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PERFORMANCE OF A MAGNETIC FLUID BASED LONGITUDINALLY ROUGH PLANE SLIDER BEARING

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Abstract:- The analysis on the performance of a longitudinally rough plane slider bearing under the presence of magnetic fluid is presented. The magnetic field is oblique to the stator and the bearing surfaces are assumed to be longitudinally rough. The roughness of the bearing surfaces is characterized by a stochastically random variable with non-zero mean, variance and skewness. The concerned Reynolds equation is averaged with respect to the random roughness parameters. This equation is solved with appropriate boundary conditions to obtain the expression for pressure distribution which is further used to calculate the load carrying capacity. The results are presented graphically. It is seen that the load carrying capacity increases due to the magnetic fluid lubricant. It is also observed that the roughness affects the bearing performance adversely. The study reveals that the negative effect of roughness can be minimized by the positive effect of magnetization.

Keywords: plane slider bearing, magnetic fluid, longitudinal roughness, Reynolds equation, load carrying capacity.

1. INTRODUCTION:

The slider bearing is the simplest and is encountered most often. Slider bearings are used for supporting transverse loads. Slider bearings have been studied for various film shapes [Pinkus and Sternlicht (1961), Hamrock (1994), Bagci and Singh (1983)]. It is a well-known fact that the bearing surfaces, particularly after having some run-in and wear, develop roughness. The contamination of lubricant is also responsible for the development of roughness through chemical degradation. The roughness appears to be random which does not seem to follow any particular structural pattern. The randomness of the roughness was recognized by several investigators [Davis (1963), Burton (1963), Tzeng and Saibel (1967)]. Christensen and Tonder (1969.a, 1969.b, 1970.a) proposed a comprehensive general analysis both for transverse as well as longitudinal surface roughness based on a general probability density function. The same approach was followed by number of investigators to discuss the effect of surface roughness [Ting (1975), Prakash and Tiwari (1982, 1983), Prajapati (1991, 1992), Guha (1993), Gupta and Deheri (1996), Andharia et al. (1999)].

All the above studies dealt with a conventional lubricant. The magnetic fluid is prepared by suspending fine magnetic grains coated with a surfactant and dispersing it in a non-conducting and magnetically passive solvent, such as benzene, hydrocarbons and fluorocarbons. These magnetic fluids remain liquid in a magnetic field and after removal of a field recover their characteristics. Particles within the liquid experience a force due to the field gradient and move through the liquid imparting drag to it causing it to flow. The advantage of magnetic fluid lubricant over the conventional ones is that the former can be retained at the desired location by an external magnetic field [Bhat (2003)].

Agarwal (1986) presented the squeeze film performance by taking a magnetic fluid as a lubricant. Bhat and Deheri (1991) analyze the squeeze film behavior between porous annular disks using a magnetic fluid lubricant with the external magnetic field oblique to the lower disk. These discussions suggested that the magnetic fluid lubricant causes sharp rise in load carrying capacity of the bearing. Deheri et al. (2005) analyzed the performance of transversely rough slider bearing with squeeze film formed by a magnetic fluid.

Here it has been proposed to study and analyze the effect of longitudinal surface roughness on the performance of a magnetic fluid based plane slider bearing when the probability density function for the random variable characterizing the surface roughness is asymmetrical with non-zero mean.

2. ANALYSIS:

The configuration of the bearing which is infinite in the Y-direction is shown in Fig. 1.

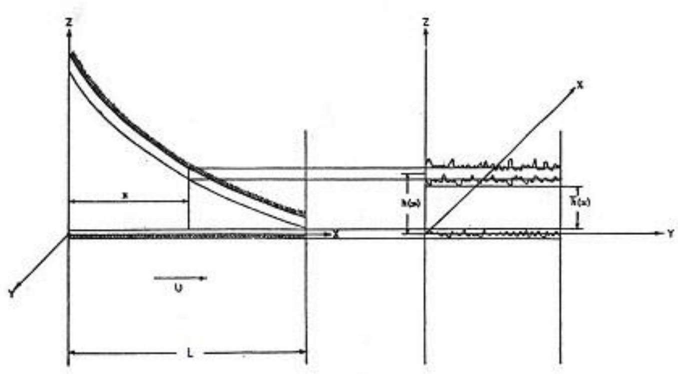

Fig.1 Bearing Configuration

The Slider moves with a uniform velocity U in the X-direction. h_0 and h_1 being the minimum and maximum film thickness respectively. The length of the bearing is L. The magnetic field is oblique to the stator as in Agrawal (1986) and its magnitude is given by

$$M^2 = x(L-x) \qquad [1]$$

The bearing surfaces are assumed to be longitudinally rough. The expression for film thickness is considered in the form of [Christensen and Tonder (1969(a), 1967(b) and 1970)]

$$h(x) = \bar{h}(x) + h_s \qquad [2]$$

where $\bar{h}(x)$ is the mean film thickness and h_s is a randomly varying portion measured from the mean level characterizing the random roughness. Here h_s is assumed to have the probability density function $f(h_s)$ over the domain [–c, c]

$$f(h_s) = \begin{cases} \dfrac{32}{35c^7}(c^2 - h_s{}^2)^3, & -c \le h_s \le c \\ 0 & , \ otherwise \end{cases} \qquad [3]$$

where c is the maximum deviation from the mean film thickness. The mean α , standard deviation σ and skewness ε associated with the random variable h_s are determined by the relations

$$\alpha = E(h_s), \qquad \sigma^2 = E[(h_s-\alpha)^2], \qquad \varepsilon = E[(h_s-\alpha)^3] \tag{4}$$

where E is the expectancy operation defined by

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

The lubricant film is considered to be isoviscous and incompressible and the flow is laminar. Under the usual assumptions of the hydrodynamic lubrication, the modified Reynolds equation [Agrawal (1986), Bhat (1978), Bhat and Patel (1981)] in the present case becomes:

$$\frac{d}{dx}\left[h^3 \frac{d}{dx}\left(p-0.5\mu_0\bar{\mu}M^2\right)\right] = 6\mu U\frac{dh}{dx} \tag{6}$$

where μ_0 is the magnetic susceptibility, $\bar{\mu}$ is the free space permeability, μ is the lubricant viscosity, M^2 is the magnitude of the magnetic field, U is the velocity of the slider in the X-direction.

Following the average process discussed in Andharia et al. (1997), Eq. (6) takes the form

$$\frac{d}{dx}\left[m(\bar{h})^{-1} \frac{d}{dx}\left(\bar{p}-0.5\mu_0\bar{\mu}M^2\right)\right] = 6\mu U\frac{d}{dx}\left[n(\bar{h})^{-1}\right] \tag{7}$$

where $\bar{p}$ is the expected value of the lubricant pressure p and

$$m(\bar{h}) = \frac{1}{\bar{h}^3}\left[1-\frac{3\alpha}{\bar{h}}+\frac{6}{\bar{h}^2}\left(\sigma^2+\alpha^2\right)-\frac{20}{\bar{h}^3}\left(\varepsilon+3\sigma^2\alpha+\alpha^3\right)\right] \quad \text{and} \tag{8}$$

$$n(\bar{h}) = \frac{1}{\bar{h}}\left[1-\frac{\alpha}{\bar{h}}+\frac{1}{\bar{h}^2}\left(\sigma^2+\alpha^2\right)-\frac{1}{\bar{h}^3}\left(\varepsilon+3\sigma^2\alpha+\alpha^3\right)\right] \tag{9}$$

Integrating Eq. (7) with respect to x gives

$$\frac{d}{dx}\left(\bar{p}-0.5\mu_0\bar{\mu}M^2\right) = 6\mu U m(\bar{h})\left[n(\bar{h})^{-1}+Q\right] \tag{10}$$

where Q is the constant of integration.

Introducing the dimensionless quantities

$$H = \frac{\bar{h}}{h_0},\quad \bar{Q} = \frac{Q}{h_0},\quad X = \frac{x}{L},\quad M(H) = h_0^{3} m(\bar{h}),\quad N(H) = h_0 n(\bar{h}),$$
$$\bar{\alpha} = \frac{\alpha}{h_0},\quad \bar{\sigma} = \frac{\sigma}{h_0},\quad \bar{\varepsilon} = \frac{\varepsilon}{h_0^{3}},\quad P = \frac{\bar{p} h_0^{2}}{\mu U L},\quad and \quad \mu^* = \frac{\mu_0 \bar{\mu} h_0^{2} L}{2\mu U} \qquad [11]$$

and using the associated boundary conditions P = 0 at X = 0, 1 [12]

the pressure distribution in non-dimensional form comes out be

$$P = \int_0^X \left\{ 6M(H)\left[\frac{1}{N(H)} + \bar{Q} \right] + \mu^*(1-2X) \right\} dX \qquad [13]$$

where $M(H) = \frac{1}{H^3}\left[1 - \frac{3\bar{\alpha}}{H} + \frac{6}{H^2}\left(\bar{\sigma}^2 + \bar{\alpha}^2 \right) - \frac{20}{H^3}\left(\bar{\varepsilon} + 3\bar{\sigma}^2 \bar{\alpha} + \bar{\alpha}^3 \right) \right]$, [14]

$$N(H) = \frac{1}{H}\left[1 - \frac{\bar{\alpha}}{H} + \frac{1}{H^2}\left(\bar{\sigma}^2 + \bar{\alpha}^2 \right) - \frac{1}{H^3}\left(\bar{\varepsilon} + 3\bar{\sigma}^2 \bar{\alpha} + \bar{\alpha}^3 \right) \right] \quad \text{and} \qquad [15]$$

$$\bar{Q} = \frac{-\int_0^1 \frac{M(H)}{N(H)} dX}{\int_0^1 M(H) dX} \qquad [16]$$

The dimensionless load carrying capacity of the bearing is given by

$$W = \frac{h_0^{2}}{\mu U L^2} w = \int_0^1 P dX \qquad [17]$$

3. RESULTS AND DISCUSSIONS:

It is observed that Eq. (13) and Eq. (17) presents the dimensionless pressure distribution and dimensionless load carrying capacity respectively. For the analysis of plane slider bearing the film thickness H is taken as 2 – X [Andharia et al. (1997)]. To have the analyses for the quantitative effect of magnetic fluid lubricant and longitudinal roughness of the bearing surfaces on its performance characteristics, Eq. (13) and Eq. (17) are numerically integrated using Simpson's 1/3 rule for different values of magnetization parameter μ^* and roughness parameters α, σ and ε. The results are presented graphically in Figures (2) – (7).

Figures (2) – (4) presents the variation of load carrying capacity with respect to μ^* for various values of α, σ and ε. These figures suggest that the load carrying capacity increases significantly due to magnetic fluid lubricant. It is seen from the Figure (5) that the load carrying capacity increases with σ, but this increase is very less. The combined effect of ε and α on the distribution of load carrying capacity is presented in Figure (6). Here, it is observed that the positively skewed roughness decreases the load carrying capacity, while load carrying capacity increases sharply due to the negatively skewed roughness. The trend of α is identically with that of trend of ε which can be seen in Figure (7). Thus, the combined effect of negatively skewed roughness and standard deviation is significantly positive [Figure (5)]. Further, decrease the load carrying capacity due to (+ve) α gets further decrease owing to positively skewed roughness [Figure (6)]. A close look at figures reveals that the negative effect of longitudinal surface roughness can be minimized by the positive effect of magnetization especially in the case of negatively skewed roughness.

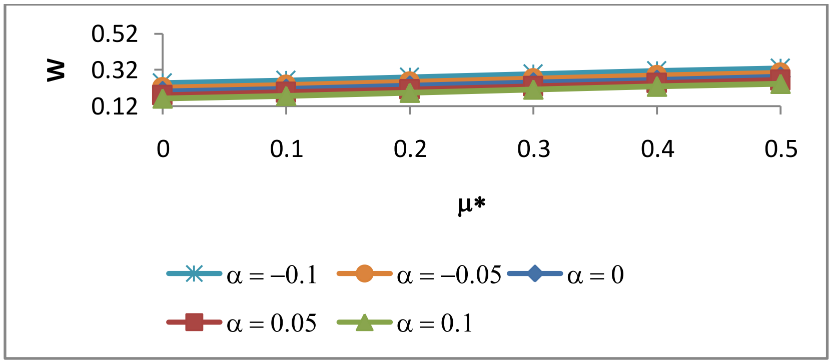

Figure 2: Variation of load carrying capacity with respect to μ^* and α.

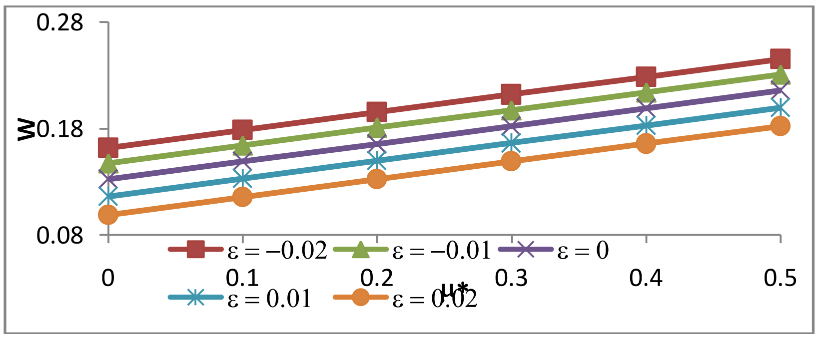

Figure 3: Variation of load carrying capacity with respect to μ^* and ε.

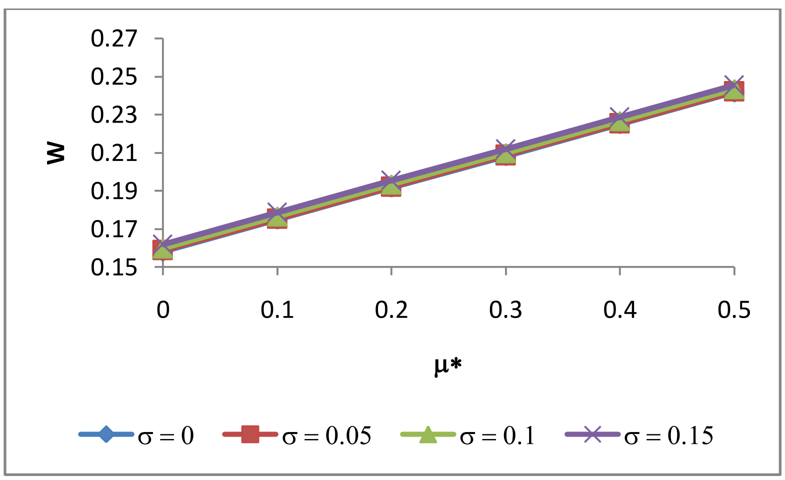

Figure 4: Variation of load carrying capacity with respect to μ^* and σ.

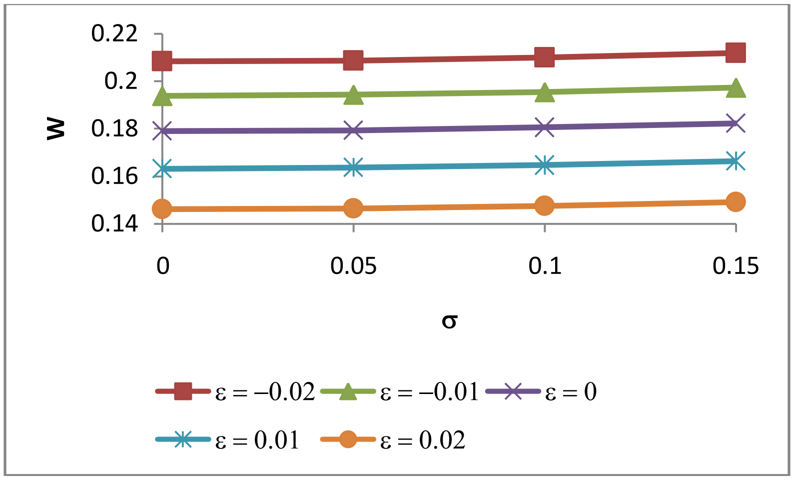

Figure 5: Variation of load carrying capacity with respect to σ and ε.

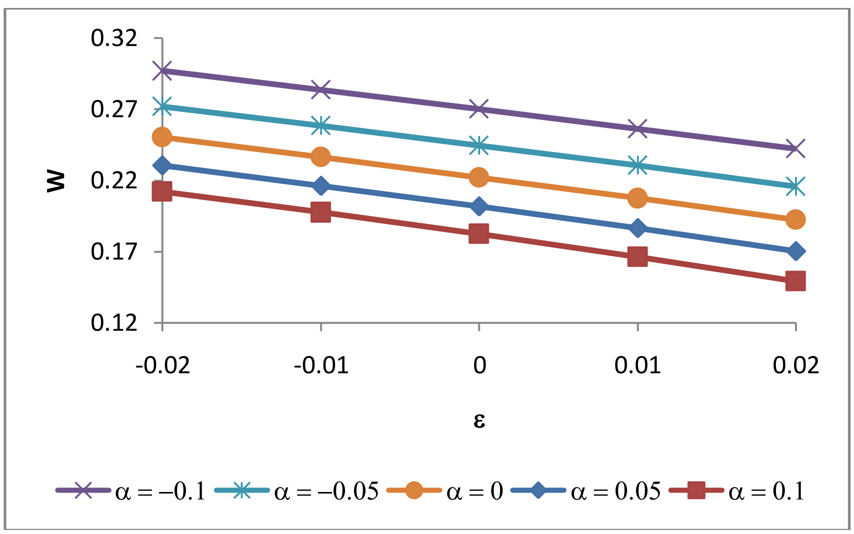

Figure 6: Variation of load carrying capacity with respect to ε and α.

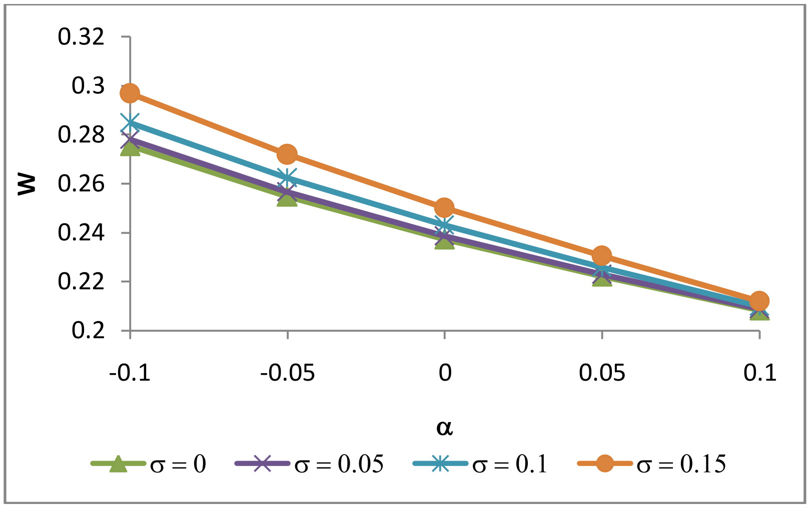

Figure 7: Variation of load carrying capacity with respect to α and σ.

4. CONCLUSION:

This study suggests that the magnetization leads to overall improved performance of the bearing system. We have seen that in the case of longitudinal roughness, standard deviation betters the performance of the slider bearing. Therefore the use of magnetic fluid as a lubricant and choosing suitable values of standard deviation in the case of longitudinal roughness, the bearing performance can be improved considerably. Further, this investigation makes it clear that the roughness must be accounted for while designing the bearing system from the life period point of view.

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