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The Performance of an Idealized Rough Porous Hydrodynamic Plane Slider Bearing

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Abstract:

This paper aims to analyze the performance of an idealized rough porous hydrodynamic plane slider bearing. The stochastic model of Christensen and Tonder has been used (with some modification of the probeblity dencity function) to study the effect of transverse surface roughness in the performance of the bearing system. The modified Darcy's law has been used to account for porosity effect. Solving the associated stochastically averaged Reynold's type equation, the pressure distribution in the bearing system has been obtained which results in the calculation of load carrying capacity. The results indicate that the transverse surface roughness induces an adverse effect on the performance which compounds further due to the negative effect of porosity. However the situation is relatively better in the cash of negatively skewed roughness, which further improves when variance (–Ve) occurs.

Keywords: Plane slider bearing, Porosity. Roughness, Pressure, Load Carrying Capacity.

Introduction:

Abramowitz[1] studied the effect of pad-surface curvature on load capacity. The center of pressure and fluid friction was determined using Reynolds' differential equation in hydrodynamic lubrication theory. The results indicated the practical use of pad curvature for centrally pivoted pads where the variation in fluid viscosity from pad inlet to outlet was negligible. Tzeng and Saibel [2]studied the effect of surface roughness on the load carrying capacity and friction force for a slider bearing. The distinction was emphasized between waviness and roughness. Kapur[3] presented a theoretical analysis for a pivoted slider bearing in the presence of an azimuthal magnetic field. The analysis took into account the pad curvature which was usually neglected. The effects of pad curvature and magnetic field were simultaneously shown in a family of curves and some interesting conclusions were drawn. Prakash and Vij[4] analyzed squeeze film behavior between porous plates of various shapes. The effect of the shape of plate and porosity on the bearing performance was estimated. Agrawal[5] discussed porous inclined slider bearing, lubricated with a magnetic fluid, in the presence of an externally applied magnetic field. It was shown that the magnetic-fluid-based porous inclined slider bearing had a performance superior to that of the conventional fluid based porous inclined slider bearing. Gupta and Kavita[6] developed a mathematical model to investigate the behavior of a plane-inclined porous slider bearing under the effect of a uniform small rotation. The Beavers-Joseph slip condition was used for the slip velocity at the porous boundary. It was shown that for a rotation (in the negative direction) the load-carrying capacity increased and the coefficient of friction decreased. The load-carrying capacity also decreased with increase in the slip velocity parameter. Bhat and Deheri[7]presented a theoretical study of a porous composite slider bearing lubricated with magnetic fluid. Magnetic fluid increased the load capacity, did not alter the friction, decreased the coefficient of friction and shifted the centre of pressure towards the inlet Das[8] presented the theoretical study of slider bearings in some general form. He considered the lubricant to be an isothermal, incompressible electrically conducting couple stress fluid in the presence of a uniform magnetic field. A comparative study of optimum load-carrying capacity for finite and infinite slider bearings was also made.. Prat et. al, [9] investigated the flow between rough surfaces in sliding motion with contacts between these surfaces. The volume averaging method was adopted to analyze the problem. An average flow model combining spatial and time average was developed using Reynolds approximation at the roughness scale.Shah and Bhat [10] made comparison between squeeze film behaviour in an infinitely long journal bearing using the ferrofluid flow model of Neuringer–Rosensweig, Jenkins and Shliomis with uniform and non-uniform magnetic fields. It was concluded that a uniform magnetic field could not produce magnetic pressure in the Neuringer–Rosensweig model, but it could affect the bearing characteristics in the Shliomis model owing to the rotational velocity. The influence of longitudinal surface roughness on the thermo hydrodynamic lubrication of an infinitely wide plane slider bearing was analyzed by Sharma and Pandey [11]. It was found that the increase in roughness parameter resulted significant reduction in load carrying capacity(thermal) of bearing due to increase in lubricant temperature. Shukla and Deheri[12] dealt with the performance of a rough hyperbolic slider bearing under the presence of a magnetic fluid lubricant. It was revealed that the negative effect of transverse roughness could be reduced to certain extent by the positive effect of magnetization in case variance(-ve) occurred. Patel and Deheri[13] investigated the performance of a transversely rough porous parallel plate slider bearing with slip velocity taking a magnetic fluid as the lubricant. It was noticed that for an improved performance, slip deserved to be kept at minimum. Patel et. al.[14] studied the effect of transverse surface roughness on a Rayleigh step bearing in the presence of a magnetic fluid. It was concluded that the adverse effect of roughness could be minimized by the positive effect of magnetization at least in the case of negatively skewed roughness. Singh and Gupta[15]presented a theoretical investigation related to the effect of ferro-fluid on the dynamic characteristics of curved slider bearings. This study made use of Shliomis model which accounted for the rotation of magnetic particles, their magnetic moments, and the volume concentration in the fluid. It was observed that the effect of rotation of magnetic particles improved the stiffness and damping capacity of the bearings.

Here, it has been proposed to analyze the performance of an idealized rough porous hydrodynamic plane slider bearing.

Analysis:

A hydrodynamic plane-slider bearing is made of two plane, non-parallel surfaces separated by a lubricant film (as shown in figure-1).

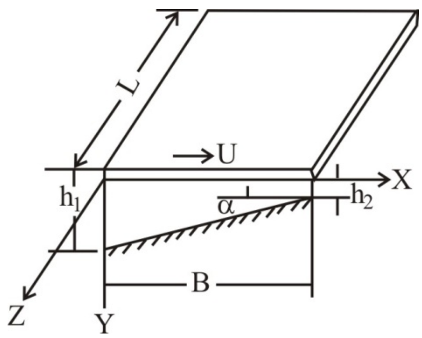

Figure: 1 Configuration of bearing system

One of the surfaces is normally stationary while the other moves with a constant speed. The direction of motion and the inclination are given in such a way that the convergent film is formed. The stationary plane can either be fixed or pivoted so that it can assume any inclination relative to the moving plane[16].

Following the stochastic modeling of surface roughness by Christensen and Tonder [17-19] the thickness $h(x)$ is considered as

$$h(x) = \overline{h}(x) + h_s \tag{1}$$

where $\bar{h}(x)$ 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 assumed to be stochastic in nature and governed by the probability density function

$$f(h_s) = \begin{cases} \dfrac{15}{16c^7}\left(c^2 - h_s^2\right)^3, -c \le h_s \le c \\ 0, Otherwise \end{cases} \tag{2}$$

where c is the maximum deviation from the mean film thickness. The mean $\bar{\alpha}$, the standard deviation $\bar{\sigma}$ and the parameter $\bar{\varepsilon}$ which is the, measure of symmetry of the random variable h_s are defined by the relationships

$\bar{\alpha} = E(h_s)$, $\bar{\sigma}^2 = E\left\{\left(h_s - \bar{\alpha}\right)^2\right\}$ and $\bar{\varepsilon} = E\left\{\left(h_s - \alpha\right)^3\right\}$

where, E denotes the expected value defined by

$$E(R) = \int_{-c}^{c} R\, f(h_s)\, dh_s \tag{3}$$

In this section hydrodynamic pressure, load carrying capacity, friction force and coefficient of fluid friction for a plane-slider are derived. For an idealized plane slider bearing (that is, there is no variation of pressure in z-direction), the Reynolds equation associated is

$$\frac{dp}{dx} = 6\eta U\left[\frac{h - h_m}{h^3}\right] \tag{4}$$

While h_m is maximum film thickness, where $\frac{dp}{dx} = 0$

$k = \dfrac{h_1 - h_2}{B}$ where $h = h_1 - kx$,

Stochastically averaging this equation along the model adopted in [17-19] one finds that the fluid film pressure is governed by a generalized form of the Reynolds' equation of the type

$$\frac{dp}{dx} = 6\eta U\left[\frac{h - h_m}{g(h)}\right], \tag{5}$$

For simplicity[16]

let $h_m = \dfrac{2nh_2}{n+1}$

And

$g(h) = h^3 + 3\alpha h^2 + 3(\sigma^2 + \alpha^2)h + 3\sigma^2\alpha + \alpha^3 + \varepsilon + 12\varphi H$

The associated boundary conditions are

p= 0 at x = 0, $h = h_1$ and p = 0 at x = B, $h = h_2$

Solution of eq.(5) in view of boundary conditions, one obtains

$$p = 6\eta U \int_0^x \left[\frac{h - h_m}{g(h)} \right] dx \tag{6}$$

Introducing the non dimensional quantities,

$$p = \frac{h_2^2}{nuB} P \ , \ n = \frac{h_1}{h_2} \ , \ \bar{\alpha} = \frac{\alpha}{h_2} \ , \ X = \frac{x}{B}$$

$$\bar{\sigma} = \frac{\sigma}{h_2}, \bar{h} = \frac{h}{h_2}, \bar{\varepsilon} = \frac{\varepsilon}{h_2^3}, \psi = \frac{\varphi H}{h_2^3},$$

The dimensionless pressure distribution turns out to be

$$P = \frac{h_2^2}{nuB} p$$

$$P = 6 \int_0^X \left(\frac{\bar{h} - \frac{2n}{n+1}}{g(\bar{h})} \right) dX \tag{7}$$

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

For parallel plate using

$\bar{h} = 1,$

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

The non-dimensional load carrying capacity is calculated from

$$W = \frac{h_2^2}{\eta U B^2 L} w, where.....w = \int P dx,$$

$$w = \int_0^1 \left(\frac{1 - \frac{2n}{n+1}}{g(1)} \right) dX \tag{8}$$

The non-dimensional pressure distribution is given by equation (7) while equation (8) accounts for dimensionless load carrying capacity. It can be easily seen that this bearing system is equivalent to a bearing system with thickness $\{g(1)\}^{1/3}$.

Result and Discussion:

It is found the expression (8) is dependent on various parameters such as $n, \bar{\alpha}, \bar{\sigma}, \bar{\varepsilon}$ and ψ. The effects of these parameters are presented graphically below:

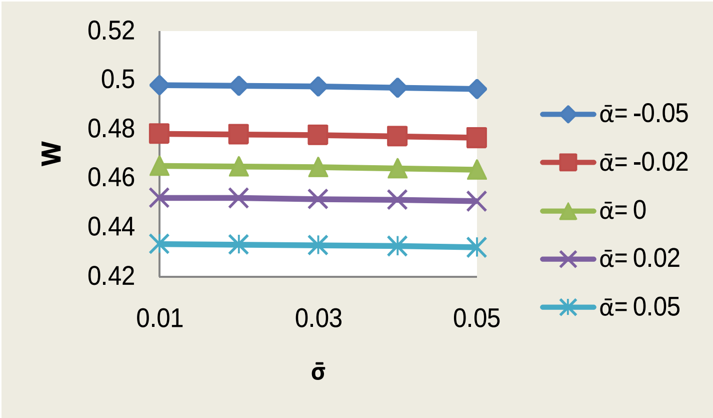

Figure: 2 Variation of Load carrying capacity with respect to $\bar{\sigma}$ for various values of $\bar{\alpha}$.

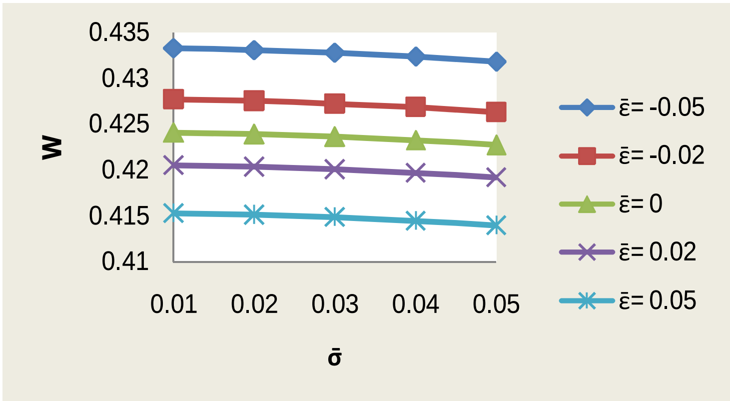

Figure: 3 Variation of Load carrying capacity with respect to $\bar{\sigma}$ for various values of $\bar{\varepsilon}$.

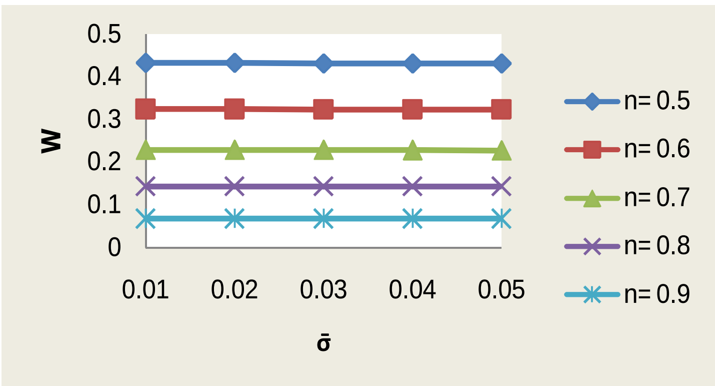

Figure: 4 Variation of Load carrying capacity with respect to $\bar{\sigma}$ for various values of n.

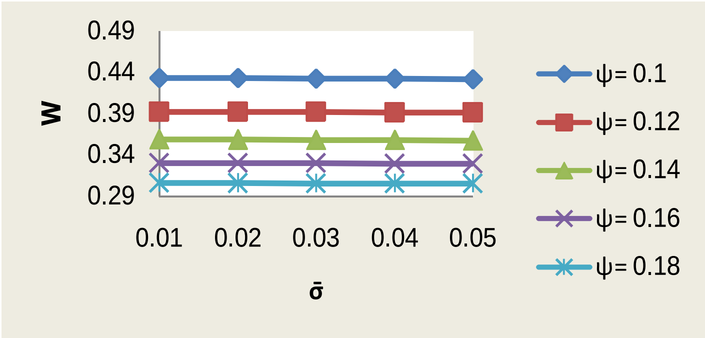

Figure: 5 Variation of Load carrying capacity with respect to $\bar{\sigma}$ for various values of ψ.

Interestingly, the standard deviation associated with roughness turns in a negligible or at the best marginal effect as it can be seen from fig.(2) to fig.(5). The nominally decreased load carrying capacity due to $\overline{\sigma}$, gets further decreased due to porosity fig.(5)

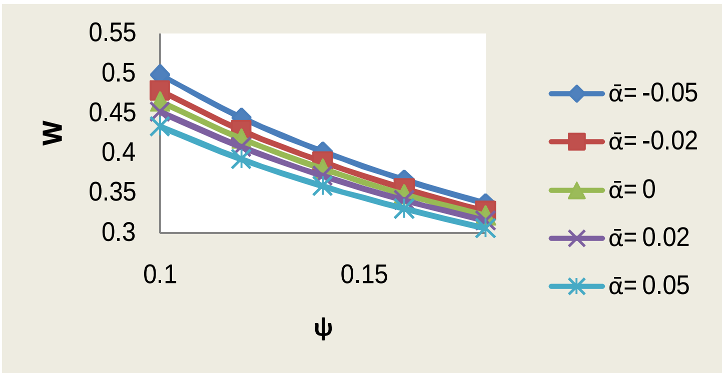

Figure: 6 Variation of Load carrying capacity with respect to ψ for various values of $\bar{\alpha}$.

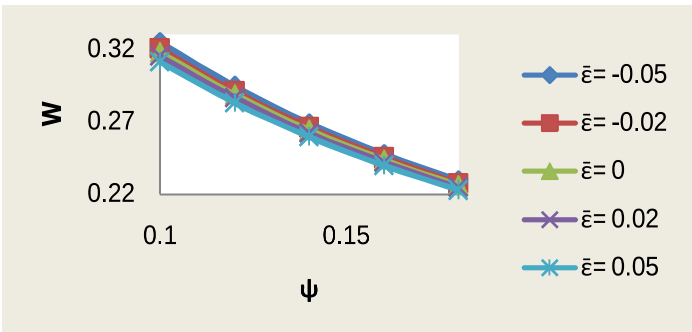

Figure: 7 Variation of Load carrying capacity with respect to ψ for various values of $\bar{\varepsilon}$.

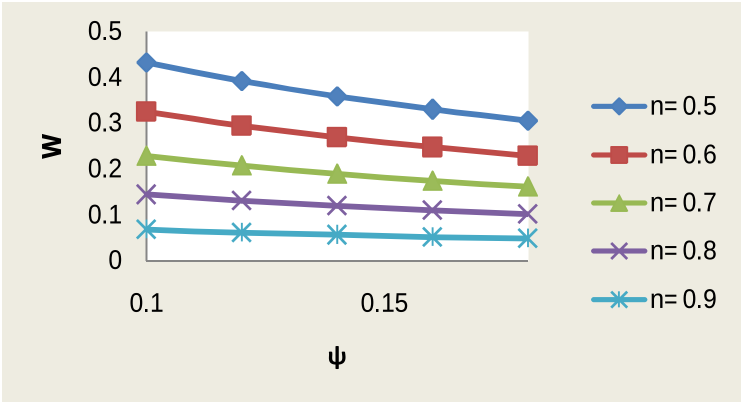

Figure: 8 Variation of Load carrying capacity with respect to ψ for various values of n.

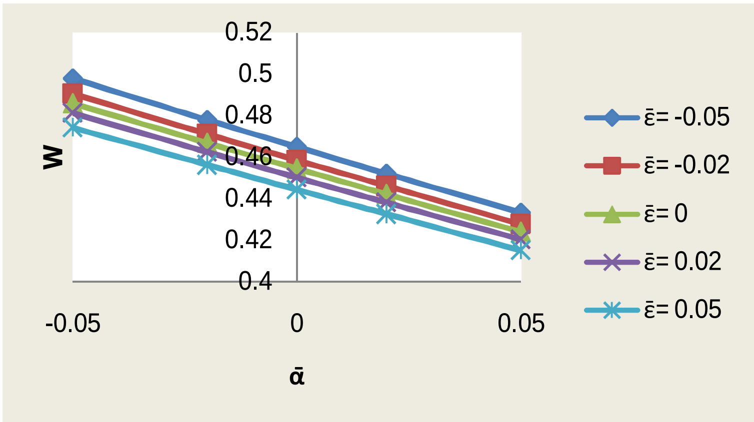

Figure: 9 Variation of Load carrying capacity with respect to $\bar{\alpha}$ for various values of $\bar{\varepsilon}$.

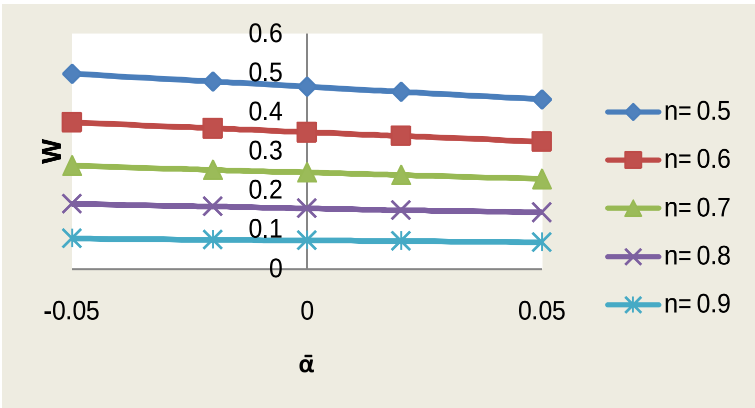

Figure: 10 Variation of Load carrying capacity with respect to $\bar{\alpha}$ for various values of n.

The load carrying capacity gets decreased due to the positive skewness while negatively skewed roughness increases the load carrying capacity [9,10].

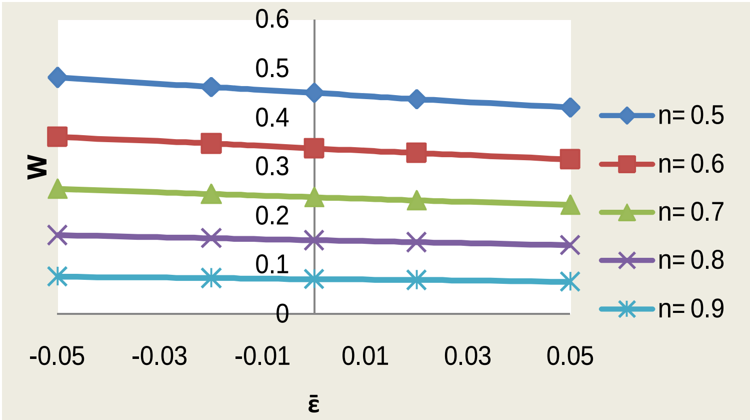

Figure: 11 Variation of Load carrying capacity with respect to $\bar{\varepsilon}$ for various values of n.

That the variance follows the path of skewness is reflected in figure (11).
Thus the combined effect of variance (-ve) and negatively skewed roughness is significantly positive. The effect of skewness on the distribution of load carrying capacity with respect to porosity is nominal in figure(7)

Conclusion:
The bearing system suffers due to transverse roughness in general. Hence this study makes it clear that the roughness aspect must be evaluated, while designing this type of bearing system. This is highly essential from bearing's life period point of view.

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