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Patel, N.D.
Deheri, G.M.

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SHLIOMIS MODEL BASED FERROFLUID LUBRICATION OF A ROUGH, POROUS PARALLEL SLIDER BEARING WITH SLIP VELOCITY

N.D. PATEL
Department of Mathematics, Sardar Patel University, V.V.Nagar-388120, Anand, Gujarat, India
G.M. DEHERI
Department of Mathematics, Sardar Patel University, V.V.Nagar-388120, Anand, Gujarat, India

Corresponding Author E-mail: ndpatel2002@gmail.com, Phone: +91-9638553619

Abstract: *This study aims to analyze the performance of a transversely rough porous parallel slider bearing under the presence of a magnetic fluid lubricant flowing as par Shliomis model, taking velocity slip in to account. The associated stochastically averaged Reynolds equation is solved to get the pressure distribution thereby leading to the calculation of load carrying capacity. Further, the friction is computed. The results show that magnetization not only increases the load carrying capacity but also increases the friction. But the slip parameter decreases the load carrying capacity as well as the friction. Besides, it is noticed that the standard deviation and the positively skewed roughness decrease the friction. It becomes clear from this investigation that the magnetization can minimize the adverse effect of roughness up to some extent when relatively small values of the slip parameter are involved. A comparison of this investigation with the Neuringer-Rosensweig model indicates that the reduction of the adverse effect of roughness is comparatively more here. This article makes it clear that the bearing can support a load even where there is no flow.*

Key-words: *Parallel Plate Slider bearing; Ferro Fluid; Roughness; Load Carrying Capacity; Porosity; Pressure.*

1. Introduction

Shukla and Kumar (1987) derived a generalized form of Reynolds equation governing the pressure in a thin film of a ferrofluid in presence of a transversely applied magneticfield by considering the rotation of particles. This equation was used to analyze the effects of applied magnetic field and the Brownian relaxation time parameter on the bearing characteristics of an infinitely long squeeze film slider bearing. Magnetic fluids are a fascinating product of colloidal sciences, a vital field of colloidal research and of investigations in other branches of science. Odenbach (1993) over viewed the research on magnetic fluids performed under micro gravity conditions and those problems leading to a close contact between different researches activities in the field of magnetic fluids. Patrick et al. (2005) investigated the anisotropy of the magneto viscous effect in ferrofluids by extensive molecular simulations. Ram et al. (2010) theoretically investigated a ferrofluid flow over a moving plate in a porous medium solving the bound layer equations with boundary conditions using Neuringer-Rosensweig model. Das (1998) conducted a theoretical study of slider bearings considering the lubricant to be an isothermal, incompressible electrically conducting couple stress fluid in the presence of a magnetic field. Further, a comparative study of optimum load carrying capacity for finite and infinite slider bearing was made. It was observed that both the values of maximum load capacity and the corresponding inlet-outlet film ratio depended on couple stress and magnetic parameters and the shape of bearings conjointly. Ramanaiah and Sarkar (1979) presented a theoretical analysis for infinitely long slider bearings assuming the lubricant to be an incompressible fluid with couple stress. Here it was shown that the load carrying capacity and the frictional force increased as the couple stress parameter increased. Gupta and Bhat (1979) studied the performance of a hydromagnetic inclined porous slider bearing with a transverse magnetic field. Here, it was concluded that the load carrying capacity and friction increased with increasing Hartmann number. Malik and Singh (1980) derived a generalized Reynolds equation for a magnetohydrodynamic journal bearing with the magnetic field perpendicular to the bearing axis and the electric current in the axial direction. Osterle and Young (1962) investigated the load carrying capacity of liquid metal lubricated slider bearings subject to an applied magnetic field transverse to the film. The optimum profile was found to be the Rayleigh step form with the riser location and step height ratio dependent on the strength of the magnetic field. Substantial load increases could be accomplished by using superconducting electromagnets.

(Ferro fluids also known as magnetic nano fluids) constitute a special category of nano materials exhibiting simultaneously liquid and super paramagnetic properties. One of the most important properties of ferrofluids / magneticfluids is the possibility to significantly influence their flow behavior by moderate magnetic fields. Thurm and Odenbach (2003) theoretically

calculated that the interaction between larger particles is strong enough to form chain-like structures, which in turn, dominate the flow behavior. Ferrofluids / magneticfluids are stable colloidal suspensions of ferromagnetic particles of about 10nm diameter in a carrier liquid. The particles are stabilized against agglomeration by coating with polymers or the electrostatic repulsion of charges upon their surface. Ferrofluids / magneticfluids exhibit no net magnetization in the absence of an external field, but they exhibit relatively large saturation magnetizations in the presence of moderate applied fields Rosensweig (1985) and Odenbach (2000).

Abramovitz (1955) analyzed the effect of pad-surface curvature on load carrying capacity, centre of pressure and fluid friction. The results established the practical use of pad curvature for centrally pivoted pads where the variation in fluid viscosity from pad inlet to outlet is negligible. Various theoretical investigations were launched into using a ferrofluid as lubricant due to its various advantages such as long life, silent operation reduced wear. Agrawal (1986) studied the performance of an inclined plane slider bearing with a ferrofluid lubricant and found that its performance was comparatively better than the corresponding bearing system with a convential lubricant. The study of Bhat and Patel (1991) regarding the performance of an exponential slider bearing with a ferrofluid lubricant concluded that the magnetic fluid lubrication caused increased load carrying capacity without altering the friction on the slider. The analyses of Bhat and Deheri (1991a) and Shah et al. (2002) suggested the positive effect of magnetic fluid lubrication over the conventional ones. Bhat and Deheri (1991b) observed the performance of a porous composite slider bearing under the presence of a magnetic fluid lubricant. Here also the magnetic fluid lubricant enhanced the performance of the bearing system.

However, the above investigations resorted to the Neuringer-Rosensweig model for a ferrofluid flow. But unlike the Shliomis (1972) which were used by Siha et al. (1993) to study the ferrofluid lubrication of cylindrical rollers the Neuringer-Rosensweig model fails to contribute to the flow field when a constant magnetic field is used. Shah and Bhat (2004) dealt with a ferrofluid based squeeze film in a long journal bearing using the Shliomis, Jenkins and Neuringer-Rosensweig model of flow. Most of the above studies assumed that there was no slip at the interface of the lubricant field and the porous matrix. Beavers and Joseph (1967) proved that such an assumption might not be true at the nominal boundary of a naturally permeable material.

Sparrow (1972) presented at simplified boundary condition for the above case. All the above contributions considered the bearing surfaces to be smooth. However, it is an established fact that the bearing surfaces tend to develop roughness after receiving some run-in and wear. In fact, due to elastic thermal and uneven wear effects the configurations encountered in practice are usually far from being smooth. By now, it is known that the roughness of the bearing surfaces adversely affects the bearing system as it retards the motion of the lubricant. Tzeng

and Saibel (1967) recognized the random character of the surface roughness and developed a stochastic approach to study the effect of surface roughness. This modeling of Tzeng and Saibel (1967) was modified by Christensen and Tonder (1969a, 1969b, 1970) to analyzed the effect of transverse surface roughness on the bearings' performance subsequently several contributors Deheri et al. (2004, 2005) and Nanduvinamani et al. (2003) made use of this stochastic mode of Christensen and Tonder to study the effect of roughness. Recently, Patel and Deheri (2011) analyzed the ferrofluid lubrication of a plane inclined rough slider bearing with velocity slip. This type of bearing systems is found to be used in super- finish treatment of ceramics and magnetic sealing of food processing machines.

The flow of ferrofluid was governed by Shliomis model. It was shown that the magnetization could minimize the adverse effect of roughness up to certain extent taking suitable values of slip parameter. Here, it has been proposed to study the Shliomis model based, hydromagnetic lubrication of a rough, porous parallel slider bearing considering slip velocity.

2. Analysis
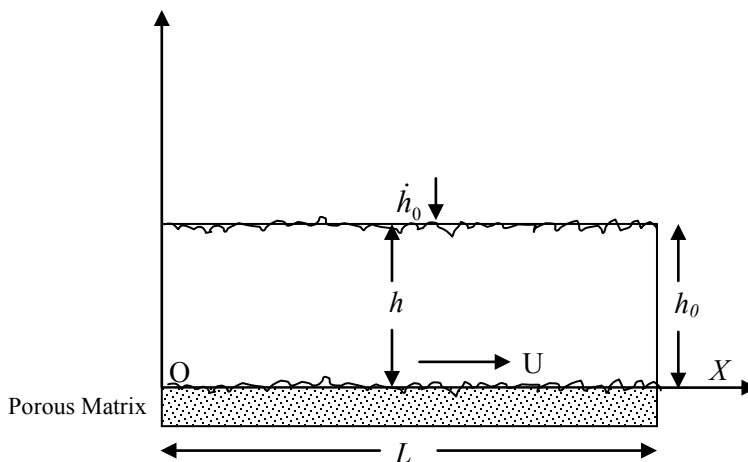
Figure 1 Parallel Plate Slider Bearing

The bearing surfaces are assumed to be the transversely rough. Following the stochastical modeling of Christensen and Tonder (1969a, 1969b, 1970) the thickness $h(\chi)$ of the lubricant film is considered 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 assumed to be stochastic in nature and governed by the probability density function $f(h_s)$, $-c \le h_s \le c$, where c is the maximum deviation from the mean film thickness. The mean α standard deviation σ and skewness ε, which is the measure of symmetry of the random variable h_s, are determined by the relationships,

$$\acute{a} = E(h_s)$$

$$\acute{o}^2 = E\left[\left(h_s - \acute{a}\right)^2\right]$$

$$\mathring{a} = E\left[h_s - \acute{a}^3\right]$$

where E is the expectancy operator given by

$$E(R) = \int_{-c}^{c} R\, f\left(h_s\right) dh_s$$

while the probability density function is represented as

$$f\left(h_s\right) = \begin{cases} \dfrac{35}{32}\left(c^2 - h_s^2\right)^3 \Big/ c^7, & -c \le h_s \le c\ ; \\ 0 & otherwise. \end{cases}$$

Shliomis (1972) reported that magnetic particles of a magnetic fluid can relax in two ways when the applied magnetic field changes. The first is by the rotation of magnetic particles in the fluid and the second by rotation of the magnetic moment with in the particles. The particle rotation is given by Brownian relaxation time parameter τ_B while the intrinsic rotational process is given by relaxation time parameter τ_S. For a detailed discussion of this two parameters τ_B and τ_S one can have a glance at Shliomis (1972) and Bhat (2003). The lower surface moves with uniform velocity (U, V, 0) and the upper with. Neglecting inertia and second derivative of the internal angular momentum, following the discussions of Bhat (2003) regarding the Shliomis model for a two dimensional flow, the Reynolds type equation is obtained in the form of

$$\frac{\partial}{\partial x}\left\{\left(h^3 + 2\,\phi H\right)\frac{\partial p}{\partial x}\right\} + \frac{\partial}{\partial y}\left\{\left(h^3 + 2\,\phi H\right)\frac{\partial p}{\partial y}\right\} = 2\,\eta_a \dot{h}_0 + 6\eta_a\left[\frac{\partial}{\partial x}(bU) + \frac{\partial}{\partial y}(bV)\right]$$

$$-\frac{N\tau_B{}^3}{6\,\eta_a}\left[\begin{array}{l}\dfrac{\partial}{\partial x}\left\{h\left(3U^2\,\dfrac{\partial p}{\partial x} + 2U\,\dfrac{\partial p}{\partial y}V + V^2\,\dfrac{\partial p}{\partial x}\right)\right\}\\[2mm]+\dfrac{\partial}{\partial y}\left\{h\left(2U\,V\dfrac{\partial p}{\partial x} + 3V^2\,\dfrac{\partial p}{\partial y} + U^2\,\dfrac{\partial p}{\partial y}\right)\right\}\end{array}\right]$$

$$-\frac{3N\tau_B{}^3}{320\eta_a{}^3}\left[\begin{array}{l}\dfrac{\partial}{\partial x}\left\{h^5\,\dfrac{\partial p}{\partial x}\left[\left(\dfrac{\partial p}{\partial x}\right)^2 + \left(\dfrac{\partial p}{\partial y}\right)^2\right]\right\}\\[2mm]+\dfrac{\partial}{\partial y}\left\{h^5\,\dfrac{\partial p}{\partial y}\left[\left(\dfrac{\partial p}{\partial x}\right)^2 + \left(\dfrac{\partial p}{\partial y}\right)^2\right]\right\}\end{array}\right] \tag{1}$$

In view of the stochastic modeling adopted by Christensen and Tonder (1969a, 1969b, 1970) for transverse surface roughness we derive the stochastically averaged Reynolds type equation incorporating slip velocity Patel and Deheri (2011), Shah and Bhat (2003) as

$$\frac{\partial}{\partial x}\left\{g(h)\frac{\partial p}{\partial x}\right\} + \frac{\partial}{\partial y}\left\{g(h)\frac{\partial p}{\partial y}\right\} = 2\,\eta_a\dot{h}_0 + 6\eta_a\left[\frac{\partial}{\partial x}(bU) + \frac{\partial}{\partial y}(bV)\right] \tag{2}$$

where

$$g(h) = \left\{h^3 + 3\alpha h^2 + 3\left(\alpha^2 + \sigma^2\right)h + 3\sigma^2\alpha + \alpha^3 + \varepsilon + 2\,\phi H\right\}\frac{(4 + bh)}{(2 + bh)} \tag{3}$$

Thus, the Reynolds type equation for a one dimensional flow as in a slider bearing with the slider moving with a uniform velocity U in the X-direction and the stator having a porous facing of thickness H^* is obtained as

$$\frac{d}{dx}\left(g(h)\frac{dp}{dx}\right)=12\,\eta_a\dot{h}_0+6\eta_a U\frac{dh}{dx}-\frac{3N\tau_B{}^3 U^2}{16\,\eta_a}\frac{d}{dx}\left(\frac{(2+sh)}{(1+sh)}h\frac{dp}{dx}\right)-\frac{3N\tau_B{}^3}{320\eta_a{}^3}\frac{d}{dx}\left(h^5\left(\frac{dp}{dx}\right)^3\right)\quad(4)$$

Using the dimensionless quantities

$$N=\mu_0 M_0 H_0\,,\quad \overline{\beta}=\frac{-UJ_0}{2L\dot{h}_0}\,,\quad \tau=\frac{N\tau_B}{4\eta}\,,\quad \eta_q=\eta(1+\tau)\,,\quad \delta=\frac{3N\tau_B{}^3 U^2}{16\,\eta(1+\tau)h_0{}^2}\,,$$

$$\overline{\alpha}=\frac{\alpha}{h_0}\,,\quad \overline{\sigma}=\frac{\sigma}{h_0}\,,\quad \tilde{a}=\frac{\mathring{a}}{h_0{}^3}\,,\quad \overline{\psi}=\frac{\phi H}{h_0{}^3}\,,\quad \overline{s}=sh_0\,,\quad \overline{h}=\frac{h}{h_0}\,,\quad X=\frac{x}{L}\,,$$

$$\eta_a=\eta+\frac{N\tau_B}{4}\,,\quad P=\frac{-h_0{}^3 p}{\eta L^2\dot{h}_0}\,,\quad W=\frac{-h_0{}^3 w}{\eta L^4\dot{h}_0}\,,\quad F=\frac{h_0{}^2 f}{\eta L^3\dot{h}_0}$$

the Equation (4) can be expressed as,

$$\frac{d}{dX}\left(g(\overline{h})\frac{dP}{dX}\right)=12\,(1+\tau)\left(-1+\overline{\beta}\frac{d\overline{h}}{dX}\right)-\delta\frac{d}{dX}\left(\frac{(2+\overline{s}\overline{h})}{(1+\overline{s}\overline{h})}\overline{h}\frac{dP}{dX}\right)-\frac{\delta}{60\,(1+\tau)^2\,\overline{\beta}^2}\frac{d}{dX}\left\{\overline{h}^5\left(\frac{dP}{dX}\right)^3\right\}\quad(5)$$

where

$$g(\overline{h})=\left\{\overline{h}^3+3\overline{\alpha}\overline{h}^2+3\left(\overline{\alpha}^2+\overline{\sigma}^2\right)\overline{h}+3\overline{\sigma}^2\overline{\alpha}+\overline{\alpha}^3+\overline{\varepsilon}+12\,\overline{\psi}\right\}\frac{(4+\overline{s}\overline{h})}{(2+\overline{s}\overline{h})}\quad(6)$$

In a parallel plate slider bearing which means =1. So taking =1 in Equation (5), one gets

$$\frac{d}{dX}\left(g(1)\frac{dP}{dX}\right)=-12\,(1+\tau)-\delta\frac{d}{dX}\left(\frac{(2+\overline{s})}{(1+\overline{s})}\frac{dP}{dX}\right)-\frac{\delta}{60\,(1+\tau)^2\,\overline{\beta}^2}\frac{d}{dX}\left\{\left(\frac{dP}{dX}\right)^3\right\}\quad(7)$$

where

$$g(1)=\left\{1+3\overline{\alpha}+3\left(\overline{\alpha}^2+\overline{\sigma}^2\right)+3\overline{\sigma}^2\overline{\alpha}+\overline{\alpha}^3+\overline{\varepsilon}+12\,\overline{\psi}\right\}\frac{(4+\overline{s})}{(2+\overline{s})}\quad(8)$$

Taking the first approximation for P (Pressure) as in Bhat (2003) and solving equation (7) under the boundary conditions

$$P(0) = P(1) = 0$$

one arrives at

$$P = \frac{(1+\tau)}{(1+\delta)g(1)}\left[6(X - X^2)\frac{(1+\overline{s}-\delta)}{(1+\overline{s})} + \frac{2\ \delta}{9\ (1+\delta)^2\ \overline{\beta}^2}\left\{(2X-1)^4 - 1\right\}\right] \quad (9)$$

The load capacity W (property of a lubricant to form a film on the lubricated surface, which resists rupture under given load conditions), frictional force F are respectively, expressed in dimensionless form as

$$W = \frac{-h_0^3 w}{\eta L^4 \dot{h}_0} = \int_0^1 P\, dX$$

$$= \frac{(1+\tau)}{(1+\delta)g(1)}\left[\frac{(1+\overline{s}-\delta)}{(1+\overline{s})} - \frac{2\ \delta}{100(1+\delta)^2\ \overline{\beta}^2}\right] \quad (10)$$

and

$$F = -\int_0^1 \left[\frac{\vec{h}}{2}\frac{dP}{dX} + \frac{1}{\overline{h}}\right]dX$$

$$= \frac{(1+\tau)L}{(1+\delta)g(1)h_2}\left[3(a - a^2)\frac{(1+\overline{s}-\delta)}{(1+\overline{s})} - \frac{2\ \delta}{160(1+\delta)^2\ \overline{\beta}^2}\left\{(2a-1)^4 - 1\right\}\right] + (a-1) \quad (11)$$

For more about the friction and related aspects one can consider Basu et al. (2005), Shah and Bhat (2003).

3. Results and Discussion

It can be easily seen that the dimensionless pressure distribution is determined by the Equation (9). The load carrying capacity is determined from Equation (10). Besides, the friction is calculated from Equation (11). A close observation of Equation (9) reveals that the magnetization results in increased load carrying capacity. Setting the roughness parameters to be zero this investigation reduces

to the performance of a porous parallel slider bearing having smooth surfaces with slip velocity under the presence of a magnetic fluid lubricant. Furthermore, taking to be zero this study essentially, tends to the discussion of a parallel plate slider bearing with slip velocity which in turn, leads to the study of Basu et al. (2005) in the absence of slip velocity. An inspection of Equation (10) indicates that the bearing supports a load even where there is no flow.

The variation of load carrying capacity with respect to magnetization parameter given in Figures 2-6 suggests that the load carrying capacity increases sharply due to the magnetization parameter.
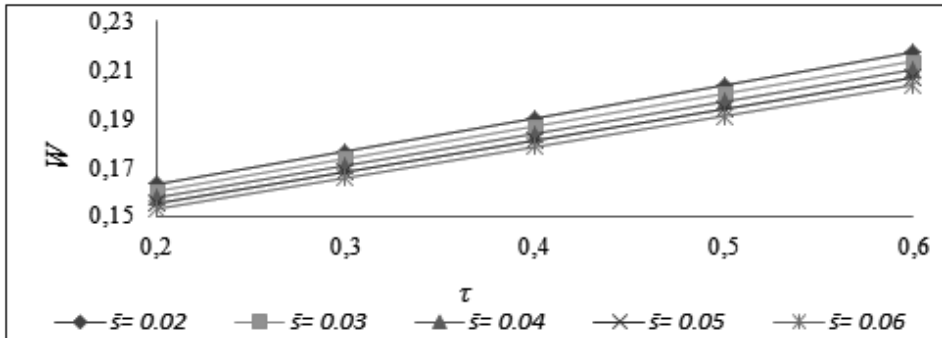
Figure 2 Variation of load carrying capacity with respect to τ and S
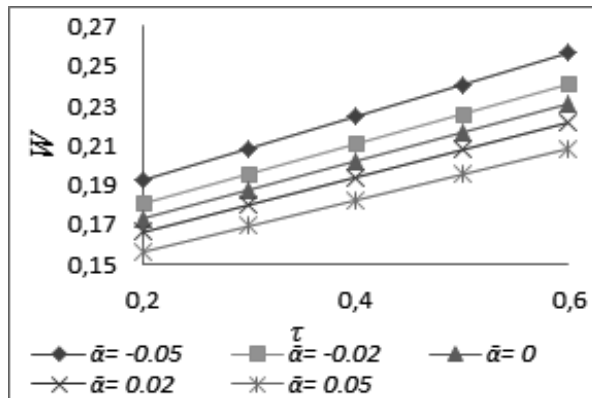
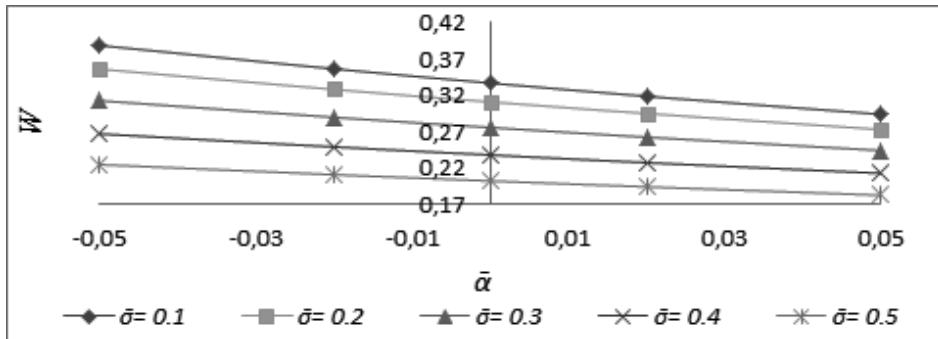Figure 3 Variation of load carrying capacity with respect to τ and $\bar{\alpha}$
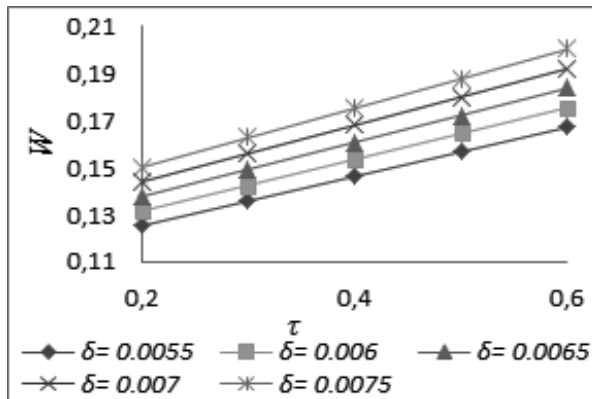
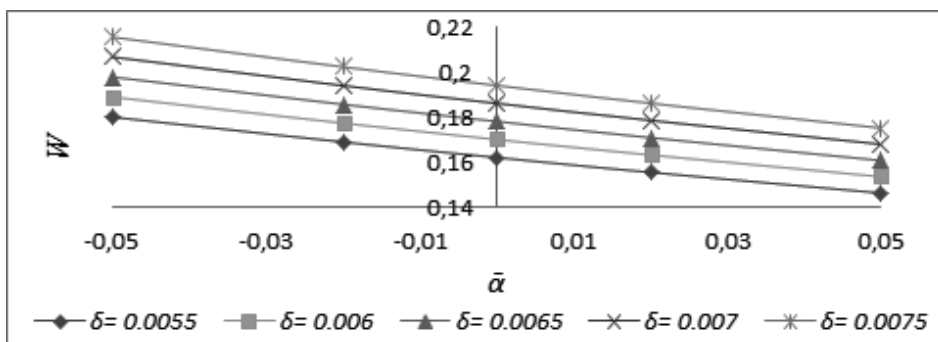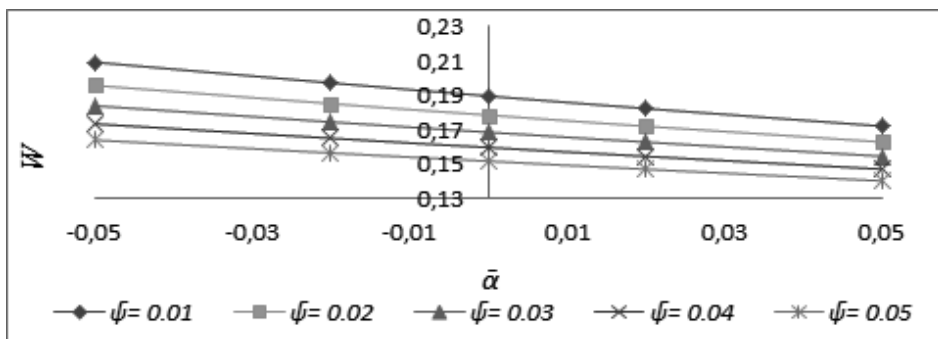Figure 4 Variation of load carrying capacity with respect to τ and $\bar{\sigma}$
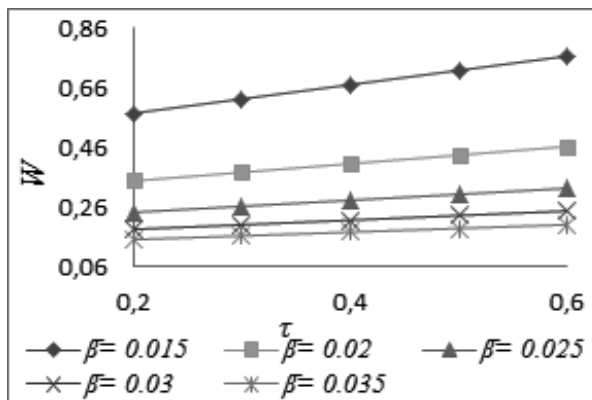
Figure 5 Variation of load carrying capacity with respect to τ and δ
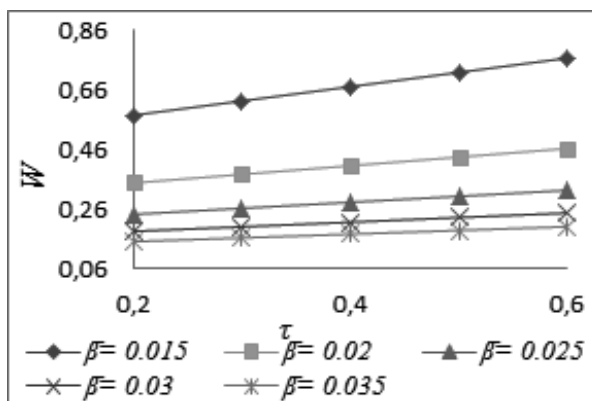
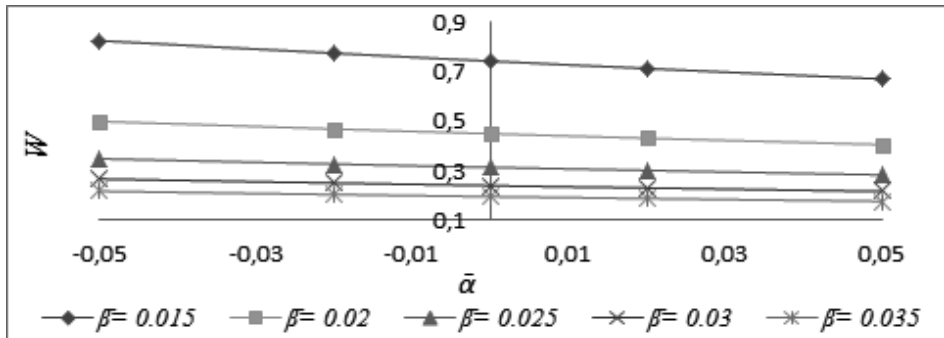Figure 6 Variation of load carrying capacity with respect to τ and β

The variation of load carrying capacity with respect to slip parameter is presented in Figures 7-9. It is clearly seen that the load carrying capacity decreases considerably with the increasing values of slip parameter.
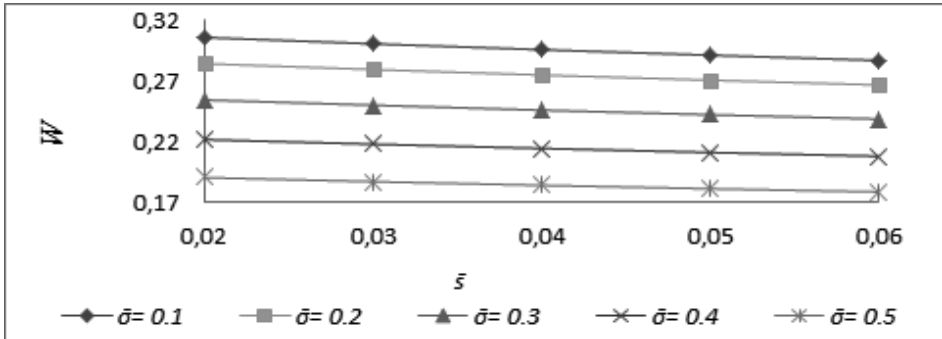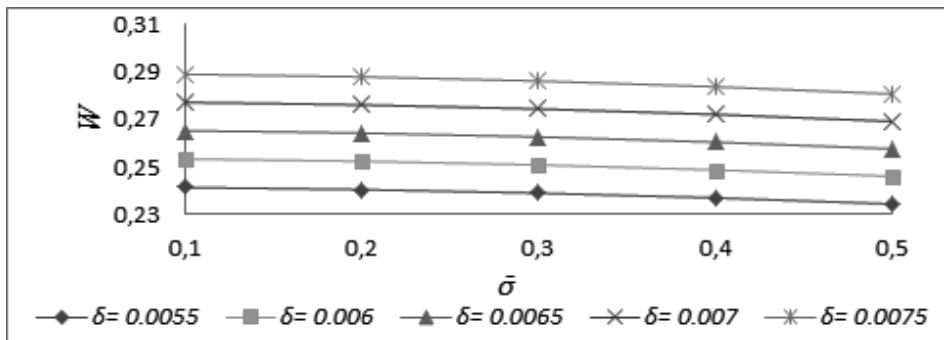

Figure 7 Variation of load carrying capacity with respect to $\bar{s}$ and $\bar{\sigma}$
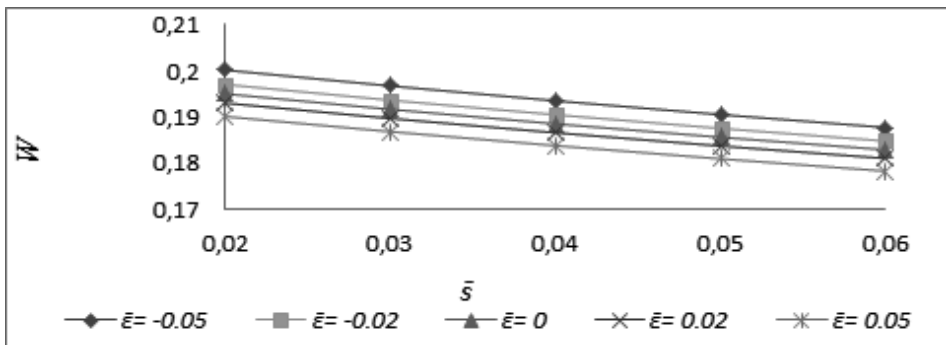
Figure 8 Variation of load carrying capacity with respect to $\bar{s}$ and $\bar{\varepsilon}$
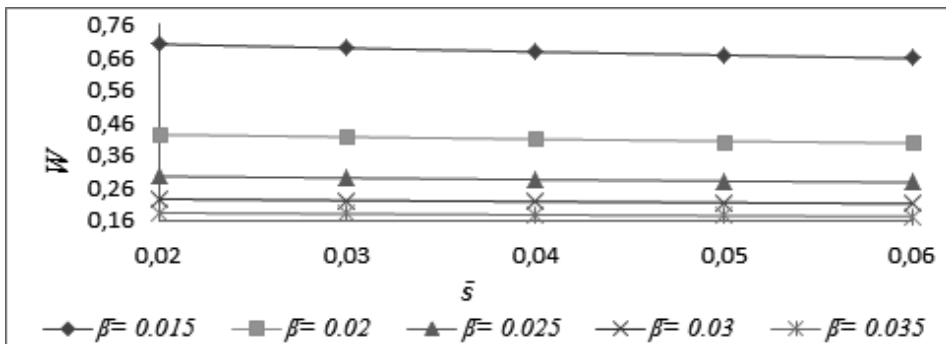
Figure 9 Variation of load carrying capacity with respect to $\bar{s}$ and β

The effect of the variance $\bar{\alpha}$ on the distribution of load carrying capacity is displayed in the Figures 10-13. It becomes clear that variance (+ve) decreases the load carrying capacity while variance (-ve) increases the load carrying capacity.

Figure 10 Variation of load carrying capacity with respect to $\bar{\alpha}$ and $\bar{\sigma}$.

Figure 11 Variation of load carrying capacity with respect to $\bar{\alpha}$ and δ.

Figure 12 Variation of load carrying capacity with respect to $\bar{\alpha}$ and $\bar{\psi}$

Figure 13 Variation of load carrying capacity with respect to $\bar{\alpha}$ and β

The effect of standard deviation depicted in Figures 14-15 suggests that the standard deviation has considerably adverse effects on the performance of the bearing system in the sense that it decreases the load carrying capacity.

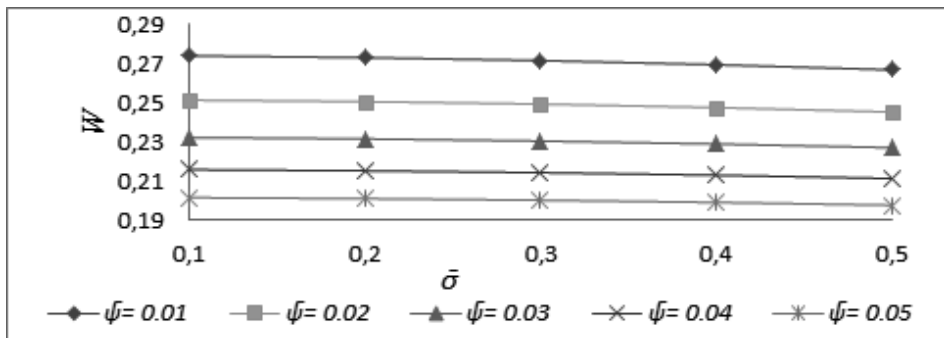Figure 14 Variation of load carrying capacity with respect to $\bar{\sigma}$ and δ.

Figure 15 Variation of load carrying capacity with respect to $\bar{\sigma}$ and $\bar{\psi}$

Figure 16 represents the effect of skewness on the load carrying capacity. Interestingly it follows the path of variance regarding the trends of load carrying capacity.
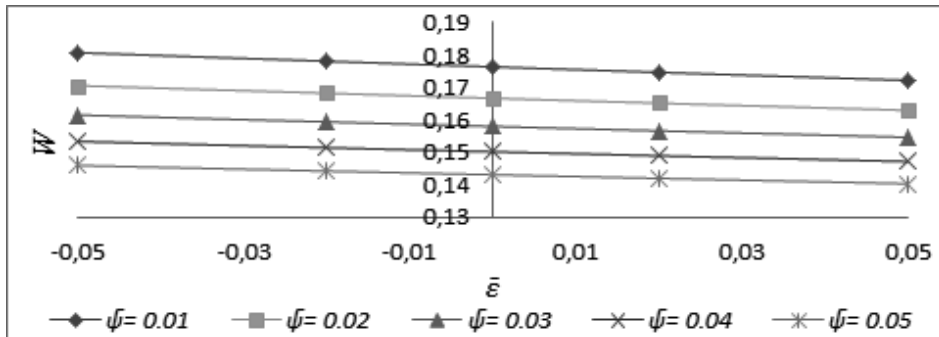
Figure 16 Variation of load carrying capacity with respect to $\bar{\varepsilon}$ and $\bar{\psi}$

The fact that the material parameter β decreases the load carrying capacity significantly can be found in Figure 17.
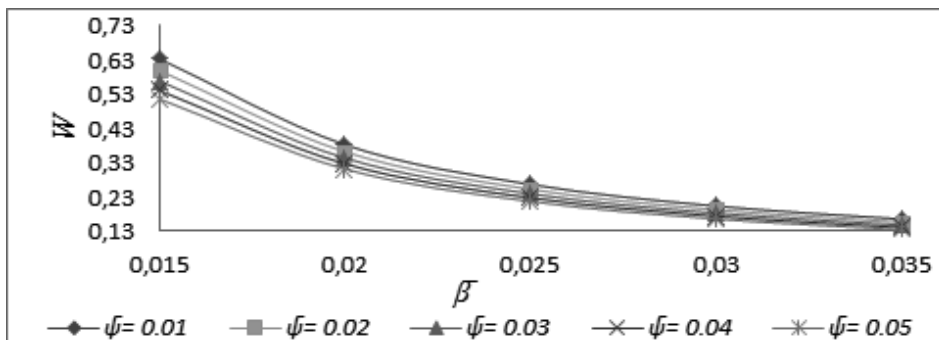

Figure 17 Variation of load carrying capacity with respect to β and $\bar{\psi}$

The effect of magnetization on the friction is presented in Figures 18-22. It is found that the friction increases due to the magnetization parameter.
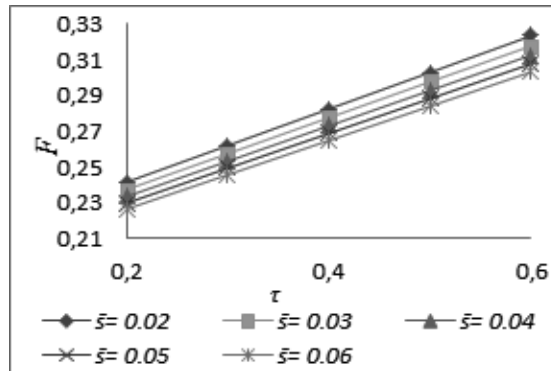
Figure 18 Variation of load carrying capacity with respect τ to and S
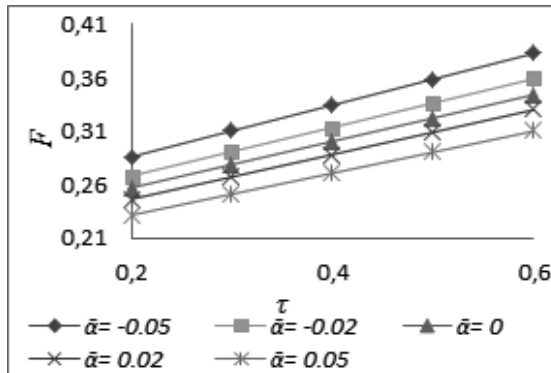

Figure 19 Variation of load carrying capacity with respect to τ and $\bar{\alpha}$
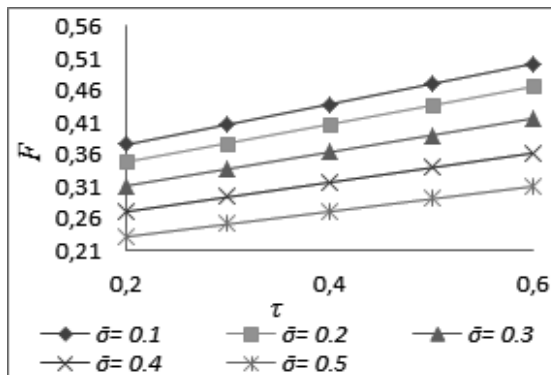

Figure 20 Variation of load carrying capacity with respect to τ and $\bar{\sigma}$
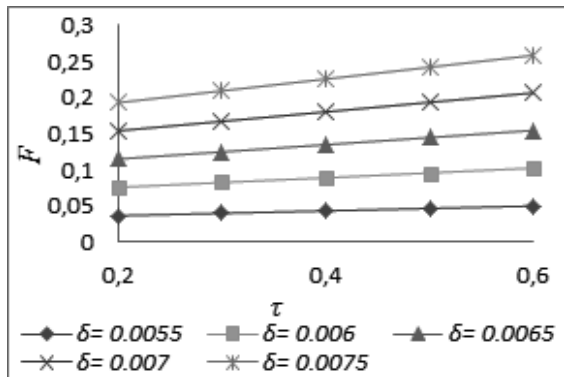
Figure 21 Variation of load carrying capacity with respect to τ and δ
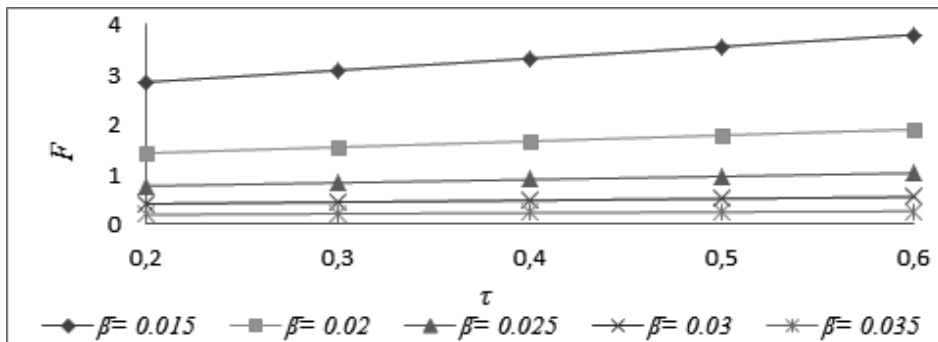

Figure 22 Variation of load carrying capacity with respect to τ and β

The effect of slip parameter on the friction depicted in Figures 23-25 makes it clear that the friction decreases due to the slip parameter. Of course, this decrease is negligible in the case of material parameter.
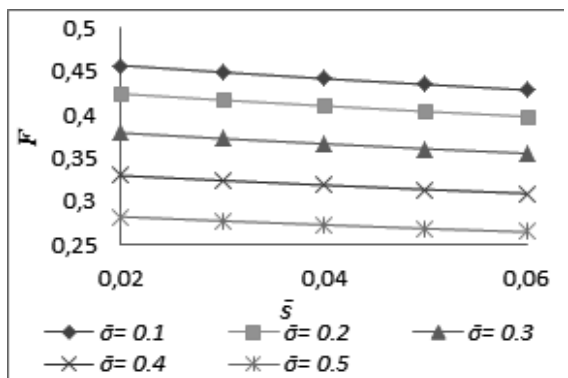

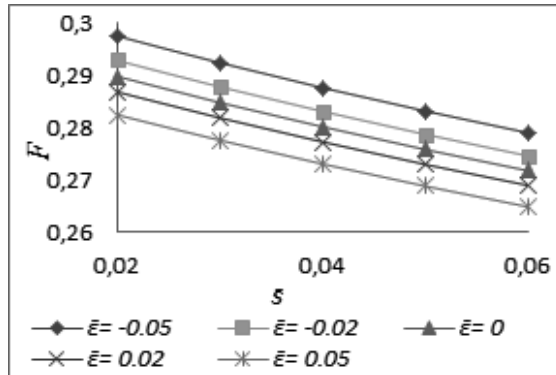Figure 23 Variation of friction with respect to $\bar{s}$ and $\bar{\sigma}$

Figure 24 Variation of friction with respect to $\bar{s}$ and $\bar{\varepsilon}$
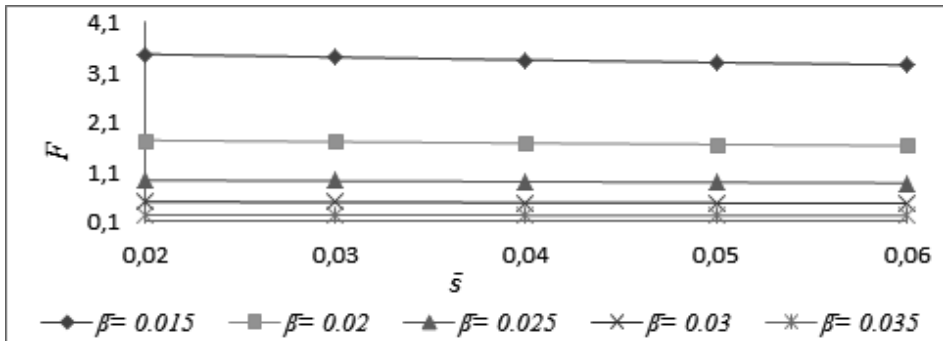

Figure 25 Variation of friction with respect to $\bar{s}$ and $\bar{\beta}$

Figures 26-29 present the effect of variance on the friction. It is clearly seen that the variance (+ve) decreases the friction while variance (-ve) increases the friction.

Figure 26 Variation of friction with respect to $\bar{\alpha}$ and $\bar{\sigma}$

Figure 27 Variation of friction with respect to $\bar{\alpha}$ and δ

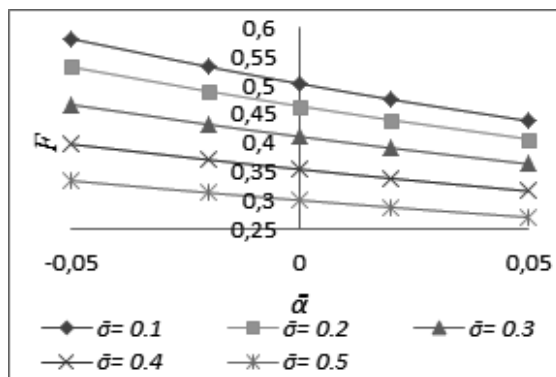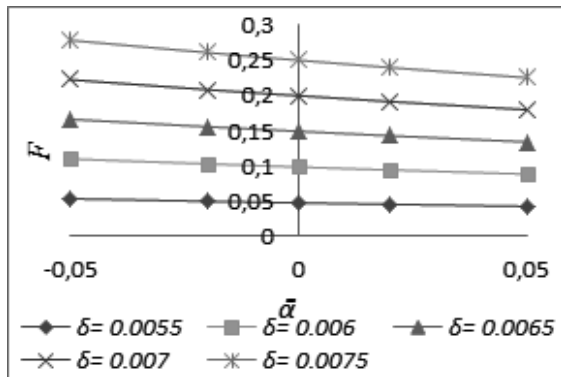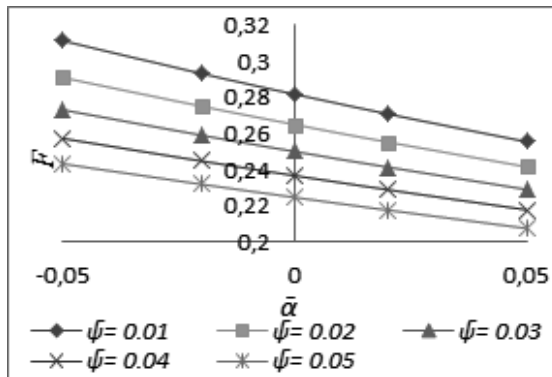Figure 28 Variation of friction with respect to $\bar{\alpha}$ and $\bar{\psi}$

Figure 29 Variation of friction with respect to $\bar{\alpha}$ and β

Figures 30-31 suggest that the friction reduces due to the standard deviation.

Figure 30 Variation of friction with respect to $\bar{\sigma}$ and δ

Figure 31 Variation of friction with respect to $\bar{\sigma}$ and $\bar{\psi}$
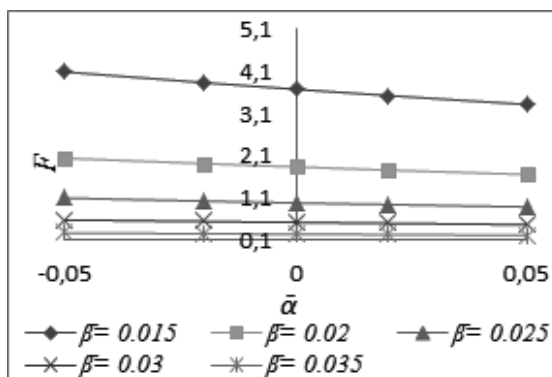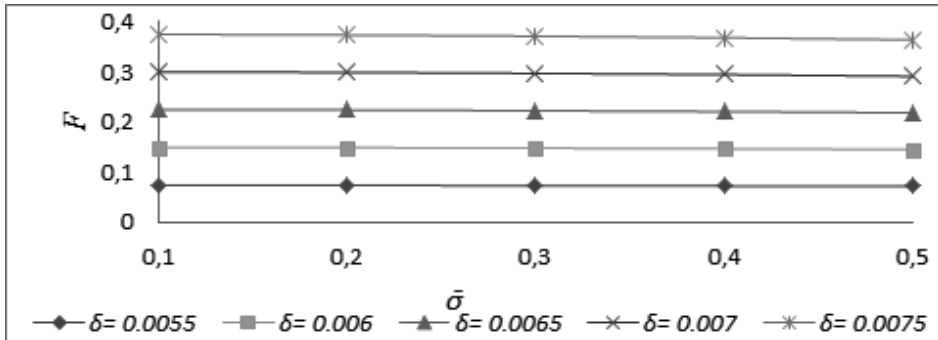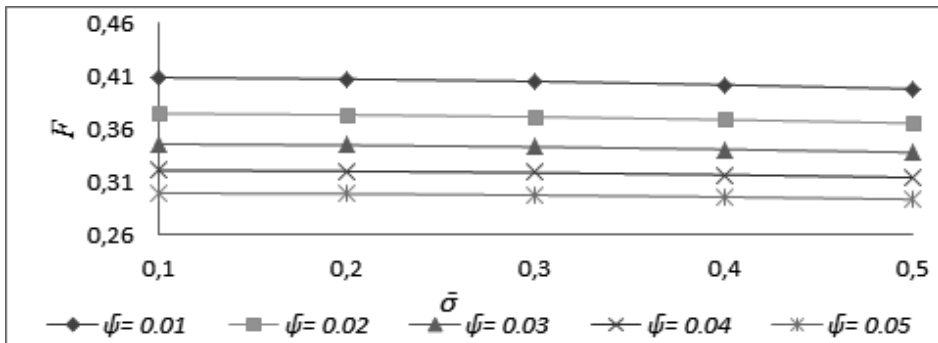
The effect of skewness on the friction is displayed in Figure 32. It is clear that skewness follows the path of variance regarding the trends of friction.
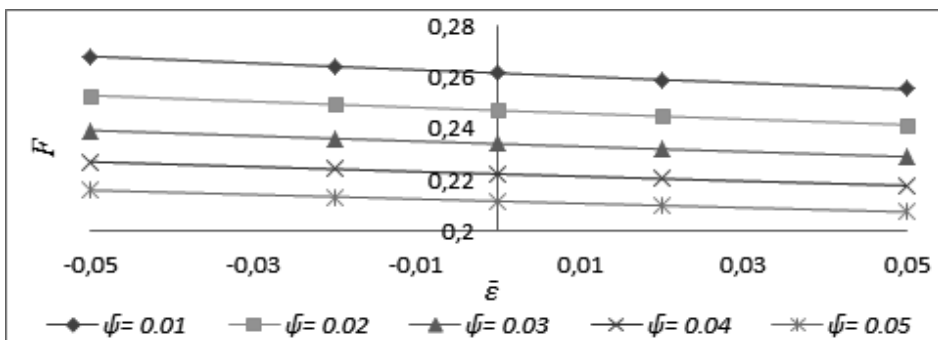

Figure 32 Variation of friction with respect to $\bar{\varepsilon}$ and $\bar{\psi}$

It is clear that material parameter decreases the friction as shown in Figure 33.
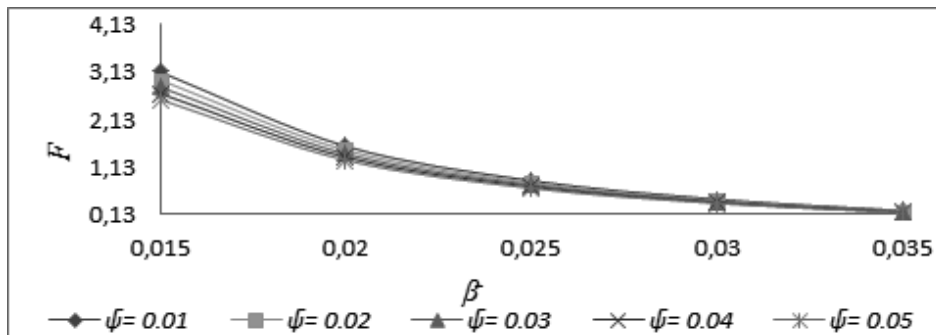
Figure 33 Variation of friction with respect to β and $\bar{\psi}$

A comparison of this study with the investigations of Shah and Bhat (2003) indicates that the load carrying capacity gets increase by 3% at least in the case of negatively skewed roughness. The effect of transverse surface roughness is adverse in general. Probably this is due to the fact that the roughness retards the motion of the lubricant.

4. Conclusion

This investigation makes it mandatory that the roughness deserves to be given due consideration while designing the ferrofluid lubricated parallel plate slider bearing system even if, the slip is negligible and magnetization has been chosen suitably. It is found that even in the absence of flow the bearing can support a load, in spite of the fact that the roughness and the slip adversely affect the bearing system.

Nomenclature

f	Frictional force
h	Fluid film thickness at any point
p	Lubricant pressure
s	Slip parameter
u	X component of film fluid velocity
w	Load carrying capacity
F	Dimensionless frictional force
H	Magnitude of the magnetic field
L	Length of the bearing

P	Dimensionless pressure
U	Velocity of slider
W	Dimensionless load carrying capacity
$\bar{s}$	Dimensionless slip parameter
	Squeeze Parameter
	Permeability of the porous matrix
	The fluid viscosity
σ	Standard deviation
α	Variance
ε	Skewness
$\bar{\psi}$	Porosity
$\bar{\sigma}$	Dimensionless standard deviation
$\bar{\alpha}$	Dimensionless variance
$\bar{\varepsilon}$	Dimensionless skewness
$\bar{\tau}$	Magnetization parameter in non dimensional form

References

Abramovitz, S., 1955. Theory for a slider bearing with a convex pad surface; side flow neglected, *Journal of the Franklin Institute*, 259, Issue 3, 221-233.

Agrawal, V., 1986. Magnetic fluid based porous inclined slider bearing, *Wear*, 107, 133-139.

Basu, S., Sengupta, S. and Ahuja, B., 2005. Fundamentals of Tribology, *Prentice-Hall of Indian Private Limited*, New Delhi.

Beavers, G. and Joseph, D., 1967. Boundary conditions at a naturally permeable wall, Journal of Fluid Mech., 30, 197-207.

Bhat, M. and Deheri, G., 1991a. Squeeze film behavior in porous annular discs lubricated with magnetic fluid, *Wear*, 151, 123-128.

Bhat, M. and Deheri, G., 1991b. Porous composite slider bearing lubricated with magnetic fluid, Jap. Journal Applied Physics, 30, 2513-2514.

Bhat, M., 2003. Lubrication with a magnetic fluid, *Team Spirit (India) Pvt. Ltd.*

Bhat, M. and Patel, R., 1991. Analysis of an exponential porous slider bearing lubricated with magnetic fluid, *Journal of Eng. and Tech.*, S.P. University., 6, 19-21.

Christensen, H. and Tonder, K., 1969a. Tribology of rough surfaces: Stochastic models of hydrodynamic lubrication, SINTEF Report No. 10/69-18.

Christensen, H. and Tonder, K., 1969b. Tribology of rough surfaces: parametric study and comparison of lubrication, SINTEF Report No. 22/69-18.

Christensen, H. and Tonder, K., 1970. The Hydrodynamic lubrication of rough bearing surfaces of finite width, *ASME-ASLE lubrication conference*, Paper No.

70. Lub.-7.

Das, N., 1998. A study of optimum load-bearing capacity for slider bearings lubricated with couple stress fluids in magnetic field, *Tribology International*, 393-400.

Deheri, G., Andharia, P. and Patel, R., 2004. Longitudinally rough slider bearings with Squeeze film formed by a magnetic fluid, *Industrial Lubrication and Tribology*, 56, No. 3, 177-187.

Deheri, G., Andharia, P. and Patel, R., 2005. Transversely rough slider bearings with squeeze film formed by a magnetic fluid, *International Journal of Applied Mechanics and Engineering*, 10, No. 1, 53-76.

Gupta, J. and Bhat, M., 1979. An inclined porous slider bearing with a transverse magnetic field, *Wear*, 55, Issue 2, 359-367.

Malik, M. and Singh, D., 1980. Analysis of finite magneto hydrodynamic journal bearings, *Wear*, 64, Issue 2, 273-280.

Nanduvinamani, N., Fathima, S. and Hiremath, P., 2003. Hydrodynamic lubrication rough Slider bearings with couple stress fluids, *Tribology International*, 36, 12, 949-959.

Odenbach, S., 1993. Magnetic fluids, *Advances in Colloid and Interface Science*, 46, 6, 263-282.

Odenbach, S., 2000. Magneto viscous and viscoelastic effects in ferrofluid, *International Journal of Modern Physics B (IJMPB)*, 14, Issue: 16, 1615-1631.

Osterle, J. and Young, F., 1962. On the load capacity of the hydro magnetically lubricated slider bearing, *Wear*, 5, Issue 3, 227-234.

Patel, N. and Deheri, G., 2011. Shliomis model based ferrofluid lubrication of a plane inclined slider bearing with slip velocity, *International journal of fluids engineering*, 3, No 3, 311-324.

Patrick, Ilg., Roger, M. and Ess, S., 2005. Anisotropy of the magneto viscous effect in ferrofluidsbstract, *Phys. Rev. E,* 71, [6 pages].

Ram, P., Sharma, K. and Bhandari, A., 2010. Effect of porosity on ferrofluid flow with rotating disk, *Int. J. of Appl. Math and Mech.*, 6 (16), 67-76.

Ramanaiah, G. and Sarkar, P., 1979. Slider bearings lubricated by fluids with couple stress, *Wear*, 52, Issue 1, 27-36.

Rosensweig, R., 1985. Ferro hydrodynamics, *Cambridge Univ. Press*, Cambridge.

Shah, R., Tripathi, S. and Bhat, M., 2002. Magnetic fluid based squeeze film between porous annular curved plates with the effect of rotational inertia, *Pramana-J. Phys.*, 58, 545-550.

Shah, R. and Bhat, M., 2004. Ferrofluid squeeze film in a long journal bearing, *Tribology International*, 37, 441-446.

Shah, R. and Bhat, M., 2003. Effect of slip velocity in a porous secant shaped slider bearing with a ferrofluid lubricant, *FIZIKA A,* 12, (1), 1-8.

Shukla, J. and Kumar, D., 1987. A theory for ferromagnetic lubrication, *Journal of Magnetism and Magnetic Materials*, 65, Issues 2-3, 1, 375-378.

Shliomis, M., 1972. Effective viscosity of magnetic suspensions, *Sov. Phys. JETP.*,

34, 1291-1294.

Sinha, P., Chandara, P. and Kumar, D., 1993. Ferrofluid lubrication of cyndrical rollers with cavitations, *Acta Mech.*, 98, 27-38.

Sparrow, E., Beavers, G. and Hwang, I., 1972. Effect of velocity slip on porous walled squeeze film, *Journal Lubrication Tech.*, 94, 260-265.

Thurm, S. and Odenbach, S., 2003. Particle size distribution as key parameter for the flow behavior of ferrofluids, *Phys. Fluids*, 15, 1658-1664.

Tzeng, S. and Saibel, E., 1967. Surface roughness effect on slider bearing lubrication, *Trans. ASME, J. Lubrication, Technology*, 10, 334-338.