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PERFORMANCE OF A ROUGH SHORT POROUS BEARING CONSIDERING THERMAL EFFECT AND SLIP VELOCITY

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ABSTRACT

Efforts have been made to analyze the performance characteristics of a rough, porous short bearing with slip velocity and thermal effect. The stochastic model of Christensen and Tonder has been used to study the effect of transverse surface roughness in the performance of the bearing system. The effect of slip velocity has been studied by Beavers and Joseph's slip model. The model of Tipei has been adopted for the thermal effect. The associated stochastically averaged Reynolds type equation is solved with appropriate boundary condition to obtain the pressure distribution, in turn, which gives the expression for Load Carrying Capacity (L.C.C.). The results shown in graphical form establish that transverse roughness and slip velocity induce an adverse effect on the bearing system. In the case of negatively-skewed roughness the situation yields better results. The thermal effect induced negative effect can be retrieved in the case of negative skewed roughness, to certain extent.

Index Terms: Short bearing, porosity, roughness, slip velocity, thermal effect, load carrying capacity.

INTRODUCTION

"Hydrodynamic lubrication" is a process by which two surfaces moving at some relative velocity with respect to each other are separated by a fluid film in which forces are generated by the relative motion only.

Lubricated bearings are well known in industrial applications and their main advantages compared with rolling and friction bearings are their low friction and high precision.

R.I.Tanner [1] studied an isothermal short journal bearing with non-newtonian lubricants and the method adopted to here was applied to predict the temperature in the short journal bearing.

Rohde and Li [2] also analyzed short bearing performance where Benerjee et al. [3] investigated the short bearing performance consider the effect of rotation about an axis across the fluid film.

Short bearings have been subjected to investigation by Patel et al. [4], Shimpi and Deheri [5], Patel and deheri [6].

However, bearing surfaces could be rough through manufacturing process, the wear and the impulsive damage, In order to understand the effect of surface roughness. Tzeng and Seibel [7] dealt with the effect of roughness. A simple statistical model was mooted by Christensen and Tonder [8,9,10] to evaluate the effect of surface roughness. Later, this approach of Christensen and Tonder was employed in many investigators Deheri and Andhria [11], Patel et al. [12].

Patir and Cheng [13] proposed an average flow model for deriving the Reynold type equation which was applicable to any general surface roughness structure. Dresse and Sinha [14] analyzed the roughness and thermal effect on different characteristics of finite rough tilted pad slider bearings. It was observed that for non-parallel slider bearings that the load carrying capacity due to combined effect was less than the load capacity due to roughness effect for both longitudinal and transverse roughness model.

The performance of an idealized rough porous hydrodynamic plane slider bearing was studied by Patel et al. [15]

Porous bearings have the features of simple structure and low cost. Porous bearings are used where non-porous bearings are impracticable owing to lack of space or inaccessibility for lubrication. The application of Porous bearings in mounting horsepower motors include vacuum cleaners, coffee grinders, hair dries, saving machines, sewing machines, water pumps, tape recorders, generators and distributors. The hydrodynamic lubrication theory of porous bearings was first studied by Morgan and Cameron [16]. Ahmad and Singh [17], Kashinath and Hanumagowda [18], Shah and Patel [19] Thakkar et.al [20] have analyzed the porous slider bearing by using Darcy's equation to model the flow of Newtonian lubricant in the porous matrix.

Tipei [21] observed that the highest temperature happened when the film thickness was least.

The thermal effect was also found to be analyzed by Kumar at al. [22] and Patel at al. [23]

Here it has been proposed to study the performance of a rough short porous bearing considering thermal effect and slip velocity.

Analysis

The geometry and configuration of the present bearing system is shown in the Fig.1. The slider moves with the uniform velocity u in the X direction.

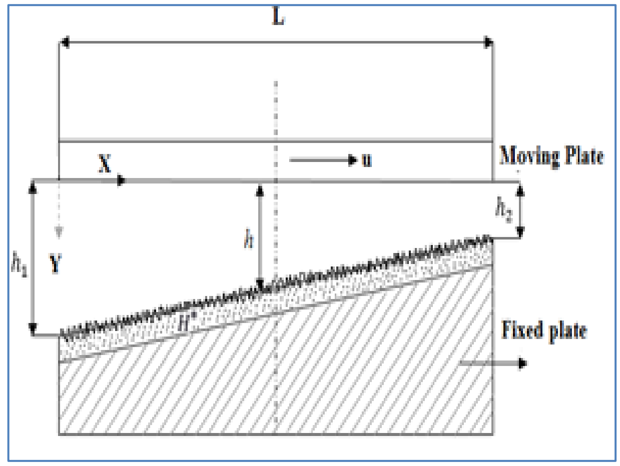

Fig. 1. Configuration of bearing system.

The length of the bearing is L and breadth B is in Z direction where $B \ll L$, The dimension of B to be very small.

It is widely known that the Reynolds pressure distribution equation for two-dimensional fluid flow by Cameron [24].

$$\frac{\partial}{\partial x}\left(h^3\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(h^3\frac{\partial p}{\partial z}\right)=6\eta u\frac{dh}{dx} \tag{1}$$

The pressure gradient ${\partial p}/{\partial Z}$ is much larger as compared to the pressure gradient ${\partial p}/{\partial X}$.

In the light of Christensen and Tonder [8,9,10] the thickness h of the lubricant film is assumed to be

$$h = \bar{h} + h_s \tag{2}$$

Where $\bar{h}$ is the mean film thickness and h_s is the deviation from the mean film thickness characterizing the random roughness of the bearing surfaces.

The details regarding the roughness characterization an application can be hold from Christensen and Tonder [8,9,10].

Under the usual assumptions of hydrodynamic lubrication and employing the Beavers and Joseph's model for slip and Christensen and

Tonder roughness model the governing Reynolds equation, Patel and deheri [6] is given by

$$\frac{d^2p}{dz^2} = \frac{6\eta u}{g(h)}\left(\frac{2+sh}{4+sh}\right)\frac{dh}{dx} \tag{3}$$

Where,
$$g(h) = h^3 + 3\alpha h^2 + 3(\sigma^2+\alpha^2)h + 3\sigma^2\alpha + \alpha^3 + \varepsilon + 12kH^*$$
and $\quad h = h_2\left\{1 + m\left(1 - \frac{x}{L}\right)\right\}$

Equation (3) transforms to the following

$$\frac{d^2p}{dz^2} = \frac{-mh_2}{L}\frac{6\eta u}{g(h)}\left(\frac{2+sh}{4+sh}\right) \tag{4}$$

The thermal effect gives the viscosity-temperature relation as $\eta = \eta_0\left(\frac{h}{h_2}\right)^q$ where, q is the thermal factor which usually lies between 0 and 1 according to the nature of the lubrication in Tipei [10].

Here, η is the fluid viscosity and at $h = h_2$ is known as η_0. The associated boundary conditions $\frac{dp}{dz} = 0\ at\ z = 0$ and $p = 0\ at\ z = \pm\frac{B}{2}$.

The following dimensionless quantities are introduced

$$A = \frac{h}{h_2}, m = \frac{h_1 - h_2}{h_2}, Z = \frac{z}{B}, X = \frac{x}{L}, \bar{s} = sh_2,$$

$$P = \frac{{h_2}^{q-3}p}{\eta_0 uB^2}, \bar{\alpha} = \frac{\alpha}{h_2}, \bar{\sigma} = \frac{\sigma}{h_2}, \bar{\varepsilon} = \frac{\varepsilon}{{h_2}^3}, \psi = \frac{kH^*}{{h_2}^3}$$

Integrating equation (4) with the associated boundary conditions and dimensionless quantities with thermal parameter, one get the pressure equation in non-dimensional form as

$$P = (g(\mathrm{A}))^{\frac{q}{3}-1}\left(\frac{3mh_2}{L}\right)\left(\frac{2+\bar{s}\mathrm{A}}{4+\bar{s}\mathrm{A}}\right)\left(\frac{1}{4} - Z^2\right) \tag{5}$$

Now the dimensionless load carrying capacity of the bearing system comes out to be

$$W = \pi\int_{-\frac{1}{2}}^{\frac{1}{2}}\int_0^1 P(X,Z)dXdZ = \frac{\pi}{2}\int_0^1 (g(\mathrm{A}))^{\frac{q}{3}-1}\left(\frac{mh_2}{L}\right)\left(\frac{2+\bar{s}\mathrm{A}}{4+\bar{s}\mathrm{A}}\right)dX \tag{6}$$

Result and discussion

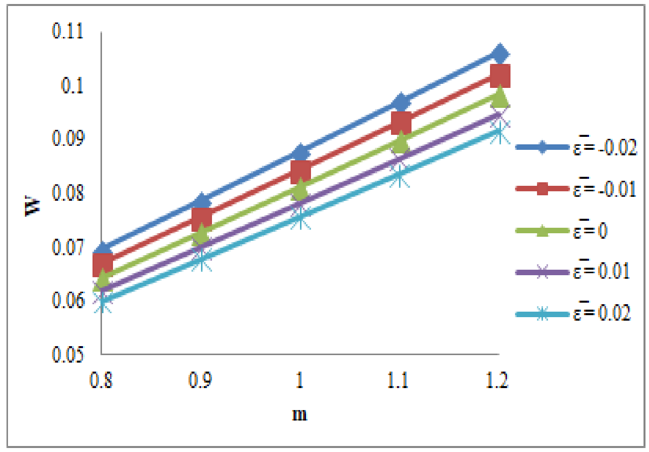

Fig. 2. Variation of load carrying capacity with respect to m to $\bar{\varepsilon}$

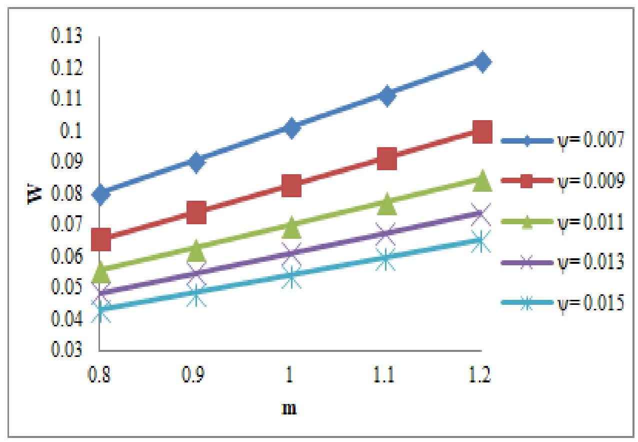

Fig. 3. Variation of load carrying capacity with respect to m to ψ

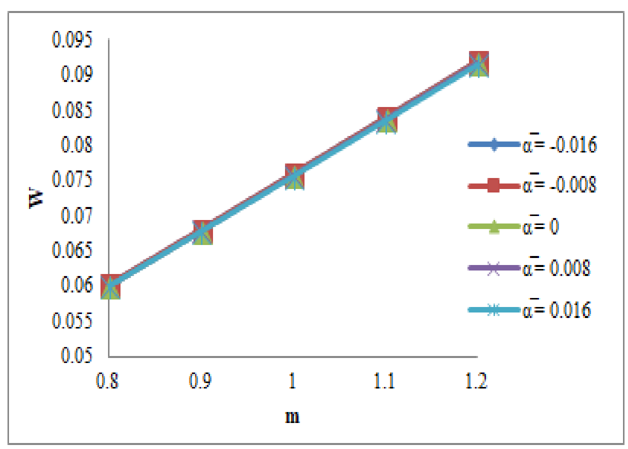

Fig. 4. Variation of load carrying capacity with respect to m to $\bar{\alpha}$

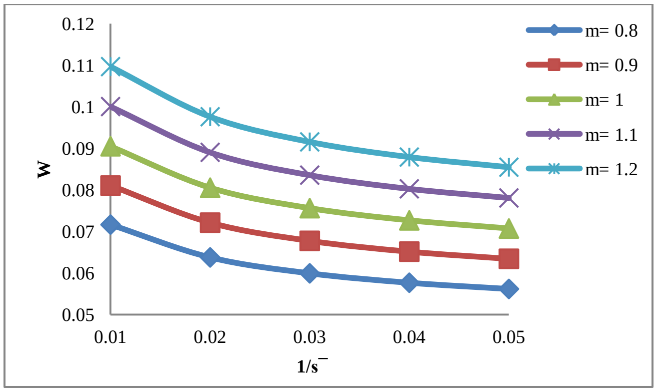

Fig. 5. Variation of load carrying capacity with respect to $^1/_{\bar{s}}$ to m

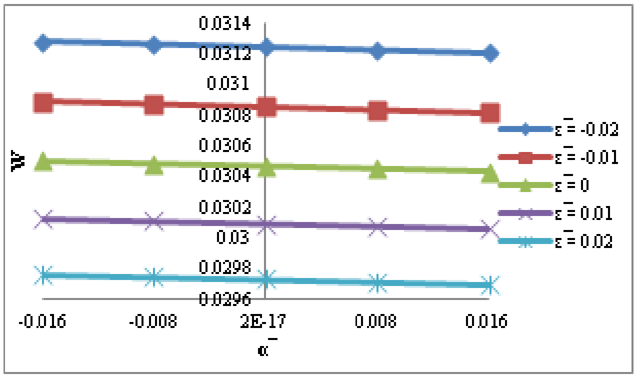

Fig. 6. Variation of load carrying capacity with respect to $\bar{\alpha}$ to $\bar{\varepsilon}$

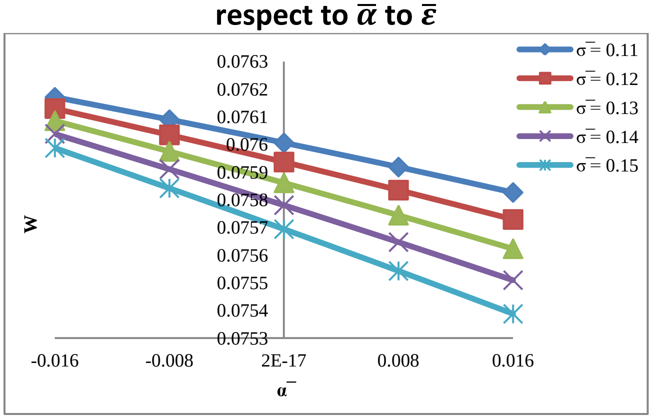

Fig. 7. Variation of load carrying capacity with respect to $\bar{\alpha}$ to $\bar{\sigma}$

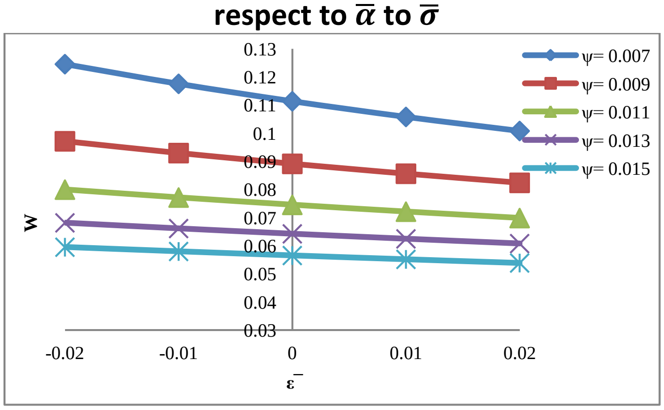

Fig. 8. Variation of load carrying capacity with respect to $\bar{\varepsilon}$ to ψ

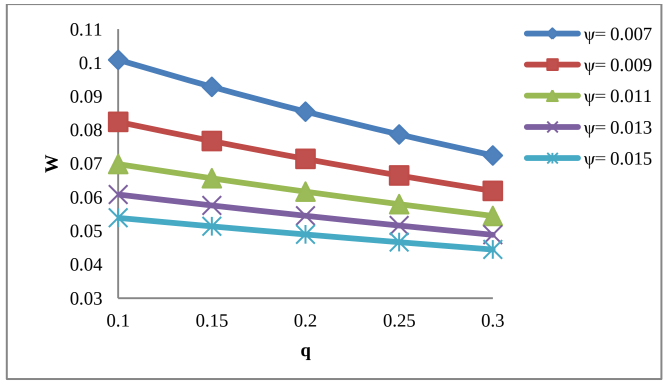

Fig. 9. Variation of load carrying capacity with respect to q to ψ

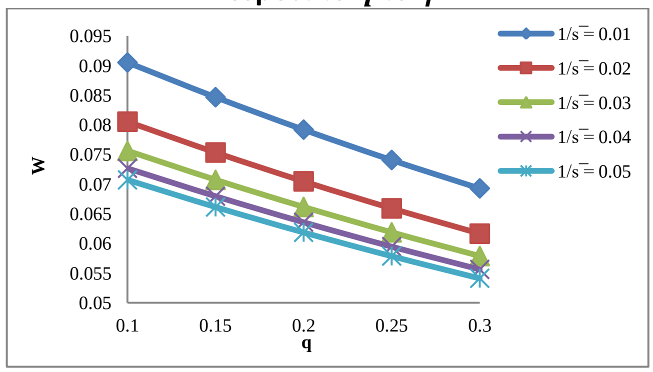

Fig. 10. Variation of load carrying capacity with respect to q to $^1/_{\bar{s}}$

The effect of aspect ratio on the load carrying capacity is presented in figs. 2-4. It is observed that the aspect ratio causes increased load carrying capacity in the sense that the load carrying capacity increases sharply.

The load carrying capacity decreases with increasing slip velocity in fig.5.

Figs. 6 and 7 underlines that the load carrying capacity decrease with respect to positive variance while the negative variance induces an increase in the load carrying capacity.

The fact that the standard deviation has considerable adverse effect on the performance of the bearing system. However, the decrease remains nominal in the case of variance while it remains negligible in the case of standard deviation as can be seen from fig.7.

The effect of skewness on the load carrying capacity follows almost the trends of the variance in figs. 6 and 8. Therefore, the increased load carrying capacity due to variance (negative) gets further increased owing to the negatively skewed roughness.

As usual form figs. 3, 8 and 9 it is noticed that the load carrying capacity decreases with increasing values of porosity.

In figs. 9 and 10 it can be seen that the load carrying capacity reduces when value of thermal parameter increases.

Conclusions

This investigation confirms that the thermal effect is quite significant in this type of bearing systems. But the situation is registered to be a tad better in the case of negative skewed roughness. The slip parameter needs to be put at a minimum value. It is suggested that the adverse effect of slip velocity can be compensated by a suitable aspect ratio value. However, if properly designed this type of bearing system can be of some help to the industry, as an aspect ratio has a prominent role to play.

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NOMENCLATURE

H^*	=	Thickness of porous layer
h_1, h_2	=	Maximum and minimum film thickness
p	=	Lubricant film pressure
P	=	Dimensionless pressure
W	=	Load carrying capacity
$\overline{W}$	=	Dimensionless load carrying capacity
$\bar{\alpha}$	=	Non-dimensional variance
s	=	Slip parameter
$\bar{\varepsilon}$	=	Skewness in dimensionless form
ρ	=	Fluid density
$\bar{\sigma}$	=	Dimensionless standard deviation
ψ	=	Porosity
u	=	Uniform velocity in X- direction
m	=	Aspect ratio
α	=	Variance
σ	=	Standard deviation
ε	=	Skewness