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MAGNETIC FLUID BASED SQUEEZE FILM IN A ROUGH POROUS PARALLEL PLATE SLIDER BEARING

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ABSTRACT: An attempt has been made to study and analyze the performance of a magnetic fluid based squeeze film in a rough porous parallel plate slider bearing. The bearing surfaces are considered to be transversely rough. The random roughness of the bearing surfaces is characterized by a stochastic random variable with non-zero mean, variance and skewness. A magnetic fluid is taken as the lubricant. The associated Reynolds' equation is averaged with respect to the random roughness parameter and the concerned averaged differential equation is solved with appropriate boundary conditions to get the pressure distribution. This is then used to obtain the load carrying capacity. The graphical representation establishes that the performance of the bearing system gets enhanced due to the magnetic fluid lubricant. Porosity decreases the load carrying capacity and the bearing suffers due to transverse surface roughness in general. However, negatively skewed roughness increases the load carrying capacity. This increased load carrying capacity gets further increased when negative variance is involved. But the effect of standard deviation is adverse while the aspect ratio increases the load carrying capacity. The present study suggests that there exist sufficient scopes for improving the performance of the bearing system by choosing suitable values of magnetization parameter and aspect ratio in the case of negatively skewed roughness, especially, when negative variance occurs. This study makes it clear that the bearing can support a load even when there is no flow. It is indicated by this investigation that the roughness must be given due consideration while designing such bearing system, from bearing's life period point of view.
KEYWORDS: Slider Bearing, Magnetic Fluid, Roughness, Porosity, Load Carrying Capacity

❖ INTRODUCTION

It is well known that bearings are an integral part of the machine and therefore, the performance characteristics of the bearing affect the performance of a machine element to a great extent. In the modern age of machines, bearing and lubrication play a crucial 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. The analysis of hydrodynamic lubrication of a non-porous slider is a classical one, for instance, one can turn to Pinkus and Sternlitcht [22]. Exact solutions of Reynolds equation for slider bearings with various simple film geometrics are presented in a number of books and research papers (Cameron [10], Archibald [3], Lord Rayleigh [19]). Prakash and Vij [24] analyzed the hydrodynamic lubrication of a plane inclined slider bearing taking different geometries into consideration. Here it was shown that porosity decreased the load carrying capacity and friction. Patel and Gupta [21] extended the above analysis of Prakash and Vij [24] 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 [24], Bhat [8], Bhat and Patel [9]) dealt with the problem of squeeze film slider bearing. The problem of squeeze film porous metal bearings has been discussed by several investigators (Murti [20], Ting [27,26], Wu[29,30]). The problem of squeeze film behavior between porous circular plates was analyzed by Murti [20]. The well known Morgan-Cameron approximation was used by Prakash and Vij [24] 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 [30].

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 [16, 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. Several investigators have adopted a stochastic approach to model the random roughness (Tzeng and Seibel [28], Christensen and Tonder [13, 12, 11]). Christensen and Tonder [13, 12, 11] presented a comprehensive general analysis for surface roughness (both transverse and longitudinal) based on a general probability density function by modifying and developing the approach of Tzeng and Seibel [28]. Subsequently, on the ground of this Christensen and Tonder's stochastic model, many researches have been carried out to study the effect of surface roughness on hydrodynamic mechanisms, such as the works in the hydrodynamics journal bearing by Guha [18] and Taranga et. al. [25], the hydrodynamic slider bearings by Christensen and Tonder [14], the squeeze film spherical bearings by Andharia et. al.[2].

The use of magnetic fluid as a lubricant results in the overall improvement of performance characteristics. Hence, from the industry point of view the study of porous metal lubrication involving the above discussed machine an element in presence of a magnetic fluid is of paramount importance.

In all the above studies conventional lubricants were used. The use of magnetic fluid as a lubricant modifying the performance of the bearing has been very well recognized. The magnetic fluid is a stable suspension of small particles of ferro-magnetic materials in a base fluid. For the purpose of ensuring the colloid stability, a surfactant of polymer (such as oleic acid) is usually introduced into the suspension. This will create around each single particle a coating layer to prevent the agglomeration of the particles by the magnetic field effect or by the molecular attraction. When a magnetic field is applied to the magnetic fluid, each particle experiences a force that depends on the magnetization of the magnetic materials of the particles and on the strength of the applied field.

Agrawal [1] considered the configuration of Prakash and Vij [24] in the presence of a magnetic fluid lubricant and found its performance better than the one with conventional lubricant. Bhat and Deheri [6] extended the analysis of Agrawal [1] by investigating a magnetic fluid 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 [7] 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.

Here it has been proposed to study and analyze the performance of a transversely rough parallel plate slider bearing, in the presence of a magnetic fluid lubricant.

❖ ANALYSIS

The geometry and configuration of the bearing system is presented in the Figure given below. In this figure 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 O. U is the velocity with which top part approaches the bottom part and the velocity of approach is $U = \frac{dh}{dt}$.

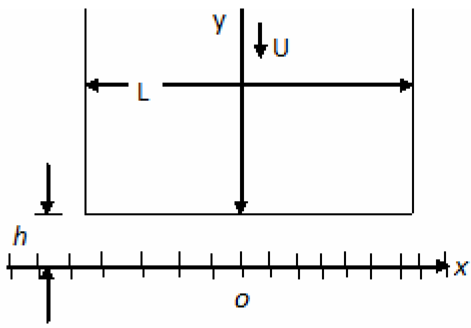

Figure 1. Configuration of bearing system

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 [1]. In Prajapati [23] the effect of various forms of the magnitude of the magnetic field on the squeeze film performance has been discussed in details. Following Prajapati [23] and Bhat [5] 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[7]].

The bearing surfaces are assumed to be transversely rough. Following Christensen and Tonder[13, 12, 11]the thickness *h(x)* of the lubricant film is considered as

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

where $\overline{h}(x)$ 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

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

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

$$\alpha = E(h_s)$$

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

where E denotes the expected value defined by

$$E(R) = \int_{-c}^{c} R f(h_s) dh_s \tag{2}$$

Thus, under the usual assumptions of hydrodynamic lubrication the governing Reynolds' equation (Bhat [5], Prajapati[23], Deheri, Andharia and Patel [17]) turns out to be

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

where $g(h) = h^3 + 3\alpha h^2 + 3h\left(\alpha^2 + \sigma^2\right) + \varepsilon + 3\sigma^2\alpha + \alpha^3 + 12\varphi H$, 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 (3) with respect to X one obtains,

$$p = -\frac{6\eta U}{g(h)} x^2 + c_1 x + c_2 \tag{4}$$

The associated boundary conditions are

$$p = 0 \quad \text{at} \quad x = \pm\frac{L}{2}$$

In view of these boundary conditions Equation (4) leads to

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

Introducing the non dimensional quantities,

$$\mu^* = \frac{\mu_0 \bar{\mu} h^3}{\eta U}, \quad P = \frac{h^3}{\eta U L^2} p, \quad X = \frac{x}{L}, \quad A = -\frac{1}{g(\bar{h})}, \quad \psi = \frac{12\varphi H}{h^3}$$

$$g(\bar{h}) = 1 + 3\bar{\alpha} + 3\left(\bar{\alpha}^2 + \bar{\sigma}^2\right) + \bar{\varepsilon} + 3\bar{\sigma}^2\bar{\alpha} + \bar{\alpha}^3 + 12\psi$$

the pressure distribution in dimensionless form is obtained as

$$P = \frac{\mu^*}{2}\left(\frac{1}{2} - X\right)\sin\left(\frac{1}{2} + X\right) + 6A\left(\frac{1}{4} - X^2\right) \tag{5}$$

The dimensionless load carrying capacity of the bearing is given by

$$W = \frac{h^3}{\eta U L^2} w = (1 - \sin 1)\frac{\mu^*}{2} + A \tag{6}$$

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

while B is finite breadth along Z-axis.

RESULTS AND DISCUSSION

It is seen clearly that the non-dimensional pressure distribution is determined from Equation (5) while Equation (6) gives the distribution of dimensionless load carrying capacity. These two Equations suggest that the pressure increase by

$$\frac{\mu^*}{2}\left(\frac{1}{2}-X\right)\sin\left(\frac{1}{2}+X\right)$$

while the increase in the load carrying capacity turns out to be $0.079264507\mu^*$ as compared to the case of conventional lubricant. Taking the roughness parameters to be zero this study reduces to the squeeze film performance of a porous parallel plates slider bearing under the presence of a magnetic fluid lubricant. Taking only μ^* to be zero this investigation leads to the squeeze film performance of a porous rough parallel plate slider bearing. Lastly, setting the roughness parameters and the magnetization parameter to be zero one obtains the deliberation carried out in Basu et. al.[4].

First of all it is seen that there is symmetry of the pressure distribution with respect to X for various values of the magnetization parameter [Figure 2]. This figure underlines that the effect of magnetization parameter is negligible up to $\mu^*=0.001$, so far as pressure distribution is concerned.

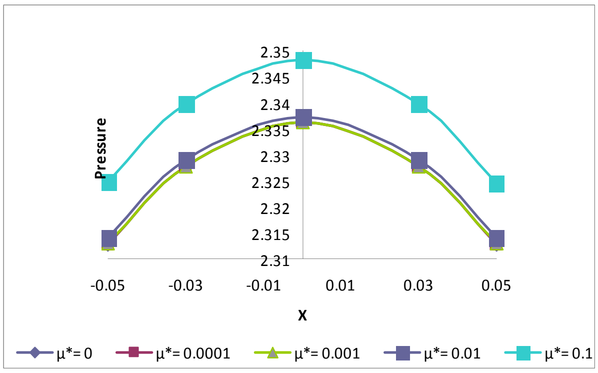

Figure 2. Non dimensional pressure distribution with respect to X and μ^*.

Figures 3-6 depict the variation of non-dimensional load carrying capacity with respect to the magnetization parameter for different values of $\bar{\alpha}$, $\bar{\sigma}$, $\bar{\varepsilon}$ and ψ. It is clearly observed from these figures that the load carrying capacity increases sharply due to the magnetization parameter. However, the porosity effect is negligible up to $\psi=0.001$.

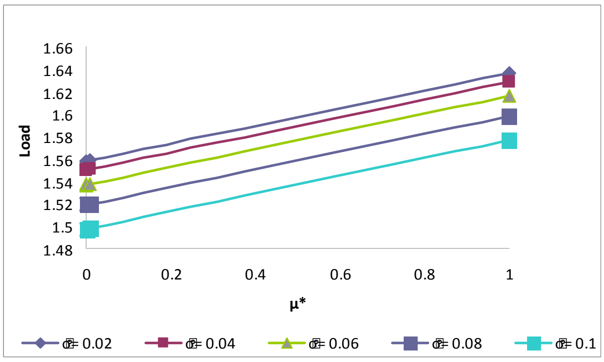

Figure 3. Variation of Load carrying capacity with respect to μ^* and $\bar{\sigma}$

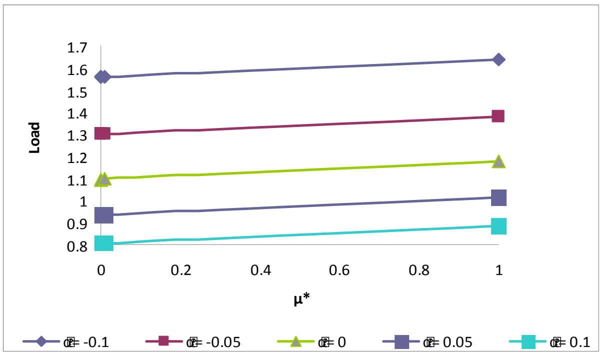

Figure 4. Variation of Load carrying capacity with respect to μ^* and $\bar{\alpha}$.

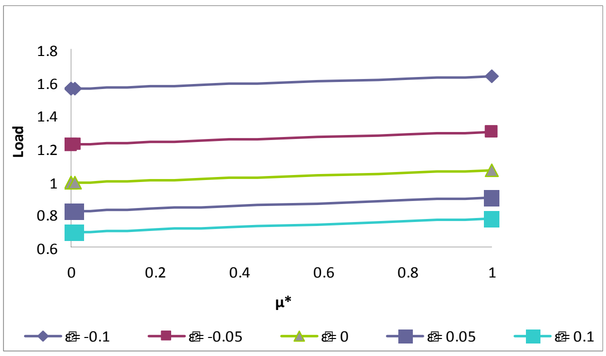

Figure 5. Variation of Load carrying capacity with respect to μ^* and $\bar{\varepsilon}$

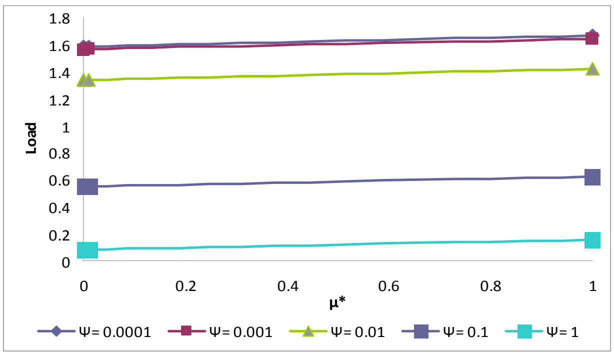

Figure 6.Variation of Load carrying capacity with respect to μ^* and ψ

In Figures 7-9 one can have the variation of load carrying capacity with respect to $\bar{\alpha}$ for various values of $\bar{\sigma}$, $\bar{\varepsilon}$ and ψ respectively. It is found that variance (+ve) decreases the load carrying capacity while variance (-ve) tends to increase the load carrying capacity. But the effect of porosity is negligible up to 0.001. Further, Figure 7 indicates that the effect of standard deviation with respect to the variance is negligible.

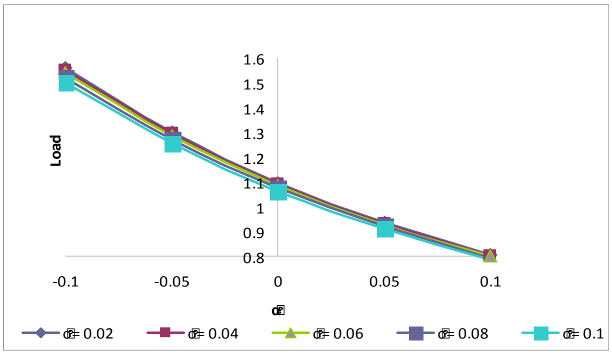

Figure 7. Variation of Load carrying capacity with respect to $\overline{\alpha}$ and $\overline{\sigma}$.

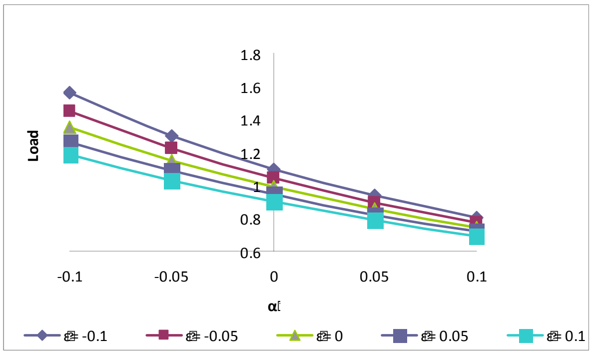

Figure 8. Variation of Load carrying capacity with respect to $\overline{\alpha}$ and $\overline{\varepsilon}$.

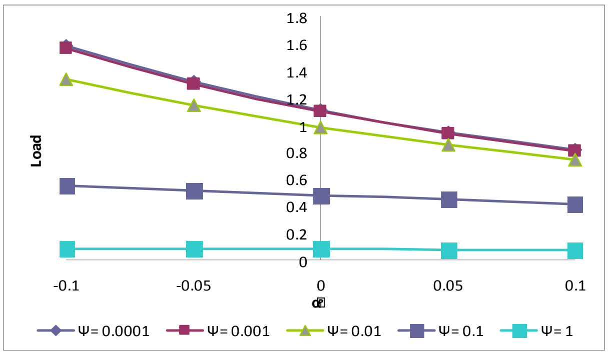

Figure 9. Variation of Load carrying capacity with respect to $\overline{\alpha}$ and ψ.

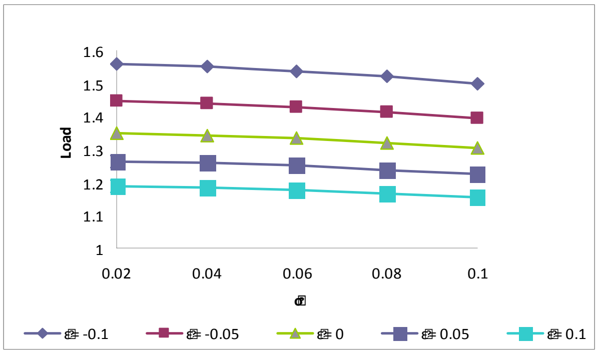

Figure 10. Variation of Load carrying capacity with respect to $\overline{\sigma}$ and $\overline{\varepsilon}$.

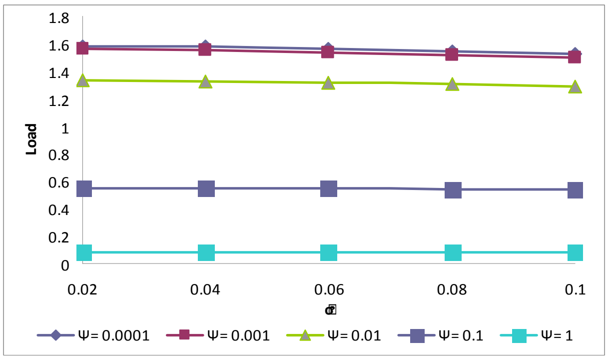

Figure 11. Variation of Load carrying capacity with respect to $\overline{\sigma}$ and ψ

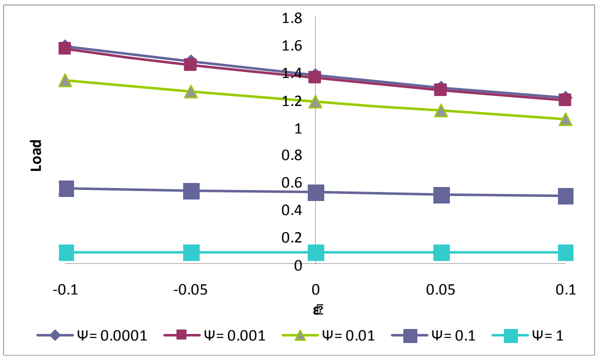

Figure 12. Variation of Load carrying capacity with respect to $\overline{\varepsilon}$ and ψ

The effect of standard deviation can be had from Figure 10 and Figure 11. It is noticed that the standard deviation has an adverse effect in the sense that the load carrying capacity decreases considerably due to the standard deviation.

Lastly, from Figure 12 and the graphs related to the skewness it is observed that the skewness follows the trends of the variance. Here also, the effect of porosity is negligible up to 0.001 which is suggested by Figure 12.

Some of these figures reveal that the decreased load carrying capacity due to porosity gets further decreased owing to the standard deviation. Besides, the negatively skewed roughness registers a considerable positive effect especially, when the negative variance occurs. Thus, this investigation makes it clear that the negative effect of the porosity and standard deviation can be compensated to a sufficiently large extent by the positive effect of the magnetization parameter in the case of negatively skewed roughness particularly, when negative variance is involved.

❖ 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.

❖ NOMENCLATURE

Symbol	Name	Symbol	Name
B	*Width of bearing*	α	*Variance*
h_1	*Maximum film thickness*	$\bar{\alpha}$	*Dimensionless Varience*
h_2	*Minimum film thickness*	ε	*Skewness*
h	*Fluid film thickness at any point*	$\bar{\varepsilon}$	*Non-dimensional skewness*
L	*Length of the bearing*	μ	*Dynamic fluid viscosity*
m	*Aspect ratio*	μ^*	*Dimensionless magnetization parameter*
M	*Magnitude of the Magnetic field*	σ	*Standard deviation*
p	*Lubricant pressure*	$\bar{\sigma}$	*Dimensionless standard deviation*
P	*Dimensionless pressure*	ψ	*Non-dimensional porosity*
w	*Load carrying capacity*	ϕ	*Permeability of the porous matrix*
W	*Dimensionless load carrying capacity*		

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