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Research Article

Combined Effect of Surface Roughness and Slip Velocity on Jenkins Model Based Magnetic Squeeze Film in Curved Rough Circular Plates

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This paper aims to discuss the effect of slip velocity and surface roughness on the performance of Jenkins model based magnetic squeeze film in curved rough circular plates. The upper plate's curvature parameter is governed by an exponential expression while a hyperbolic form describes the curvature of lower plates. The stochastic model of Christensen and Tonder has been adopted to study the effect of transverse surface roughness of the bearing surfaces. Beavers and Joseph's slip model has been employed here. The associated Reynolds type equation is solved to obtain the pressure distribution culminating in the calculation of load carrying capacity. The computed results show that the Jenkins model modifies the performance of the bearing system as compared to Neuringer-Rosensweig model, but this model provides little support to the negatively skewed roughness for overcoming the adverse effect of standard deviation and slip velocity even if curvature parameters are suitably chosen. This study establishes that for any type of improvement in the performance characteristics the slip parameter is required to be reduced even if variance (−ve) occurs and suitable magnetic strength is in force.

1. Introduction

Nowadays, magnetohydrodynamic flow of a fluid in squeeze film lubrication is of interest, because it prevents the unexpected variation of lubricant viscosity with temperature under various operating conditions. The effects of magnetic fluid in squeeze film lubrication have been encouraging because magnetic fluid has important applications in the industry with obvious relevance to technology-based world. Owing to the development of modern technology, the increasing use of magnetic fluids as lubricants has been highlighted. Magnetic fluids can be controlled and located at some preferred places in the presence of an external magnetic field. Because of these prominent phenomena, ferrofluids are widely used in different fields of sciences and technology, for instance, dampers, seals, sensors, loudspeakers, steppers and coating systems, ink-jet printing, and filtering.

Neuringer and Rosensweig [1] proposed a simple flow model to describe the steady flow of magnetic fluids in the presence of slowly changing external magnetic fields. Numerous papers are available in the literature for the study of different types of bearing using Neuringer and Rosensweig flow model, for example, Tipei [2] in short bearing, Agrawal [3] and Shah and Bhat [4] in slider bearing, journal bearing by Nada and Osman [5] and Patel el al. [6], and circular plates by Shah and Bhat [7] and Deheri and Abhangi [8]. Later on, the flow model of Neuringer and Rosensweig was modified by Jenkins [9] with Maugin's modification. It was found that Neuringer-Rosensweig model modified pressure while Jenkins flow model modified both the pressure and the velocity of the ferrofluid. The steady-state performance of bearings with Jenkins model based magnetic fluids was discussed by Agrawal [3], Ram and Verma [10], Shah and Bhat [11], and Ahmad and Singh [12]. It was concluded that the load carrying capacity of the bearing system increased with increasing magnetization of the magnetic fluid.

Squeezing flow between parallel walls accrues in many industrial and biological systems, such as machine elements, approaching gears, braking units, hydraulic dampers, skeletal bearings, synovial joints, moving pistons in engines, and chocolate filler, and many other devices are based on the principle of flow between contracting domains. To develop this equipment and machines, better understanding of such flow models which describe the squeezing flow between parallel walls is always needed. Analysis of squeeze film performance assumes that the lubricant behaves essentially as a Newtonian viscous fluid although, to establish the flow properties and to increase the lubricating quantities, the use of ferrofluid has been emphasized. Also, the flow pattern corresponds to the slip flow and the fluid presents a loss of adhesion at the welted wall making the fluid slide along the wall in several applications. Flow with slip becomes useful for problems in chemical engineering, for example, flow through pipes in which chemical reactions occur at the walls, two phase flows in bearing system.

When the gap between two mating surfaces becomes smaller, the effects of roughness become more important. In recent years, surface roughness has been studied with much interest because all bearing surfaces are rough to some extent. Also, to increase the performance of hydrodynamic lubrication in various bearings, it is important to evaluate the influence of surface roughness. Tzeng and Saibel [13] and Christensen and Tonder [14–16] proposed the model of surface roughness within the framework of the stochastic theory. Christensen and Tonder's [14–16] stochastic model assumes that the probability density function for the random variable characterizing the roughness is symmetric with the mean of the random variable equal to zero. According to this model, there are two types of roughness patterns which are of special interest in the roughness theory: one is transverse roughness and the other one is longitudinal roughness. In the literature, many authors (Ting [17], Prakash and Tiwari [18], Guha [19], Gupta and Deheri [20], and Chiang et al. [21]) have adopted this model to study the effect of surface roughness. Deheri et al. [22] studied the behaviour of ferrofluid based squeeze film between porous circular plates with porous matrix of variable thickness. Here the magnetization had a significant positive effect by considering a suitable thickness ratio. Bujurke et al. [23] analyzed the effect of surface roughness and couple stress effect on squeeze film behaviour in porous circular disks. Although the transverse roughness had an adverse effect, the couple stress effect improved the performance of the squeeze film. Shimpi and Deheri [24] considered the combined effect of surface roughness and elastic deformation on the behaviour of a ferrofluid based squeeze film between rotating porous circular plates with a concentric circular pocket. In spite of the fact that the combined effect of surface roughness and deformation was adverse the situation remained better due to the magnetization with a suitable choice of pocket radius. Patel et al. [25] extended the discussion of Deheri et al. [22] by incorporating the effect of surface roughness. Here the load carrying capacity was relatively higher when variance (−ve) was involved. Abhangi and Deheri [26] embarked on numerical modelling of squeeze film performance between rotating transversely rough curved circular plates under the presence of a magnetic fluid lubricant. Here it was established that the negatively skewed roughness turned in an augmented performance with suitable choice of rotational inertia and proper choice of curvature parameters. Kudenatti et al. [27] derived a numerical solution of MHD Reynolds equation for squeeze film lubrication between porous and rough rectangular plates. The load was enhanced because of MHD effect. All the above authors investigated that roughness played a crucial role in improving the performance of bearing system. Patel and Deheri [28] analyzed the effect of a ferrofluid lubricated rough porous inclined slider bearing considering slip velocity. It was found that the performance of the bearing system could be made to improve by suitably choosing the magnetization parameter and slip coefficient in the case of negatively skewed roughness. The performance of velocity slip and viscosity variation in squeeze film lubrication of two circular plates was investigated by Rao et al. [29]. Patel and Deheri [30] discussed the effect of various porous structures on the performance of Shliomis model based ferrofluid lubrication of a squeeze film in rotating rough porous curved circular plates. It was manifest that the adverse effect of transverse roughness could be overcome by the positive effect of ferrofluid lubrication in the case of negatively skewed roughness by suitably choosing curvature parameters and rotational inertia. Recently, Patel and Deheri [31] theoretically investigated the effect of Shliomis model based ferrofluid lubrication on the squeeze film between curved rough annular plates with comparison between two different porous structures. It was found that the effect of morphology parameter and volume concentration parameter increased the load carrying capacity.

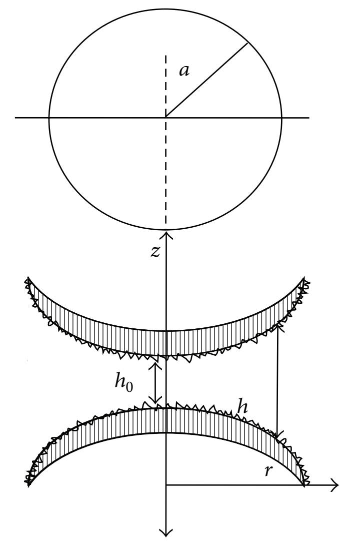

Figure 1: Configuration of the bearing system.

The aim of this paper is to analyze the effect of roughness and slip velocity on the performance of a Jenkins model based magnetic squeeze film in curved rough circular plates.

2. Analysis

The configuration of the bearing system is displayed in Figure 1 (Bhat [32]). The bearing system consists of two circular plates, each of radius a.

In 1972, the flow model of a ferrofluid was discussed by Jenkins. According to this paper the magnetizable liquid was regarded as an anisotropic fluid. Further, to complete the description of the material the vector magnetization density was added to the motion and the temperature. The use of local magnetization as an independent variable allowed Jenkins to treat static and dynamic situation in a uniform fashion and to make a natural distinction between paramagnetic and ferromagnetic fluids. A uniqueness theorem was established for incompressible paramagnetic fluids and determined that in these materials the magnetization vanished with the applied magnetic field.

Using Maugin's modifications, equations of the model for steady flow are (Jenkins [9] and Ram and Verma [10])

$$\rho\left(\overline{q}\cdot\nabla\right)\overline{q} = -\nabla_p + \eta\nabla^2\overline{q} + \mu_0\left(\overline{M}\cdot\nabla\right)\overline{H} + \frac{\rho A^2}{2}\nabla \times\left[\frac{\overline{M}}{M}\times\left\{\left(\nabla\times\overline{q}\right)\times\overline{M}\right\}\right] \tag{1}$$

together with

$$\begin{aligned}\nabla\cdot\overline{q} &= 0,\\ \nabla\times\overline{H} &= 0,\\ \overline{M} &= \overline{\mu}\overline{H},\\ \nabla\cdot\left(\overline{H}+\overline{M}\right) &= 0\end{aligned} \tag{2}$$

(Bhat [32]), where ρ denotes the fluid density, $\overline{q}$ represents the fluid velocity in the film region, $\overline{H}$ is external magnetic field, $\overline{\mu}$ denotes magnetic susceptibility of the magnetic field, p represents the film pressure, η is the fluid viscosity, μ_0 denotes the permeability of the free space, and A is a material constant. The details of these parameters have been discussed by Bhat [32] and Prajapati [33]. From the above equation it is noticed that Jenkins model is a generalization of Neuringer-Rosensweig model with an additional term:

$$\begin{aligned}&\frac{\rho A^2}{2}\nabla\times\left[\frac{\overline{M}}{M}\times\left\{\left(\nabla\times\overline{q}\right)\times\overline{M}\right\}\right]\\ &\quad= \frac{\rho A^2\overline{\mu}}{2}\nabla\times\left[\frac{\overline{H}}{H}\times\left\{\left(\nabla\times\overline{q}\right)\times\overline{H}\right\}\right],\end{aligned} \tag{3}$$

which modifies the velocity of the fluid. Neuringer-Rosensweig model modifies the pressure while Jenkins model modifies both the pressure and the velocity of the magnetic fluid.

Let (u, v, w) be the velocity of the fluid at any point (r, θ, z) between two solid surfaces, with z-axis. Making use of the assumptions of hydrodynamic lubrication and remembering that the flow is steady and axially symmetric, the equations of motion take the form

$$\left(1-\frac{\rho A^2\overline{\mu}H}{2\eta}\right)\frac{\partial^2 u}{\partial z^2} = \frac{1}{\eta}\frac{d}{dr}\left(p-\frac{\mu_0\overline{\mu}}{2}H^2\right) \tag{4}$$

$$\frac{1}{r}\frac{\partial}{\partial r}(ru) + \frac{\partial w}{\partial z} = 0. \tag{5}$$

In view of the boundary conditions,

$$u = 0 \quad \text{at } z = 0, h, \tag{6}$$

the solution of (4) can be obtained as

$$u = \frac{z(z-h)}{2\eta\left(1-\left(\left(\rho A^2\overline{\mu}H\right)/(2\eta)\right)\right)}\frac{d}{dr}\left(p-\frac{\mu_0\overline{\mu}}{2}H^2\right). \tag{7}$$

By substituting the value of u in (5) and integrating it with respect to z over the interval $(0, h)$ one can get Reynolds type equation for film pressure:

$$\begin{aligned}&\frac{1}{r}\frac{d}{dr}\left(\frac{h^3}{\left(1-\left(\left(\rho A^2\overline{\mu}H\right)/(2\eta)\right)\right)}r\frac{d}{dr}\left(p-\frac{\mu_0\overline{\mu}}{2}H^2\right)\right)\\ &\quad= 12\eta\dot{h}_0.\end{aligned} \tag{8}$$

Here the bearing surfaces are considered transversely rough. According to the stochastic model of Christensen and Tonder [14–16], the thickness h of the lubricant film is assumed as

$$h = \overline{h} + h_s. \tag{9}$$

In this equation $\overline{h}$ denotes the mean film thickness and h_s represents the deviation from the mean film thickness characterizing the random roughness of the bearing surfaces. h_s is governed by the probability density function:

$$f\left(h_s\right) = \begin{cases}\dfrac{35}{32c^7}\left(c^2-{h_s}^2\right)^3, & -c \le h_s \le c\\ 0, & \text{elsewhere}\end{cases}, \tag{10}$$

wherein c is the maximum deviation from the mean film thickness. The mean α, the standard deviation σ, and the parameter ε, which is the measure of symmetry of the random variable h_s, are defined by the relationships

$$\begin{aligned}\alpha &= E\left(h_s\right),\\ \sigma^2 &= E\left[\left(h_s-\alpha\right)^2\right],\\ \varepsilon &= E\left[\left(h_s-\alpha\right)^3\right],\end{aligned} \tag{11}$$

where E denotes the expected value defined by

$$E(R) = \int_{-c}^{c} Rf\left(h_s\right)ds. \tag{12}$$

Making use of the discussions of Bhat [32], Abhangi and Deheri [26], and Patel and Deheri [34], it is considered that the upper plate lying along the surface determined by the relation

$$z_u = h_0\exp\left(-\beta r^2\right); \quad 0 \le r \le a \tag{13}$$

approaches with normal velocity $\dot{h}_0$ the lower plate lying along the surface given by

$$z_l = h_0\left[\frac{1}{1+\gamma r}-1\right]; \quad 0 \le r \le a, \tag{14}$$

where β and γ are the curvature parameters of the corresponding plates and h_0 is the central film thickness. The film thickness $h(r)$, then, is defined by (Bhat [32] and Abhangi and Deheri [26])

$$h(r) = h_0\left[\exp\left(-\beta r^2\right) - \frac{1}{1+\gamma r} + 1\right]; \quad 0 \le r \le a. \tag{15}$$

Christensen and Tonder [14–16] proposed a method for the stochastic averaging of the above differential equation. Here an attempt has been made to deploy this technique, which on certain simplifications yields, under the usual assumptions of hydromagnetic lubrication (Bhat [32], Prajapati [33], and Deheri et al. [35]), the modified Reynolds equation,

$$\begin{aligned}\frac{1}{r}\frac{d}{dr}\left(\frac{g(h)}{\left(1-\left(\left(\rho A^2\overline{\mu}H\right)/(2\eta)\right)\right)}r\frac{d}{dr}\left(p-\frac{\mu_0\overline{\mu}}{2}H^2\right)\right) \\ = 12\eta\dot{h}_0,\end{aligned} \tag{16}$$

where

$$\begin{aligned}g(h) = &\left(h^3 + 3h^2\alpha + 3\left(\sigma^2+\alpha^2\right)h + 3\sigma^2\alpha + \alpha^3 + \varepsilon\right) \\ &\cdot\left(\frac{4+sh}{2+sh}\right).\end{aligned} \tag{17}$$

Introducing the nondimensional quantities,

$$\begin{aligned}\overline{h} &= \frac{h}{h_0}, \\ R &= \frac{r}{a}, \\ P &= -\frac{h_0^3 p}{\eta a^2 \dot{h}_0}, \\ B &= \beta a^2, \\ C &= \gamma a, \\ H^2 &= kr^2\frac{(a-r)}{a}, \\ \mu^* &= -\frac{k\mu_0\overline{\mu}h_0^3}{\eta\dot{h}_0}, \\ \overline{A}^2 &= \frac{\rho A^2\overline{\mu}\sqrt{k}a}{2\eta}, \\ \overline{\sigma} &= \frac{\sigma}{h_0}, \\ \overline{\alpha} &= \frac{\alpha}{h_0}, \\ \overline{\varepsilon} &= \frac{\varepsilon}{h_0^3}, \\ \overline{s} &= sh_0\end{aligned} \tag{18}$$

and making use of (18), (16) reduces to

$$\begin{aligned}\frac{1}{R}\frac{d}{dR}\left(\frac{g\left(\overline{h}\right)}{\left(1-\overline{A}^2R\sqrt{1-R}\right)}R\frac{d}{dR}\left(P-\frac{1}{2}\mu^*R^2(1-R)\right)\right) \\ = -12,\end{aligned} \tag{19}$$

where

$$\begin{aligned}g\left(\overline{h}\right) = &\left(\overline{h}^3 + 3\overline{h}^2\overline{\alpha} + 3\left(\overline{\sigma}^2+\overline{\alpha}^2\right)\overline{h} + 3\overline{\sigma}^2\overline{\alpha} + \overline{\alpha}^3 + \overline{\varepsilon}\right) \\ &\cdot\left(\frac{4+\overline{s}\overline{h}}{2+\overline{s}\overline{h}}\right).\end{aligned} \tag{20}$$

Solving the above expression, under the boundary conditions

$$\begin{aligned}P(1) &= 0, \\ \left(\frac{dP}{dR}\right)_{R=0} &= 0,\end{aligned} \tag{21}$$

one can find the expression for dimensionless pressure as

$$\begin{aligned}P = &\frac{1}{2}\mu^*R^2(1-R) \\ &- 6\int_1^R \frac{R}{g\left(\overline{h}\right)}\left(1-\overline{A}^2R\sqrt{1-R}\right)dR.\end{aligned} \tag{22}$$

In view of the classical result of Riemann, following the method of Bhat [32, p. 84], the load carrying capacity of the bearing system in nondimensional form can be obtained from

$$\begin{aligned}W &= -\frac{h_0^3}{2\pi\eta a^4\dot{h}_0}w = \int_0^1 RPdR \\ &= \frac{\mu^*}{40} + 3\int_0^1 \frac{R^3}{g\left(\overline{h}\right)}\left(1-\overline{A}^2R\sqrt{1-R}\right)dR.\end{aligned} \tag{23}$$

The time t taken by the upper plate to reach a film thickness h_0 starting from an initial film thickness h_2 can be obtained in dimensionless form as

$$\begin{aligned}\overline{t} &= \frac{h_2^2Wt}{\eta a^4} \\ &= \frac{3\left(\left(1/\overline{h}_0^2\right)-1\right)\int_0^1\left(R^3/g\left(\overline{h}\right)\right)\left(1-\overline{A}^2R\sqrt{1-R}\right)dR}{\left((1/\pi)-\left(\mu_1^*/20\right)\right)},\end{aligned} \tag{24}$$

where

$$\begin{aligned}\overline{h}_0 &= \frac{h_0}{h_2}, \\ \mu_1^* &= \frac{k\mu_0\overline{\mu}a^4}{W}.\end{aligned} \tag{25}$$

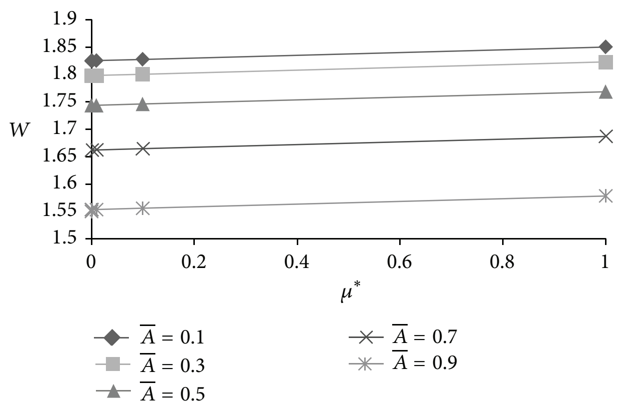

Figure 2: Variation of load carrying capacity with respect to μ^* and $\overline{A}$.

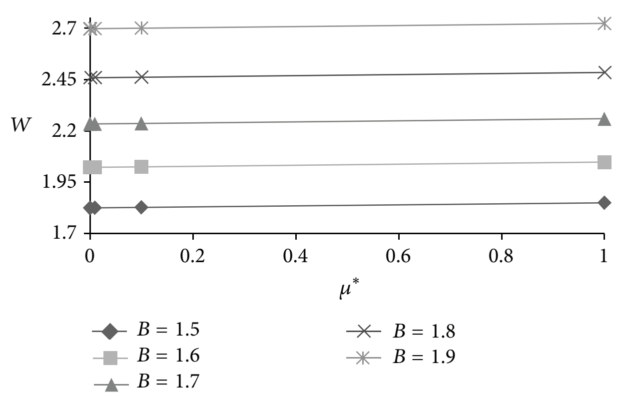

Figure 3: Variation of load carrying capacity with respect to μ^* and B.

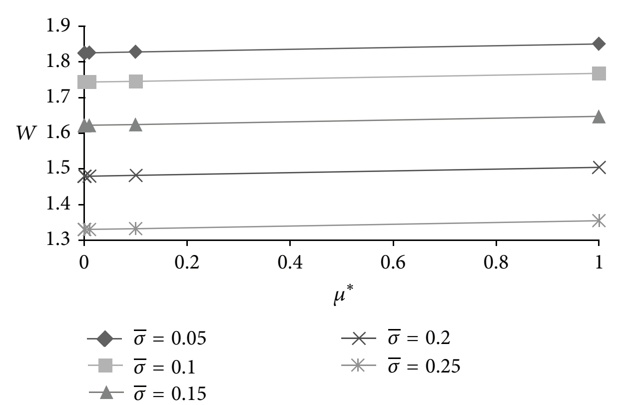

Figure 4: Variation of load carrying capacity with respect to μ^* and $\overline{\sigma}$.

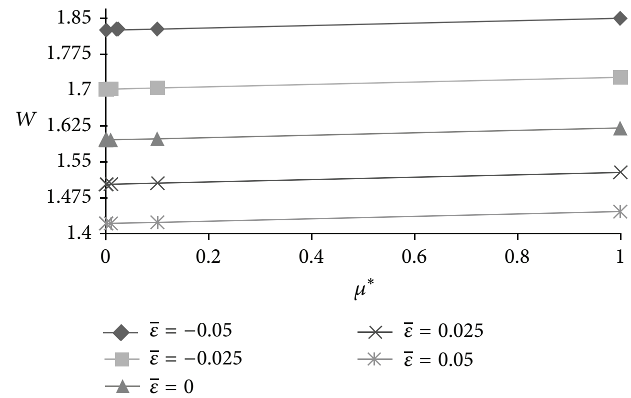

Figure 5: Variation of load carrying capacity with respect to μ^* and $\overline{\varepsilon}$.

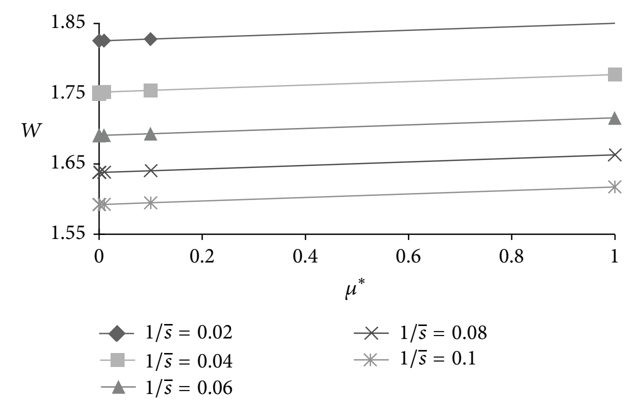

Figure 6: Variation of load carrying capacity with respect to μ^* and $1/\overline{s}$.

3. Results and Discussions

Equation of load carrying capacity offers the suggestion that the load carrying capacity gets increased by

$$\frac{\mu^*}{40} \tag{26}$$

as compared to conventional lubricant based bearing system. Probably, this may be due to the fact that the viscosity gets enhanced due to magnetization. It is noticed that the expression involved in (23) is linear with respect to the magnetization parameter. Accordingly an increase in the magnetization parameter eventually results in increased load carrying capacity.

The fact that the load carrying capacity gets marginally affected by the magnetization is displayed in Figures 2, 3, 4, 5, and 6. The increase in the load carrying capacity with respect to magnetization is caused because the magnetization induces an increase in the viscosity of the lubricant leading to increased pressure.

Figures 7, 8, 9, and 10 indicate that the load carrying capacity decreases significantly with the increase in the material constant parameter. Besides, the combined effect of material constant parameter and slip parameter causes severe reduction in load carrying capacity. The effect of material constant parameter modifies the velocity of the ferrofluid and consequently leads to decreased pressure resulting in reduced load carrying capacity.

The effect of curvature parameters is presented in Figures 11, 12, 13, 14, 15, and 16. As can be seen the trends of load carrying capacity with respect to the upper plate's curvature parameter are opposite to that of the lower plate's curvature parameter. These figures also underline the fact that the

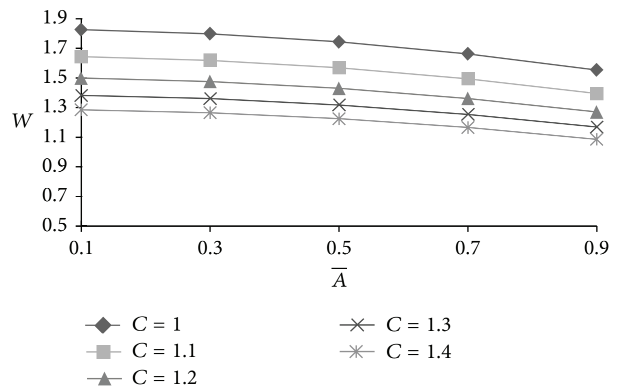

Figure 7: Variation of load carrying capacity with respect to $\overline{A}$ and C.

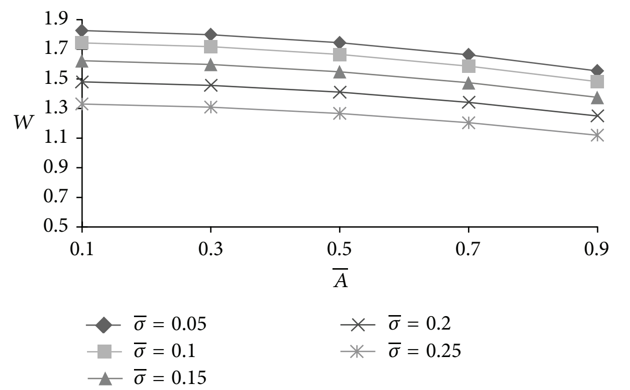

Figure 8: Variation of load carrying capacity with respect to $\overline{A}$ and $\overline{\sigma}$.

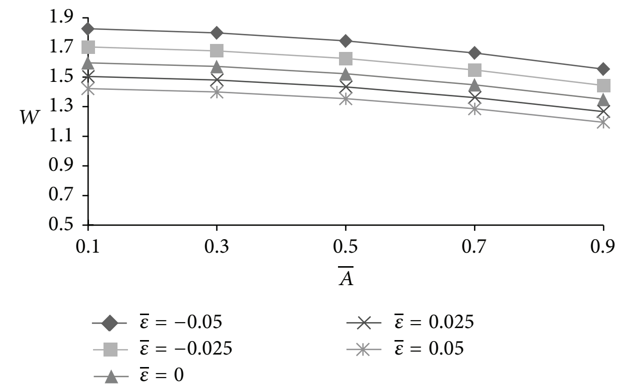

Figure 9: Variation of load carrying capacity with respect to $\overline{A}$ and $\overline{\varepsilon}$.

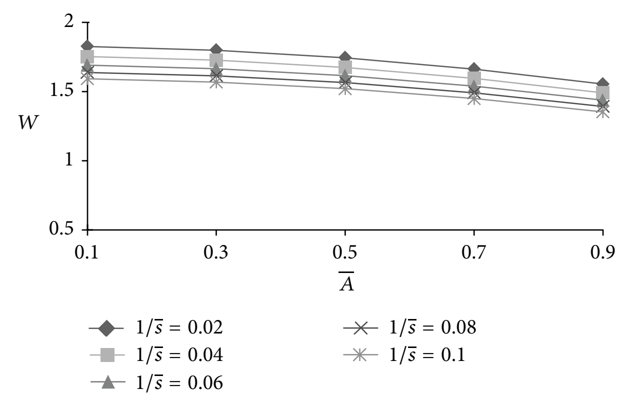

Figure 10: Variation of load carrying capacity with respect to $\overline{A}$ and $1/\overline{s}$.

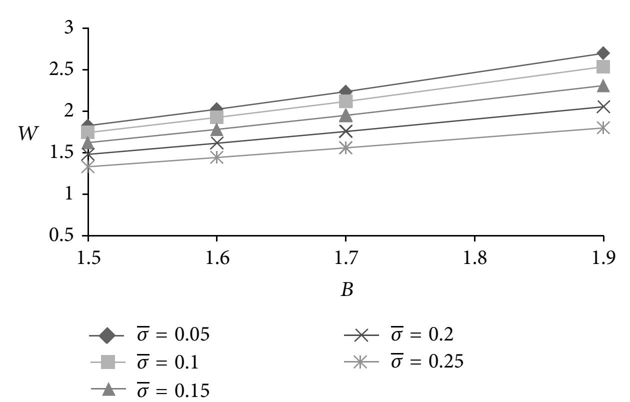

Figure 11: Variation of load carrying capacity with respect to B and $\overline{\sigma}$.

lower plate's curvature parameter should be kept at minimum for obtaining a better performance when moderate to large values of upper plate's curvature parameters are involved.

Figures 17, 18, and 19 convey that the standard deviation has a considerable adverse effect on the performance of the bearing system. This is not surprising because the motion of the fluid gets retarded by roughness. This means the combined effect of standard deviation and material constant has a strong adverse effect on the performance of the bearing system.

It is seen from Figures 20 and 21 that the load carrying capacity decreases as positive skewness increases while the negatively skewed roughness increases the load carrying capacity. Variance follows the path of skewness so far as the trends of load carrying capacity are concerned (Figure 22). Roughness retards the motion of the lubricant and hence causes reduced pressure leading to decreased load carrying capacity.

The slip parameter has a strong adverse influence on the performance of the bearing system because the fluid presents a loss of adhesion at the welted wall leading to decreased pressure. Some of the graphical representations make it clear that the combined effect of negatively skewed roughness and variance (−ve) may offer some help for getting an improved performance. It is observed that the squeeze time turns out to be relatively better as compared to the case of Neuringer-Rosensweig model based magnetic fluid lubrication.

Some of the conclusions are validated by comparing the present results with the already known results (Abhangi and Deheri [26] and Shimpi and Deheri [24]) in the case of Neuringer-Rosensweig model. Close scrutiny of the comparison reveals that there is at least 0.75% increase in the

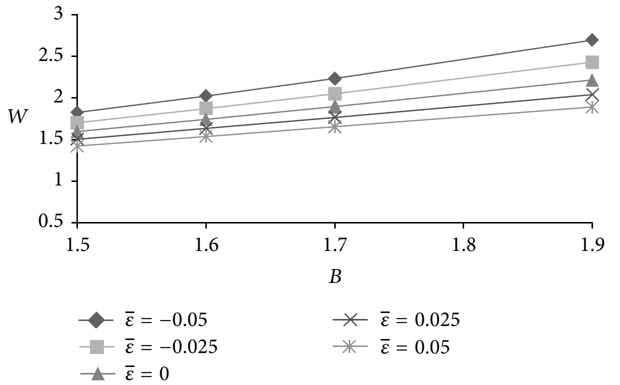

Figure 12: Variation of load carrying capacity with respect to B and $\bar{\varepsilon}$.

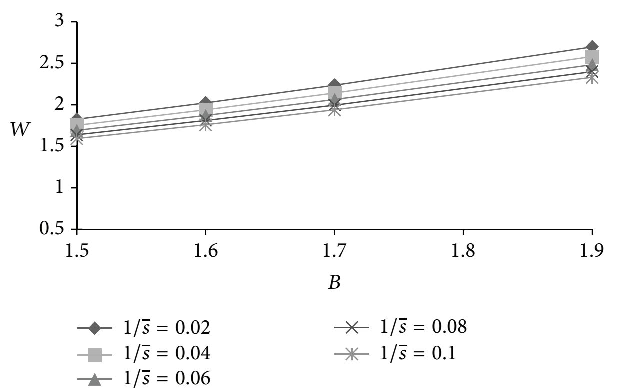

Figure 13: Variation of load carrying capacity with respect to B and $1/\bar{s}$.

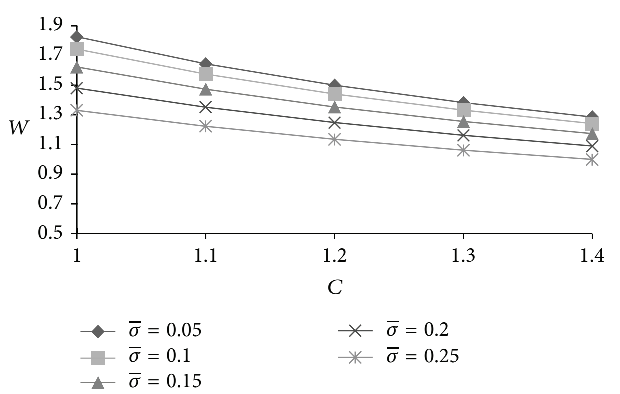

Figure 14: Variation of load carrying capacity with respect to C and $\bar{\sigma}$.

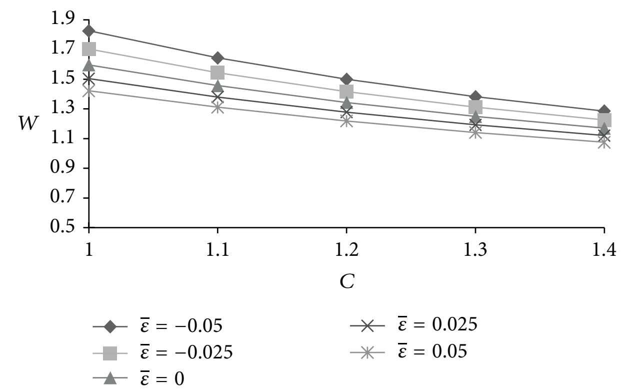

Figure 15: Variation of load carrying capacity with respect to C and $\bar{\varepsilon}$.

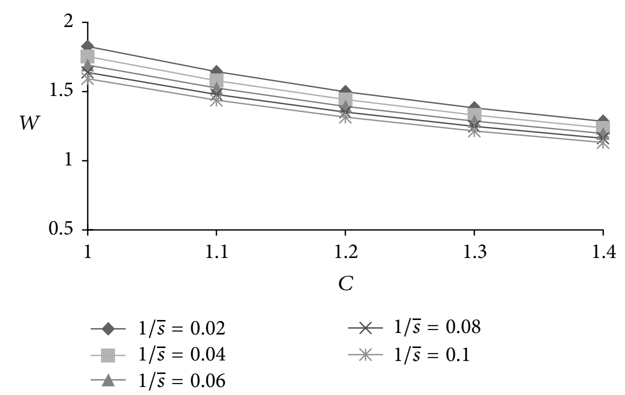

Figure 16: Variation of load carrying capacity with respect to C and $1/\bar{s}$.

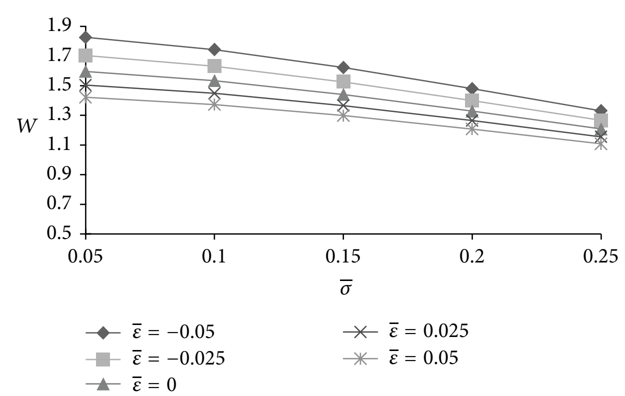

Figure 17: Variation of load carrying capacity with respect to $\bar{\sigma}$ and $\bar{\varepsilon}$.

load carrying capacity in comparison with the Neuringer-Rosensweig model based fluid flow, although roughness and slip velocity bring down the load carrying capacity. The effect of slip velocity is, however, nominally better as compared to Neuringer-Rosensweig model.

4. Conclusion

This paper reveals that the magnetization has a limited option for reducing the adverse effect of roughness and slip. However, for a better performance the slip parameter must be kept at minimum even if negatively skewed roughness is involved and variance (−ve) occurs. This investigation thus makes it mandatory to account for roughness while designing the bearing system. The curvature parameters provide an additional degree of freedom from design point of view. Lastly, it is observed that even in the absence of flow the

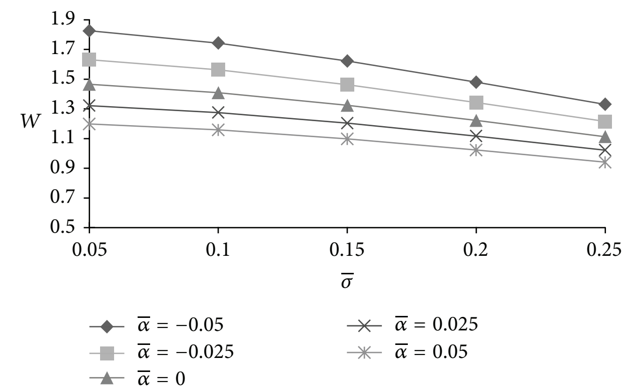

Figure 18: Variation of load carrying capacity with respect to $\overline{\sigma}$ and $\overline{\alpha}$.

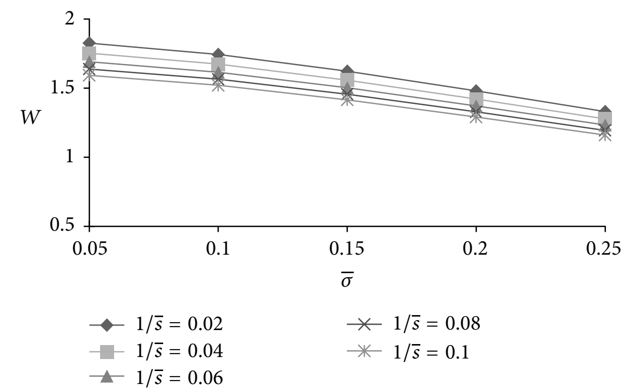

Figure 19: Variation of load carrying capacity with respect to $\overline{\sigma}$ and $1/\overline{s}$.

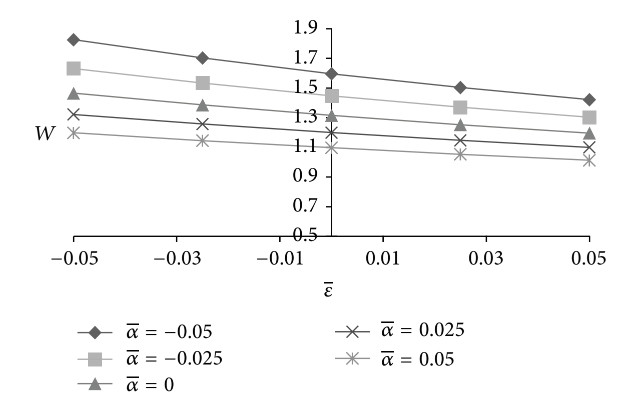

Figure 20: Variation of load carrying capacity with respect to $\overline{\varepsilon}$ and $\overline{\alpha}$.

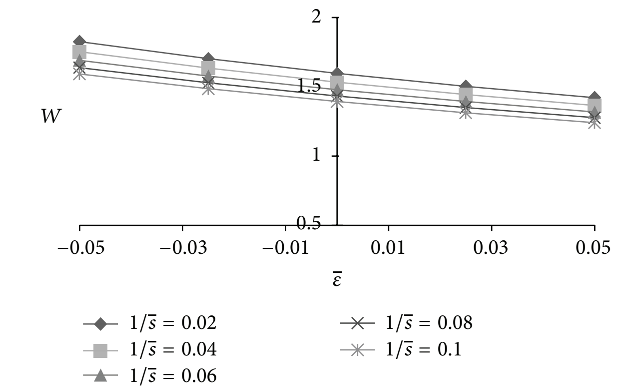

Figure 21: Variation of load carrying capacity with respect to $\overline{\varepsilon}$ and $1/\overline{s}$.

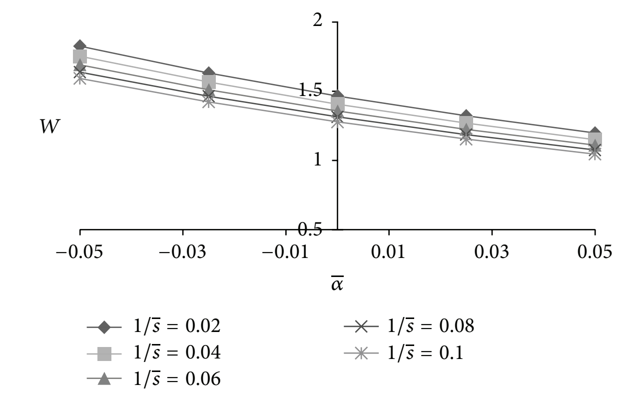

Figure 22: Variation of load carrying capacity with respect to $\overline{\alpha}$ and $1/\overline{s}$.

bearing system supports certain amount of load, unlike the case of conventional lubricants based bearing system.

Conflict of Interests

The authors declare that there is no conflict of interests regarding the publication of this paper.

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