S/mm^2)
R_b: Bearing radius (mm)
R_j: Journal radius (mm)
O_b: Bearing center
O_j: Journal center
e: Eccentricity (mm).

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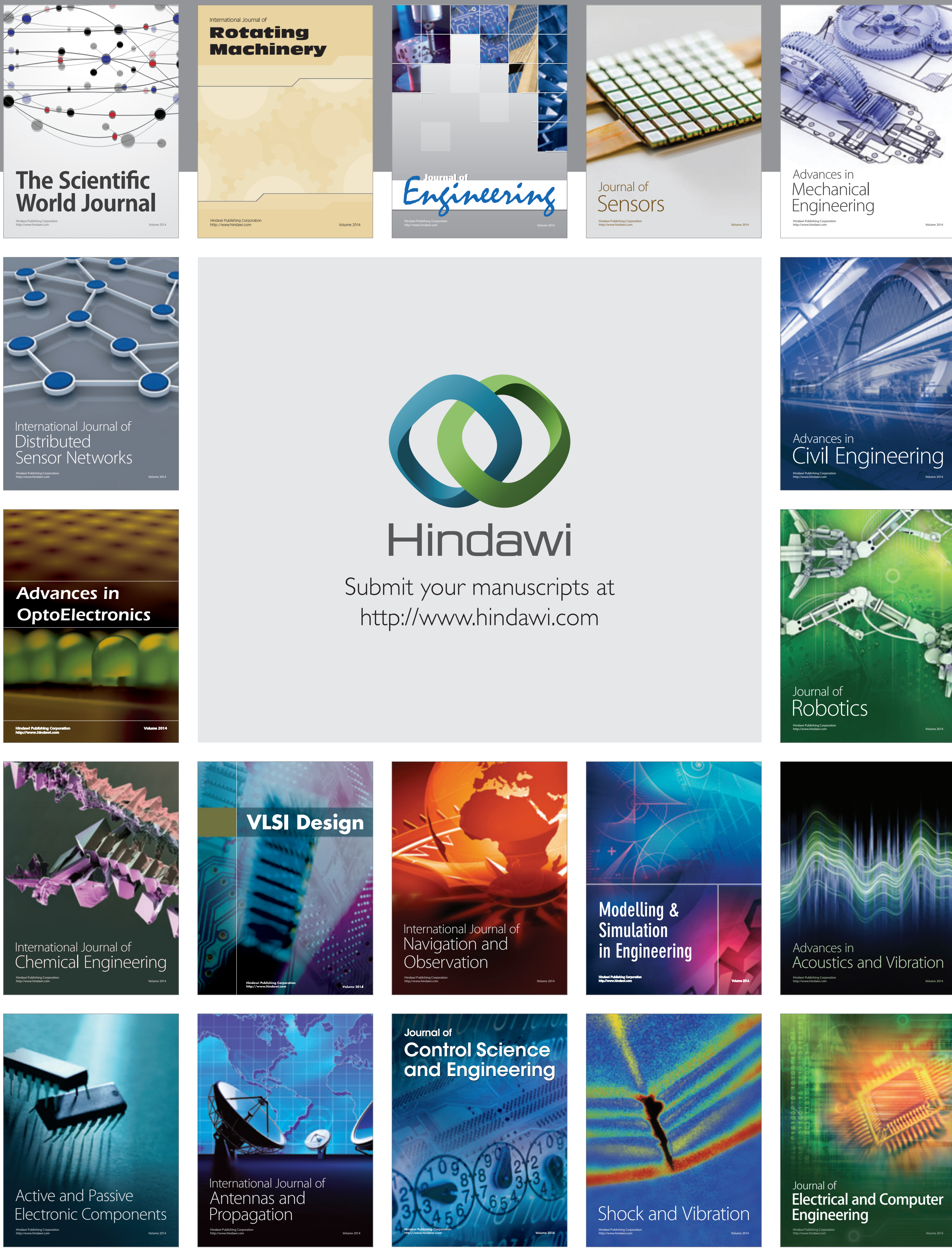
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