International Journal of Applied Engineering Research ISSN 0973-4562 Volume 14, Number 10 (2019) pp. 2408-2416
© Research India Publications. http://www.ripublication.com

A Study of Thermal Effect on the Ferrofluid Lubrication of a Rough Porous Exponential Slider Bearing Considering Slip Velocity

Sheetal Amrutbhai Patel[1], **Ashokkumar Revidas Patel**[2] **& Gunamani Deheri**[3]

[1]*PhD Scholar in Gujarat Technological University, Chandkheda, Ahmedabad, Gujarat, India.*

[2] *Department of Mathematics, Vishwakarma Government Engineering College, Chandkheda, Ahmedabad, Gujarat, India.*

[3] *Department of Mathematics, Sardar Patel University, Vallabh Vidyanagar, Gujarat, India.*

Abstract

This investigation discusses the Ferrofluid lubrication of a rough porous exponential slider bearing considering thermal effect and slip velocity. The model of Christensen and tonder has been taken into consideration for evaluating the effect of transverse roughness. The Ferrofluid flow model of Neuringer - Rosensweig has been invoked for the development of modified Reynolds equation. The model of Tipei has been adopted for the thermal effect. Beavers and Joseph's slip model used for slip velocity. The load bearing capacity has been obtained numerically. The graphical representation shows that there is a steep hike in the load carrying capacity due to magnetization which becomes nearly enough to overcome the thermal effects when the porosity is at reduce level. Of course, the transverse roughness registers a negative effect, But the situation remains relatively better in the case of negatively skewed roughness. However, in better performance the slip is desired to be less.

Keywords: Exponential slider bearing, porosity, roughness, Slip velocity, Thermal effect, load carrying capacity.

1 INTRODUCTION

In a short time ago the application of Ferrofluid lubrication of rough surfaces has achieve much observation by research approach. Generally the bearing surfaces are not smooth in nature.When the film thickness between the surface is very microscopic,the surface asperities establish to interface with the lubricant.The impact of surface roughness taken in to account to study the performance of lubricant system,where the flow becomes narrow.The tribology investigation has been used to create of modern machine equipment. The surface roughness effect has been evaluated by Christensen and Tonder [1, 2, 3]. The model of Christensen and Tonder has been used in many investigators Andharia and Patel [4], Chawla and Bhardwaj [5], Patel and Deheri [6] these studies confirm that the load carrying capacity reduces due to roughness effect. Also, magnetic fluid lubrication used by many research [7], [8], [9].

Investigators managing hydrodynamic lubrication have thought about consistency, as steady. Though, it depends on temperature and pressure. The variation of viscosity with temperature remains crucial in numerous tribological applications especially where a wide range of temperature is required.

Tipei [10] observed that the highest temperature happened when the film thickness was least. The combined impact of surface roughness and variation of viscosity because of added substances on long journal bearing was analyzed by Siddangouda et al. [11] From the different designs of bearings, Cameron [12] is Proposed the shape which was approximately near to the shape of an exponential form of the slider.

Bhat [13] considered the oil based lubrication of an exponential permeable slider bearing and inferred that utilization of porous matrix diminished the load limit and friction force on slider bearing.

Bhat and Patel [14] studied an exponential permeable slider bearing, considering a Ferro liquid as lubricant with the flow represented by Neuringer-Rosensweig's model. It was discovered that due to Ferrofluid lubricant the load limit of the bearing expanded without influencing the friction force.

Shah and Bhat [15] dealt with a permeable exponential slider bearing lubricated with a Ferrofluid considering slip velocity. It was seen that an exponential porous slider bearing had more burden bearing limit and friction than the related inclined plane permeable slider bearing.

Chi-Ren Hung [16] explored the dynamic qualities of slider bearing for an exponential film profile and found the higher estimation of firmness coefficient, load bearing limit and damping co-efficient.

Patel et al. [17] investigated the Ferrofluid lubrication of a journal bearing considering the thermal effect. The load bearing capacity got reduced due to thermal effect. It was found that for as type of improvement in the performance characteristics the slip velocity was required to be minimized.

In order to modified and develop the analysis contained in Patel et.al. [18].

In the present paper a Ferrofluid has been taken as a lubricant to study an exponential slider bearing taking in to account the effect of roughness, slip velocity and porosity considering thermal effect.

2 ANALYSIS

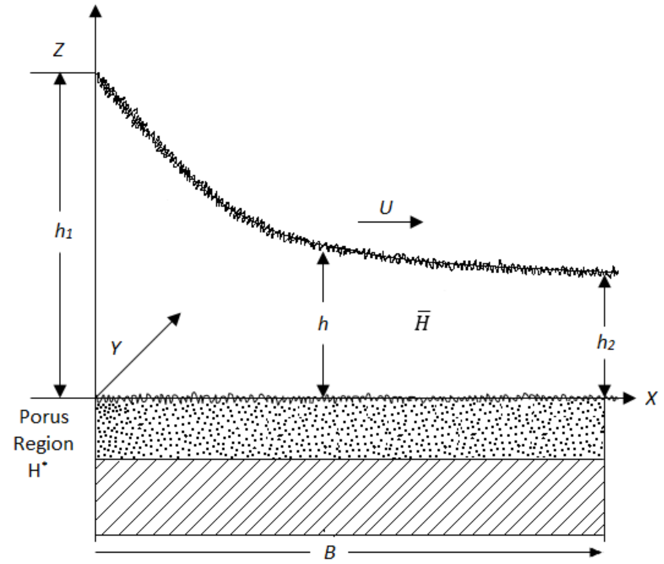

Figure 1. Configuration of an Exponential Bearing.

The film thickness is taken as

$$h = h_1 e^{\frac{-(x\ln a)}{B}}; 0 \le x \le B$$

As can be seen the bearing length is B.

The random roughness of the bearing surfaces is characterized by a random variable with non-zero mean α, the standard deviation σ and the Skewness ε. They are defined as:

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

where E denotes the expected value given by

$$E(R) = \int_{-c}^{c} Rf(h_s)\, ds$$

In the light of christensen and tonder [1, 2, 3] the thickness h of the lubricant film is assumed to be

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

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. Also h_s is governed by the probability density function

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

where c is the maximum deviation from the mean film thickness. The details can be obtained from christensen and tonder [1, 2, 3].

The basic flow equations of magnetic fluid flow as provided by Neuringer-Rosensweig model [19] are

$$\rho\left[\frac{\partial \bar{q}}{\partial t} + (\bar{q} \cdot \nabla)\bar{q}\right] = -\nabla p + \eta \nabla^2 \bar{q} + \mu_0 (\bar{M} \cdot \nabla)\bar{H} \tag{3}$$

$$\nabla \cdot \bar{q} = 0, \nabla \times \bar{H} = 0, \bar{M} = \bar{\mu}\bar{H}, \nabla \cdot (\bar{H} + \bar{M}) = 0 \tag{4}$$

Where in

$$\bar{q} = ui + vj + wk \tag{5}$$

u, v, w are components of fluid film velocity in x, y, z directions respectively.

With the use of Eq. (3), (4) and (5) the lubricant flow is determined by

$$\frac{\partial^2 u}{\partial z^2} = \frac{1}{\eta}\frac{\partial}{\partial x}\left(p - \frac{1}{2}\mu_0 \bar{\mu} H^2\right) \tag{6}$$

Here, η is the fluid viscosity,p is the film pressure, $\mu_0$is the permeability of free space and $\bar{\mu}$ is the magnetic susceptibility.

Sprrow et.al. [20] have considered the following boundary conditions:

$$u = V \text{ when } z = h,$$

$$u = \left(\frac{1}{s}\frac{\partial u}{\partial z}\right)_{z=0} \text{ when } z = 0 \tag{7}$$

where s is slip constant.

Solving Eq. (6) under the boundary conditions (7), 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 of (Agrawal [21], Shah and Bhat [15] Patel et.al. [22]

$$\frac{d}{dx}\left[\left\{12kH^* + \frac{h^3(4+sh)}{(1+sh)}\right\}\frac{d}{dx}\left(p - \frac{\mu_0 \bar{\mu} H^2}{2}\right)\right] = 6\eta V \frac{d}{dx}\left[\frac{h(2+sh)}{(1+sh)}\right] + 12\eta V \tag{8}$$

Here, the magnetic field is oblique to the lower surface whose magnitude is taken as Bhat (23)

$$H^2 = kx(B - x)$$

where $k(A^2 m^{-4})$ is chosen for a suitable dimension.

Where roughness

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

when the viscosity η_0 at $h = h_2$ is known,

The thermal effect gives the viscosity-temperature relation as

$\eta = \eta_0 \left(\frac{h}{h_2}\right)^q$ where q is the thermal factor which usually lies between 0 and 1 according the nature of the lubrication. (Tipei) [10]

In corporation of thermal effect and surface roughness effect transform to Eq. (8)

$$\frac{d}{dx}\left[\left\{\frac{12\psi}{(g(\bar{h}))^{\frac{q}{3}}} + \frac{g(\bar{h})(4+\bar{s}\bar{h})}{(g(\bar{h}))^{\frac{q}{3}}(1+\bar{s}\bar{h})}\right\}\frac{d}{dx}\left(\bar{p} - \frac{\mu^* X(1-X)}{2}\right)\right]$$

$$= \frac{d}{dx}\left\{\frac{(g(\bar{h}))^{\frac{1}{3}}(2+\bar{s}\bar{h})}{(1+\bar{s}\bar{h})}\right\} + 12 \tag{9}$$

Induction of non-dimensional quantities

$$X = \frac{x}{B}\ ,\ \bar{h} = \frac{h}{h_2},\ \bar{p} = \frac{{h_2}^3 p}{\eta_0 V B^2}\ ,\ \mu^* = \frac{\mu_0 \bar{\mu} k {h_2}^3}{\eta_0 V}\ ,\ \bar{s} = h_2 s\ ,$$

$$\psi = \frac{kH^*}{{h_2}^3},\ \bar{\sigma} = \frac{\sigma}{h_2},\ \bar{\alpha} = \frac{\alpha}{h_2},\ \bar{\varepsilon} = \frac{\varepsilon}{{h_2}^3}\ ,\ a = \frac{h_1}{h_2}$$

Eq. (9) transforms to:

$$\frac{\partial}{\partial X}\left[C\frac{\partial}{\partial X}\left(\bar{p} - \frac{1}{2}\mu^* X(1-X)\right)\right] = \frac{\partial M}{\partial X} + 12 \qquad (10)$$

where, $C = \left(\frac{12\psi}{(g(\bar{h}))^{\frac{q}{3}}} + \frac{g(\bar{h})^{\frac{3-q}{3}}(4+\bar{s}\bar{h})}{(1+\bar{s}\bar{h})}\right)$, $M = \frac{6(g(\bar{h}))^{\frac{1}{3}}(2+\bar{s}\bar{h})}{(1+\bar{s}\bar{h})}$

As usually discuss the associated boundary conditions are

$$\bar{p} = 0 \text{ when } X = 0,1$$

Now, taking $\quad \bar{h} = ae^{-Xlna}, 0 \le X \le 1 \qquad (11)$

And integrating Eq. (10) with the above boundary conditions, one gets

$$\bar{p} = \frac{1}{2}\mu^* X(1-X) + \int_1^X \frac{M-N+12X}{C}\,dX \qquad (12)$$

Where, $N = \frac{\int_0^1 \frac{M}{C}dX + 12\int_0^1 \frac{X}{C}dX}{\int_0^1 \frac{1}{C}dX}$

Lastly the load carrying capacity is determined by the relation

$$\bar{W} = \int_0^1 \bar{p}\,dX$$

$$\bar{W} = \frac{\mu^*}{12} + \int_0^1 \frac{(N-M+12X)}{C}XdX \qquad (13)$$

3 RESULT AND DISCUSSIONS

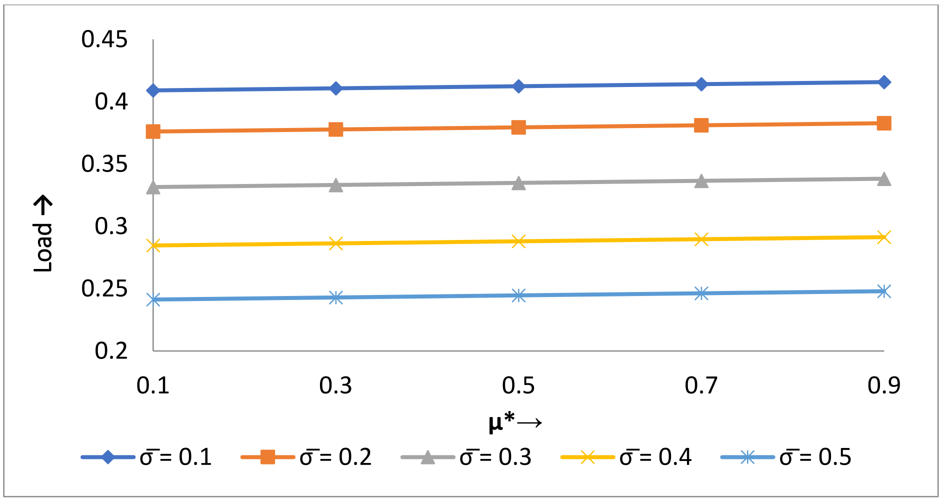

Figure 2. Variation of W with respect to $\mu^*$and $\bar{\sigma}$.

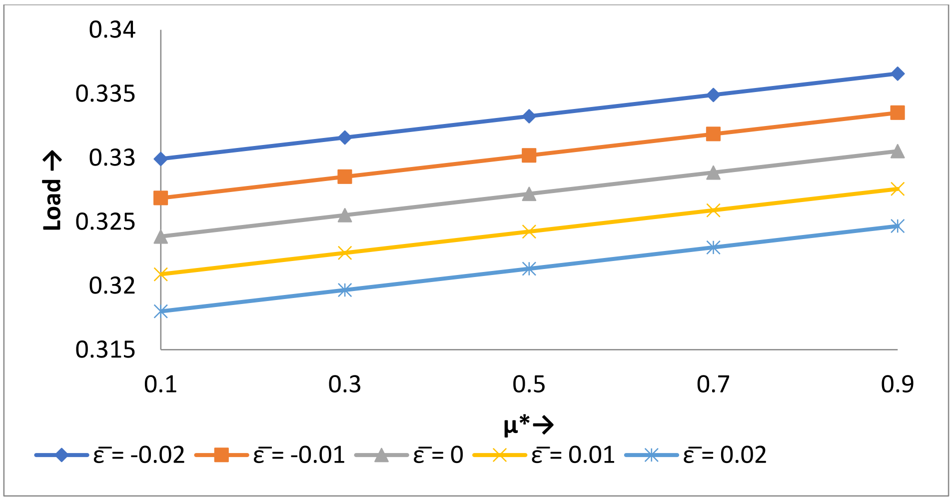

Figure 3. Variation of W with respect to $\mu^*$and $\bar{\varepsilon}$.

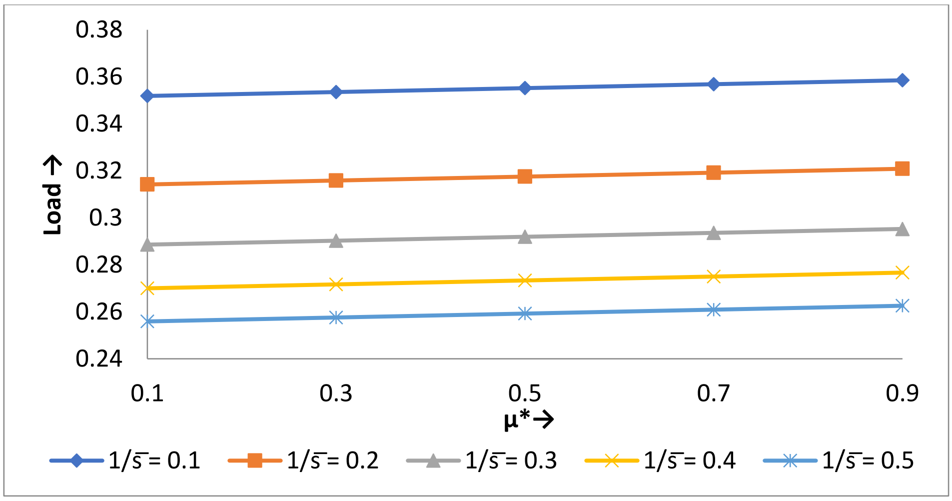

Figure 4. Variation of W with respect to μ^* and$1/\bar{s}$.

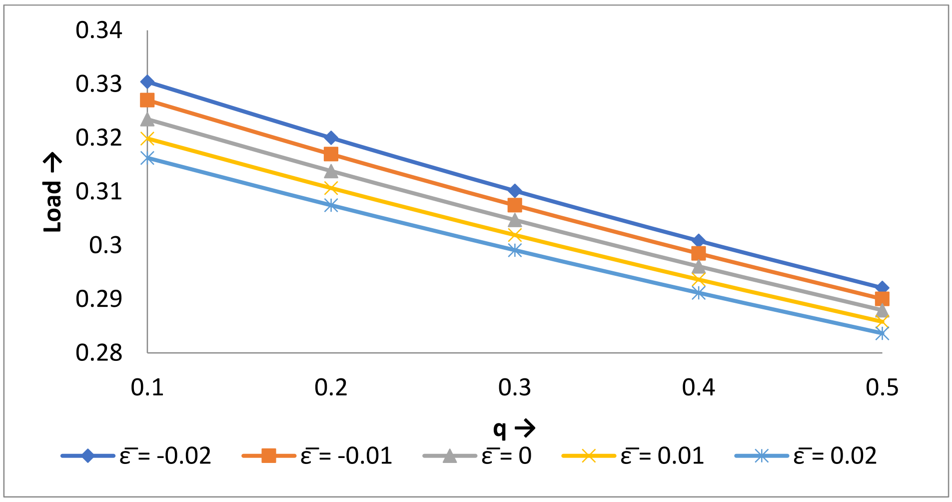

Figure 5. Variation of W with respect to q and $\bar{\varepsilon}$.

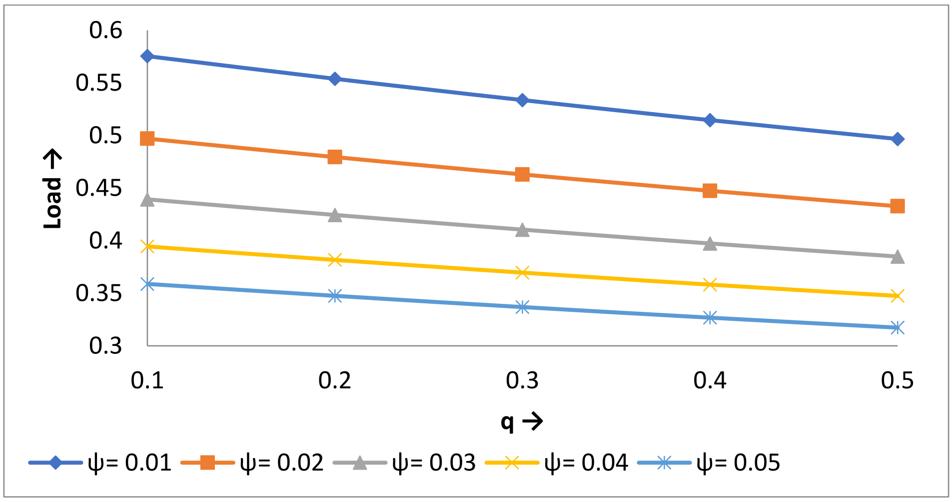

Figure 6. Variation of W with respect to q and ψ.

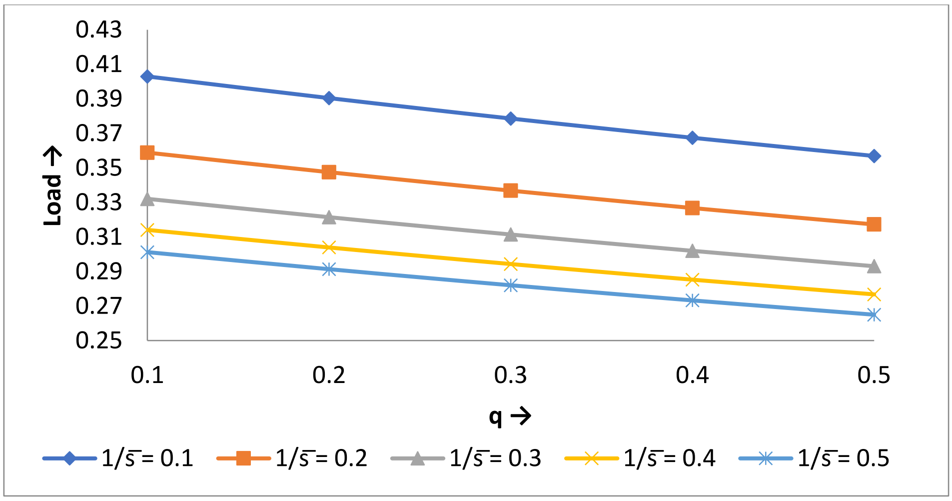

Figure 7. Variation of W with respect to q and $1/\bar{s}$.

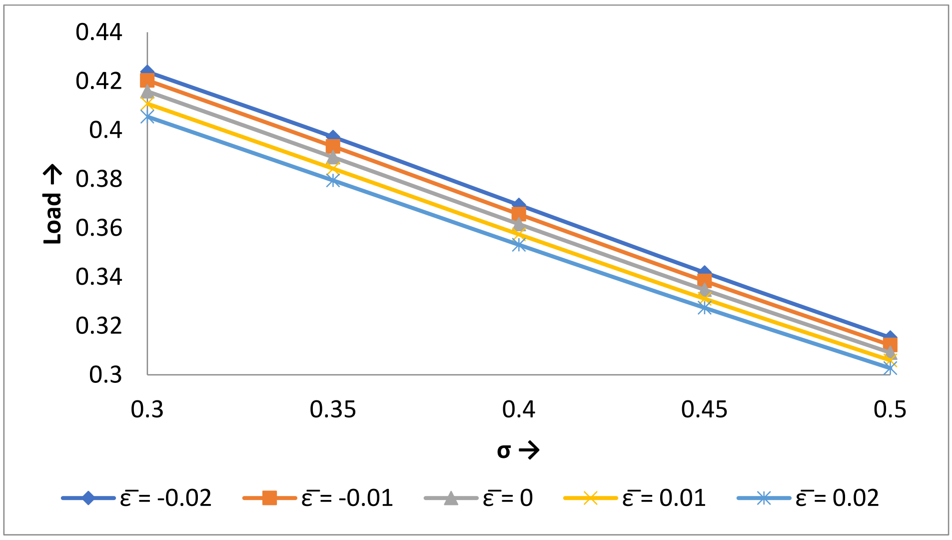

Figure 8. Variation of W with respect to σ and $\bar{\varepsilon}$.

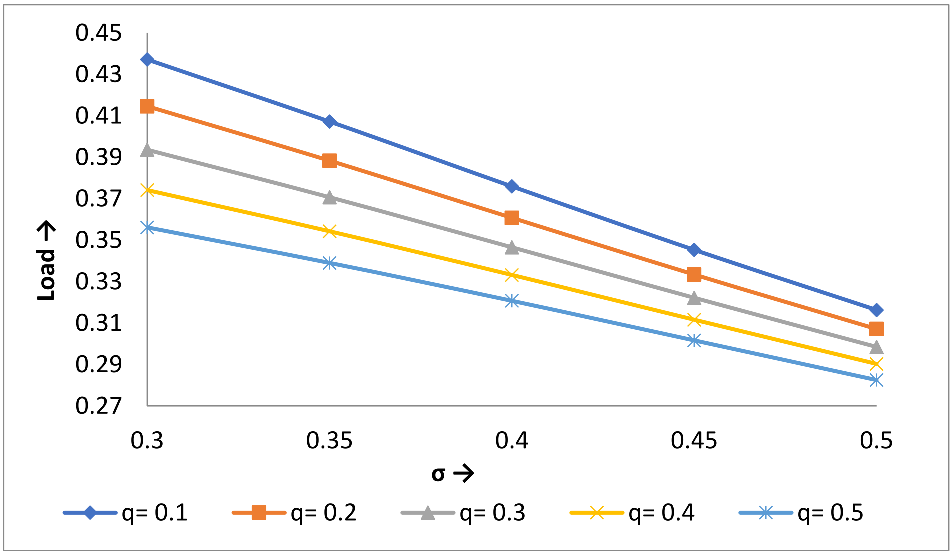

Figure 9. Variation of W with respect to σ and q.

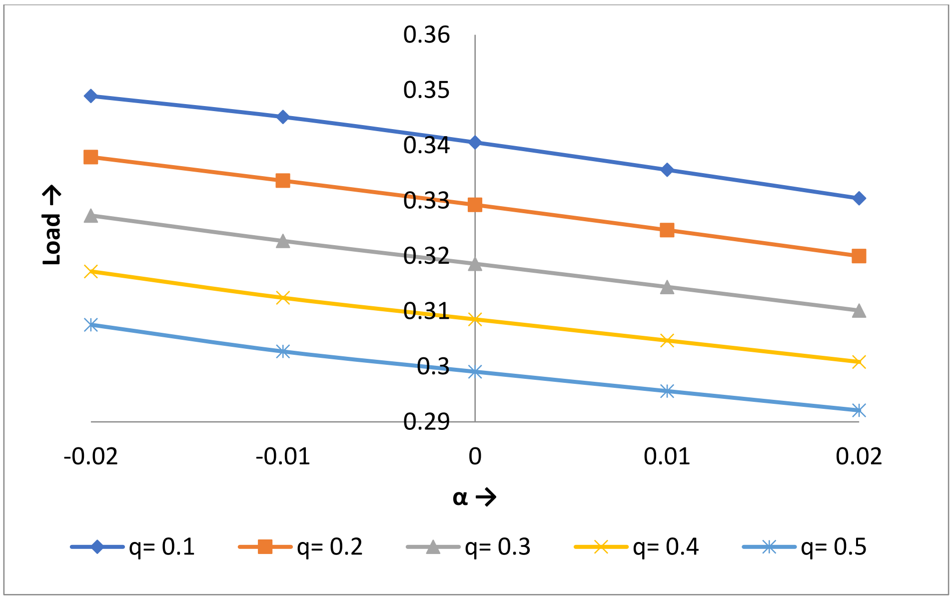

Figure 10. Variation of W with respect to α and q.

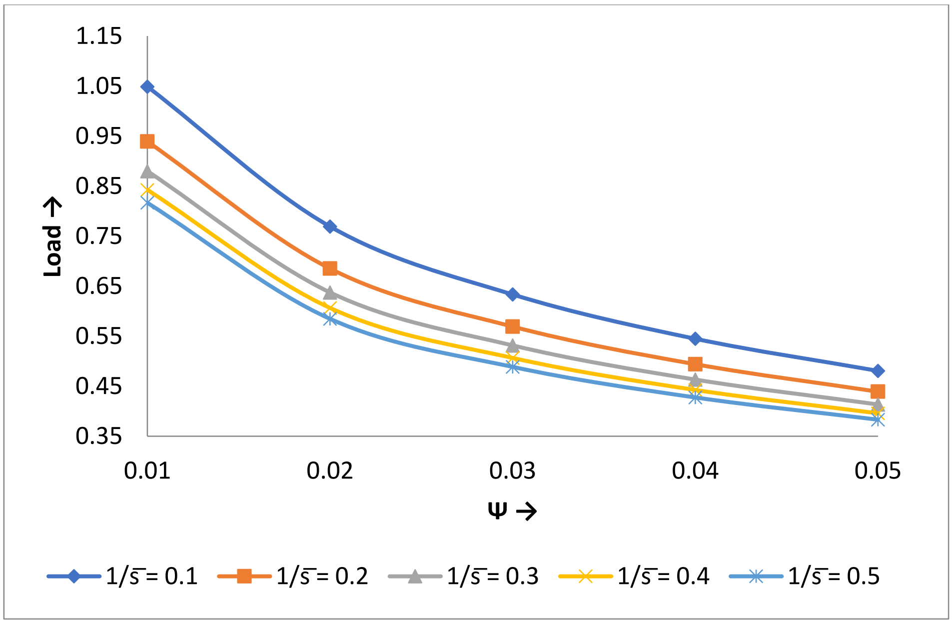

Figure 11. Variation of W with respect to ψ and $1/\bar{s}$.

It is seen from Eq. (12) and (13) that the dimensionless pressure increases by $\frac{1}{2}\mu^* X(1-X)$ while the non-dimensional form load carrying capacity moves up by $\mu^*/12$ in comparison with the usual fluid-based bearing system. Further, an increase in the magnetization parameter increases the load carrying capacity which can be seen from fig. (2-4). When the thermal effect increases as the load is decreased in fig. (5-7). The standard deviation causes load reduction which can be seen from (8-9). Better load can be observed for the negative values of skewness in fig (8). The trends of L.C.C. with respect to variance are similar to that of skewness in fig. (10). The significant effect of slip velocity on the performance characteristic is presented in fig (11). Where it is easily seen that an increased in slip velocity can cause considerable load reduction. This effect is less when negative variance occurs. In addition the magnetic fluid lubrication prevents much load reduction due to slip effect as compare the case of traditional lubrication.

Validation

The comparison Tables 1-6 presents the effect of magnetization, porosity, roughness parameter and slip parameter on load carrying capacity.

There is a considerable amount of thermal effect as can be seen from tables 2-5. However, the thermal effect is registered to be less due to the magnetic effect which is reflected in table-1.

Table 1. Comparison of load carrying capacity for μ^*.

Parameter	$\bar{\alpha}$=-0.05, $\bar{\sigma}$=0.4, $\bar{\varepsilon}$=-0.05, ψ=0.03, $1/\bar{s}$=0.3	
μ^*	**W with thermal effect**	**W without thermal effect**
0.1	0.308464	0.311004
0.3	0.31013	0.312671
0.5	0.311797	0.314338
0.7	0.313464	0.316004
0.9	0.31513	0.317671

Table 2 Comparison of load carrying capacity for ψ

Parameter	μ^*=0.1, $\bar{\alpha}$=-0.05, $\bar{\sigma}$=0.3, $\bar{\varepsilon}$=-0.05, $1/\bar{s}$=0.3	
ψ	**W with thermal effect**	**W without thermal effect**
0.01	0.437641	0.445178
0.02	0.397853	0.404526
0.03	0.364965	0.370954
0.04	0.337289	0.342725
0.05	0.313652	0.318628

Table 3. Comparison of load carrying capacity for $1/\bar{s}$

Parameter	μ^*=0.1, $\bar{\alpha}$=-0.05,$\bar{\sigma}$=0.3, $\bar{\varepsilon}$=-0.05, ψ=0.03	
$1/\bar{s}$	**W with thermal effect**	**W without thermal effect**
0.1	0.456604	0.464042
0.2	0.401466	0.408031
0.3	0.364965	0.370954
0.4	0.338989	0.344571
0.5	0.319547	0.324824

Table 4. Comparison of load carrying capacity for $\bar{\sigma}$

Parameter	μ^*=0.1, $\bar{\alpha}$=-0.05,$1/\bar{s}$ =0.3, $\bar{\varepsilon}$=-0.05, ψ=0.03	
$\bar{\sigma}$	**W with thermal effect**	**W without thermal effect**
0.1	0.461156	0.474542
0.2	0.419902	0.429899
0.3	0.364965	0.370954
0.4	0.308464	0.311004
0.5	0.257475	0.257501

Table 5. Comparison of load carrying capacity for $\bar{\varepsilon}$

Parameter	μ^*=0.1, $\bar{\alpha}$=-0.05,$\bar{\sigma}$ =0.3, $1/\bar{s} = 0.3$,ψ=0.03	
$\bar{\varepsilon}$	**W with thermal effect**	**W without thermal effect**
-0.02	0.353553	0.358742
-0.01	0.349894	0.354833
0	0.346305	0.351003
0.01	0.342787	0.347251
0.02	0.339336	0.343575

Table 6. Comparison of load carrying capacity for $\bar{\alpha}$

Parameter	μ^*=0.1, $\bar{\alpha}$=-0.05,$1/\bar{s}$ =0.3, $\bar{\varepsilon}$=-0.05, ψ=0.03	
$\bar{\alpha}$	**W with thermal effect**	**W without thermal effect**
-0.02	0.339934	0.344369
-0.01	0.332014	0.335982
0	0.324299	0.327824
0.01	0.316784	0.319887
0.02	0.309464	0.312167

4. CONCLUSION

This investigation establishes that performance of the bearing system remains a little better in the case of negatively skewed roughness. with suitable magnetic strength this type of bearing system can be made to perform better especially when this slip is law. further, when the thermal effect is at reduced level magnetization helps in counting the effect of surface roughness.

In addition some amount of load is supported by the bearing system even in the absence of flow. this doesn't happen in the case of traditional lubrication. Lastly this investigation may be some help for reduced the temperature rise if the designed is develop properly.

5. NOMENCLATURE

H = Magnitude of the magnetic field
H^* = Thickness of porous layer
$\bar{H}$ = Applied magnetic field
h_1, h_2 = Maximum and minimum value of h
$\bar{M}$ = Magnetization vector
k = Permeability of porous matrix
P = Lubricant film pressure
$\bar{p}$ = Dimensionless pressure
$\bar{q}$ = Magnetic fluid velocity
W = Load carrying capacity
$\bar{W}$ = Dimensionless load carrying capacity
$\bar{\alpha}$ = Non-dimensional variance
s = Slip parameter
$\bar{\varepsilon}$ = Skewness in dimensionless form
ξ = Slip coefficient
ρ = Magnetic fluid density
$\bar{\sigma}$ = Dimensionless standard deviation
ψ = Porosity

REFERENCES

[1] Christensen, H. and Tonder, K. C., *Tribology of rough surfaces:* stochastic models of hydrodynamic lubrication, SINTEF Report 10, 1969.

[2] Christensen, H. and Tonder, K.C., *Tribology of rough surfaces:* parametric study and comparison of lubrication models, SINTEF Report 22, 1969.

[3] Christensen, H. and Tonder, K.C., *The hydrodynamic lubrication of rough bearing surfaces of finite width*, in Proceedings of the ASME-ASLE Lubrication Conference, Paper no. 70-lub-7, Cincinnati, Ohio, USA, October 1970.

[4] Andharia, P.I. & Patel, M., *The Surface Roughness Effect of Transverse patterns on the Performance of Short Bearing*, International Journal of Scientific & Engineering Research,6(5), pp.82-87, 2015.

[5] Chawla, M. & Bhardwaj, R., *Surface Roughness Effect on Couple Stress Fluid Lubricated Porous Pivoted Slider Bearing*, Journal of Information and Optimization Sciences,37(1), pp.13-22,2016.

[6] Patel, N.D. & Deheri, G.M., *Hydromagnetic Lubrication of a rough Porous Parabolic Slider Bearing with Slip Velocity*, Journal of Applied Mechanical Engineering,3(3),pp.1-8. 2014 b.

[7] Shimpi, M.E. & Deheri, G.M., *Surface Roughness and Deformation Effect on the Performance of a Squeeze Film in Rotating Curved Porous Circular Plates with a Magnetic Fluid Lubricant*, Journal of Tribology and Surface Engineering,2(3/4),175-195.2011.

[8] Kato, K. Industrial *Tribology in the Past and Future*, Tribology Online, 6(1), pp.1-9, 2011. (Journal)

[9] Panchal, G.C., Patel, H.C., Patel, N.C. & Deheri, G.M., *The Effect of Magnetic Fluid Together with Transverse Roughness Pattern Parameters on the Performance of a Plane Slider Bearing*, Proceedings of the 1st International Conference on Numerical Modelling in Engineering, pp.151-167, 2018.

[10] Tipei, N., Theory of lubrication,3, Stanford, CA, Stanford University Press,1962. (Book)

[11] Siddangouda, A, Biradar, T.V.& Naduvinamani, N.B, *Combined effects of surface roughness and viscosity variation due to additives on long journal bearing*, Tribology-Materials, surface &Interfaces,7(1),21-35.2013.

[12] Cameron, A. *Basic Lubrication Theory*, Wiley Eastern Ltd. 1987, pp. 62.

[13] Bhat, M.V., *Hydrodynamic lubrication of an exponential porous slider bearing*, S.P. Uni. Res. J., 1, pp. 93-96.1982.

[14] Bhat, M.V. & Patel, R.R., *Analysis of an exponential porous slider bearing lubricated with magnetic fluid*, J. Eng. Technol. S.P. Uni., 6, pp. 19-21,1991.

[15] Shah, R.C & Bhat, M.V., *Analysis of a porous exponential slider bearing lubricated with a ferrofluid considering slip velocity*, J.BRAZ. Soc. Mech. Sci & Eng,25(3) Rio de Janeiro July/Sept.2003.

[16] Lin, Jaw-Ren & Hung, Chi-Ren, *Analysis of dynamic characteristics for wide slider bearings with an exponential film profile*, Journal of marine science and Technology,12(3), pp. 217-221, 2004.

[17] Patel, J.R., Deheri, G.M. & Patel, P.A., *Ferro-fluid Lubrication of Journal Bearing with Thermal Effects,* Mathematics Today, 34(A) (special Issue), 92-99. April. 2018.

[18] Patel, Sheetal A., Patel, A.R. & Deheri, G.M., *Combined Impact of Roughness and Slip Velocity on the Ferrofluid Lubrication of the porous exponential slider bearing*, International Journal of Advanced Research in Engineering and Technology, 9(3), pp.221-232, 2018.

[19] Neuringer, J. L. & Rosensweig, R. E., *Magnetic Fluid, Physics of fluids*, 7(12), pp. 1927-1938,1964.

[20] Sparrow E.M, Beavers G.S, & Hwang I.T., *Effects of velocity slip on porous walled squeeze films*, Journal of

Lubrication Technology. 94, pp. 260-265.1972.

[21] Agrawal, V.K., *Magnetic fluid based porous inclined slider bearing*. Wear,107:133-139,1986.

[22] Patel, Jimitkumar, Deheri, G.M. & Patel, S.J., *Ferrofluid lubrication of a rough porous secant-shaped slider bearing with slip velocity*, Journal of the Serbien Society for Computational Mechanics.11(1), pp. 69-81. 2017.

[23] Bhat, M.V. *Lubrication with A Magnetic Fluid*, Team Spirit India, Pvt, Ltd. India, 2003