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RESEARCH

Ferrofluid Lubrication of a Rough Porous Hyperbolic Slider Bearing with Slip Velocity

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A B S T R A C T

An attempt has been made to study and analyze the combined effect of roughness and slip velocity on the performance of a ferrofluid based rough porous hyperbolic slider bearing. Beavers and Joseph's slip model has been used to evaluate the effect of slip velocity. The stochastic modeling of Christensen and Tonder has been employed to estimate the effect of transverse surface roughness. The associated Reynolds' type equation is solved with suitable boundary conditions to obtain the pressure distribution leading to the calculation of load carrying capacity. Further, the expressions for friction and position of centre of pressure are obtained. It is observed that the bearing system gets adversely affected by the transverse surface roughness. The slip velocity further enhances this adverse effect. However, the magnetization saves the situation in the case of negatively skewed roughness when variance (-ve) occurs. But for an overall improved performance of the bearing the lower values of slip parameter may be favorable.

1. INTRODUCTION

Fluids with strong magnetic properties have attracted considerable attentions during the last decade. The use of the magnetic fluid as a lubricant for the bearing system in technical applications in the domain of nano scale science and technology has made significant progress. Thus, the use of magnetic fluid lubrication adds an additional importance from nano science point of view. Magnetic fluid consists of colloidal magnetic nano particles dispersed with the aid of surfactants in a carrier liquid. In reality magnetic fluid is a hybrid of soft material and the nano particles. The average diameter of the dispersed particles ranges from 5 to 10 nm. The ferrofluids contain enormous magnetic nanoparticles and therefore, can be influenced by either parallel or perpendicular magnetic field.

The use of magnetic fluids has resulted in the development of many energy devices and instruments. Computer disks drives, semiconductors and high precision speakers are commercially available and based on magnetic fluids effects. The most important property of a magnetic fluid is that it can be retained at a desired location with the aid of a magnetic field. When a magnetic field is applied each and every particle experiences a body force causing it to flow.

Since the work of Hughes [1] on the hydromagnetic lubrication of inclined slider bearing with a transverse magnetic field, a good deal of work has been done on the performance of magnetohydrodynamic slider bearings. For instance, one can have a look at Bhat [2]. Sparrow et al. [3] studied the effect of velocity slip on porous-walled squeeze film. This paper proved that substantially faster response could be attained by the use of porous materials which accentuated velocity slip. Andharia et al. [4] studied the effects of longitudinal surface roughness on the performance of a hyperbolic slider bearing. It was shown that the longitudinal roughness turned in a better performance under a suitable range of skewness.

Andharia et al. [5] considered the effect of transverse surface roughness on the hydrodynamic lubrication of slider bearings with various film shapes. Here, for the hyperbolic slider bearing it was found that the combined effect of positively skewed roughness and standard deviation caused severe load reduction. Shah and Bhat [6] studied the performance of a porous exponential slider bearing with a ferrofluid lubricant, whose flow was governed by Jenkins model, considering slip velocity at the porous interface. The slip parameter decreased the load carrying capacity without significantly affecting the centre of pressure. Deheri et al. [7] embarked on the magnetic fluid lubrication of longitudinally rough slider bearings with squeeze film. Here also various film geometries were considered including that of hyperbolic slider bearing. This analysis established that the bearing performance was significantly affected by all the three roughness parameters but the magnetic fluid lubrication minimized the adverse effect of longitudinal roughness. Later on, Deheri et al. [8] developed and modified the analysis of above problem considering a magnetic fluid as the lubricant. The magnetic fluid lubrication not only increased the load carrying capacity but also diminished the adverse effect of transverse roughness up to considerable extent.

Bayrakçeken and Yürüsoy [9] presented a study on an infinitely wide lubricated slider bearing consisting of connected surfaces with third grade fluid as a lubricant. To illustrate the mathematical model, for the calculation of pressure distribution, inclined and parabolic slider bearing were adopted. Ochonski [10] presented some new designs of sliding bearings lubricated with magnetic fluids and explored the possibility of using them in modern bearing technology, in new computer and audiovisual equipments. Patel and Deheri [11] investigated the characteristics of lubrication at nano scale on the performance of a transversely rough slider bearing. This paper underlined the importance of lubrication at nano scale to improve the performance of the bearing system. Oladeinde and Edokpia [12] embarked on the performance modeling of infinitely wide exponential slider bearing lubricated with a couple stress fluid. It was found that the effect of couple stresses enhanced the load carrying capacity of the bearing. It was also noticed that compared to the Newtonian case, the pressure development in the clearance zone of the bearing was augmented.

Singh and Ahmad [13] analyzed the behavior of a porous inclined slider bearing lubricated with magnetic fluid considering thermal effects with slip velocity. The effect of slip velocity on load capacity and mean temperature field was graphically represented. Patel and Deheri [14] investigated the performance of a transversely rough porous parallel plate slider bearing under the presence of a magnetic fluid, considering the slip velocity. For a better performance it was established that the slip parameter deserved to be kept at minimum, even if suitable magnetic strength was in force. Shimpi and Deheri [15] discussed the effect of deformation in magnetic fluid based transversely rough short bearings. Here it was established that the effect of deformation could be curtailed by suitable magnetic strength particularly when negatively skewed roughness occurred. Shimpi and Deheri [16] evaluated the effect of deformation and surface roughness on the ferrofluid squeeze film in rotating curved porous circular plates. The ferrofluid lubrication had a very limited option in reducing the adverse effect of porosity, deformation and standard deviation even in the case of negatively skewed roughness. Singh et al. [17] presented a theoretical analysis to investigate the effect of non-Newtonian pseudoplastic and dilatant lubricants (lubricant blended with viscosity index improver)-Rabinowitsch fluid model on the dynamic stiffness and damping characteristics of pivoted curved slider bearings. It was concluded that load carrying capacity, stiffness, centre of pressure, dynamic stiffness and damping characteristics varied significantly with the non-Newtonian behavior of the fluid consistent with the real nature of the problem.

Patel and Deheri [18] analyzed the comparison of various porous structures on the performance of a magnetic fluid based transversely rough short bearing. It was found that the magnetization affected the bearing system positively while the bearing suffered owing to the transverse roughness. Hsu et al. [19] dealt with the lubrication performance of short journal bearings considering the effect of surface roughness and magnetic field. The effect of longitudinal roughness significantly modified the performance even for moderate values of magnetic strength. Myshkin and Grigoriev [20] presented a review of theoretical and practical problems in tribology involving concept of rough surface. Advantages and Disadvantages of traditional and modern approaches of surface analysis based on concepts of roughness and texture were discussed. Rao et. al. [21] obtained a generalized form of Reynolds' equation for two symmetrical surfaces by considering the effects of velocity slip and viscosity variation for squeeze film lubrication of two circular plates. The load carrying capacity and squeezing time decreased due to slip. Ghalme et al. [22] investigated the effect of surface roughness and lubricant viscosity on coefficient of friction in silicon nitride-steel rolling contact. The analysis of experimental results using taguchi technique presented a strong interaction between surface roughness and lubricant viscosity on friction in rolling contact. Recently, Siddangouda et al. [23] analyzed the combined effects of micro polarity and surface roughness on the hydrodynamic lubrication of slider bearings with different film shapes including hyperbolic slider bearing. Here, the negatively skewed roughness increased the load carrying capacity and temperature.

Here, it has been proposed to discuss the effect of slip velocity and roughness on the magnetic fluid lubrication of a squeeze film in hyperbolic slider bearing.

2. ANALYSIS

The configuration of the bearing system is presented below (Fig. 1.).

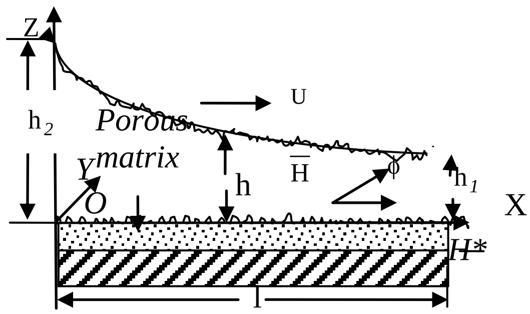

Fig. 1. Porous hyperbolic slider bearing.

The bearing system consists of a slider in hyperbolic form moving with a uniform velocity U in the X-direction and a stator with a porous matrix of thickness H^*.

Here l is the length and b is the breath of the bearing with $l << b$. In view of the discussions of Cameron [24] the film thickness h is defined as

$$h = \frac{h_2}{1+\left(\frac{x\ln a}{l}\right)}, 0 \le x \le l \tag{1}$$

where $a = \frac{h_2}{h_1}$;

h_2 is the maximum value of h and h_1 is the minimum value of h.

Following the investigations of Agrawal [25] the magnitude H of the magnetic field $\overline{H}$ is taken as

$$H^2 = Kx(l-x) \tag{2}$$

K being a quantity chosen to suit the dimensions so as to manufacture a magnetic field of required strength. The inclination angle of the magnetic field is determined as in Agrawal [25]. The study carried out in Verma [26] indicates that the equation of fluid flow in the film region is:

$$\frac{\partial^2 u}{\partial z^2} = \frac{1}{\mu}\frac{\partial}{\partial x}\left(p - \frac{\mu_0 \overline{\mu} H^2}{2}\right) \tag{3}$$

where u is the x-component of the fluid velocity, μ is the fluid viscosity, p is the film pressure, μ_0 is the permeability of free space and $\bar{\mu}$ is the magnetic susceptibility.

Sparrow et al. [3] mooted the following boundary conditions,

$$u = U$$

when

$$z = h, u = \left(\frac{1}{s}\frac{\partial u}{\partial z}\right)_{z=0} \tag{4}$$

where *s* is the slip constant.

Solving equation (3) under the boundary conditions (4), substituting the value of u in the integral form of the continuity equation for the film region, using continuity of velocity components of the fluid in the film region and porous matrix across the surface $z=0$, one arrives at the Reynolds type equation governing the film pressure:

$$\frac{d}{dx}\left[\left\{12kH^* + \frac{h^3(4+sh)}{(1+sh)}\right\}\frac{d}{dx}\left(p - \frac{\mu_0\bar{\mu}H^2}{2}\right)\right] = 6\mu U\frac{d}{dx}\left[\frac{h(2+sh)}{1+sh}\right] \tag{5}$$

where k is the permeability of the porous material; in view of the method of Agrawal [25].

The random roughness of the bearing surfaces is characterized by a random variable with non zero mean, variance and skewness. In line with the discussions of Christensen and Tonder [27-29] the thickness $h(x)$ of the lubricant film is taken as:

$$h(x) = \bar{h}(x) + h_s$$

where $\bar{h}(x)$ 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 considered to be stochastic in nature and governed by the probability density function:

$$f(h_s) = \begin{cases} \frac{35}{32c}\left(1 - \frac{h_s^2}{c^2}\right)^3, -c \le h_s \le c \\ 0, \text{elsewhere} \end{cases}$$

Where in 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 and treated in Christensen and Tonder [27-29].

Now, resorting to the stochastic averaging of Christensen and Tonder one gets the following equation from equation (5):

$$\frac{d}{dx}\left[\left\{12kH^* + \frac{g(h)(4+sh)}{(1+sh)}\right\}\frac{d}{dx}\left(p - \frac{\mu_0\bar{\mu}H^2}{2}\right)\right] = 6\mu U\frac{d}{dx}\left[\frac{(g(h))^{\frac{1}{3}}(2+sh)}{1+sh}\right] \tag{6}$$

where

$$g(h) = h^3 + 3\alpha h^2 + 3(\sigma^2 + \alpha^2)h + \alpha^3 + 3\sigma^2\alpha + \varepsilon$$

Making use of equations (1) and (2) and dimensionless quantities:

$$X = \frac{x}{l}, \psi = \frac{kH^*}{h_1^3}, \bar{h} = \frac{h}{h_1}, \bar{s} = sh_1, P = \frac{h_1^2 p}{\mu U l},$$

$$\mu^* = \frac{\mu_0\bar{\mu}klh_1^2}{\mu U}, \bar{\alpha} = \frac{\alpha}{h_1}, \bar{\sigma} = \frac{\sigma}{h_1}, \bar{\varepsilon} = \frac{\varepsilon}{h_1^3}$$

equation (6) transforms to:

$$\frac{d}{dX}\left[G\frac{d}{dX}\left\{P - \frac{1}{2}\mu^* X(1-X)\right\}\right] = \frac{dE}{dX} \tag{7}$$

where

$$\bar{h} = \frac{a}{1 + X\ln a}, G = 12\psi + \frac{g(\bar{h})(4 + \bar{s}\bar{h})}{(1 + \bar{s}\bar{h})},$$

$$E = \frac{6(g(\bar{h}))^{\frac{1}{3}}(2 + \bar{s}\bar{h})}{1 + \bar{s}\bar{h}} \tag{8}$$

where in

$$g(\bar{h}) = \bar{h}^3 + 3\bar{\alpha}\bar{h}^2 + 3\left(\bar{\sigma}^2 + \bar{\alpha}^2\right)\bar{h} + \bar{\alpha}^3 + 3\bar{\sigma}^2\bar{\alpha} + \bar{\varepsilon}$$

Solving equation (7) under the boundary conditions:

$$P = 0, \text{ when } X = 0,1 \tag{9}$$

one obtains the expression for non dimensional pressure distribution as:

$$P = \frac{1}{2}\mu^* X(1-X) + \int_1^x \frac{E-Q}{G}dX \tag{10}$$

where

$$Q = \frac{\int_0^1 \frac{E}{G} dX}{\int_0^1 \frac{1}{G} dX}$$

Then, the non dimensional form of the load carrying capacity w is expressed as:

$$W = \frac{h_1^2 w}{\mu U l^2 B} = \frac{\mu^*}{12} + \int_0^1 X \frac{Q-E}{G} dX \quad (11)$$

The dimensionless friction force F on the slider and the coefficient of friction f are derived from:

$$\bar{F} = \frac{h_1 F}{\mu U l B} = \int_0^1 \left[\frac{\bar{s}}{(1+\bar{s}\bar{h})} + \frac{(g(\bar{h}))^{\frac{1}{3}}(2+\bar{s}\bar{h})(E-Q)}{2G(1+\bar{s}\bar{h})} \right] dX \quad (12)$$

and

$$\bar{f} = \frac{lf}{h_2} = \frac{\bar{F}}{W} \quad (13)$$

respectively.

Lastly, the dimensionless X- coordinate $\bar{Y}$ of centre of pressure $\bar{X}$ can be expressed as

$$\bar{Y} = \frac{\bar{X}}{l} = \frac{1}{W}\left[\frac{\mu^*}{24} - \frac{1}{2}\int_0^1 X^2 \frac{E-Q}{G} dX \right] \quad (14)$$

3. RESULTS AND DISCUSSIONS

It is clearly seen from equation (11) that the non dimensional load carrying capacity enhances by

$$\frac{\mu^*}{12}$$

as compared to the case of conventional lubricant based bearing system. Further, the equation (11) suggests that the expression involved is linear with respect to the magnetization parameter. Accordingly, the load carrying capacity would increase with increasing magnetization. This may be due to the fact that the magnetization increases the viscosity of the lubricant, which results in increased pressure and consequently, the load carrying capacity. The fact that the load carrying capacity increases sharply with increasing magnetization can be seen from Figs. 2- 4.

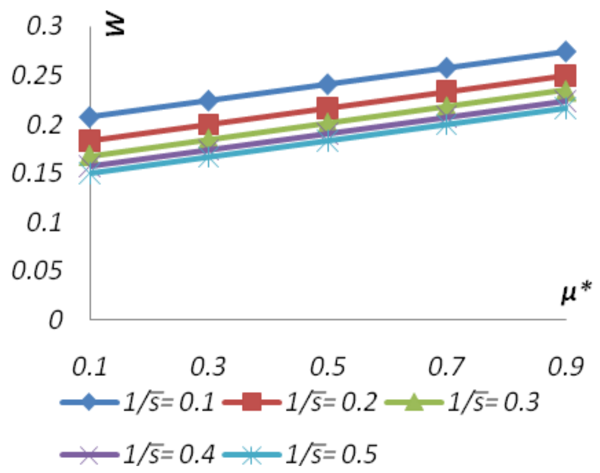

Fig. 2. Variation of Load Carrying capacity with respect to μ^* and $\frac{1}{\bar{s}}$

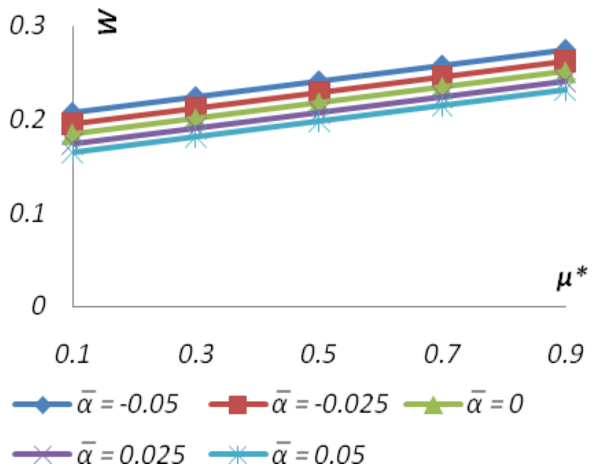

Fig. 3. Variation of Load Carrying capacity with respect to μ^* and $\bar{\alpha}$.

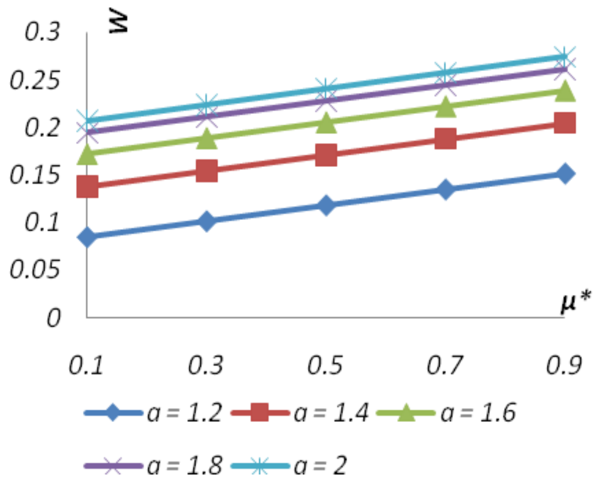

Fig. 4. Variation of Load Carrying capacity with respect to μ^* and a.

Further, the slip parameter causes reduced load carrying capacity as can be obtained from Figs. 5- 8.

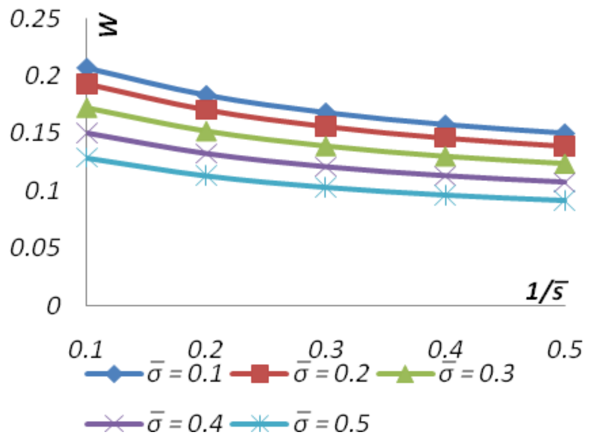

Fig. 5. Variation of Load Carrying capacity with respect to $\frac{1}{\bar{s}}$ and $\bar{\sigma}$.

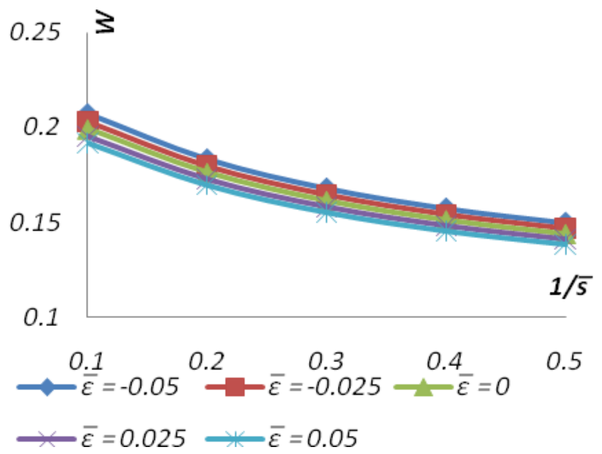

Fig. 6. Variation of Load Carrying capacity with respect to $\frac{1}{\bar{s}}$ and $\bar{\varepsilon}$.

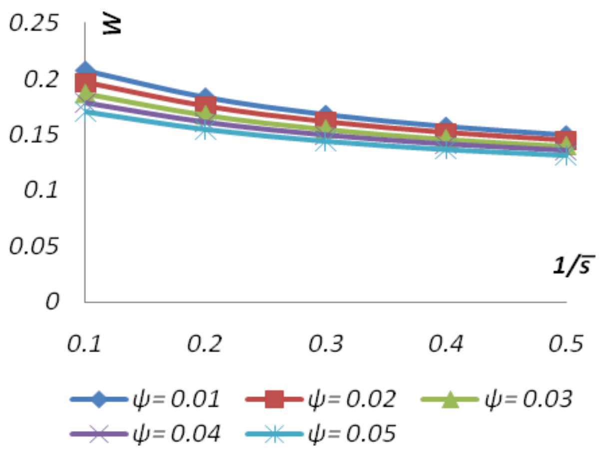

Fig. 7. Variation of Load Carrying capacity with respect to $\frac{1}{\bar{s}}$ and ψ.

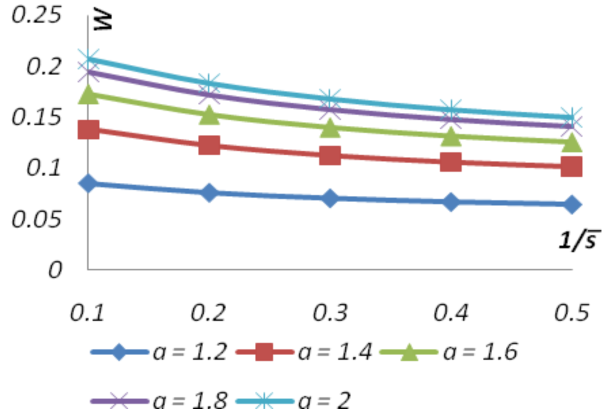

Fig. 8. Variation of Load Carrying capacity with respect to $\frac{1}{\bar{s}}$ and a.

The effect of standard deviation presented in Figs. 9-11 makes it clear that the load carrying capacity reduces when the standard deviation gets increased. Thus, the combined effect of slip parameter and the standard deviation associated with roughness turns in an adverse performance.

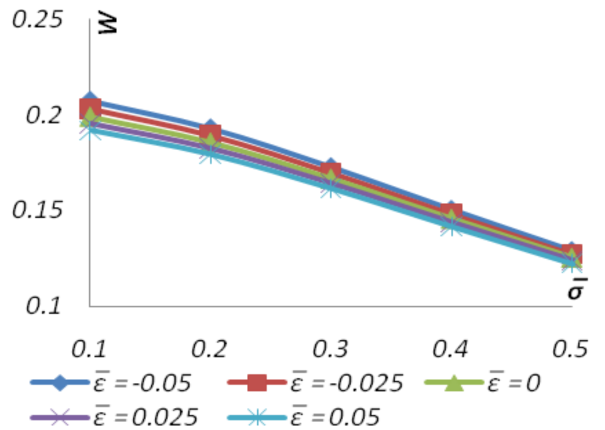

Fig. 9. Variation of Load Carrying capacity with respect to $\bar{\sigma}$ and $\bar{\varepsilon}$.

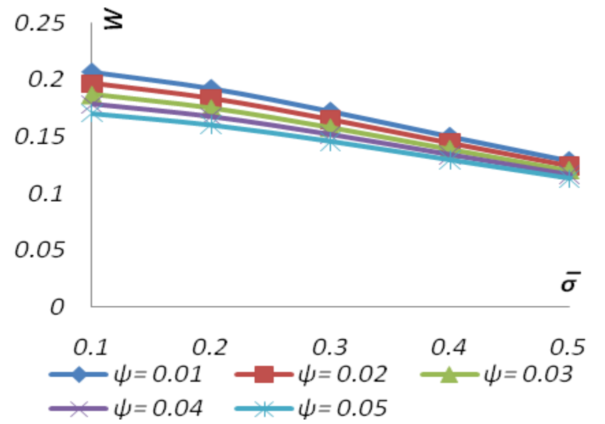

Fig. 10. Variation of Load Carrying capacity with respect to $\bar{\sigma}$ and ψ.

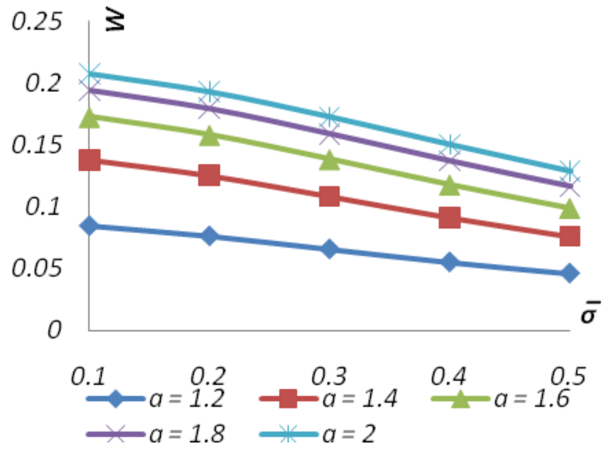

Fig. 11. Variation of Load Carrying capacity with respect to $\bar{\sigma}$ and a .

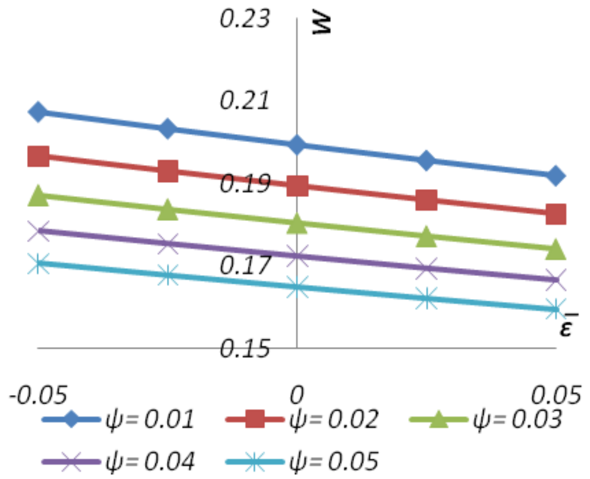

Fig. 12. Variation of Load Carrying capacity with respect to $\bar{\varepsilon}$ and ψ .

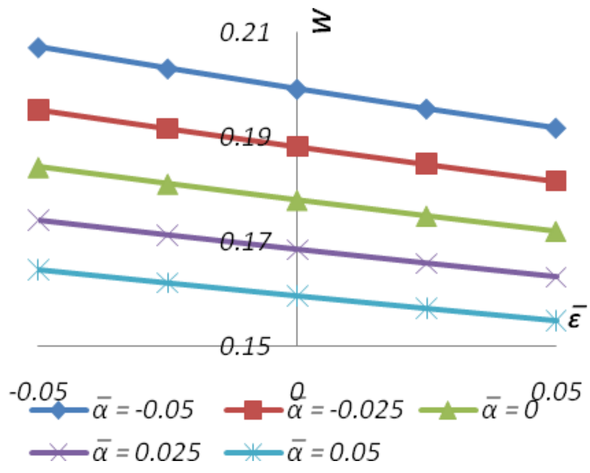

Fig. 13. Variation of Load Carrying capacity with respect to $\bar{\varepsilon}$ and $\bar{\alpha}$.

The effect of skewness and variance on the load carrying capacity is shown in Figs. 12-15. The positive skewness decreases the load carrying capacity, while the load carrying capacity gets increased due to negatively skewed roughness. The variance follows the path of skewness so far as the trends of load carrying capacity are concerned. Of course, the porosity causes reduction in load carrying capacity.

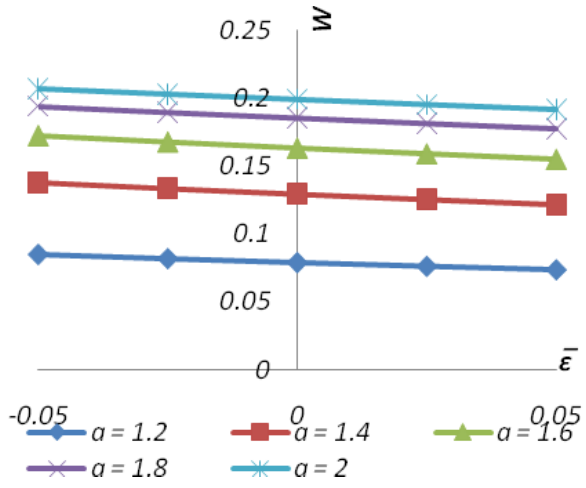

Fig. 14. Variation of Load Carrying capacity with respect to $\bar{\varepsilon}$ and a .

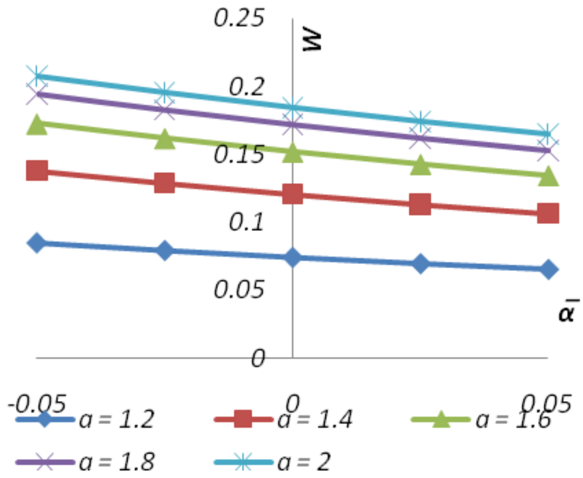

Fig. 15. Variation of Load Carrying capacity with respect to $\bar{\alpha}$ and a .

The profile of the variation of friction is presented in Fig. 16 - 21. The combined effect of porosity, slip and standard deviation reduces the friction heavily. Even the magnetization reduces the coefficient of friction. However, the variance and skewness increase the friction. Besides, the effect of skewness on the variation of friction with respect to standard deviation is not that significant for lower to moderate values of skewness. It is interesting to note that the friction increases with respect to the slip parameter up to the value of slip parameter 0.15 and after words it decreases.

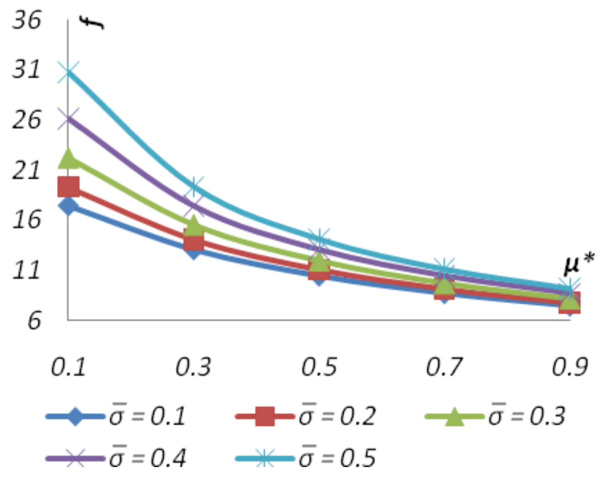

Fig. 16. Variation of friction with respect to μ^* and $\bar{\sigma}$.

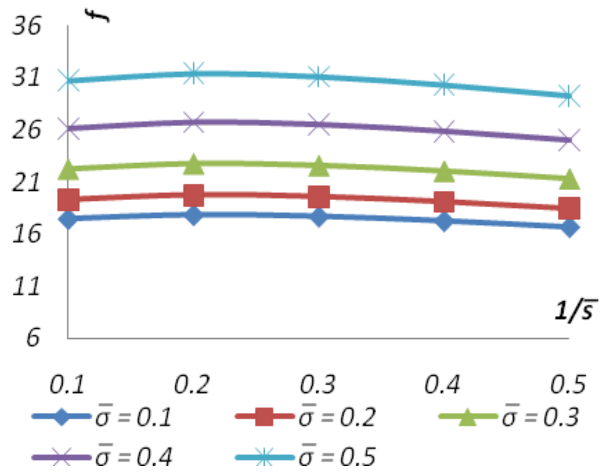

Fig. 17. Variation of friction with respect to $\frac{1}{\bar{s}}$ and $\bar{\sigma}$.

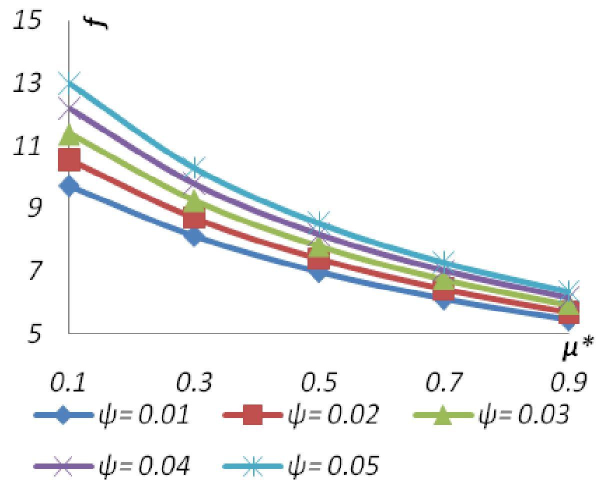

Fig. 18. Variation of friction with respect to μ^* and ψ.

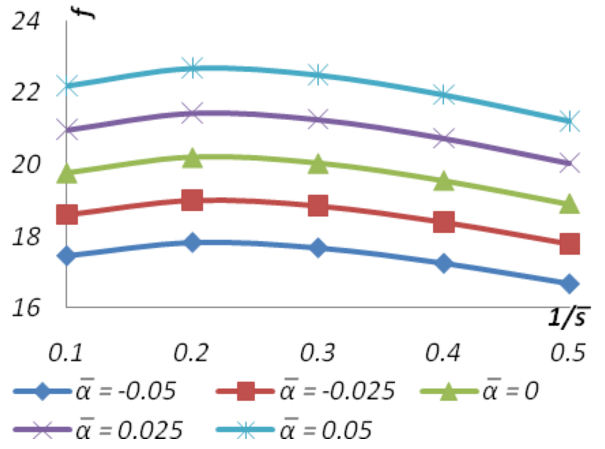

Fig. 19. Variation of friction with respect to $\frac{1}{\bar{s}}$ and $\bar{\alpha}$.

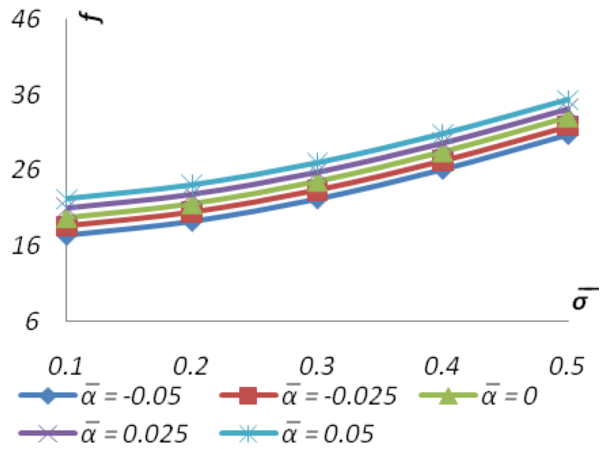

Fig. 20. Variation of friction with respect to $\bar{\sigma}$ and $\bar{\alpha}$.

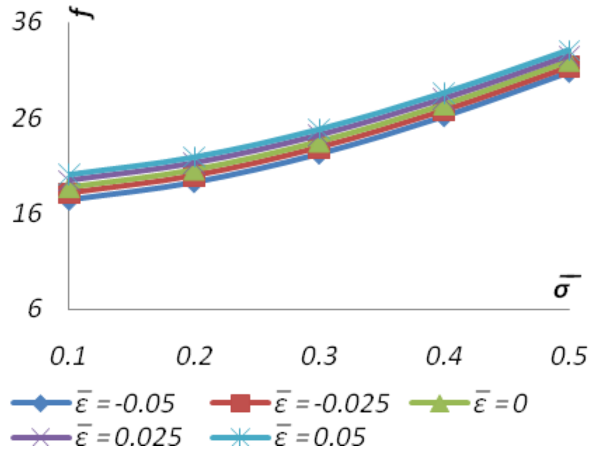

Fig. 21. Variation of friction with respect to $\bar{\sigma}$ and $\bar{\varepsilon}$.

Lastly, the variation of centre of pressure is presented in Figs. 22-25. Mostly, the centre of pressure shifts towards the out let edge. It is clearly seen that the effect of skewness on the change of position of centre of pressure is at the most nominal. Although, the effect of transverse roughness is adverse in general, the situation can be made better by suitably choosing the magnetic strength, at least in the case of negatively skewed roughness. Further, the negative effect of porosity and standard deviation can be compensated up to some extent by the magnetic fluid lubrication, when variance (-ve) occurs.

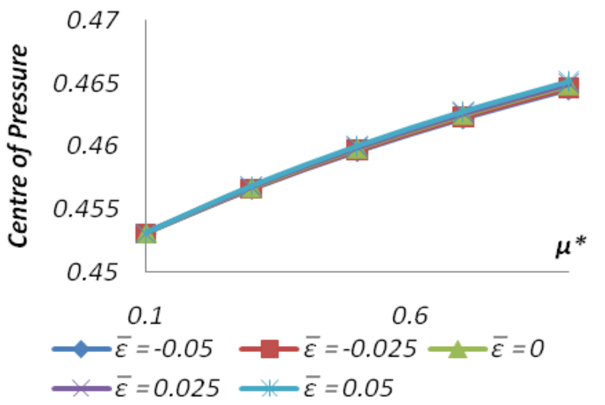

Fig. 22. Variation of centre of pressure with respect to μ^* and $\bar{\varepsilon}$.

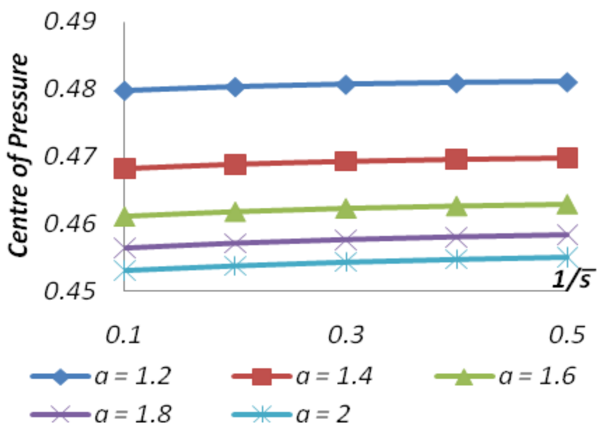

Fig. 23. Variation of centre of pressure with respect to $\frac{1}{\bar{s}}$ and a.

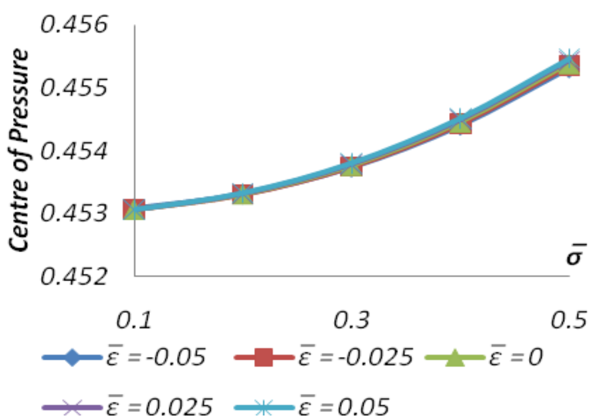

Fig. 24. Variation of centre of pressure with respect to $\bar{\sigma}$ and $\bar{\varepsilon}$.

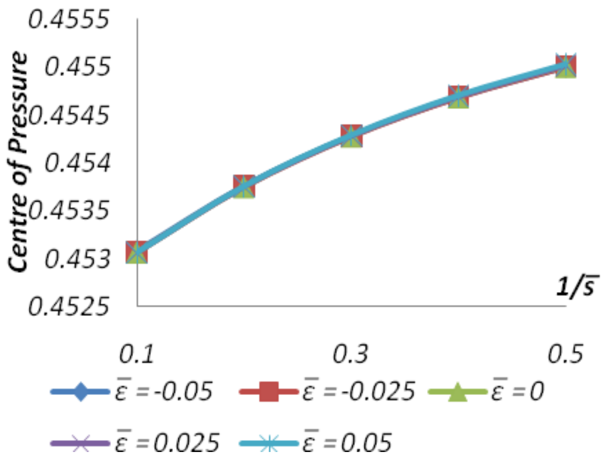

Fig. 25. Variation of centre of pressure with respect to $\frac{1}{\bar{s}}$ and $\bar{\varepsilon}$.

4. CONCLUSION

This article offers the suggestion that for an overall improved performance of the bearing system the slip parameter is required to be kept at minimum. Hence, this study indicates that while designing the bearing system, one is required to consider the roughness aspect carefully. In spite of the fact that there are several parameters causing load reduction, this type of bearing system supports a good amount of load even in the absence of flow, which fails to happen in the case of conventional lubricant, based bearing system.

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