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RESEARCH ARTICLE

ROUGH TILTED PAD SLIDER BEARING LUBRICATED WITH A MAGNETIC FLUID

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ABSTRACT

The performance of a transversely rough tilted pad slider bearing was analyzed in the presence of a magnetic fluid lubricant. A mathematical model based on stochastic averaging process was developed. The expressions for non dimensional pressure, load carrying capacity and friction were derived. The computed results made it clear that the bearing working with magnetic fluid as a lubricant had a better performance than that of an identical bearing working with a conventional lubricant. In addition, it was concluded that the adverse effect of transverse roughness could be reduced to some extent by the positive effect of the magnetic fluid lubricant. However, this reduction got larger due to negatively skewed roughness.

Key words: Titled pad slider bearing, magnetic fluid, roughness, load carrying capacity, friction.

INTRODUCTION

The first titled pad bearing in service was probably the one built under the guidance of Mitchell in 1907. By 1913, the great merits of the titled- bearing had been recognized for marine applications. To overcome stability problems experienced in machines with high speed rotors often tilting pad bearings are preferred (such bearings are widely used in the design of large percentage of world's high speed, multi stage turbo machinery). The tilting pad bearing differs from the multi-lobed bearing in that each pad rotates about a pivot enabling each pad to have higher degree of freedom corresponding to movement about the pivot point. The pad tilts such that its centre of curvature moves to create a converging pad film thickness. When the bearing is in operation, the rotating part of the bearing carries fresh oil into the pad area through viscous drag. Fluid pressure causes the pad to tilt slightly, creating a narrow construction between the shoe and other bearing surface. A wedge of pressurized fluid builds behind this construction, separating the moving part. The optimum design of tilted pad slider with one-dimensional continuous surface profiles has been studied by (Singh, 1982). (Royal and Raizada, 1965) embarked on the numerical analysis of effects of tilt, sliding and squeeze action on externally pressurized oil-film bearings. This paper illustrated the application of the results to dynamical problems. (Kennedy, 1989) investigated the transient characteristics of an infinite tilted-pad thrust bearing for a step change in the slider velocity. The numerical solutions indicated that the transient characteristics depend on the initial condition of the bearing and on the terminal speed of

the slider. Magnetic fluids are stable colloidal system of fine single-domain particles that are suspended in liquid carrier such as water, mineral oil, paraffin and kerosene. However, the inclusion of a magnetically active additive to a lubricant decrease motion, resistance and consequently improves the dynamics of the bearing system, which in turn, leads to the increase in the durability of the whole machine. Actually, magnetic aid can cause a reduction of the surface stresses and when the field is sufficiently strong it makes it possible to avoid the direct contacting of mating surfaces. Therefore, in the magnetic- aided bearing systems reduction of the friction co-efficient and wear rate is possible and simultaneously rotational velocities can be higher. Magnetic medium as lubricant mainly performs three functions: decreasing friction losses, reducing the wear and damping out vibration and mid range resonance (Bhat, 2003).

(Verma and Agrwal, 1986) studied the squeeze film performance by taking a magnetic fluid as a lubricant. Also, (Bhat and Deheri, 1991) modified the analysis of (Agrwal, 1986) by considering a magnetic fluid based porous composite slider bearing with its slider consisting of an inclined pad and a flat pad. It was shown that the magnetic fluid lubrication increased the load carrying capacity, unaltered the friction and shifted the centre of pressure towards the inlet. (Shah and Bhat, 2003) analyzed an exponential porous slider bearing with a ferro-fluid lubricant, the flow being governed by Jenkins flow behavior considering the slip velocity. It was observed that the load carrying capacity as well as friction decreased and co-efficient of friction increased when the slip parameter increased. (Lin *et al.*, 2006) dealt with the dynamic characteristics of a wide exponential film shape slider bearing

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lubricated with a non-Newtonian couple stress fluid. Here it was shown that the effect of couple stress decreased the friction as well as the volume flow rate. Further, the couple stress fluid increased the load carrying capacity.

The bearing surfaces particularly, after receiving some run-in and wear often exhibit a one dimensional type of roughness, running in the direction of sliding or longitudinal direction. Rough surfaces usually wear more quickly and have higher friction coefficient than smooth surfaces. The roughness appears to be random in character without following any particular structural pattern. (Christensen and Tonder, 1969(a), 1969(b), 1970) developed the stochastic Reynolds equation for transverse and longitudinal roughness. This approach has been employed by many researchers to predict the effect of surface roughness (Gupta and Deheri, 1996, Prakash and Tiwari, 1983, Ting, 1975, Tonder, 1972 and Tzeng, 1967). Sinha and Adamu, (2009) summarized the thermal and roughness effects on different characteristics of an infinite tilted pad slider bearing. It was observed that for a non-parallel slider bearing the load carrying capacity due to the combined effect was less than the load carrying capacity due to the roughness effect. However, the reverse was true in the case of parallel slider bearing. (Deheri *et al.* 2004 and 2005) discussed the effects of longitudinal as well as transverse surface roughness on the performance with squeeze film formed by a magnetic fluid. It was concluded that the magnetization introduced compensation for the adverse effect of surface roughness. The situation was slightly better in the case of longitudinal surface roughness.

Analysis

The geometry and pattern of the bearing structure is given in Fig. 1.

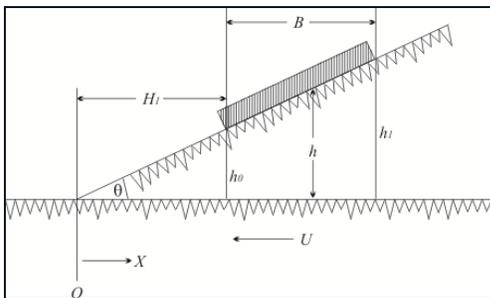


Fig.1.The configuration of the bearing system

The gap h increases with increasing x , hence the runner has to move towards the origin with its velocity is $-U$. The position of h_0 is a distance H_1 from the origin and h_1 is $(H_1 + B)$ away. By similarity of triangles giving

$$h = \frac{x h_1}{H_1}$$

$$H_1 = B \left(\frac{h_1}{h_1 - h_0} \right) = \frac{B}{K}$$

where

$$K = \frac{h_1 - h_0}{h_0}$$

The expression for film thickness is considered in the form of (Christensen and Tonder, 1969(a), 1969(b), 1970)

$$h(x) = \bar{h}(x) + h_z$$

where \bar{h} is the mean film thickness, while h_z is a randomly varying portion measured from the mean level characterizing the random roughness. Here h_z is assumed to have the probability density function $F(h_z)$ over the domain $[-b, b]$

$$F(h_z) = \begin{cases} \frac{32}{35b} \left(1 - \frac{h_z^2}{b^2} \right)^3 & -b \leq h_z \leq b \\ 0 & \text{elsewhere} \end{cases}$$

The details regarding the mean α , deviation σ , and skewness ϵ associated with the characterization of roughness can be seen from (Christensen and Tonder, 1969(a), 1969(b), 1970), where

$$\begin{aligned} \alpha &= E(h_z) \\ \sigma^2 &= E[(h_z - \alpha)^2] \\ \epsilon &= E[(h_z - \alpha)^3] \end{aligned}$$

while E is the expectancy operator defined by

$$E(R) = \int_{-b}^b R F(h_z) dh_z$$

The magnetic field is oblique to the stator and its magnitude is given by

$$H^2 = kL^2 \sin^2\left(\frac{\pi x}{L}\right)$$

where $k = 10^{14} A^2 m^{-4}$ chosen so as to have a magnetic field of strength over 10^5 (Bhat and Deheri, 1991, Prajapati, 1991, 1992 and Verma, 1986). Taking into account the usual assumptions of the hydromagnetic lubrication with the aid of the modeling resorted to by (Andharia *et al.*, 1997, 1999) one gets

$$\frac{d}{dx} \left(P - \frac{\mu_0 \bar{\mu} H^2}{2} \right) = 6U\eta \frac{h - \bar{h}}{A(h)} \tag{1}$$

where

$$A(h) = h^3 + 3\alpha h^2 + 3(\sigma^2 + \alpha^2)h + \epsilon + 3\sigma^2\alpha + \alpha^3$$

The concerned boundary conditions are

$$P = 0, \quad x = -L \text{ and } 0$$

Using the following dimensionless scheme

$$x^* = \frac{x}{L}, \quad P^* = \frac{h_0^2 P}{6U\eta B}, \quad \mu^* = \frac{k\mu_0 \bar{\mu}}{U\eta} \tag{2}$$

$$A_1 = \epsilon^* + 3\sigma^{*2}\alpha^* + \alpha^{*3}, \quad A_2 = 3(\sigma^{*2} + \alpha^{*2}), \quad A_3 = 3\alpha^*$$

$$\epsilon^* = \frac{\epsilon}{h_0^3}, \quad \sigma^* = \frac{\sigma}{h_0}, \quad \alpha^* = \frac{\alpha}{h_0}, \quad h_1^* = \frac{h_1}{h_0}$$

and fixing the following symbols,

$$S = 1 + A_1 + A_2 + A_3, \quad \frac{dh}{dx} = \tan \theta,$$

one obtains the pressure distribution in dimensionless form as

$$P^* = h^*(h^* + 1) \frac{\mu^*}{2}$$

$$+ \frac{1}{5K} \left[\frac{1}{h^*} - \frac{K+1}{h^{*2}(K+2)} - \frac{1}{K+2} \right] \quad (3)$$

Substitution of the pressure field above leads to dimensionless load carrying capacity, given by

$$W^* = \frac{h_0^2}{6\eta U L B^2} W - \int_0^1 P^* dx$$

which means

$$W^* = \frac{\mu^*}{12} [2(K+1)^2 + 5(K+2)] + \frac{1}{5K^2} \left[\ln(K+1) - \frac{2K}{(K+2)} \right] \quad (4)$$

The total frictional force is

$$F = \int_0^L \int_{H_1}^{B+H_1} \tau dx dy$$

The stress on the surface $z = h$ is $\tau_{h,0}$ and $z = 0$ is $\tau_{0,0}$. Following the discussions in (Cameron, 1972) one finds that

$$F_{h,0} = \int_0^L \int_{H_1}^{B+H_1} \tau_{h,0} dx dy = \int_0^L \int_{H_1}^{B+H_1} \left[\left(\pm \frac{\partial}{\partial x} \left(P - \frac{\mu_0 \bar{\mu} H^2}{2} \right) \frac{h}{2} \right) + \frac{\eta U}{h} \right] dx dy$$

Following (Cameron, 1972) the non-dimensional friction is calculated, as

$$F^* = \frac{h_0}{\eta U B L} F = \frac{\tan \theta}{2} W^* + \frac{\tan \theta}{2} \frac{\mu^*}{12} [2(K+1)^2 + 5(K+2)] - \frac{\mu^*}{48} \left[\frac{3(K+2)((K+1)^2 + 1)}{+4((K+1)^2 + K+2)} + \frac{\ln(K+1)}{K} \right] \quad (6)$$

RESULTS AND DISCUSSION

Equations (3), (4) and (6) gave the expression respectively for dimensionless pressure, load carrying capacity and friction. It was observed from Equation (3) that the non-dimensional pressure was increased by

$$h^*(h^* + 1) \frac{\mu^*}{2}$$

while increase in the load carrying capacity was

$$\frac{\mu^*}{12} [2(K+1)^2 + 5(K+2)]$$

as could be seen from Equation (4) as, compared to the case of conventional lubricants.

Setting μ^* to be zero one got the performance of the corresponding conventional rough bearing system. Further, taking roughness parameters to be zero, one obtained the discussion carried out in (Cameron, 1972).

Figures (2)-(5) demonstrated the variation of dimensionless load carrying capacity with respect to the magnetization parameter for various values of σ^* , ϵ^* , α^* and K respectively. It was clearly noticed that the load carrying capacity was increased sharply due to the magnetization. However, this increase was more when considered with the case of K. Figures (6)-(8) presented the non-dimensional load carrying capacity with respect to the standard deviation σ made it clear that the standard deviation had a considerable adverse effect on the performance of the bearing system as the load carrying capacity decreased substantially with respect to the increasing the value of standard deviation. Figure (9) described the combined effect of variance and skewness on the distribution of load carrying capacity. Positively skewed roughness decreased the load carrying capacity while the load carrying capacity got increased due to negatively skewed roughness. Variance followed the trends of skewness. Figures (10)-(11) underlined that the role of K was equally significant for improving the performance of the bearing system. It was clearly seen that a negative effect was induced by the standard deviation. However, it could be minimized to a large extent by the positive effect of the magnetic fluid lubricant in the case of negatively skewed roughness. This effect was more pronounced when negative variance was involved. The computed results for the friction were presented graphically in Figures (12)-(21). It was clearly noticed that the magnetization and the thickness ratio K reduced the friction considerably and this reduction in the friction got increased when considered with the effect of standard deviation. Also, it could be seen that the positively skewed roughness tended to decrease the friction. Similar was the case of α^* (+ve). Some of the graphs suggested that the adverse effect of transverse surface roughness could be compensated up to a large extent by the effect of magnetic fluid lubricant especially, in the case of negatively skewed roughness.

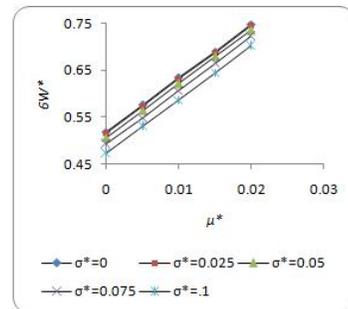


Fig. 2. The variation of load carrying capacity with respect to μ^* and σ^*

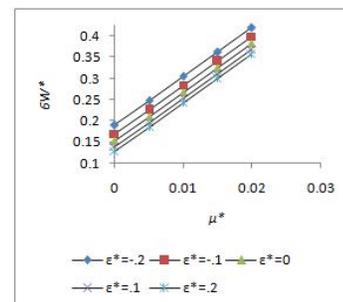


Fig. 3. The variation of load carrying capacity with respect to μ^* and ϵ^*

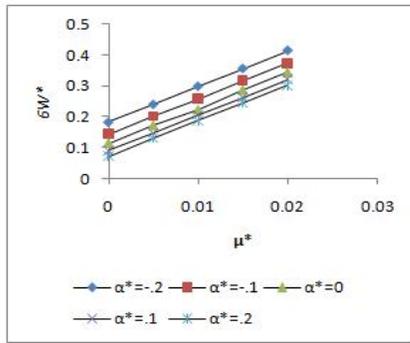


Fig.4.The variation of load carrying capacity with respect to μ^* and α^*

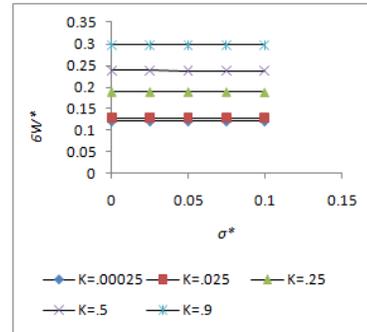


Fig.8.The variation of load carrying capacity with respect to σ^* and K

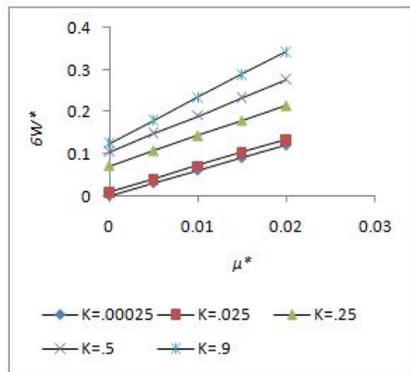


Fig.5.The variation of load carrying capacity with respect to μ^* and K

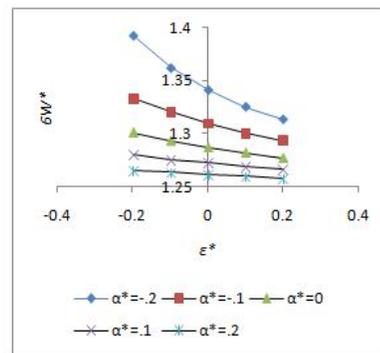


Fig.9.The variation of load carrying capacity with respect to ϵ^* and α^*

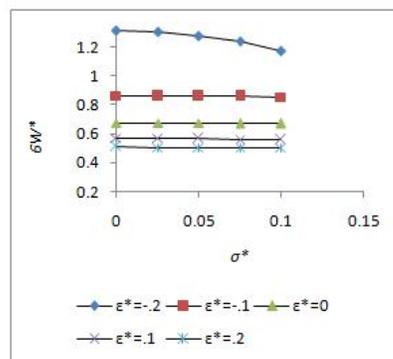


Fig.6.The variation of load carrying capacity with respect to σ^* and ϵ^*

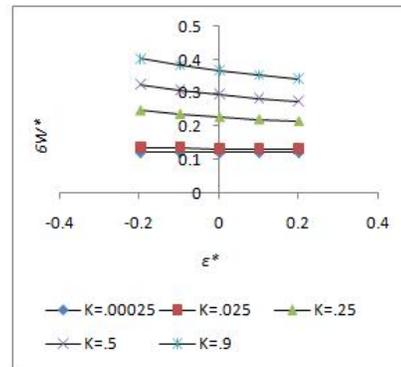


Fig.10.The variation of load carrying capacity with respect to ϵ^* and K

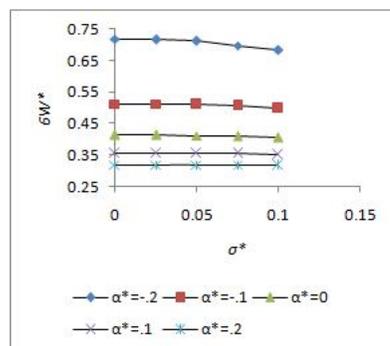


Fig.7.The variation of load carrying capacity with respect to σ^* and α^*

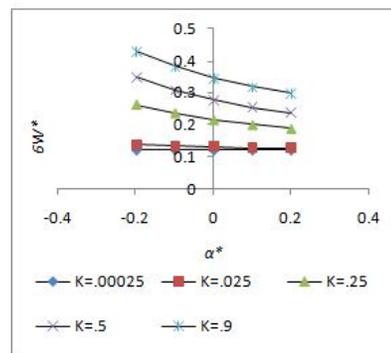


Fig.11.The variation of load carrying capacity with respect to α^* and K

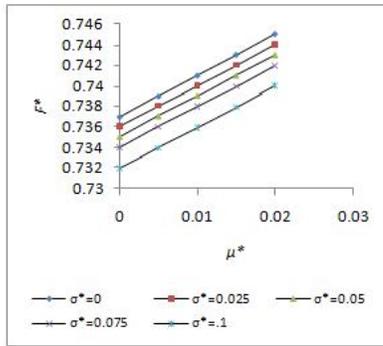


Fig.12.The variation of friction with respect to μ^* and σ^*

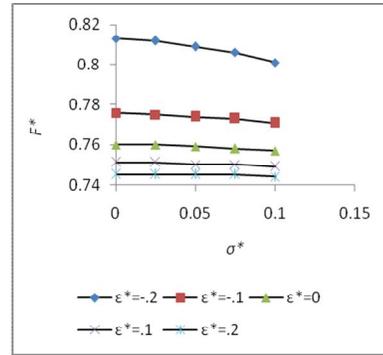


Fig.16.The variation of friction with respect to σ^* and ϵ^*

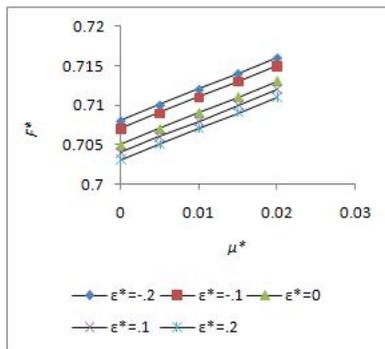


Fig.13.The variation of friction with respect to μ^* and ϵ^*

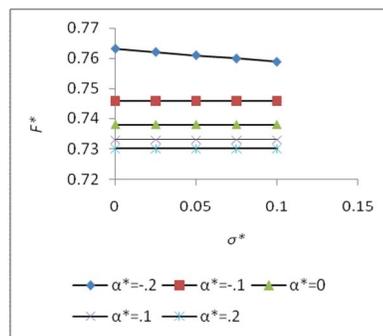


Fig.17.The variation of friction with respect to σ^* and α^*

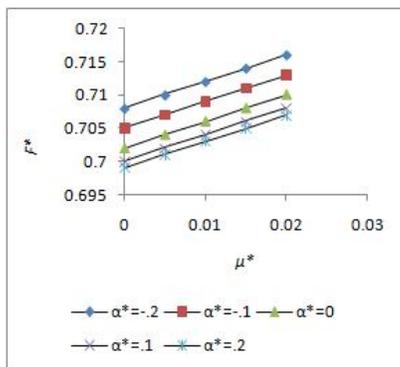


Fig.14.The variation of friction with respect to μ^* and α^*

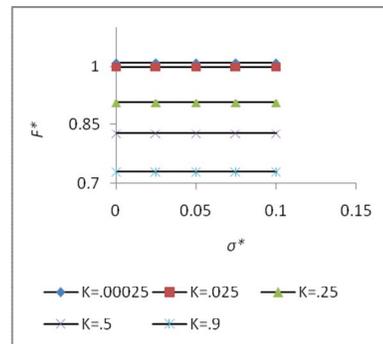


Fig.18.The variation of friction with respect to σ^* and K

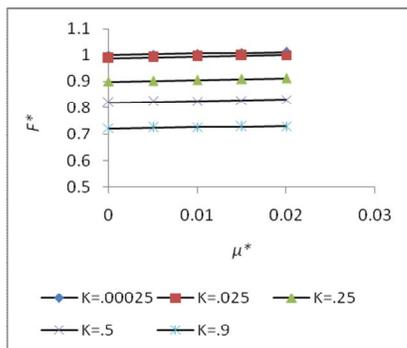


Fig.15.The variation of friction with respect to μ^* and K

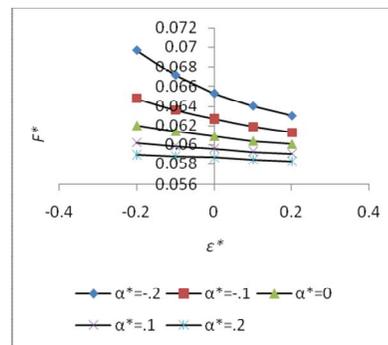


Fig.19.The variation of friction with respect to ϵ^* and α^*

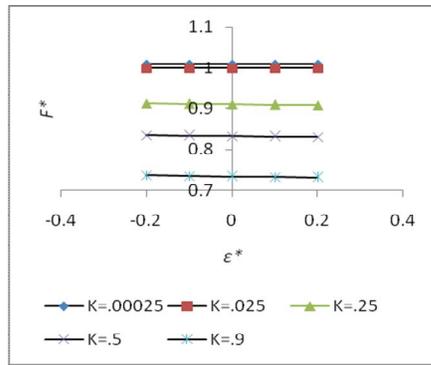


Fig.20.The variation of friction with respect to ϵ^* and K

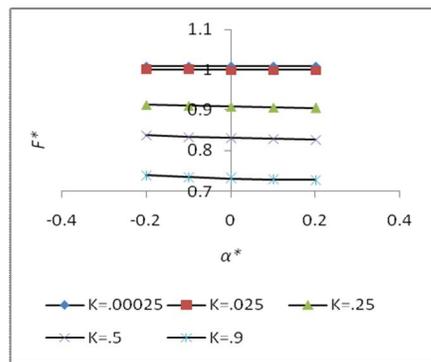


Fig.21.The variation of friction with respect to α^* and K

Conclusion

This study revealed that the roughness deserved to be addressed while designing the bearing system even if a strong magnetic field was in force. Further, this study suggested that in the absence of flow also the bearing could be observed to support a good amount of load.

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