Performance of a Magnetic Fluid-based Short Bearing

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Abstract: An effort has been made to study and analyze the performance of a magnetic fluid based infinitely short hydrodynamic slider bearing. The Reynolds' equation is solved with appropriate boundary conditions. The expressions for various performance characteristics such as pressure, load carrying capacity and friction are obtained. Results are presented graphically. It is clearly seen that the load carrying capacity increases considerably due to the magnetic fluid lubricant. Further, the film thickness ratio increases the load carrying capacity. It is found that the load carrying capacity increases as the ratio of the length to outlet film thickness increases while it decreases with respect to the increasing values of the ratio of the width to the outlet film thickness. In addition, it is investigated that the magnetic fluid lubricant unalters the friction. Lastly, this article makes it clear that the negative effect induced by the ratio of the width to the outlet film thickness can be neutralized up to a considerably large extent by the combined positive effect of the magnetization parameter, the film thickness ratio and the ratio of the length to outlet film thickness. This study provides ample scope for extending the life period of the bearing system.

Keywords: short bearing; magnetic fluid; Reynolds' equation; pressure; load carrying capacity

1 Introduction

The classical analysis of the hydrodynamic lubrication of slider bearings was presented by Pinkus and Sternlicht [1]. Exact solutions of Reynolds' equation for slider bearings with various simple film geometries were discussed in a number of books and research papers (Cameron [2], Archibald [3], Lord Rayleigh [4], Charnes and Saibel [5], Basu, Sengupta and Ahuja [6], Majumdar [7], Hamrock [8], Gross, Matsch, Castelli, Eshel, Vohr and Wildmann [9]). Prakash and Vij [10] analyzed the hydrodynamic lubrication of a plane slider bearing taking different geometries into consideration. Bagci and Singh [11] dealt with the optimal designs for hydrodynamic lubrication of finite slider bearings considering the effect of one dimensional film shape. Osterle and Saibel [12] studied the effect of bearing deformation in slider bearing lubrication. Here it was concluded that the performance was mostly adversely affected in the sense that the load carrying capacity decreased. Patel and Gupta [13] considered the effect of slip velocity on hydrodynamic lubrication of a porous slider bearing and proved that the load carrying capacity decreased due to the velocity slip. Abramovitz [14] investigated the performance of a pivoted slider bearing with convex pad surfaces. Here it was concluded that the load carrying capacity was more for a convex pad than for a flat one and that such a bearing might be centrally pivoted. Maday [15] extended the analysis of Rayleigh [4] to verify that the optimum slider contains only one step. The design of the optimum one dimensional slider bearing in terms of load carrying capacity was also investigated by McAllister, Rohde and McAllister [16]. Morgan and Cameron [17] were the first to present an approximate analytical solution for the performance characteristics of a porous metal bearing. Later on, Rouleau [18] obtained an exact solution of the above problem.

All these above studies considered conventional lubricants. Agrawal [19] dealt with the configuration of Prakash and Vij [10] with a magnetic fluid lubricant and found its performance better than the one with conventional lubricant. Bhat and Deheri [20] modified and extended the analysis of Agrawal [19] by considering a magnetic fluid based porous composite slider bearing with its slider consisting of an inclined pad and a flat pad. Here it was established that magnetic fluid increased the load carrying capacity, unaltered the friction, decreased the coefficient of friction and shifted the centre of pressure towards the inlet. Prajapati [21] investigated the performance of a magnetic fluid based porous inclined slider bearing with velocity slip and concluded that the magnetic fluid lubricant minimized the negative effect of the velocity slip. Also, Prajapati [22] analyzed the hydrodynamic lubrication of an inclined porous slider bearing with variable porous matrix thickness. The analysis presented was modified and extended by Deheri, Patel and Patel [23] to study the effect of transverse surface roughness on the above configuration in the presence of a magnetic fluid lubricant.

Here an attempt has been made to study and analyze the performance of a magnetic fluid based short bearing wherein, the magnitude associated with the magnetic field is represented by a trigonometric function.

2 Analysis

The configuration of the bearing which is infinitely short in Z – direction is shown below.

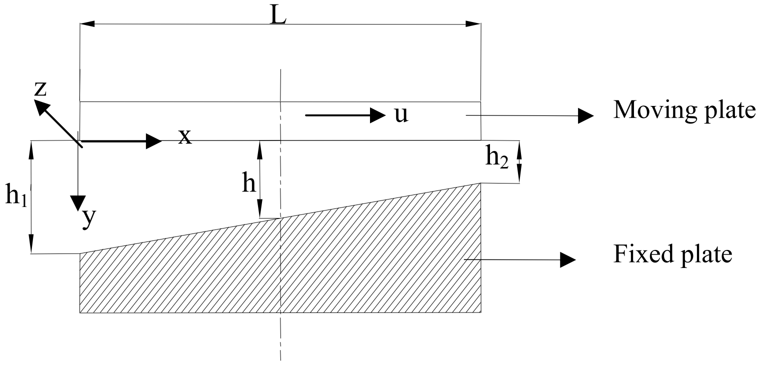

Figure 1
Configuration of the bearing system

The slider moves with the uniform velocity u in the X – direction. The length of the bearing is L and the breadth B is in Z – direction, where B<<L. The dimension B being very small the pressure gradient $\partial p/\partial z$ is much larger than the pressure gradient $\partial p/\partial x$ and hence the latter can be neglected. The lubricant film is considered to be isoviscous and incompressible and the flow is laminar. The magnetic fluid is a suspension of solid magnetic particles of diameter approximately 3-15 nanometer stabilized by a surfactant in a liquid carrier. By applying an external magnetic field these fluids can be confined, positioned, shaped and controlled as desired. The magnetic field is oblique to the stator as in Agrawal [19]. The magnetic field is

$$\overline{H} = (H(z)\cos\phi, 0, H(z)\sin\phi);\ \phi = \phi(x, z) \tag{1}$$

and the inclination angle is determined from

$$\cot\phi\frac{\partial\phi}{\partial z} + \frac{\partial\phi}{\partial x} = -\frac{1}{H}\frac{dH}{dz} \tag{2}$$

Following the discussions conducted in Verma [25], Prajapati [22] and Bhat and Deheri [26] we consider the magnitude expressed in the form of

$$H^2 = kB^2\cos\left(\frac{\pi z}{B}\right) \tag{3}$$

where k is chosen to suit the dimensions of both sides and the strength of the magnetic field. Under usual assumptions of magnetohydrodynamic lubrication the governing Reynolds' equation (Agrawal [19], Prajapati [22], Bhat and Deheri [27]) turns out to be

$$\frac{d^2}{dz^2}\left(p-\frac{\mu_0\bar{\mu}H^2}{2}\right)=\frac{6\mu u}{h_2^3\left\{1+m\left(1-\frac{x}{L}\right)\right\}^3}\frac{dh}{dx} \tag{4}$$

where μ_0 is the magnetic susceptibility, $\bar{\mu}$ is free space permeability, μ is lubricant viscosity and m is the aspect ratio. The associated boundary conditions are

$$p = 0 \text{ at } z = \pm(B/2)$$

and

$$\frac{dp}{dz}=0 \text{ at } z = 0 \tag{5}$$

Integrating equation (1) with the boundary conditions (2) one obtains the pressure distribution as

$$p=\frac{\mu_0\bar{\mu}kB^2\cos\left(\frac{\pi z}{B}\right)}{2}+\frac{3\mu muh_2}{Lh_2^3\left\{1+m\left(1-\frac{x}{L}\right)\right\}^3}\left(\frac{B^2}{4}-z^2\right) \tag{6}$$

Introducing the dimensionless quantities

$$m=\frac{h_1-h_2}{h_2} \qquad X=\frac{x}{L} \qquad P=\frac{h_2^3p}{\mu uB^2}$$

$$\mu^*=\frac{h_2^3k\mu_0\bar{\mu}}{\mu u} \qquad Y=\frac{y}{h} \qquad Z=\frac{z}{B}$$

one gets the pressure distribution in dimensionless form as

$$P=\frac{\mu^*\cos(\pi Z)}{2}+\frac{3mh_2}{L\{1+m(1-X)\}^3}\left(\frac{1}{4}-Z^2\right) \tag{7}$$

The non-dimensional load carrying capacity is given by

$$W=\frac{h_2^3w\pi}{\mu uB^4}=\pi\int_{-\frac{1}{2}}^{\frac{1}{2}}\int_0^1 P(X,Z)\,dX\,dZ \tag{8}$$

Therefore, the dimensionless load carrying capacity of the bearing is given by

$$W = \frac{\mu^{*} L}{h_2} \frac{h_2}{B} + \frac{\pi h_2}{4B}\left[1 - \frac{h_2^2}{h_1^2}\right] \tag{9}$$

The frictional force $\bar{F}$ per unit width of the lower plane of the moving plate is obtained as

$$\bar{F} = \int_{-1/2}^{1/2} \bar{\tau}\, dZ \tag{10}$$

where

$$\bar{\tau} = \left(\frac{h_2}{\mu u}\right)\tau \text{ is non-dimensional shearing stress} \tag{11}$$

while

$$\tau = \frac{dp}{dz}\left(y - \frac{h}{2}\right) + \frac{\mu u}{h} \tag{12}$$

On simplifications this yields,

$$\bar{\tau} = \frac{dP}{dZ}\frac{B}{h_2}\{1 + m(1 - X)\}\left(Y - \frac{1}{2}\right) + \frac{1}{\{1 + m(1 - X)\}} \tag{13}$$

At Y = 0 (at moving plate), we find that

$$\bar{\tau} = \frac{B\mu^{*}\pi}{4h_2}\{1 + m(1 - X)\}\sin(\pi Z) + \frac{3mh_2 ZB}{Lh_2\{1 + m(1 - X)\}^2} + \frac{1}{\{1 + m(1 - X)\}} \tag{14}$$

Thus, in non-dimensional form the frictional force assumes the form

$$F_0 = \frac{1}{\{1 + m(1 - X)\}} \tag{15}$$

Further, at Y = 1 (at fixed plate), one obtains that

$$\bar{\tau} = -\frac{B\mu^{*}\pi}{4h_2}\{1 + m(1 - X)\}\sin(\pi Z) - \frac{3mh_2 ZB}{Lh_2\{1 + m(1 - X)\}^2} + \frac{1}{\{1 + m(1 - X)\}} \tag{16}$$

Lastly, in dimensionless form the frictional force comes out to be

$$F_1 = \frac{1}{\{1 + m(1 - X)\}} \tag{17}$$

3 Results and Discussion

Equations (7) and (9) present the variation of pressure distribution and load carrying capacity while the frictional force is determined from Equation (10). A comparison with the conventional lubricants indicates that the non-dimensional pressure increases by

$$\frac{\mu^{*}\cos(\pi Z)}{2}$$

while the load carrying capacity enhances by

$$\frac{\mu^{*}\left(L/h_{2}\right)}{\left(B/h_{2}\right)}.$$

Furthermore, a closed scrutiny of the results presented here and the results of the investigation carried out by Patel [24] reveals that the load carrying capacity is approximately four times more here. Besides, the magnetic fluid lubricant unalters the friction which is a clear message from Equations (15) and (17).

In Figures 2-4 we have the analytical non-dimensional pressure distribution with respect to Z for different values of aspect ratio, the ratio of length to outlet film thickness and X respectively. These figures suggest that the pressure is more in the case of ratio of the length to outlet film thickness in the sense that the non-dimensional pressure is more with respect to the ratio of length to outlet film thickness as compared to that of m and X. Besides, it is observed that the pressure increases significantly with respect to the aspect ratio up to the value 0.75 and then it increases marginally.

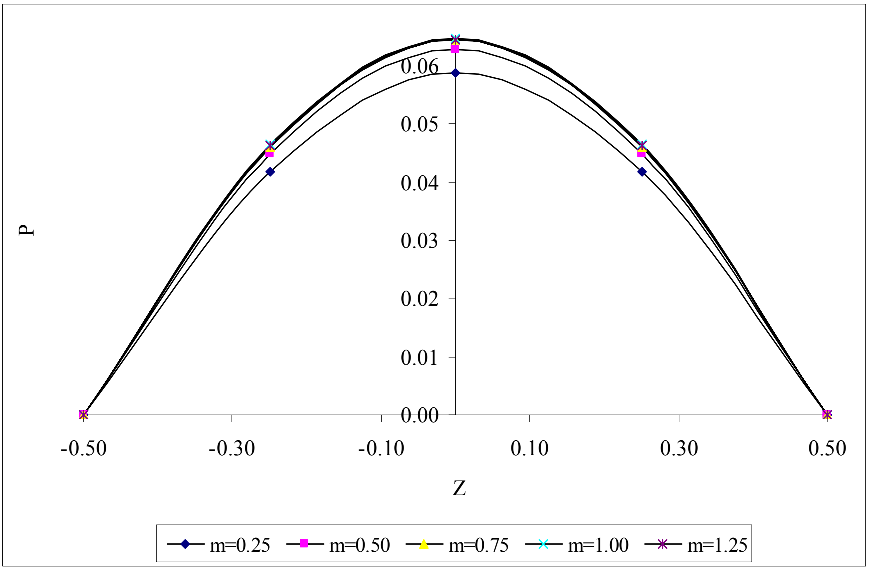

Figure 2
Variation of pressure with respect to Z and m for $\mu^* = 0.001$

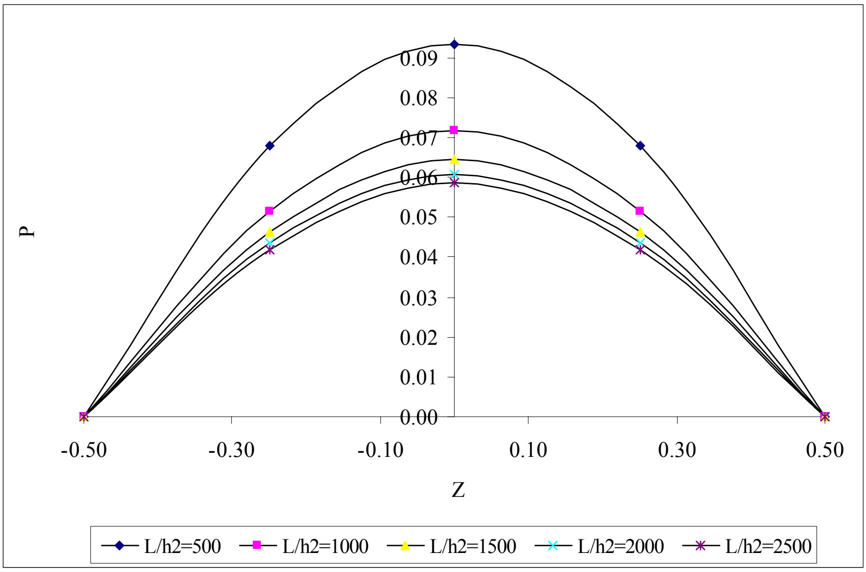

Figure 3
Variation of pressure with respect to Z and L/h_2 for $\mu^* = 0.001$

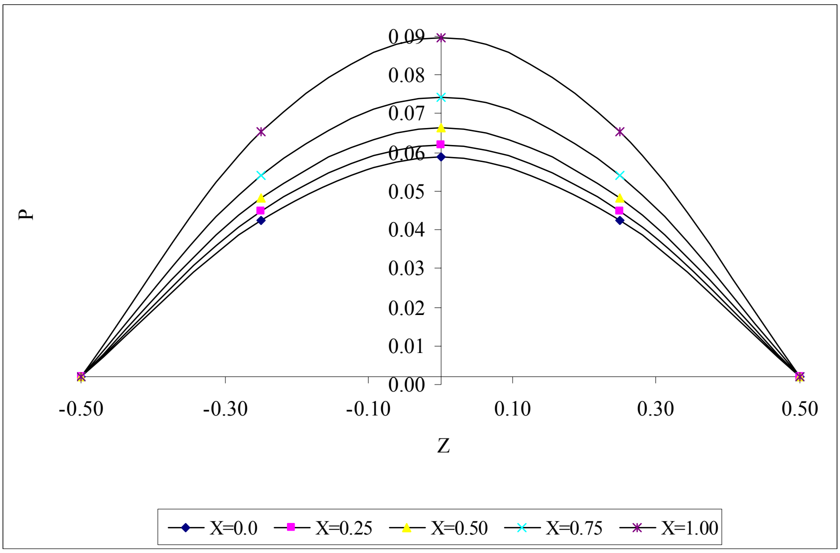

Figure 4
Variation of pressure with respect to Z and X for $\mu^* = 0.001$

In Figures 5-7 the magnitude of the pressure at a given Z coordinate is depicted in function of μ^* for various values of m, L/h_2 and X. These figures reveal that the effect of the ratio of length to outlet film thickness, X and m are negligible with respect to the magnetization parameter.

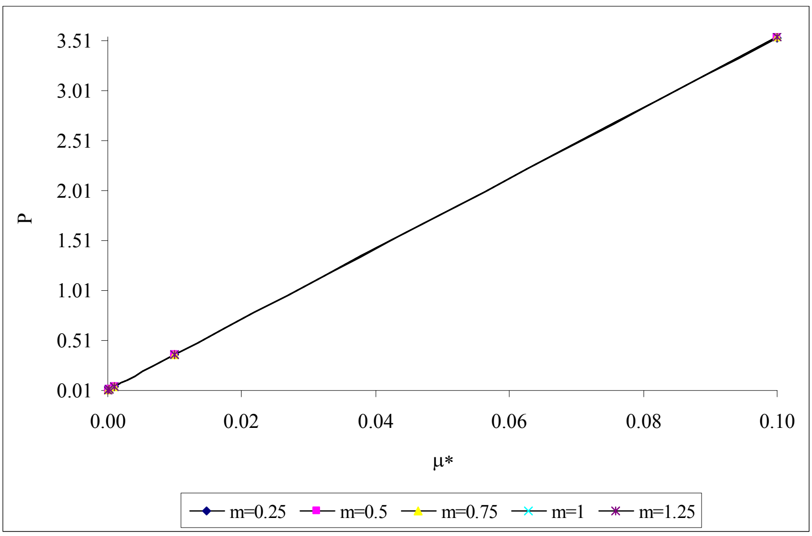

Figure 5
Variation of pressure with respect to μ^* and m for Z = 0.25

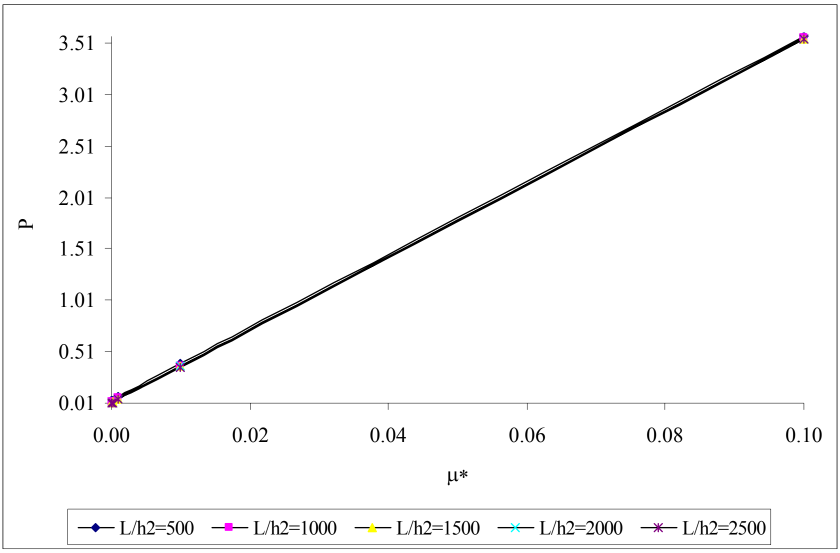

Figure 6
Variation of pressure with respect to μ* and L/h_2 for Z = 0.25

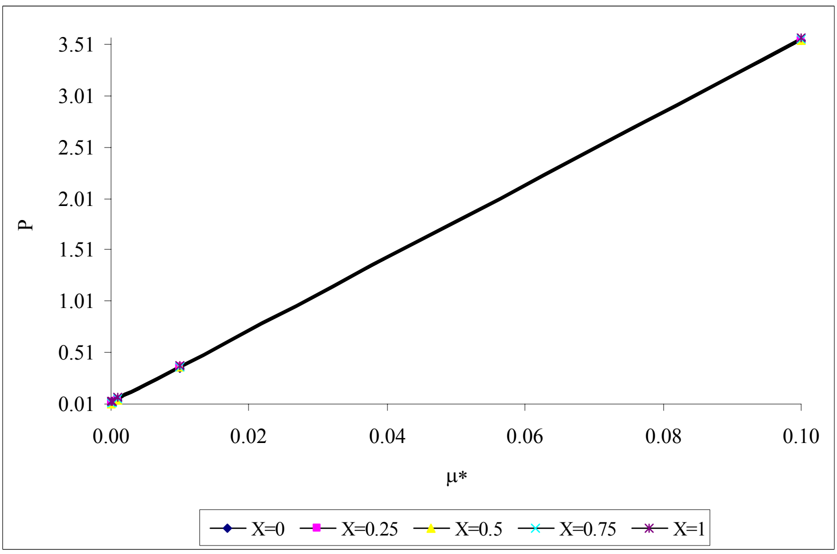

Figure 7
Variation of pressure with respect to μ* and X for Z = 0.25

The variation of load carrying capacity with respect to the magnetization parameter for various values of the ratio of the length to outlet film thickness, the ratio of width to outlet film thickness and the film thickness ratio is presented in

Figures 8-10 respectively. It is clearly seen that the load increases significantly with respect to the magnetization parameter thereby telling that the magnetism induces a positive effect. However, the effect of the film thickness ratio with respect to the magnetization parameter is negligible as indicated by Figure 10.

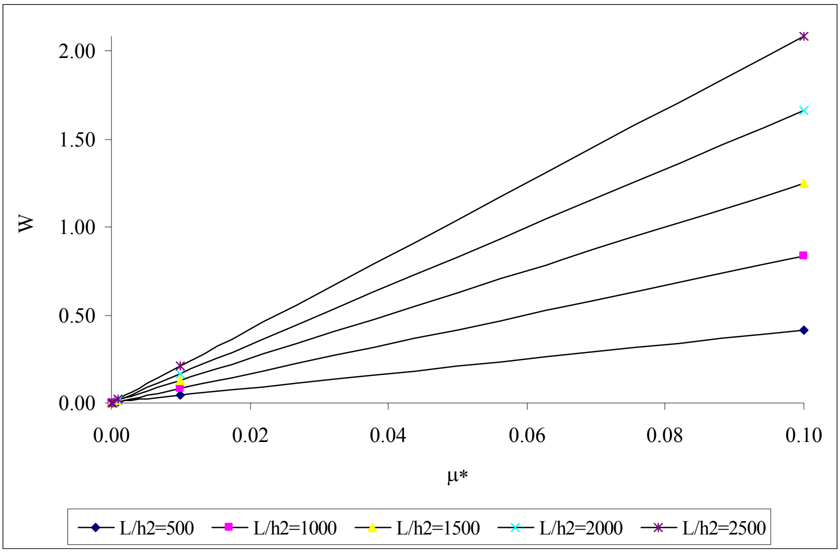

Figure 8

Variation of load carrying capacity with respect to μ* and L/h_2

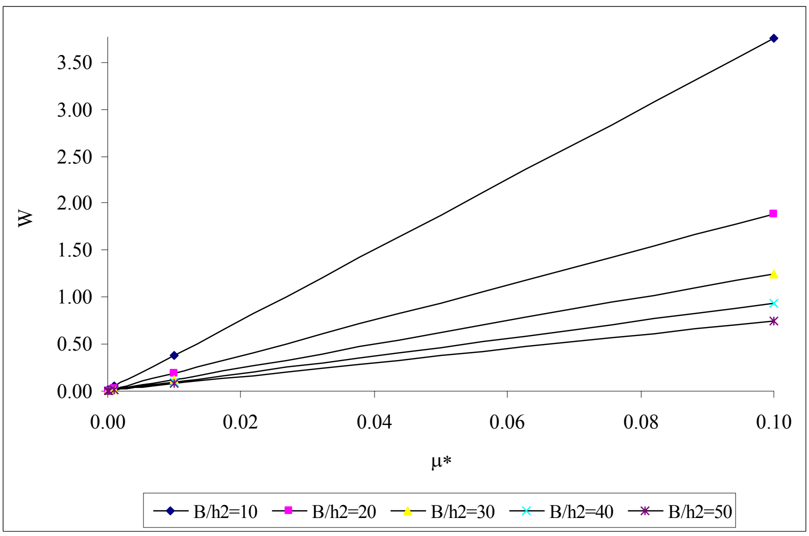

Figure 9

Variation of load carrying capacity with respect to μ* and B/h_2

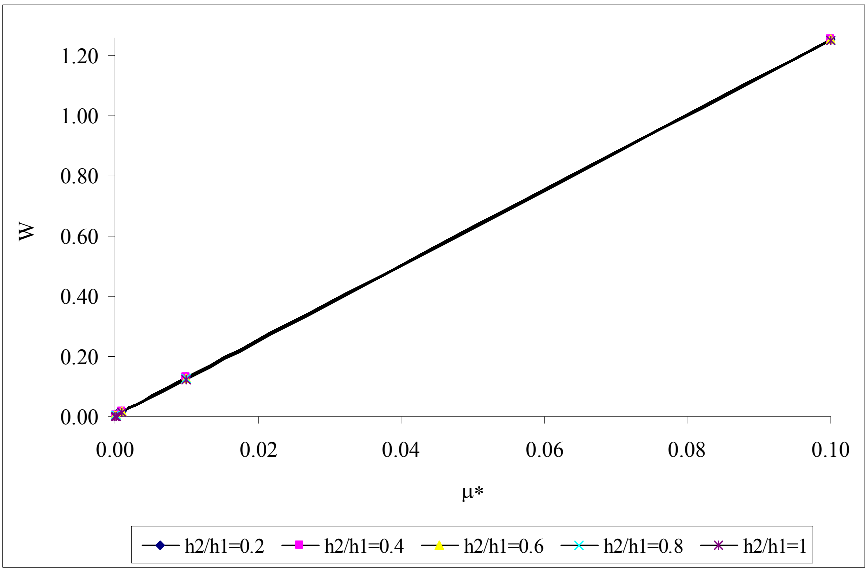

Figure 10

Variation of load carrying capacity with respect to μ* and h_2/h_1

Figures 11 and 12 describe the profile for load carrying capacity with respect to the ratio of the length to outlet film thickness for several values of the ratio of width to outlet film thickness and the film thickness ratio. These two figures tell that the load carrying capacity increases considerably due to the ratio L/h_2. Further, it is observed that the ratio of width to outlet film thickness and the film thickness ratio h_2/h_1 decrease the load carrying capacity. In addition, the rate of increase in load carrying capacity with respect to L/h_2 remains uniform in the case of the film thickness ratio while it increases in the case of the ratio of width to outlet film thickness. Thus, the aspect ratio increases the load carrying capacity considerably.

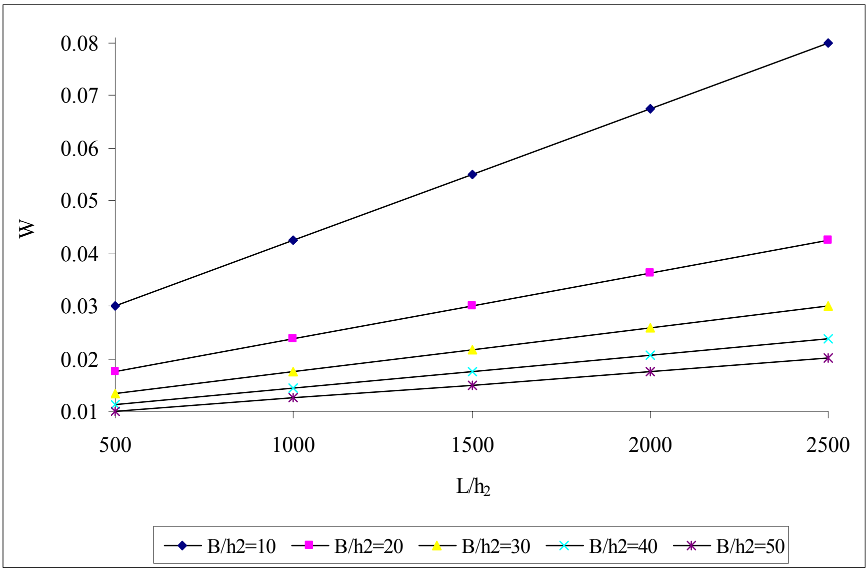

Figure 11
Variation of load carrying capacity with respect to L/h_2 and B/h_2

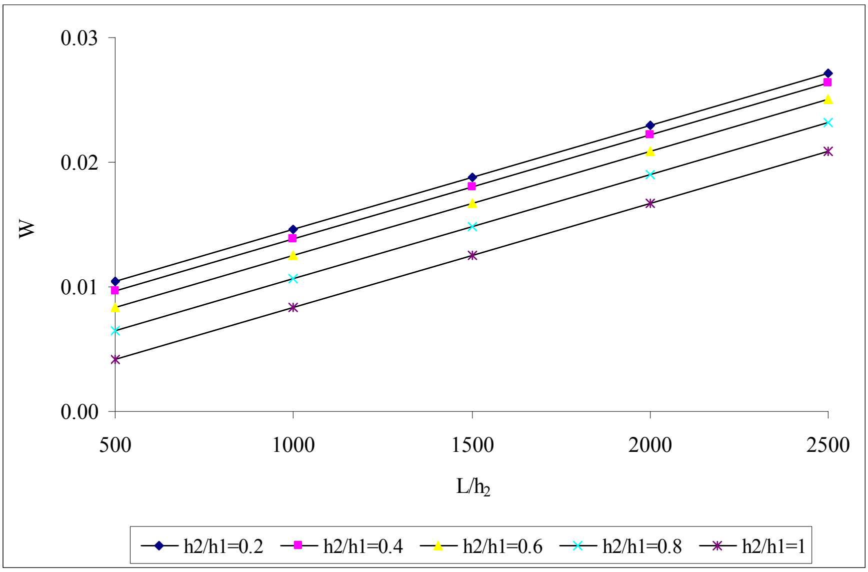

Figure 12
Variation of load carrying capacity with respect to L/h_2 and h_2/h_1

Lastly, Figure 13 presents the variation of load carrying capacity with respect to the ratio of width to outlet film thickness for various values of film thickness ratio h_2/h_1. It is clearly visible that the combined effect of these two parameters namely B/h_2 and h_2/h_1, is considerably adverse. The decrease in the load carrying capacity due to B/h_2 is more at the beginning while the reverse is true with respect to h_2/h_1.

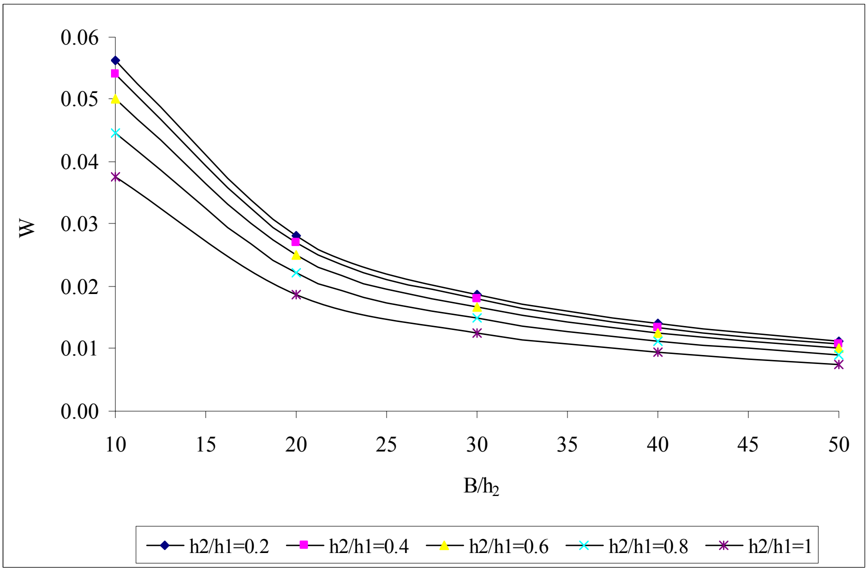

Figure 13

Variation of load carrying capacity with respect to B/h_2 and h_2/h_1

This article reveals that the negative effect induced by B/h_2 can be neutralized almost completely by the combined positive effect of magnetization parameter, L/h_2 and the aspect ratio.

Conclusion

A comparison of this present article with those of the studies carried out by Patel [24] and Bhat and Deheri [27] makes it clear that there is at least a four-time increase in the load carrying capacity here. The importance of this work lies in the fact that it offers an additional degree of freedom from a magnitude point of view in light of the investigations conducted by Prajapati [21] and Verma [25]. Moreover, the present article may open a new route towards the optimal performance of the bearing system. In addition, the results presented here tend to suggest that there exists enough scope for extending the life period of the bearing system by controlling lubricant loss.

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Nomenclature:

h	Fluid film thickness at any point (mm)
h_1	Maximum film thickness (mm)
h_2	Minimum film thickness (mm)
H^2	Magnetic field
k	Suitably chosen constant associated with the magnetic field
L	Length of the bearing (mm)

B	Breadth of the bearing (mm)
m	Aspect ratio
p	Lubricant pressure (N/mm^2)
P	Dimensionless pressure
u	Uniform velocity in X direction
w	Load carrying capacity (N)
W	Non-dimensional load carrying capacity
ϕ	Inclination angle of the magnetic field
μ	Lubricant viscosity ($N.s/mm^2$)
μ_0	magnetic susceptibility
$\bar{\mu}$	Free space permeability
μ^*	Dimensionless magnetization parameter
τ	Shear stress (N/mm^2)
$\bar{\tau}$	Dimensionless shear stress
F	Frictional force (N)
$\bar{F}$	Dimensionless frictional force
F_0	Non-dimensional frictional force (at moving plate)
F_1	Non-dimensional frictional force (at fixed plate)
$\bar{H}$	Magnetic field