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Ferrofluid Lubrication of a Double Layer Porous Rough Slider Bearing

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Abstract: A study on the performance of a ferrofluid based double layer porous rough slider bearing is presented. The stochastic model of Christensen and Tonder has been used to evaluate the effect of surface roughness. The expressions for pressure distribution, load carrying capacity, friction coefficient, centre of pressure and temperature rise are obtained. The effect of double layer increases the load carrying capacity and decreases the coefficient of friction. It is appealing to see that the positive effect of double layered gets enhanced by the magnetic fluid lubrication. The negatively skewed roughness further augments this performance. However, for lower values of magnetization parameter the effect of temperature rise is not that significant. The centre of pressure shifts towards the outlet edge for lower to moderate values of magnetization.

Keywords: Slider bearing; roughness; magnetic fluid; double layered porous.

1. Introduction

Amongst the hydrodynamic bearings, slider bearing is the simplest and frequently encountered because the expression of film thickness is simple and boundary conditions to be required zero at the bearing ends are not that complicated. The fundamental aspect in a hydrodynamic slider bearing is the formation of a converging wedge of the lubricant. The hydrodynamic slider may be constructed to provide this converging wedge in a number of ways. Major applications of slider bearing include factory automation, machine tools, semiconductor, printing, automotive assembly and aerospace. Prakash and Vij [1] analyzed a plane slider where in one of the surface was having a porous facing.

The effect of porosity decreases the load carrying capacity but the load carrying capacity of the bearing can be augmented by reducing the seepage into the wall of the bearing. This can be obtained by lowering the permeability. Unfortunately, this is impractical as reduction in the permeability results in the reduction of porosity and hence reduction of the fluid contained within the bearing material. A double layer porous facing may be useful because it may not only increase the load carrying capacity, but also help to bring the fluid between the surfaces, thereby, improving the performance of the bearing system when it is not completely saturated with the fluid. Bujurke *et al*. [2] theoretically studied the performance of a porous slider bearing in the presence of a couple stress fluid. The couple stress effect yielded an increase in load carrying capacity but ensured the decrease in the coefficient of friction. Malvano and Vatta [3] dealt with the problem of determining the influence of pressure at the bearing entry on load carrying capacity of a plane slider bearing. It

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was established that pivoted pad could be employed where a shaft was also to be rotated in the reverse direction.

The analyses of porous slider bearings have been based upon the Darcy's model, where Darcy's equations were applied to guide fluid motion through the porous medium [4-8]. Cusano [9] obtained an analytical solution for the performance characteristics of a two layer porous bearing using the short bearing approximation. Shrinivasan [10] launched an investigation into the performance of a double layered porous slider bearing. The results showed that the existence of double layered increased the load carrying capacity but decreased the coefficient of friction. Bujurke *et al*. [11] investigated the influence of couple stresses on the dynamic properties of a double layer porous slider bearing. Chebakov [12] embarked on an analysis of a two layer cylindrical bearing. The performance was found to be a little bit better than that of a single layer cylindrical bearing. Later on, Rao *et al*. [13] presented an analysis of a long journal bearing with a double layered porous lubricant film using couple stress and Newtonian fluids. In all the above studies a double layered porous lubricant film configuration increased the load carrying capacity and reduced the coefficient of friction in the bearing.

All surfaces are more or less rough and the roughness has strong effects on many interesting problems like wetting, adhesion, friction, light scattering and surface growth that are at the fore front of science and technology. Bujurke and Patil [14] studied the effects of variable permeability and surface roughness on the hydrodynamic characteristics of squeeze film porous bearings. It was established that the load carrying capacity decreased for higher values of the permeability parameter and that the optimal load shifted for off-squared plates. The squeeze time for the anisotropic case was found to be quite large as compared to the isotropic case. Ozalp and Umur [15] presented an optimization study to propose an innovative surface profile design by implementing a wavy form on the upper surface without varying the physical limits of the complete slider bearing structure. The computations indicated that friction coefficient values decreased with wave amplitude in suitable pad inclination ranges. Sinha and Adamu [16] analyzed the thermal and roughness effects on performance characteristics of an infinite tilted pad slider bearing. It was observed that for a non parallel slider bearing the load carrying capacity due to the combined effect was less than the load capacity due to the roughness effect. However, in the case of parallel pad slider bearing the reverse was true, although, the load capacity was not that significant. Adamu and Sinha [17] investigated the thermal and roughness effects in a tilted pad slider bearing considering the heat condition through the pad and the slider. In order to derive modified Reynolds equation the irregular domain of the fluid due to roughness was made into a regular domain and numerical methods were applied. Tauvqirrahman *et al*. [18] made a systematic comparison with various surface conditions, that is texturing slip and the combination of those configurations with respect to the performance of flat classical (no slip) contact. Optimal values of design parameters were provided for allowing maximum load carrying capacity.

Always lubrication occurs in engines and machines to reduce friction between the moving plates. All the above studies used various kinds of fluids as lubricants. During the last decade, the use of magnetic fluid as a lubricant has received considerable attention. Use of magnetic fluid as a lubricant modifying the performance of the bearings has been very well established. The application of magnetic fluid as a lubricant was investigated by many authors [19-29]. In all these studies it has been established that the performance of bearing system could be improved by using a magnetic fluid as the lubricant. Andharia and Deheri [30] studied the effect of longitudinal roughness on the performance of a plane slider bearing in the presence of a ferrofluid. This study recorded that the negative effect of roughness could be minimized to a large extent by the positive effect of magnetization. Recently, Patel and Deheri [31] studied the performance of a magnetic fluid based infinitely long rough bearing with slip velocity. It was accomplished that keeping the slip coefficient at minimum, the magnetization might compensate the adverse effect of the standard deviation.

The present study deals with the analytical solution of the performance of double layer porous, rough, slider bearing using a ferrofluid as the lubricant.

2. Analysis

The bearing configuration consists of two surfaces separated by a lubricant film. The X-axis is taken along the lower plane across its length, while the Z-axis is taken across the lubricant film (Figure 1). The lower surface moves with a velocity U in the X-direction while the upper surface is stationary. Both the surfaces are considered to be infinitely wide in Y-direction. The shape of the lubricant film formed between the surfaces is convergent so that pressure generating mechanism surfaces. The lubricant film is assumed to be isoviscous and incompressible and the flow is laminar.

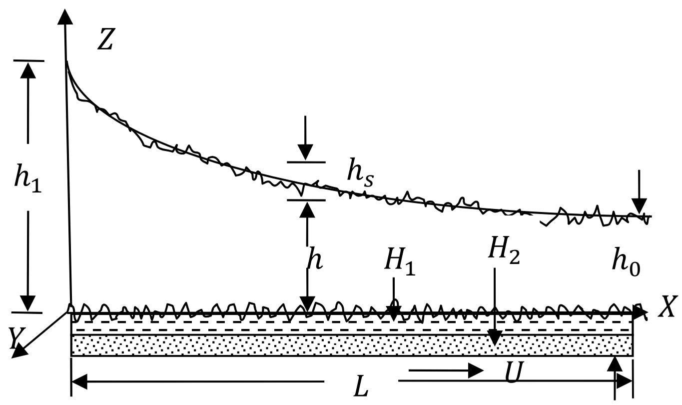


Figure 1. Configuration of the bearing system

The bearing surfaces are considered to be transversely rough. In view of the deliberations of Christensen and Tonder [32-34], the film thickness h of the lubricant film is taken as

$$h = \bar{h} + h_s \tag{1}$$

where $\bar{h}$ is the mean film thickness and h_s is the deviation from the mean film thickness characterizing the random roughness of the bearing surfaces. h_s is governed by the probability density function

$$f(h_s) = \begin{cases} \dfrac{35}{32c_1}\left(1 - \dfrac{{h_s}^2}{c_1^2}\right)^3, & -c_1 \le h_s \le c_1 \\ 0, & elsewhere \end{cases}.$$

wherein c_1 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 governed by the relationships:

$$\alpha = E(h_s), \sigma^2 = E[(h_s - \alpha)^2], \varepsilon = E[(h_s - \alpha)^3]$$

where E denotes the expectency operator defined by

$$E(*) = \int_{-c}^{c} (*) f(h_s) dh_s$$

The details can be seen from Christensen and Tonder [32-34].

Agrawal [19] investigated the magnetic fluid lubrication effect by taking the magnetic field oblique to the stator. The effect of various forms of magnitude of the magnetic field has been discussed by Bhat [35] and Prajapati [36]. Following these discussions the magnitude of the magnetic field is assumed to be

$$H^{*2} = kX(1 - X) \tag{2}$$

where H^* denotes magnitude of $\bar{H}$ and k is a suitably chosen constant from dimensionless point of view so as to produce a magnetic field of strength over 10^{-23} as discussed in Bhat and Deheri [20].

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

$$\rho(\bar{q}\ \nabla)\bar{q} = -\nabla p + \eta\ \nabla^2 \bar{q} + \mu_0(\bar{M}\ \nabla)\bar{H} \tag{2a}$$

$$\nabla \bar{q} = 0 \tag{2b}$$

$$\nabla \times \bar{H} = 0 \tag{2c}$$

$$\bar{M} = \bar{\mu}\, \bar{H} \tag{2d}$$

$$\nabla\ (\bar{H} + \bar{M}) = 0 \tag{2e}$$

where ρ represents the fluid density, $\bar{q}$ is the fluid velocity in the film region, $\boldsymbol{p}$ is the film pressure, η represents the fluid viscosity, μ_0 denotes the permeability of free space, $\bar{M}$was the magnetization vector, $\bar{H}$denotes the external magnetic field and $\bar{\mu}$ is the magnetic susceptibility of the magnetic particles.

Using above equations (2b) to (2e), equation (2a) assumed the form

$$\rho(\bar{q}\ \nabla)\bar{q} = -\nabla\left(p - \frac{\mu_0 \bar{\mu}}{2} H^{*2}\right) + \eta\ \nabla^2 \bar{q}$$

This suggestes that an extra pressure $\frac{\mu_0 \bar{\mu}}{2} H^{*2}$ is introduced in to the Navier-Stokes equations when a magnetic fluid is used as a lubricant.

According to the usual assumptions of hydro-magnetic lubrication [35, 36, 22], the modified Reynolds equation governing the pressure distribution in the present case turns out to be

$$\frac{d}{dx}\left[g(h)\frac{d}{dx}\left(p - \frac{\mu_0\, \bar{\mu} H^{*2}}{2}\right)\right] = 6\mu U \frac{dh}{dx} \tag{3}$$

Where

$$g(h) = h^3 + 3h^2\alpha + 3(\sigma^2 + \alpha^2)h + 3\sigma^2\alpha + \alpha^3 + \varepsilon + 12\phi_1 H_1 + 12\phi_2 H_2.$$

while μ is the lubricant viscosity and ϕ_1 and ϕ_2 are permeability of porous regions, and H_1 and H_2 are layer thicknesses.

The concerned boundary conditions are

$$P\ = 0 \text{ at } X = 0, 1 \tag{4}$$

Introducing the dimensionless quantities

$$H = \frac{h}{\Delta h} = H_0 + a(1 - X), \qquad H_m = \frac{h_m}{\Delta h}, \qquad X = \frac{x}{L},$$

$$\bar{\sigma} = \frac{\sigma}{\Delta h}, \qquad \bar{\alpha} = \frac{\alpha}{\Delta h}, \qquad \bar{\varepsilon} = \frac{\varepsilon}{(\Delta h)^3},$$

$$G(H) = \frac{g(h)}{(\Delta h)^3} = H^3 + 3H^2\bar{\alpha} + 3(\bar{\sigma}^2 + \bar{\alpha}^2)H + 3\bar{\sigma}^2\bar{\alpha} + \bar{\alpha}^3 + \bar{\varepsilon} + 12\bar{\psi},$$

$$\bar{\psi} = \bar{\psi}_1 + \bar{\psi}_2, \qquad \bar{\psi}_1 = \frac{\phi_1 H_1}{(\Delta h)^3}, \qquad \bar{\psi}_2 = \frac{\phi_2 H_2}{(\Delta h)^3}, \qquad P = \frac{(\Delta h)^2}{U\mu L}p,$$

$$\Delta h = h_1 - h_0, \qquad \mu^* = \frac{k\mu_0\bar{\mu}(\Delta h)^2}{U\mu}, \qquad a = 3\bar{\alpha},$$

$$b = 3(\bar{\alpha}^2 + \bar{\sigma}^2), \qquad c = \bar{\alpha}^3 + 3\bar{\sigma}^2\bar{\alpha} + \bar{\varepsilon} + 12\bar{\psi},$$

$$J = \sqrt[3]{-2a^3 + 3\sqrt{3}K_1 + 9ab - 27c},$$

$$K_1 = \sqrt{4a^3c - a^2b^2 - 18abc + 4b^3 + 27c^2},$$

$$J_1 = \frac{J}{3\sqrt[3]{2}} - \frac{\sqrt[3]{2}(3b - a^2)}{3J} - \frac{a}{3},$$

$$Q = \frac{J}{6\sqrt[3]{2}}, \qquad A = \frac{J_1}{S}, \qquad R = \frac{(3b - a^2)}{32^{2/3}J},$$

$$J_2 = -2Q + 2R - \frac{2a}{3}, \qquad T_1 = \tan^{-1}\left(\frac{2(H_0 + \beta) - J_2}{\sqrt{4J_3 - J_2^2}}\right),$$

$$J_3 = 4Q^2 + 4QR + 4R^2 + 2Q\frac{a}{3} - 2R\frac{a}{3} + \frac{a^2}{9}, \qquad B = -A, S = J_1^2 - J_1J_2 + J_3, C = \frac{J_3}{S},$$

$$A_1 = \frac{1}{S}, \qquad B_1 = -A_1, \qquad C_1 = \frac{J_2 - J_1}{S}, \qquad D = A - H_mA_1, \qquad E = \frac{1}{2}(B - H_mB_1),$$

$$F = \frac{(2C + BJ_2) - H_m(2C_1 + B_1J_2)}{\sqrt{4J_3 - J_2^2}}, L_1 = \ln(H_0 + \beta - J_1), L_3 = \ln\left({H_0}^2 - J_2H_0 + J_3\right),$$

$$L_1 = \ln(H_0 + \beta - J_1),$$

$$L_2 = \ln((H_0 + \beta)^2 - J_2(H_0 + \beta) + J_3), \qquad T_2 = \tan^{-1}\left(\frac{2H_0 - J_2}{\sqrt{4J_3 - J_2^2}}\right),$$

$$L_3 = \ln\left({H_0}^2 - J_2H_0 + J_3\right),$$

$$H_m = \frac{AL_1 - A\ln(H_0 - J_1) + \frac{1}{2}B(L_2 - L_3) + \left(\frac{2C + BJ_2}{\sqrt{4J_3 - J_2^2}}\right)(T_1 - T_2)}{A_1L_1 - A_1\ln(H_0 - J_1) + \frac{1}{2}B_1(L_2 - L_3) + \left(\frac{2C_1 + B_1J_2}{\sqrt{4J_3 - J_2^2}}\right)(T_1 - T_2)} \tag{5}$$

where β, h_m, h_1, h_0 are film shape parameters in dimensionless form, the film thickness at which the pressure is maximum, inlet film thickness and outlet film thickness respectively, using the boundary conditions (4), the non dimensional form of the pressure distribution is derived as

$$P = \frac{\mu^*}{2}X(1-X) + \frac{6}{m}\left[DL_1 - D\ln(H-J_1) + EL_2 + FT_1 - E\ln(H^2 - J_2H + J_3) - F\tan^{-1}\left(\frac{2H-J_2}{\sqrt{4J_3-J_2^2}}\right)\right] \quad (6)$$

The non dimensional load carrying capacity of the bearing system then is determined by

$$W = \frac{(\Delta h)^2}{U\mu L^2}w = \int_0^1 PdX \quad (7)$$

w being the load carrying capacity. Consequently, the expression for the dimensionless load carrying capacity turns out to be

$$W = \frac{\mu^*}{12} + \frac{6}{\beta^2}[\beta(DL_1 + EL_2 + FT_1) + DI_1 + EI_2 + FI_3] \quad (8)$$

where in

$$I_1 = (H_0 - J_1)\ln(H_0 - J_1) - (H_0 + \beta - J_1)\ln(H_0 + \beta - J_1) + \beta$$

$$I_2 = 2\beta + \sqrt{4J_3 - J_2^2}\,(T_2 - T_1) + \left(\frac{2H_0 - J_2}{2}\right)L_3 - \left(\frac{2(H_0+\beta) - J_2}{2}\right)L_2$$

$$I_3 = \left(\frac{-J_2}{2} + H_0\right)(T_2 - T_1) - \beta T_1 + \frac{1}{4}\sqrt{4J_3 - J_2^2}(L_2 - L_3)$$

The frictional force F per unit width on the lower plane defined as

$$\overline{F} = \mu\int_0^L \left(\frac{\partial u}{\partial z}\right)_{z=0} dX = \frac{F\Delta h}{\mu UL} = -\int_0^1 \left[\frac{H}{2}\frac{dP}{dX} + \frac{1}{H}\right]dX$$

where u is the lubricant velocity in X- direction, in dimensionless form takes the form

$$\bar{F} = -\frac{\mu^*\beta}{24} + \frac{3D}{\beta}F_1 - \frac{3E}{\beta}F_2 + \frac{6F}{4}F_3 + \frac{1}{\beta}\ln\left(\frac{H_0}{H_0+\beta}\right) \quad (9)$$

where

$$F_1 = [(-J_1L_1 - \beta) + J_1\ln(H_0 - J_1)], F_2 = \left[\frac{J_2}{2}(L_2 - L_3) + \sqrt{4J_3 - J_2^2}(T_2 - T_1) + 2\beta\right]$$

and

$$F_3 = \left[\frac{1}{2}(L_3 - L_2) + \frac{J_2}{\sqrt{4J_3 - J_2^2}}(T_2 - T_1)\right]$$

The friction coefficient is governed by the relation

$$f = -\frac{\bar{F}}{W} \quad (10)$$

The location of centre of pressure, where the resultant force acts, in non dimensional form is

$$\bar{X}=\frac{1}{W}\int_0^1 PXdX=\frac{\mu^*}{24W}+\frac{6}{\beta W}\left[\frac{1}{2}(DL_1+EL_2+FT_1)-\frac{D}{4}Y_1-\frac{E}{4}Y_2-\frac{F}{8}Y_3\right] \quad (11)$$

where

$$Y_1=-1+2\ln(H_0-J_1)-\frac{2(\beta+H_0-J_1)}{\beta}-\frac{2(\beta+H_0-J_1)^2}{\beta^2}\ln(-H_0+J_1)+Y_1^{'}$$

$$Y_1^{'}=\frac{2(\beta+H_0-J_1)^2}{\beta^2}\ln(-(H_0+\beta)+J_1)$$

$$Y_2=\frac{2(-2(\beta+H_0)+J_2)}{\beta}+\frac{\{2(H_0+\beta)^2-2J_2(H_0+\beta)+({J_2}^2-2J_3)\}}{\beta^2}(L_2-L_3)+Y_2^{'}$$

$$Y_2^{'}=-2+2L_3+\frac{2(-2(\beta+H_0)+J_2)\sqrt{4J_3-J_2^2}}{\beta^2}(T_2-T_1)$$

$$Y_3=4T_2+\frac{\{2(H_0+\beta)^2-2J_2(H_0+\beta)+({J_2}^2-2J_3)\}}{\beta^2}(T_1-T_2)+\frac{2\sqrt{4J_3-J_2^2}}{a}+Y_3^{'}$$

and

$$Y_3^{'}=\frac{(2(\beta+H_0)-J_2)\sqrt{4J_3-J_2^2}}{\beta^2}(L_3-L_2)$$

The work done against the viscous stress gives rise to temperature rise of the lubricant which may be transferred by conduction, radiation or the convection. The bulk of heat flow is carried through the convection by the lubricant. In view of the derivation of Andharia [37], following Hamrock [38], the temperature rise Δt in dimensionless form is given by

$$\Delta T=\frac{gJ^{'}\rho C^{'}(\Delta h)^2}{2\mu UL}=-\frac{F}{H_m} \quad (12)$$

where $g, J^{'}, C^{'}$ and ρ are the gravitational acceleration, the Joule's mechanical equivalent of heat, specific heat and density of the lubricant respectively.

3. Results And Discussions

It is easily seen from equation (6) and equation (8) that the non-dimensional pressure increases by $\frac{\mu^*}{2}X(1-X)$ while the increase in the load carrying capacity turns out to be $\frac{\mu^*}{12}$ as compared to the case of conventional lubricant. Further, it is easily seen from equation (8) that this type of bearing system sustains certain amount of load even when there is no flow which does not happen in the case of traditional lubricants. Besides, the expression involved in equation (8) is linear with respect to the magnetization parameter. Accordingly, an increase in the magnetization will lead to increased load carrying capacity. This is not surprising as the magnetization increases the viscosity of the lubricant. Equation (9) indicates that the friction changes by $-\frac{\mu^*\beta}{24}$. Accordingly, the coefficient of friction decreases as compared to usual conventional fluid lubrication. For a smooth bearing this type of bearing system can be thought of as an equivalent bearing system with film thickness $h^3+12(\bar{\psi}_1+\bar{\psi}_2)$ in the absence of magnetization. Since roughness obstructs the motion of the lubricant pressure gets decreased and hence load carrying capacity becomes less.

It is seen from Figures 2 and 3 that an increase in the film shape parameter lowers the load carrying capacity. It is noted that positively skewed roughness tends to decrease the load carrying capacity while load carrying capacity gets enhanced due to negatively skewed roughness. Undoubtedly the porosity reduces the load carrying capacity as can be seen from Figures 2 and 4. Figure 5 suggests that the combined effect of standard deviation and positively skewed roughness is relatively adverse while the negatively skewed roughness increases the load carrying capacity while the effect of variance on the distribution of load carrying capacity with respect to positive skewness is almost negligible.

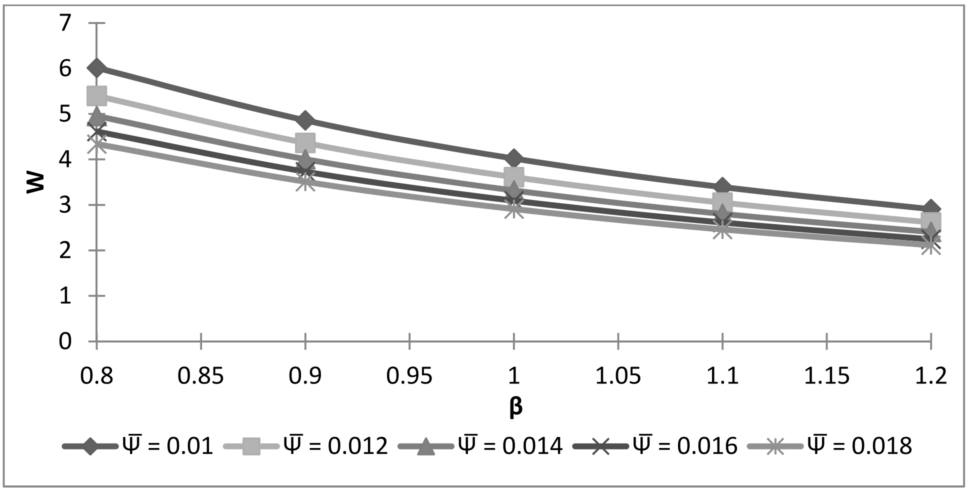


Figure 2. Variation of Load carrying capacity with respect to β and $\bar{\psi}$

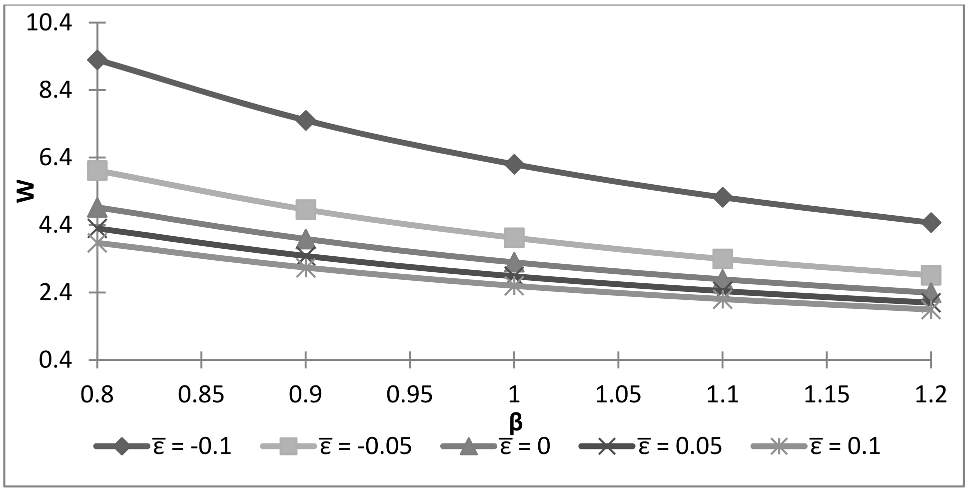


Figure 3. Variation of Load carrying capacity with respect to β and $\bar{\varepsilon}$

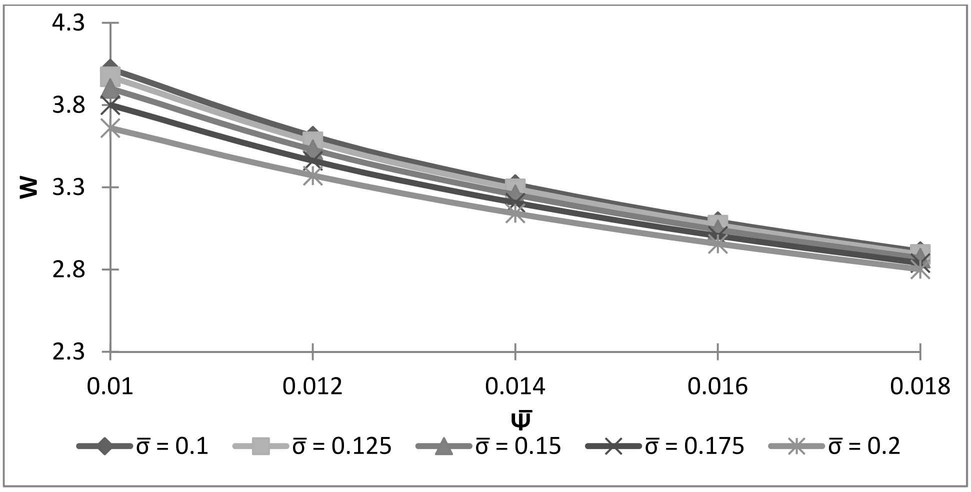


Figure 4. Variation of Load carrying capacity with respect to $\bar{\psi}$ and $\bar{\sigma}$.

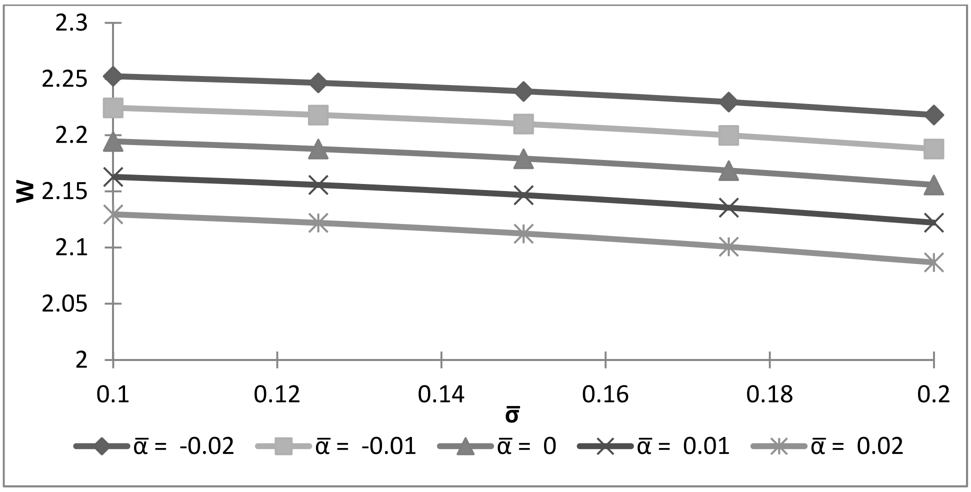


Figure 5. Variation of Load carrying capacity with respect to $\bar{\sigma}$ and $\bar{\alpha}$.

The variation of coefficient of friction presented in Figures 6 to 9 makes it clear that the coefficient of friction decreases for material constant parameter while it increases with respect to the increasing values of porosity and roughness parameters. Also this study suggests that the coefficient of friction decreases because of the use of ferrofluid lubrication.

Mostly the centre of pressure shifts towards the outlet edge as can be had from Figures 10 to 12. But, the effect of standard deviation on variation of centre of pressure with respect to porosity is almost negligible.

It is observed that the temperature rise decreases barring the case of variance (-ve) and negatively skewed roughness (Figures 13 to 15). However, the effect of skewness on the variation of temperature rise with respect to film shape parameter remains more or less negligible for smaller values of the film shape parameter.

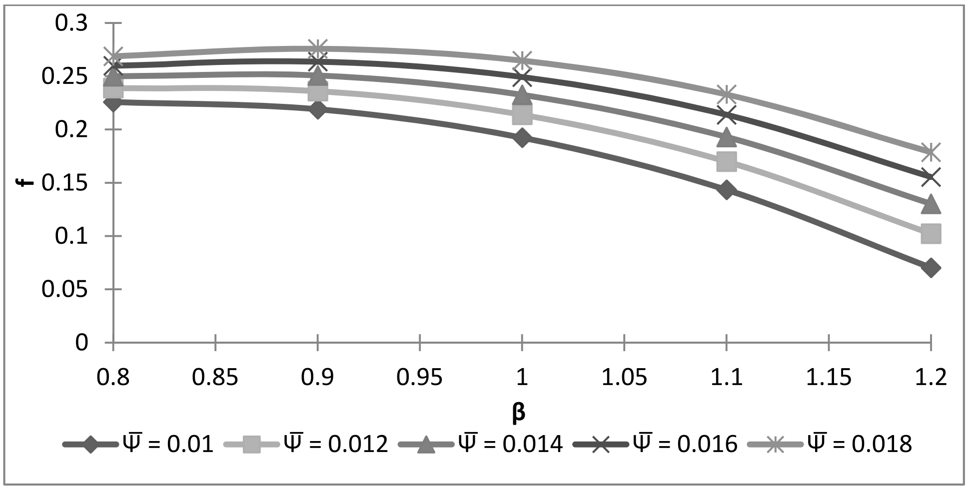
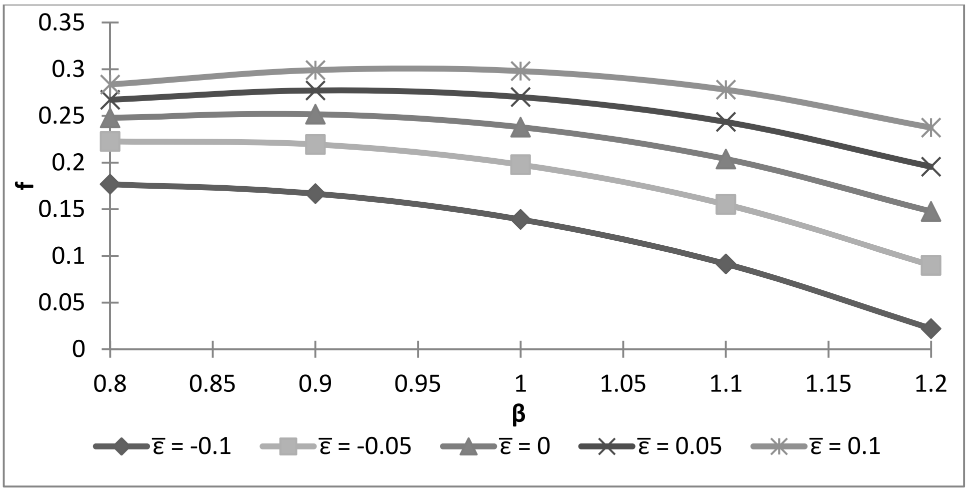


Figure 6. Variation of coefficient of friction with respect to β and $\bar{\psi}$.



Figue 7. Variation of coefficient of friction with respect to β and $\bar{\varepsilon}$.

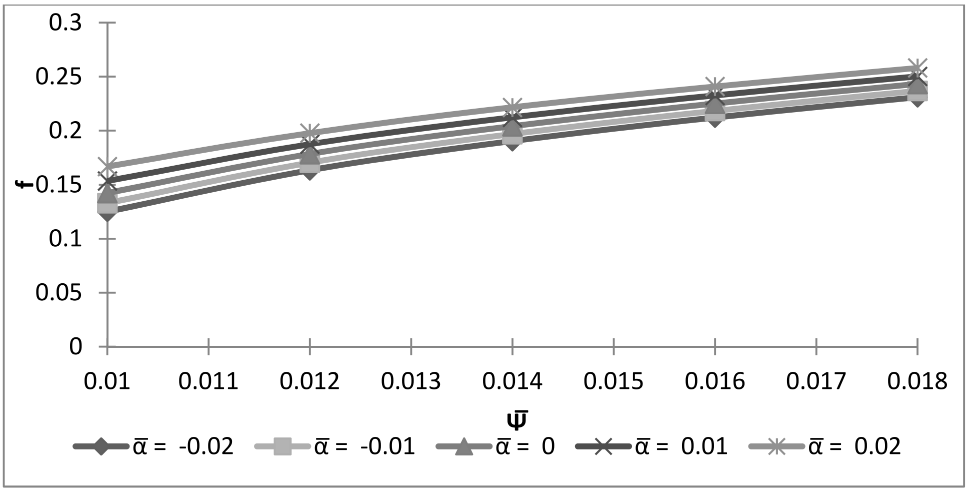


Figure 8. Variation of coefficient of friction with respect to $\bar{\psi}$ and $\bar{\alpha}$.

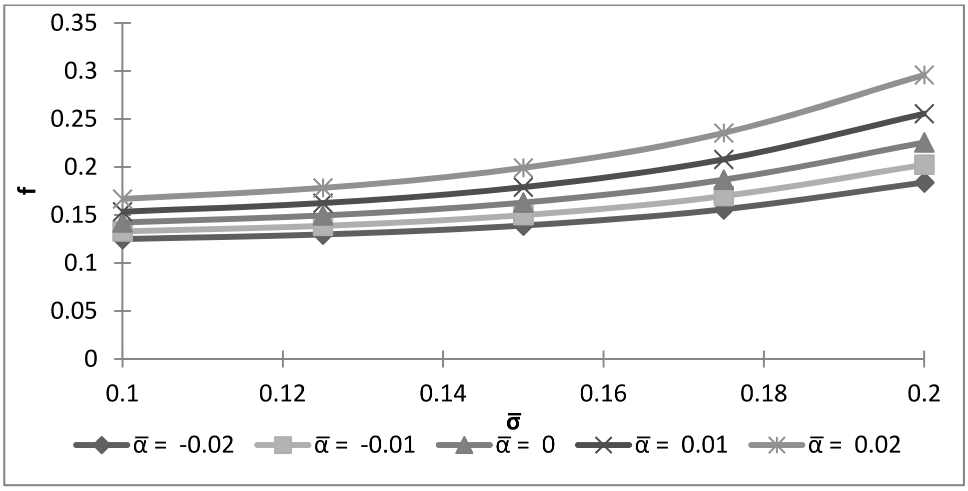


Figure 9. Variation of coefficient of friction with respect to $\bar{\sigma}$ and $\bar{\alpha}$.

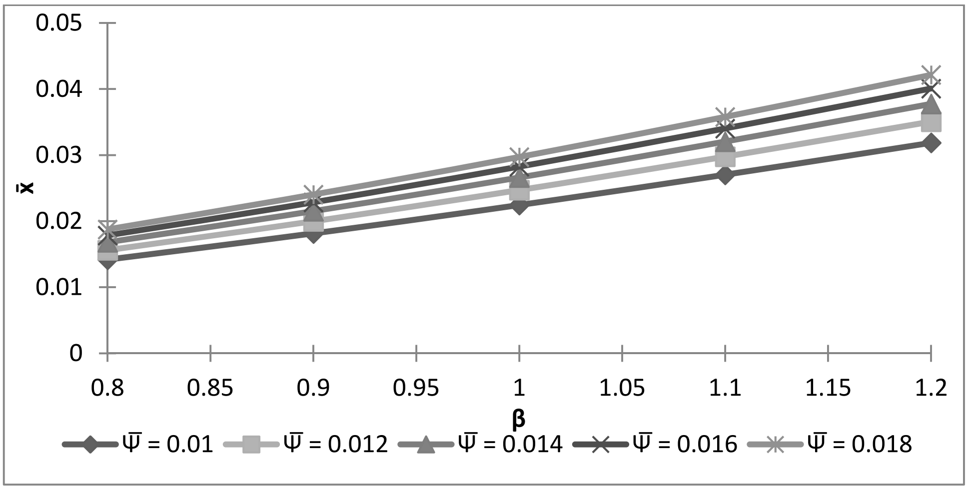


Figure 10. Variation of centre of pressure with respect to β and $\bar{\psi}$.

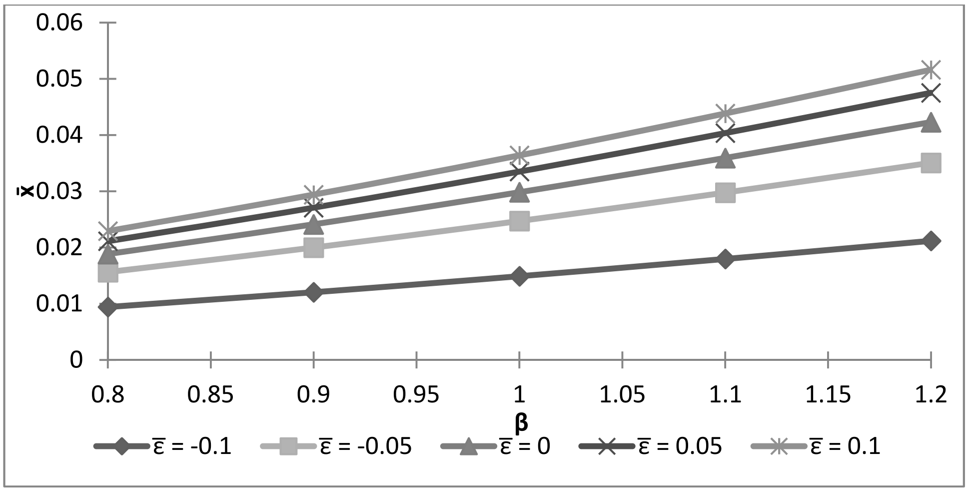


Figure 11. Variation of centre of pressure with respect to β and $\bar{\varepsilon}$.

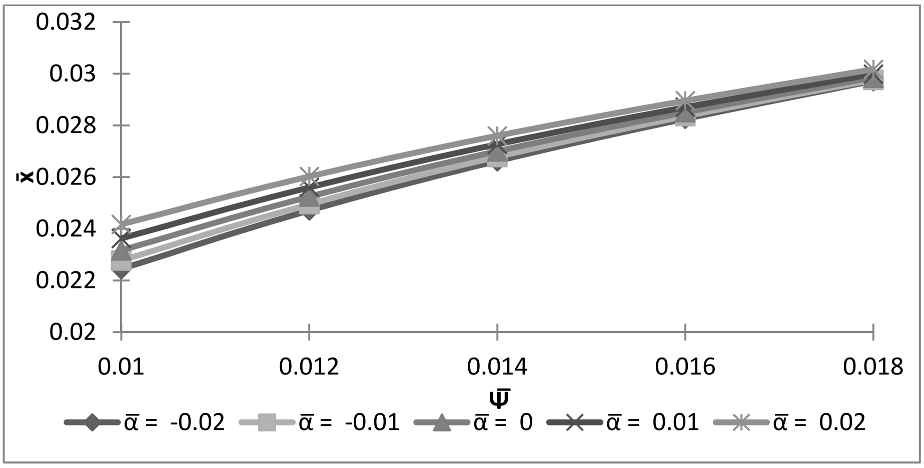


Figure 12. Variation of centre of pressure with respect to $\bar{\psi}$ and $\bar{\alpha}$.

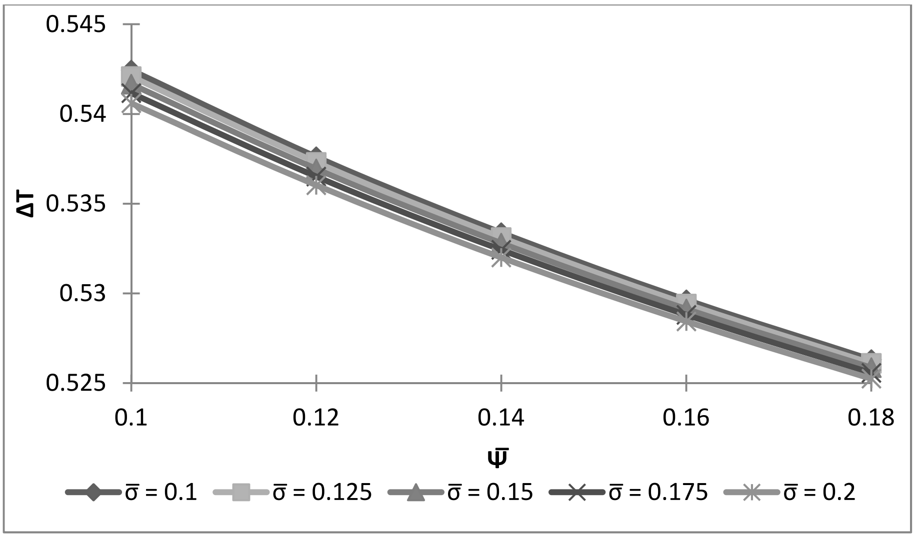


Figure 13. Variation of Temperature rise with respect to $\bar{\psi}$ and $\bar{\sigma}$.

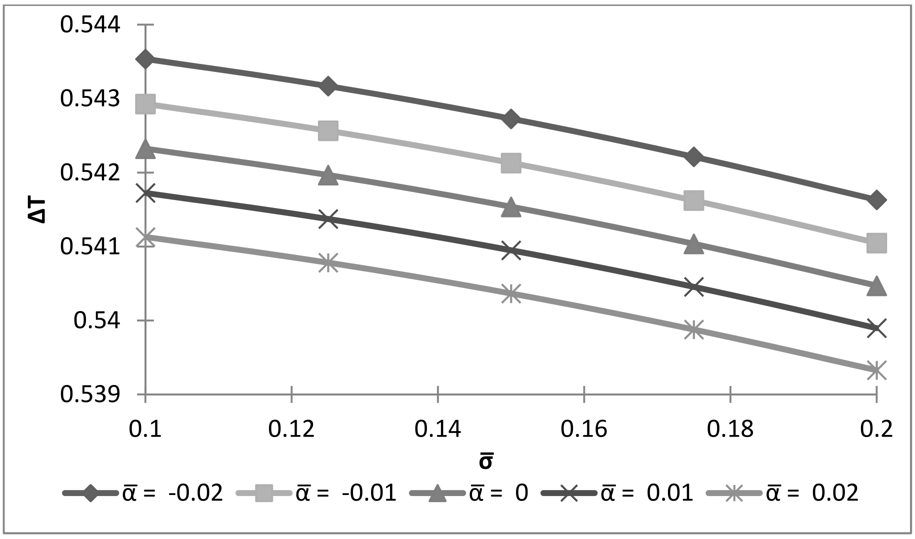


Figure 14. Variation of Temperature rise with respect to $\bar{\sigma}$ and $\bar{\alpha}$.

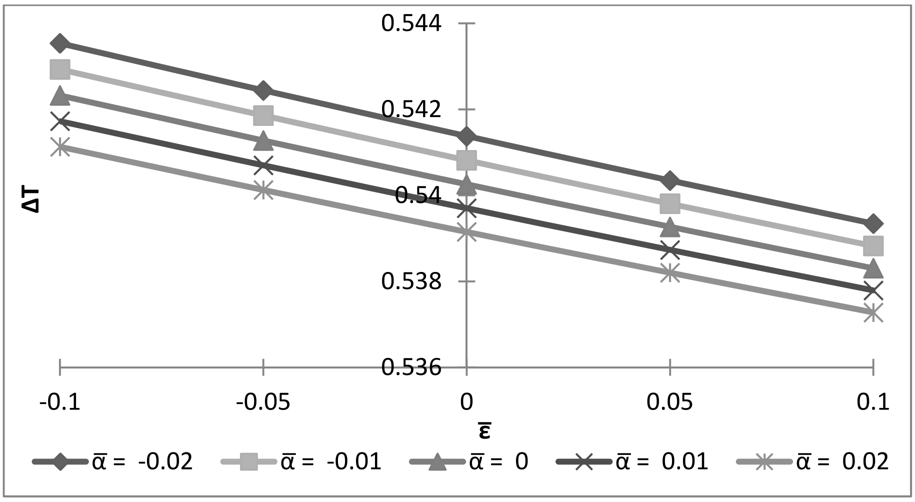


Figure 15. Variation of Temperature rise with respect to $\bar{\varepsilon}$ and $\bar{\alpha}$.

Some of the Figures reveal that the adverse effect of porosity and standard deviation can be reduced to a large extent by the positive effect of magnetization at least in the case of negatively skewed roughness. This compensation enhances because of the double layered particularly when variance (-ve) occurs. When variance (-ve) is in the place, friction coefficient is found to be significantly decreased and the rate of temperature rise is observed to be less.

4. Conclusion

This article reveals that the roughness aspects must be addressed carefully while designing the bearing system even if a double layer is considered and suitable magnetic strength is in force. The presence of double layer and magnetization prevent the load carrying capacity to fall rapidly. Needless to say that this type of bearing system sustains a good amount of load even in the absence of flow which never happens in the case of conventional fluid based bearing systems. For this type of bearing system the existence of double layer effectively modifies the performance .This investigation provides a better option from application point of view as the temperature rise fails to introduce an adverse effect on overall performance of the bearing system.

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