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Ferro Fluid Squeeze Film in Rough Porous Circular plates considering the effect of viscosity variation and velocity Slip

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Abstract

An attempt has been made to study the combined effects of roughness and velocity slip on the magnetic fluid based squeeze film in porous circular plates considering viscosity variation. The velocity slip model of Beavers and Joseph has been deployed to evaluate the effect of slip velocity. To calculate the effect of transverse surface roughness the stochastic model of Christensen and Tonder has been used. The associated stochastically averaged generalized Reynolds' type equation is solved with proper boundary conditions to get the pressure distributions, leading to the calculation of load carrying capacity. It is found that the magnetization has a limited option in reducing the adverse effect of roughness, porosity and velocity slip. However, the situation remains better owing to negatively skewed roughness. But for any type of improvement in the bearing performance the slip is required to be kept at nominal level.

Keywords: Circular Plates, Squeeze film, Roughness, Velocity Slip, Magnetization

INTRODUCTION:

It is well-known that the effects of thermal compressibility, viscosity variation, slip at the surfaces, inertia and surface roughness were neglected while deriving the famous classical equation of Reynolds' in (1). It was Cope (2) who modified this Reynolds' equation including viscosity and density variation across the fluid firm. The viscosity variation was discussed by Zienkiewicz (3) and Cameron (4). Here it was pointed out that viscosity variation and temperature could not be ignored.

Sinha et. al. (5) dealt with viscosity variation considering cavitation in a bearing under the presence of a lubricant containing additives. Rao and Prasad (6) investigated the combined effects of velocity slip and viscosity variation in rolling and normal motion. Patel and Deheri (7) studied the magnetic fluid based squeeze film be and analyzed the effect of roughness and rotation in porous circular plates with a concentric circular pocket. Deheri et al (8) dealt with the of ferro fluid squeeze film in porous circular plates with the consideration of porous matrix of variable thickness. Hydro-magnetic squeeze film between two conducting rough porous circular plates was subjected to investigation by Vadher et. al. (9). Patel et.al (10) analyzed the effect of transverse surface roughness on the performance of a magnetic fluid based squeeze film behaviour between rotating porous circular plates with concentric circular pocket considering slip velocity. Shimpi and Deheri (11) discussed the combined effect of surface roughness and deformation on a magnetic squeeze film between rotating porous circular plates with concentric circular pockets. The analysis incorporated in this paper was modified and developed by Shimpi and Deheri (12) to include the effect of curvature of the surfaces.

Here it has been proposed to study and analyze the combined effect of velocity slip and viscosity variation for ferro fluid squeeze film in transverse rough circular plates.

ANALYSIS:

The squeeze film lubrication between parallel plates are shown in figure I.

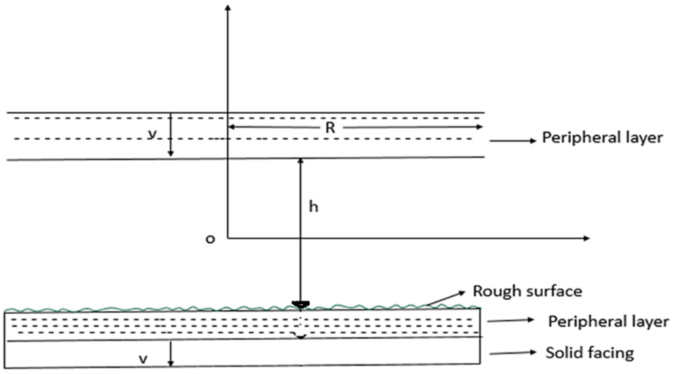

Figure I: Geometrical configuration of the bearing system

The governing squeeze film equation of the flow for the lubricant is given by

$$\frac{d}{dx}\left[F_4\frac{dp}{dx}\right] = -V \qquad \ldots(1)$$

where

$$F_4 = \frac{1}{12\mu}\left[\frac{(h-a)^3(k-1)+h^3}{k}+\frac{6h^2}{\beta}\right]$$

H is the film thickness of the lubricant between two plates, V is squeeze velocity, a is thickness of the layer, k is a ratio of viscosity, μ is a viscosity of the middle layer and β is the sleep parameter. Introducing the dimensionless quantities

$$\bar{a} = \frac{a}{l}, \qquad \bar{h} = \frac{h}{l}, \qquad \bar{\beta} = \frac{\beta}{\left(\frac{\mu}{l}\right)}.$$

One gets

$$\frac{d}{dx}\left[\overline{F_4}\frac{dp}{dx}\right] = -V \qquad \ldots(2)$$

where

$$\overline{F_4} = \frac{l^3}{12\mu}\left[\frac{\left(\bar{h}-\bar{a}\right)^3(k-1)+\bar{h}^3}{k} + \frac{6\bar{h}^2}{\bar{\beta}}\right]$$

The flux flow Q of the lubricant for equation number (1) is written as

$$Q = 2b\left[F_4\frac{dp}{dx}\right] \qquad \ldots(3)$$

where b is width of the bearing. For the circular plates perimeter b = 2πr
The flux Q obtained by using continuity equation as

$$Q = 4\pi r^2 V \qquad \ldots(4)$$

From equation (3) and (4) modified Reynolds’ equation governing the film pressure P is obtained as

$$\frac{d}{dr}\left(p - 0.5\mu_0\bar{\mu}H^2\right) = -\frac{Vr}{F_4} \qquad \ldots(5)$$

Where

$$H^2 = k\,r\,(R - r),\ 0 \le r \le R$$

K being suitable chosen constant from dimensionless point of view, μ_0 is permeability of free space, $\bar{\mu}$ is magnetic susceptibility, ϕ permeability of porous matrix. H_0 is a thickness of porous facing.

Now the integration of equation (5) with use of boundary conditions

$$p = 0 \text{ at } r = R \qquad \ldots(6)$$

leads to expression for pressure distribution as

$$p = \frac{v}{2F_4}\left(R^2 - r^2\right) - 0.5\mu_0\bar{\mu}\,kr(R - r) \qquad \ldots(7)$$

The load carrying capacity w is found to be

$$\begin{aligned} w &= 2\pi\int_0^R P\,r\,dr \\ &= \frac{\pi\, v}{F_4}\,\frac{R^4}{2} - \frac{\pi\mu_0\bar{\mu}kR^4}{12} \\ &= \frac{\pi R^4}{12}\left[\frac{6v}{F_4} - \pi\mu_0\bar{\mu}\,k\right] \end{aligned} \qquad \ldots(8)$$

In view of equation (8) the non-dimensional load carrying capacity is obtained as

$$\overline{w} = \frac{w}{p_0 l^2} = \pi P_s^* \left[\frac{6}{\overline{F_4}} + \frac{\pi}{12}\mu^* \right] \qquad \ldots (9)$$

Where

$$\mu^* = \frac{-\mu_0\, \overline{\mu}\, k}{\mu\, v}$$

and

$$p_s^* = \frac{R^4 \mu v}{p_0 l^5}$$

$$\overline{F_4} = \frac{1}{l^3}\left[\frac{g(h)(k-1) - 3ag(h)^{\frac{2}{3}}(k-1) + 3a^2 g(h)^{\frac{1}{3}}(k-1) - a^3(k-1) + g(h)}{k} + \frac{6g(h)^{\frac{2}{3}}}{\beta} \right]$$

RESULT AND DISCUSSION:

It is clearly observed that the d less persevered distribution while equation (9) represented the variation of load carrying capacity in non-dimensional form. It is not difficult to the load carrying capacity enhances by

$$\frac{\pi^2}{12} P_s^* \mu^*$$

As compared to the case of conventional lubricant based bearing systems. This is because of the fact that the magnetization increases the viscosity of the lubricant, causing increased presence and hence increased load carrying capacity. However the effect of initial values of porosity on the variation of load carrying capacity with respect to μ^* remains nominal. (Figures (1)-(2)).

The effect of standard deviation is to bring down the load carrying capacity which can be seen from Figures (3)-(6). But the effect of slip velocity on the distribution of load carrying capacity with respect to standard deviation is not that significant for lower values of slip parameter.

The fact that the variance (-ve) introduces an increase in the load carrying capacity can be seen from Figures (7)-(9). Also, the positive variance causes reduces load carrying capacity.

Here also, the effect of lower values of slip parameter on the distribution of load carrying capacity with respect to variance remains negligible.

Figures (10) – (11) established that the trends of load carrying capacity with respect to skewness are similar to those of variance. In other words, the negatively skewed roughness, may play an important role for bettering the load carrying capacity especially when variance (-ve) is involved.

As usual increased in porosity leads to decreased load carrying capacity which can be observed from figures (12)-(13). Further, the effect of slip velocity remains almost negligible.

Here the combined effect of viscosity ratio parameter and pressure P_0 introduces highly increased load carrying capacity which may play a decisive role in bearing design. (figure (14)-(15)). Lastly from figure (16), it is noticed that the load carrying capacity rises sharply due to P_0 but the effect of lower values of slip on load carrying capacity remains nominal.

In order to improve the bearing performance the viscosity ratio may play an important role when moderate to higher magnetic strength is there.

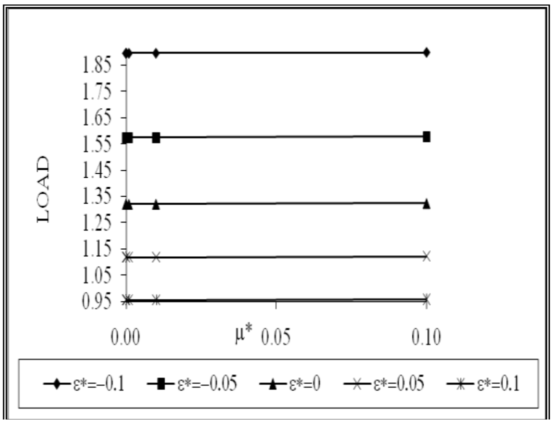

Figure: I Variation of load carrying capacity with respect to μ* and ε∗.

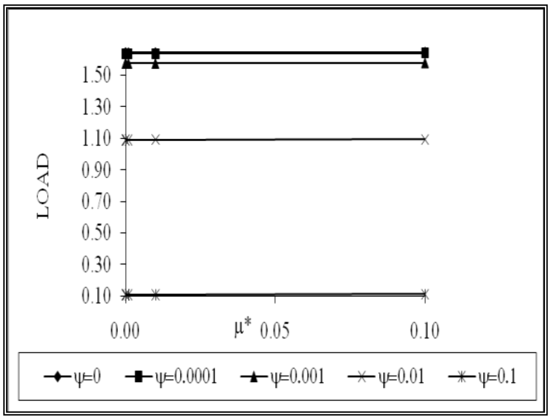

Figure: 2 Variation of load carrying capacity with respect to μ* and ψ.

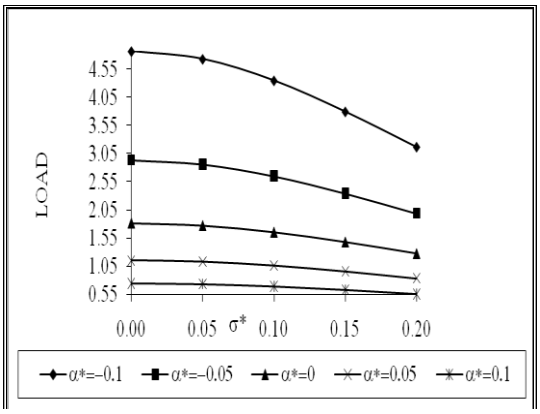

Figure: 3 Variation of load carrying capacity with respect to σ∗ and α∗.

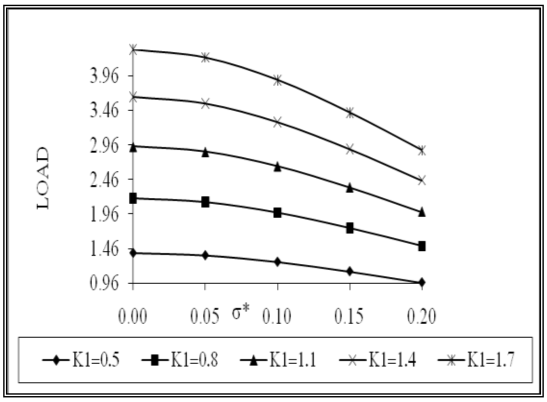

Figure: 4 Variation of load carrying capacity with respect to σ∗ and K_1.

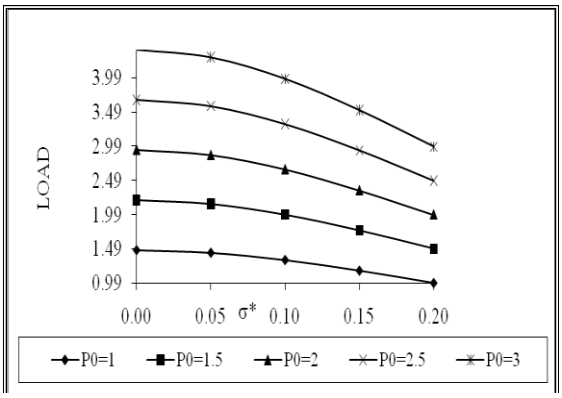

Figure: 5 Variation of load carrying capacity with respect to $\sigma*$ and P_0.

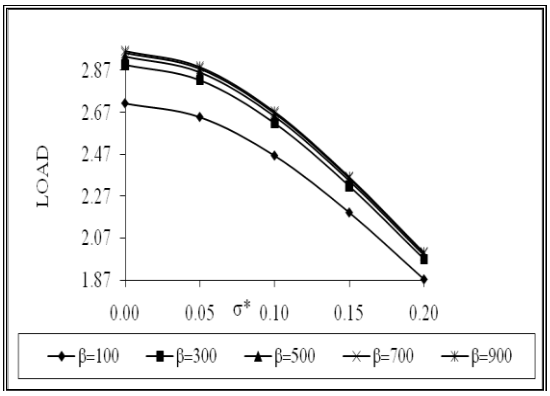

Figure: 6 Variation of load carrying capacity with respect to $\sigma*$ and β.

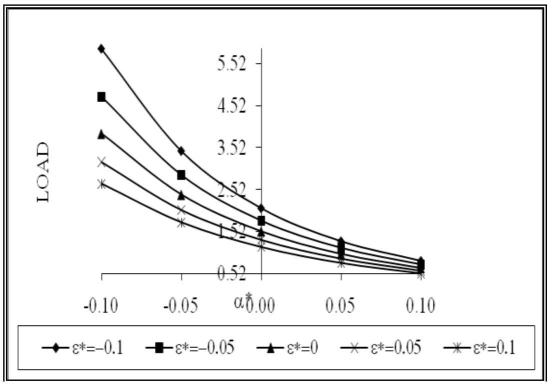

Figure: 7 Variation of load carrying capacity with respect to $\alpha*$ and $\varepsilon*$.

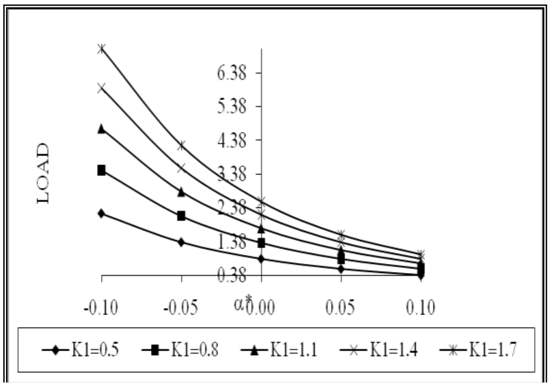

Figure: 8 Variation of load carrying capacity with respect to $\alpha*$ and K_1.

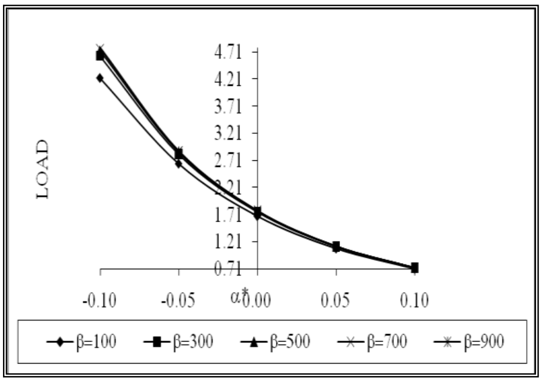

Figure: 9 Variation of load carrying capacity with respect to $\alpha*$ and β.

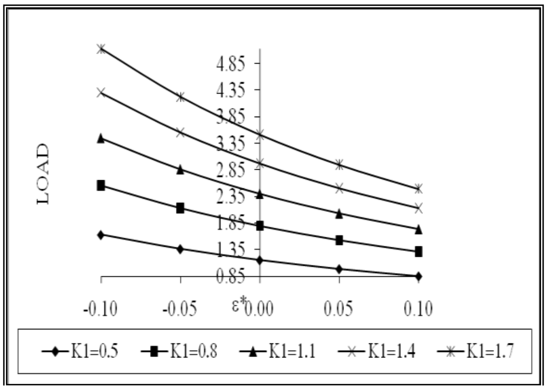

Figure: 10 Variation of load carrying capacity with respect to $\varepsilon*$ and K_1.

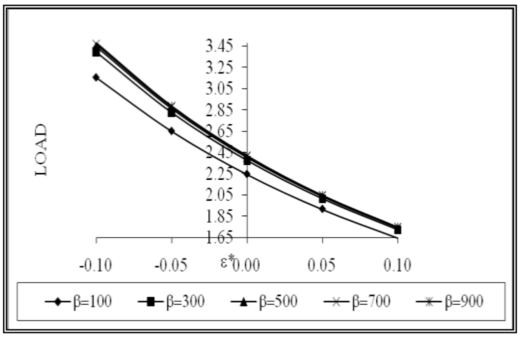

Figure: 11 Variation of load carrying capacity with respect to ε* and β.

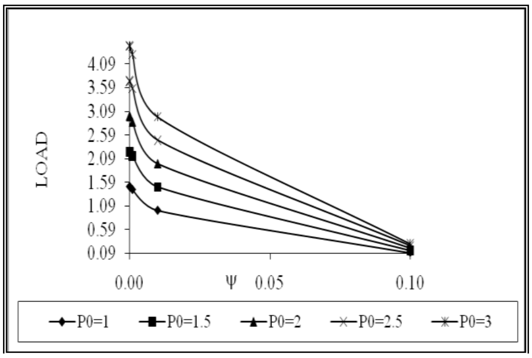

Figure: 12 Variation of load carrying capacity with respect to ψ and P_0.

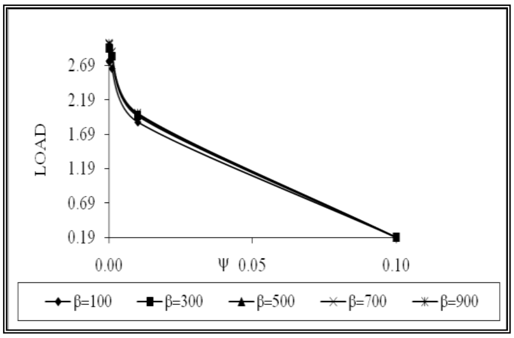

Figure: 13 Variation of load carrying capacity with respect to ψ and β.

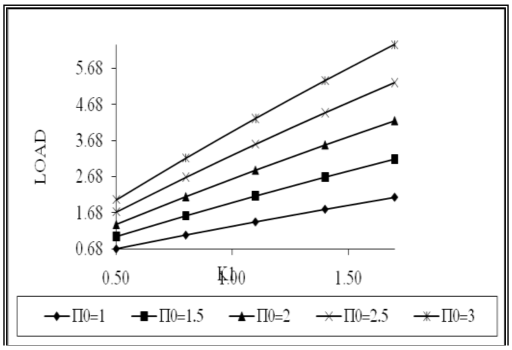

Figure: 14 Variation of load carrying capacity with respect to K_1 and P_0.

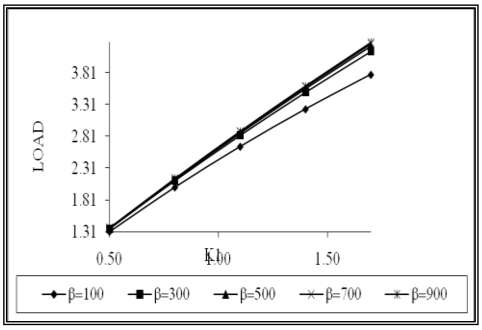

Figure: 15 Variation of load carrying capacity with respect to K_1 and β.

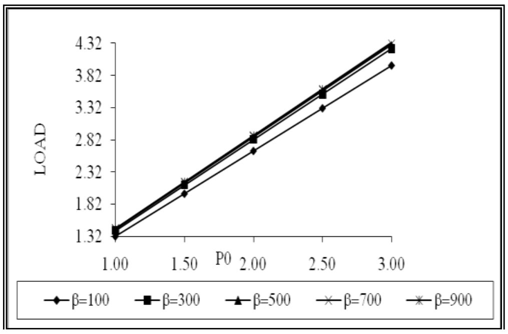

Figure: 16Variation of load carrying capacity with respect to P_0 and β.

CONCLUSION:

This investigation confirms that ever in the absence of flow this type of bearing system supports a good amount of load, in spite of the fact that slip velocity, porosity and roughness turn in adverse effect. The circumstances when negatively skewed roughness occurs remaining favourable in the presence of variance (-ve). Therefore, this study suggests that while designing the bearing system the roughness aspect must be treated carefully.

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