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Performance of a Magnetic Fluid Based Squeeze Film in a Longitudinally Rough Parallel Plate Slider Bearing

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Abstract— **The purpose of this paper is to study the performance of a magnetic fluid based squeeze film between rough parallel plate slider bearing. The lubricant used here is a magnetic fluid and the external magnetic field is oblique to the lower plate. The bearing surfaces are assumed to be longitudinally rough. The roughness of the bearing surfaces is modeled by a stochastic random variable with nonzero mean, variance and skewness. Efforts have been made to average the associated Reynolds equation with respect to the random roughness parameter. The concerned non-dimensional equation is solved with appropriate boundary conditions in dimensionless form to obtain the pressure distribution. This is then used to get the expression for load carrying capacity. The results are presented in tabular form as well as graphically. The results shows that load carrying capacity decreases with respect to roughness mean and skewed roughness. It establishes that the performance of the bearing system gets enhanced due to the magnetic fluid lubricant. However, negatively skewed roughness increases the load carrying capacity. This increased load carrying capacity gets further increased when negative variance is involved. The study reveals that there is ample opportunities for improving the performance of the bearing system by choosing suitable values of magnetization parameter. This study makes it clear that the negative effect of the roughness can be minimized by the positive effect of the magnetization. Thus, while designing such bearing system, the roughness must be given sufficient consideration**.

Index Terms— **Load carrying capacity, Longitudinal roughness, Magnetic fluid, Parallel plates, Reynolds equation, Squeeze film.**

I. INTRODUCTION

As we know that bearing is a vital part of the machine and therefore, the performance of the bearing is directly connected to the performance of a machine element to a great extent. In the current age of technology of machines, lubrication and bearing play a pivotal role from the view of longevity of the machine and conservation of energy. Bearing surfaces, particularly after having some run in and wear, develop roughness. In many practical situations involving mechanical elements, in addition to the roughness of the contact surface pockets are also encountered in the form of dents and cavities which result from the wearing of material due to rotator motion in several cases.

The slider bearing is the simplest and frequently encountered among the hydrodynamic bearings. Probably, this is due to the fact that the expression for film thickness is simple and the boundary conditions required to be zero at the bearing ends are less complicated. In slider bearings, the film is non-diverging and continuous as a result of which the problem of negative pressure does not arise. Such bearings are designed to support the axial loads. Exact solutions of Reynolds equation for slider bearings with various simple film geometrics are presented in a number of books and research papers (Cameron [1], Archibald [2], Lord Rayleigh [4]). Prakash and Vij [5] analyzed the hydrodynamic lubrication of a plane inclined slider bearing taking different geometries into consideration. Patel and Gupta [6] extended the above analysis of Prakash and Vij [5] by incorporating slip velocity. They proved that in order to increase the performance of the bearing system the value of the slip parameter deserved to be minimized.

Because of the use of squeeze film slider bearings in clutch plates, automobile transmissions and domestic appliances many investigators (Prakash and Vij [5], Bhat [7], Bhat and Patel [8]) dealt with the problem of squeeze film slider bearing. The problem of squeeze film behavior between porous circular plates was analyzed by Murti [9]. The well known Morgan- Cameron approximation was used by Prakash and Vij [5] to study the problem of the squeeze film wherein various geometries were involved. The squeeze film behavior between rotating porous annular disks was studied by Wu [13].

The squeeze film lubrication phenomenon is observed in several applications such as gears, bearings, machine tools, rolling elements and automotive engines. The squeeze film action is also seen during approach of faces of disc clutches under lubricated condition. The squeeze film phenomenon arises when the two lubricating surfaces move towards each other in the normal direction and generates a positive pressure, and hence supports a load. This is due to the fact that a viscous lubricant present between the two surfaces cannot be instantaneously squeezed out when the two surfaces move towards each other and this action provides a cushioning effect in bearings. The squeeze film lubrication between two infinitely long parallel plates is studied by Cameron[21].

However, bearing surfaces could be roughened through the manufacturing process, the wear and the impulsive damage. To account for the effect of surface roughness, Christensen [14, 15] utilized a stochastic concept and introduced an averaging film model to lubricated surfaces with straightened roughness. The stochastic Reynolds type equations of rough bearing was derived and applied to investigate the effects of surface roughness on the bearing performance characteristics.

Agrawal [16] considered the configuration of Prakash and Vij [5] in the presence of a magnetic fluid lubricant and found its performance better than the one with conventional lubricant. Bhat and Deheri [17] extended the analysis of Agrawal [16] by investigating a magnetic fluid

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based porous composite slider bearing with its slider consisting of an inclined pad and a flat pad. Magnetic fluid increased the load carrying capacity and unaltered the friction and shifted the centre of pressure towards the inlet. Bhat and Deheri [17] discussed a general porous slider bearing with squeeze film formed by a magnetic fluid. Here also, it was found that the load carrying capacity rose sharply owing to the magnetic fluid lubricant. This is an attempt to study the performance of a magnetic fluid based squeeze film in a longitudinally rough parallel plate slider bearing.

II. ANALYSIS

The figure represents geometry and configuration of the bearing system. Here h stands for film thickness, X and Y are two axes and Z-axis is perpendicular to the plane of the paper at the origin.

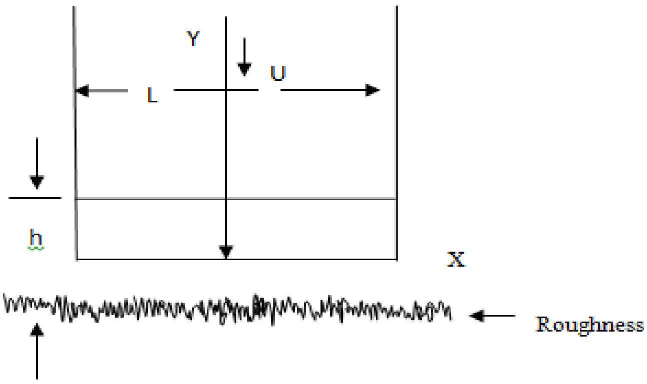

Fig.1 Bearing configuration

The lubricant film is considered to be isoviscous and incompressible and the flow is laminar. A magnetic fluid is used as the lubricant. The magnetic field is oblique to the lower plate as considered in Agrawal [16] and Deheri and Jimit [3]. In Prajapati [18] the effect of various forms of the magnitude of the magnetic field on the squeeze film performance has been discussed in details. Following Prajapati [18] and Bhat [19] the magnitude of the magnetic field is considered as

$$M^2 = KL^2\left(\frac{1}{2}-X\right)\sin\left(\frac{1}{2}+X\right) \tag{1}$$

where K is suitably chosen so as to have a magnetic field of strength over 10^5 Bhat and Deheri[8]. The bearing surfaces are assumed to be longitudinally rough. Under the usual assumptions of hydrodynamic lubrication the governing Reynolds' equation

(Bhat [7], Prajapati[18], Deheri, Andharia and Patel [21]) in the present case is given by

$$\frac{\partial}{\partial x}\left(h^3\frac{\partial}{\partial x}\left(p-\frac{\mu_0\bar{\mu}M^2}{2}\right)\right)+\frac{\partial}{\partial z}\left(h^3\frac{\partial}{\partial z}\left(p-\frac{\mu_0\bar{\mu}M^2}{2}\right)\right)=-12\eta U \tag{2}$$

Following Christensen and Tonder [15,16] the thickness h(x) of the lubricant film is considered as

$$h(x) = \bar{h}(x) + h_s \tag{3}$$

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

$$f(h_s) = \begin{cases}\frac{35}{32}\left(1-\frac{h_s^2}{c^2}\right)^3 & , -c \le h_s \le c\\ 0 & , elsewhere\end{cases} \tag{4}$$

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

$$\alpha = E(h_s),$$

$$\sigma^2 = E[(h_s-\alpha)^2] \text{ and } \varepsilon = E[(h_s-\alpha)^3]$$

where E denotes the expected value defined by

$$E(R) = \int_{-c}^{c} R\, f(h_s)\, dh_s \tag{5}$$

Applying averaging process discussed above the Reynolds' equation turns out to be

$$\frac{\partial}{\partial x}\left(\frac{1}{m(h)}\frac{\partial}{\partial x}\left(p-\frac{\mu_0\bar{\mu}M^2}{2}\right)\right)+\frac{\partial}{\partial z}\left(\frac{1}{m(h)}\frac{\partial}{\partial z}\left(p-\frac{\mu_0\bar{\mu}M^2}{2}\right)\right)=-12\eta U \tag{6}$$

where

$$m(h) = h^{-3} - 3h^{-4}\alpha + 6h^{-5}(\sigma^2+\alpha^2) - 10h^{-3}(\varepsilon + 3\sigma^2\alpha + \alpha^3) \tag{7}$$

while η is the lubricant viscosity, μ_0 is the magnetic susceptibility, $\bar{\mu}$ is the free space permeability.

One can conclude from the above equation that the magnetic fluid lubricant produces pressure in the film even in the absence of squeeze velocity unlike the conventional lubricant.

Assuming infinite breadth in the direction Z and integrating Equation (6) with respect to x one obtains,

$$p = \frac{\mu_0\bar{\mu}M^2}{2} - 6\eta U m(h)x^2 + c_1 x + c_2 \tag{8}$$

where c_1 and c_2 are constants of integrations.

The associated boundary conditions are $p = 0$ at $x = \pm\frac{L}{2}$

Using these conditions and finding integrating constants , we have

$$p = \frac{\mu_0\bar{\mu}M^2}{2} + 6\eta U m(h)L^2\left(\frac{1}{4}-\frac{x^2}{L^2}\right) \tag{9}$$

Introducing the non dimensional quantities,

$$\mu^* = \frac{\mu_0\bar{\mu}h^3}{\eta U}, P = \frac{h^3}{\eta U L^2}, X = \frac{x}{L}, m(h)h^3 = M(\bar{h})$$

$$M(\bar{h}) = 1 - 3\bar{\alpha} + 6(\bar{\sigma}^2+\bar{\alpha}^2) - 10(\bar{\varepsilon} + 3\bar{\sigma}^2\bar{\alpha} + \bar{\alpha}^3)$$

the pressure distribution in dimensionless form is obtained as

$$P = \frac{\mu^*}{2}\left(\frac{1}{2}-X\right)\sin\left(\frac{1}{2}+X\right) + 6\,M(\bar{h})\left(\frac{1}{4}-X^2\right) \tag{10}$$

The dimensionless load carrying capacity of the bearing is given by

$$W = \frac{h^3}{\eta U L^2}w = \frac{\mu^*}{2}(1-\sin 1) + M(\bar{h}) \tag{11}$$

where $w = \beta\int_{-\frac{1}{2}}^{\frac{1}{2}} p\, dx$

III. RESULTS AND DISCUSSIONS

It is clear that the non-dimensional pressure distribution is determined from Equation (10) while the distribution of dimensionless load carrying capacity is given by the equation (11). From this equation it is clearly seen that the dimensionless load carrying capacity is increased by $\frac{\mu^*}{2}(1-\sin 1)$ due to magnetic fluid lubricant.

To analyze quantitative effect of various parameter such as magnetization parameter μ^*, roughness parameters α , σ and ε on the performance of the bearing, dimensionless load carrying capacity is computed numerically for different values of these parameters. Results are presented in tables I, II and III and graphically in figures 2 to 7.

Table I shows the effect of roughness mean α on the dimensionless load carrying capacity for the various value of standard deviation parameter σ. This suggests that the load carrying capacity decreases sharply as value of σ and α increases. Table II gives the load carrying capacity for various values of σ and μ^*.This suggests that the load carrying capacity slightly increases as value of σ and μ^*increases. Table III gives the load carrying capacity for various values of α and μ^*.This suggests that the load carrying capacity decreases sharply as value of α and μ^*increases.

Table I
Variation of load carrying capacity with respect to σ and α

σ	α =-0.02	α =-0.01	α =0	α =0.01	α =0.02
0.02	2.178763	1.627563	1.256363	1.005163	0.813963
0.04	2.193163	1.638363	1.263563	1.008763	0.813963
0.06	2.217163	1.656363	1.275563	1.014763	0.813963
0.08	2.250763	1.681563	1.292363	1.023163	0.813963
0.1	2.293963	1.713963	1.313963	1.033963	0.813963

Table II
Variation of load carrying capacity with respect to σ and μ^*

σ	μ^*=0	μ^* =0.025	μ^*=0.05	μ^* =0.075	μ^*=0.1
0.02	0.81	0.811981	0.813963	0.815944	0.817925
0.04	0.81	0.811981	0.813963	0.815944	0.817925
0.06	0.81	0.811981	0.813963	0.815944	0.817925
0.08	0.81	0.811981	0.813963	0.815944	0.817925
0.1	0.81	0.811981	0.813963	0.815944	0.817925

Table III Variation of load carrying capacity with respect to α and μ^*

α	μ^*=0	μ^*=0.025	μ^*=0.05	μ^*=0.075	μ^*=0.1
-0.2	2.2132	2.2052	2.217163	2.219144	2.221125
-0.1	1.6524	1.6514	1.656363	1.658344	1.660325
0	1.2716	1.2716	1.275563	1.277544	1.279525
0.1	1.0108	1.0118	1.014763	1.016744	1.018725
0.2	0.81	0.818	0.813963	0.815944	0.817925

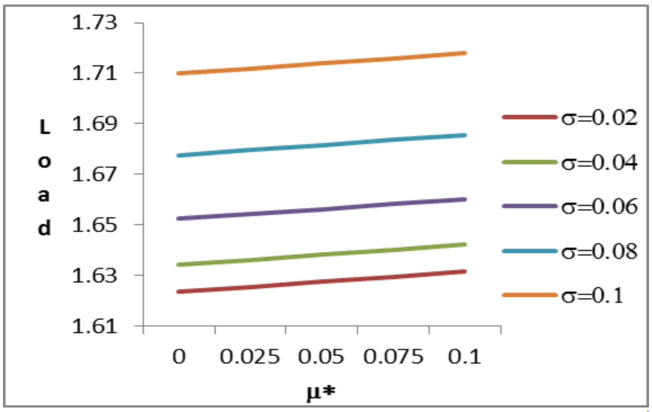

Fig. 2. Variation of load carrying capacity with respect to μ^* and σ

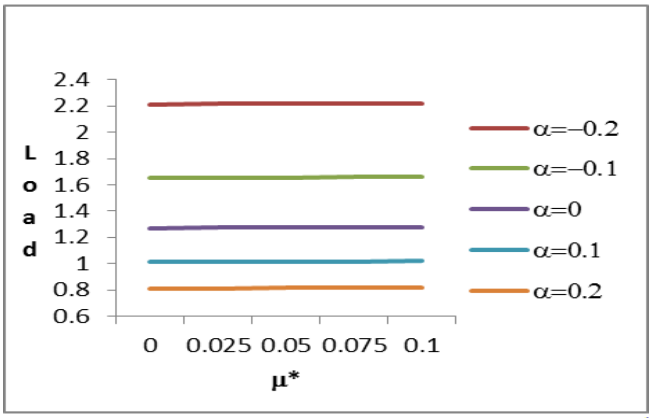

Fig. 3. Variation of load carrying capacity with respect to μ^* and α

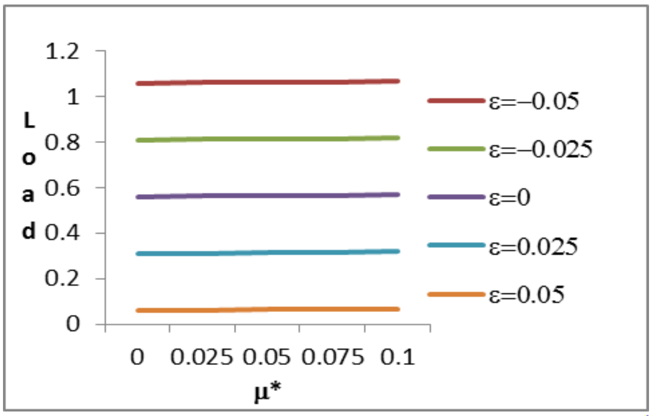

Fig. 4. Variation of load carrying capacity with respect to μ^* and ε

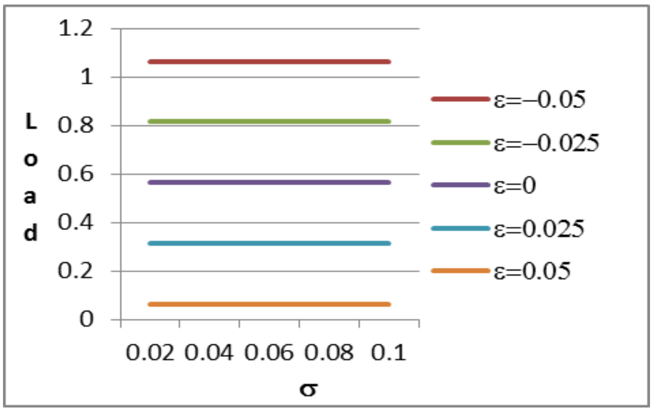

Fig. 5. Variation of load carrying capacity with respect to σ and ε

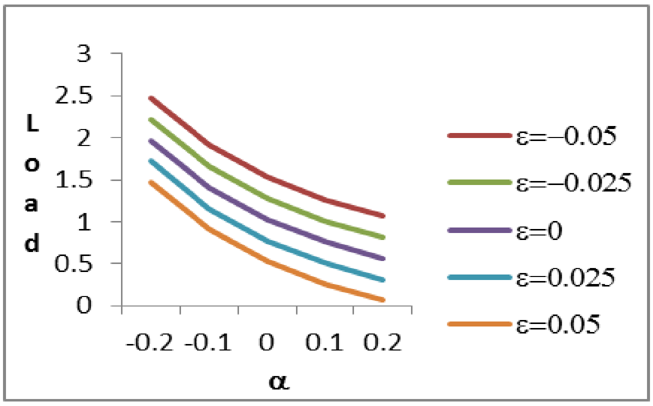

Fig. 6. Variation of load carrying capacity with respect to α and ε

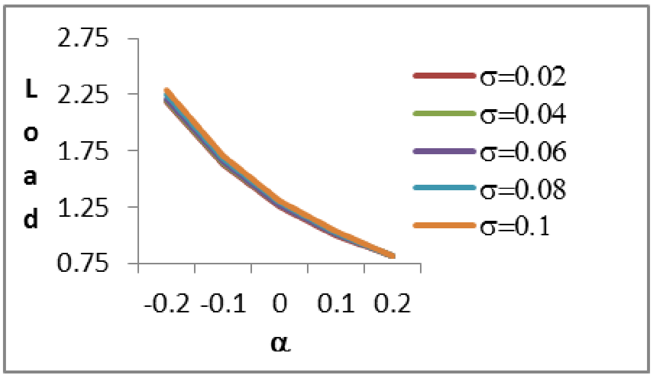

Fig. 7. Variation of load carrying capacity with respect to α and σ

Figure 2 represents the profile of load carrying capacity with respect to μ^* for various values of σ. It shows that load carrying capacity increases significantly for various values of μ^* and σ. Figures 3 and 4 present the profile of load carrying capacity with respect to μ^* for various values of α and ε. These figures suggest that the effect of roughness mean and skewness is almost negligible as far as load carrying capacity is concerned. Figures 5 also represents the profile of load carrying capacity with respect to α for various values of ε. These figure suggest that the effect of skewness is almost negligible as far as load carrying capacity is concerned. Figures 6 and 7 gives the effect of α on load carrying capacity for various values of σ and ε. These figure suggest that the load carrying capacity decreases sharply due to roughness mean α. This shows that the variation of non- dimensional load carrying capacity with respect to the magnetization parameter for different values of α , σ and ε. It is seen from these figures that the load carrying capacity increases sharply with increasing value of roughness parameter α , σ and ε.

The effect of standard deviation can be observed from Figure 6 and Figure 7. It should be observed that the standard deviation has an adverse effect in the sense that the load carrying capacity decreases considerably due to the standard deviation.

IV. CONCLUSIONS:

This study makes it mandatory that the roughness must be given due consideration while designing the bearing system even if suitable value of the magnetization parameter is chosen. This is all the more necessary from the bearings' life period point of view. This investigation further reveals that the bearing can support a load even when there is no flow.

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