Chapter-5

The Performance of an Idealized Rough Porous Hydrodynamic Plane Slider Bearing

This chapter 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 probability density 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.

5.1 Introduction

Abramowitz (1955) 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 (1967) 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 (1969) 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 (1973) 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 (1986) 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 (1986) 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 (1991) 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 (1998) 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, (2002) 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 (2004 b) 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 (2005). 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 (2011 b) 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 (2011) 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. (2012) 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 (2012) presented a theoretical investigation related to the effect of ferro-fluid on the dynamic characteristics of curved slider bearings. They used, 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. Naduvinamani, et. al. (2014) studied Combined effects of surface roughness and viscosity-pressure dependency on the couple stress squeeze film characteristics of parallel circular plates. They found that the effects of couple stresses and viscosity-pressure dependency were to increased the load carrying capacity, and squeeze film time for both types of roughness patterns. Nabhani and Khlifi (2015) presented a numerical study of magneto hydrodynamic (MHD) infinitely wide plane inclined slider bearing including both fluid inertia and non-Newtonian couple stress effect. It was found that the couple stress effects of fluid inertia forces, MHD, and non-Newtonian couple stresses provided a significant improvement in the slider bearing load capacity compared to the case of the non inertia Newtonian non conducting lubricant. The use of conducting lubricant diminished the negative effect of inertia forces on the friction coefficient. Panchal et al. (2016 b) analyzed the influence of roughness parameters on the pressure and load carrying capacity in a rough finite plane slider bearing for longitudinally rough surfaces by taking account of the influence of surface roughness through a series of flow factor. It was found that the increment in the measure of longitudinal roughness caused the decrease in load carrying capacity of the bearing. Rao and Agarwal (2016) studied the inclined slider bearing with porous layer attached to slider as well as stator including effects of slip parameter in the X-direction for the upper and lower porous regions with different permeability parameters and couple stresses. The modified Reynolds equation was solved analytically and closed form expressions were obtained for the fluid film pressure, load capacity, friction on the slider and coefficient of friction to examine possible effects on the system.

5.2 Analysis

A hydrodynamic plane-slider bearing is made of two plane, non-parallel surfaces separated by a lubricant film (as shown in figure-5,1).

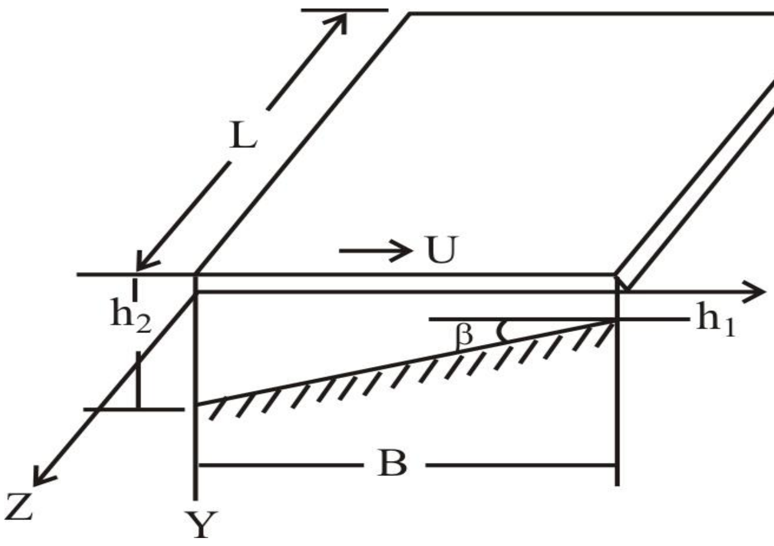

Fig. 5.1 : Configuration of bearing system

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 (Majumdar-2008).

The stochastic modeling of surface roughness by Christensen and Tonder (1969 a, 1969 b, 1970) is discussed in chapter 4 in detail.

In this chapter hydrodynamic pressure and load carrying capacity 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

$$\frac{dp}{dx} = 6\eta U \left[\frac{h - h_m}{h^3} \right] \tag{5.1}$$

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

$$\beta = \frac{h_1 - h_2}{B} \text{ where } h = h_1 - \beta x,$$

Stochastically averaging this equation along the model adopted by by Christensen and Tonder (1969 a, 1969 b, 1970), one finds that the fluid film pressure is governed by a generalized form of the Reynolds' equation of the type

$$\frac{dp}{dx} = 6\eta U \left[\frac{h - h_m}{g(h)} \right] \tag{5.2}$$

For sake of simplicity (Majumdar-2008), Let

$$h_m = \frac{2nh_2}{n+1}$$

and

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

The associated boundary conditions are

$$p = 0 \text{ at } x = 0,\ h = h_1 \text{ and } p = 0 \text{ at } x = \text{B},\ h = h_2$$

Solution of equation(5.2) in view of boundary conditions, one obtains

$$p = 6\eta U \int_0^x \left[\frac{h - h_m}{g(h)} \right] dx \tag{5.3}$$

Introducing the non dimensional quantities,

$$P = \frac{h_2^2}{\eta UB} p,\ n = \frac{h_1}{h_2},\ \bar{\alpha} = \frac{\alpha}{h_2},\ X = \frac{x}{B}$$
$$\bar{\sigma} = \frac{\sigma}{h_2},\ \bar{h} = \frac{h}{h_2},\ \bar{\varepsilon} = \frac{\varepsilon}{h_2^3},\ \psi = \frac{\phi H}{h_2^3},$$

The dimensionless pressure distribution turns out to be

$$P = \frac{h_2^2}{\eta UB} p$$

$$P = 6\int_0^X \left(\frac{\bar{h} - \frac{2n}{n+1}}{g(\bar{h})} \right) dX \tag{5.4}$$

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

$$\bar{h} = 1,$$

one avails that,

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

The non-dimensional load carrying capacity is calculated from

$$W = \frac{h_2^2}{\eta U B^2 L} w$$

where

$$w = \int P dx$$

which yields,

$$W = \int_0^1 \left(\frac{1 - \frac{2n}{n+1}}{g(1)} \right) dX \tag{5.5}$$

The non-dimensional pressure distribution is given by equation (5.4) while equation (5.5) accounts for dimensionless load carrying capacity. It can be easily seen that this bearing system is equivalent to a bearing system with thickness $\{g(1)\}^{1/3}$.

5.3 Result and Discussion

It is found the expression (5.5) is dependent on various parameters such as $n, \bar{\alpha}, \bar{\sigma}, \bar{\varepsilon}, \bar{\psi}$. The effects of these parameters are presented graphically below:

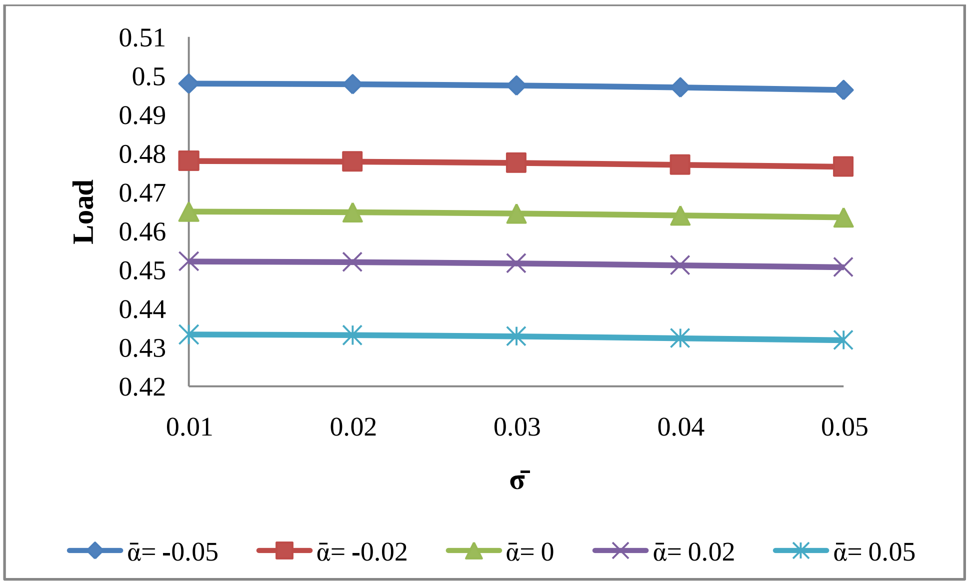

Fig. 5.2 : Variation of Load carrying capacity with respect to $\bar{\sigma}$ and $\bar{\alpha}$

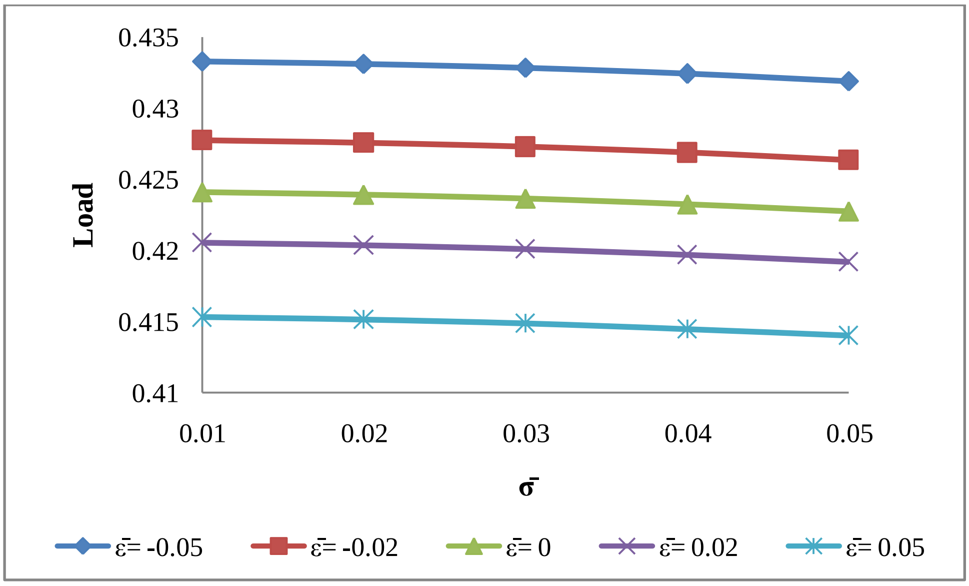

Fig. 5.3 : Variation of Load carrying capacity with respect to $\bar{\sigma}$ and $\bar{\varepsilon}$

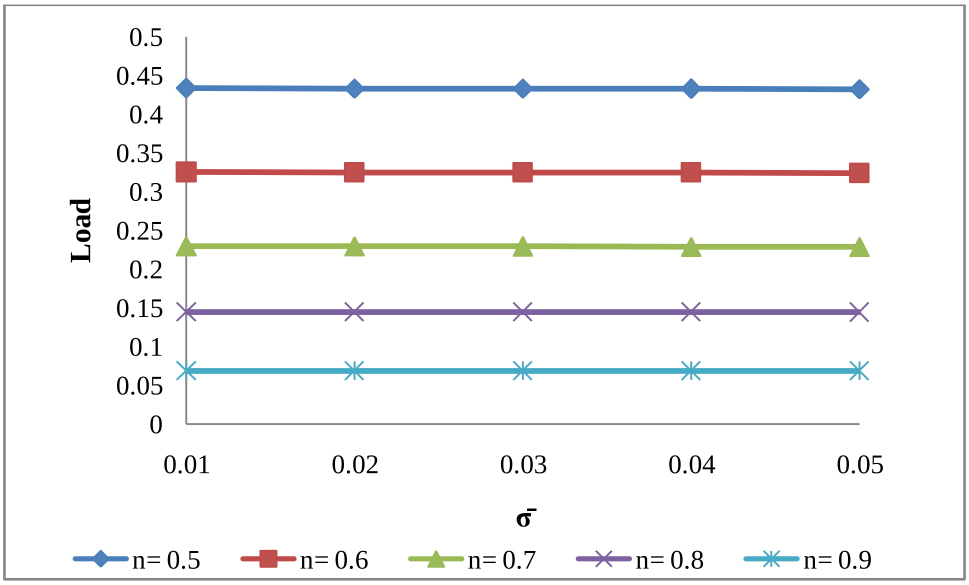

Fig. 5.4 : Variation of Load carrying capacity with respect to $\bar{\sigma}$ and *n*

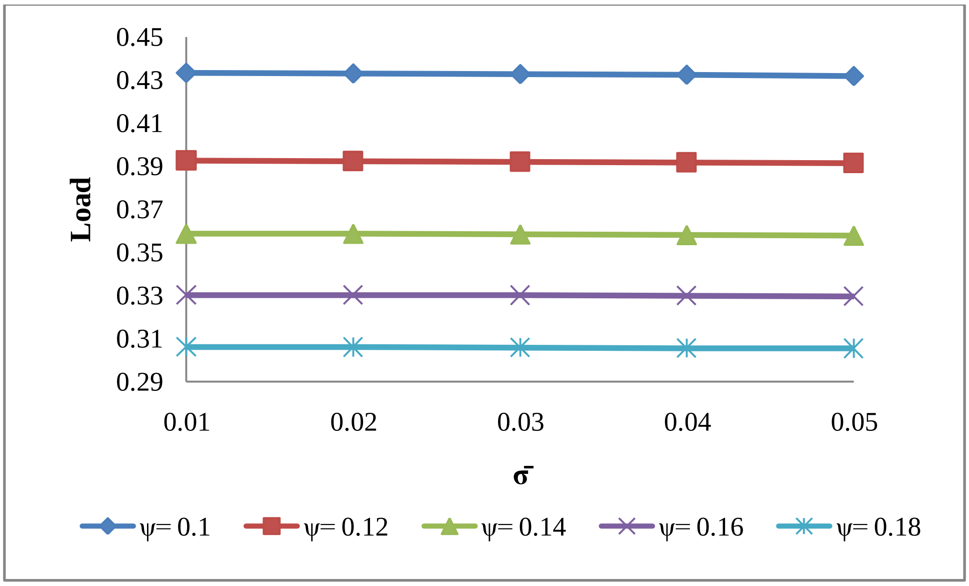

Fig. 5.5 : Variation of Load carrying capacity with respect to $\bar{\sigma}$ and ψ

Interestingly, the standard deviation associated with roughness turns in a negligible or at the best marginal effect as it can be seen from figure 5.2 to figure 5.5. The nominally decreased load carrying capacity due to $\bar{\sigma}$.

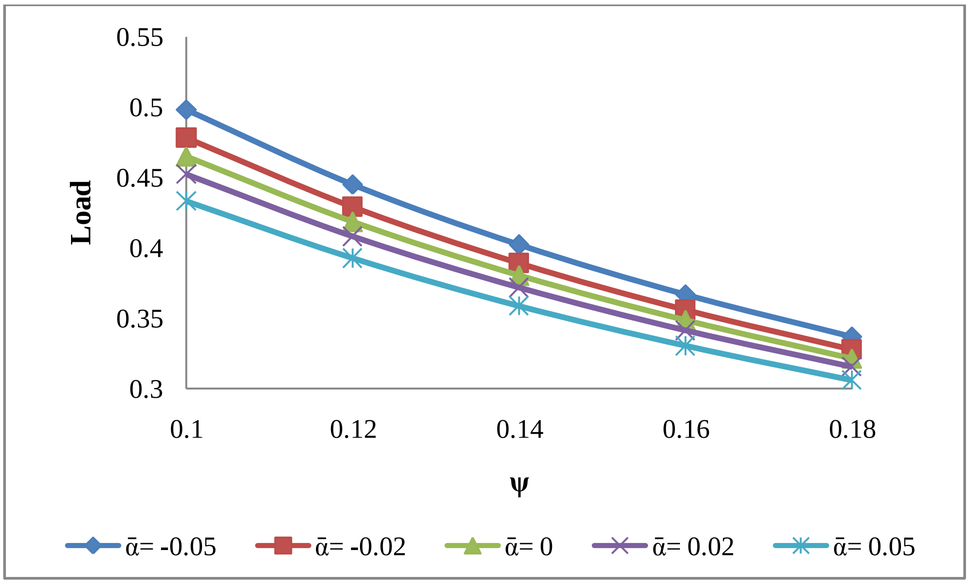

Fig. 5.6 : Variation of Load carrying capacity with respect to ψ and $\bar{\alpha}$

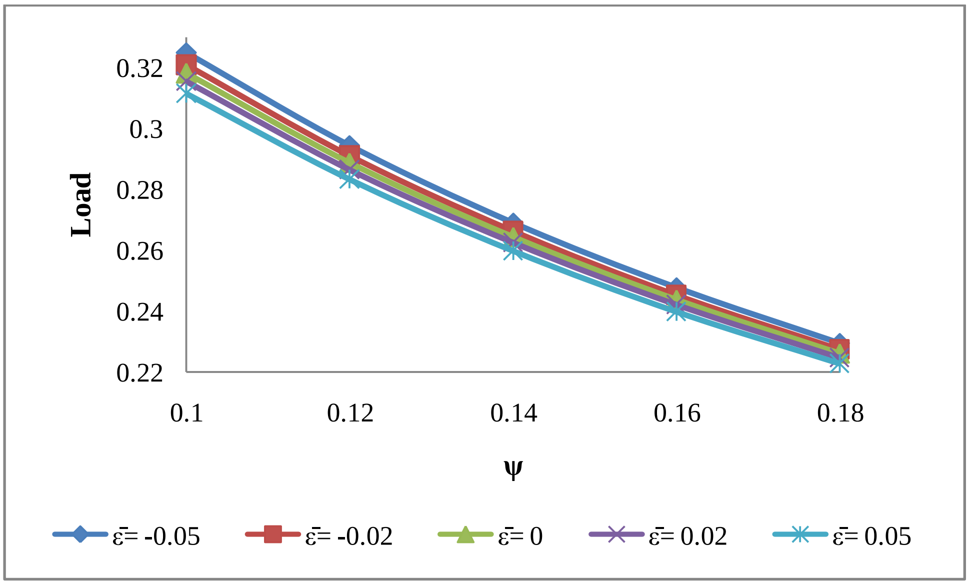

Fig. 5.7 : Variation of Load carrying capacity with respect to ψ and $\bar{\varepsilon}$

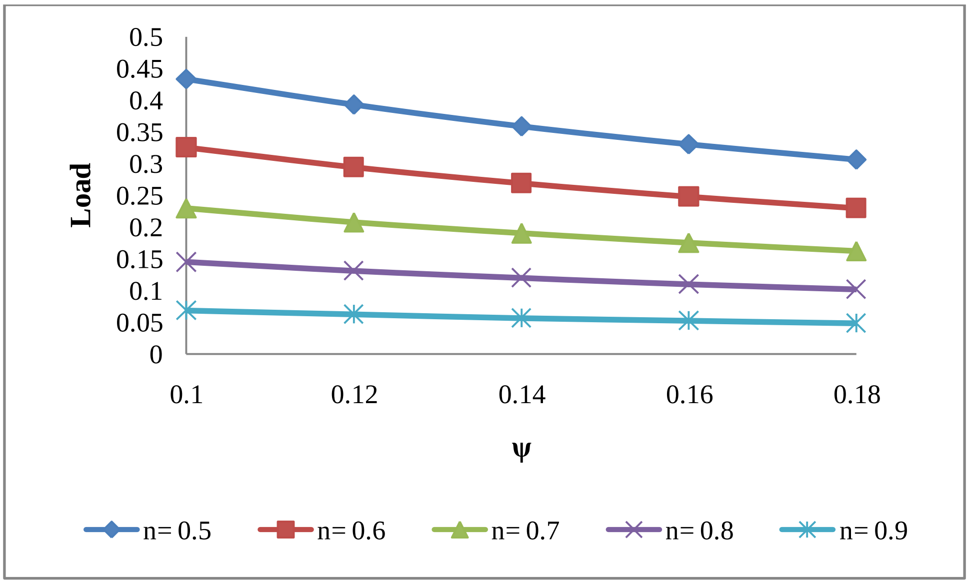

Fig. 5.8 : Variation of Load carrying capacity with respect to ψ and *n*

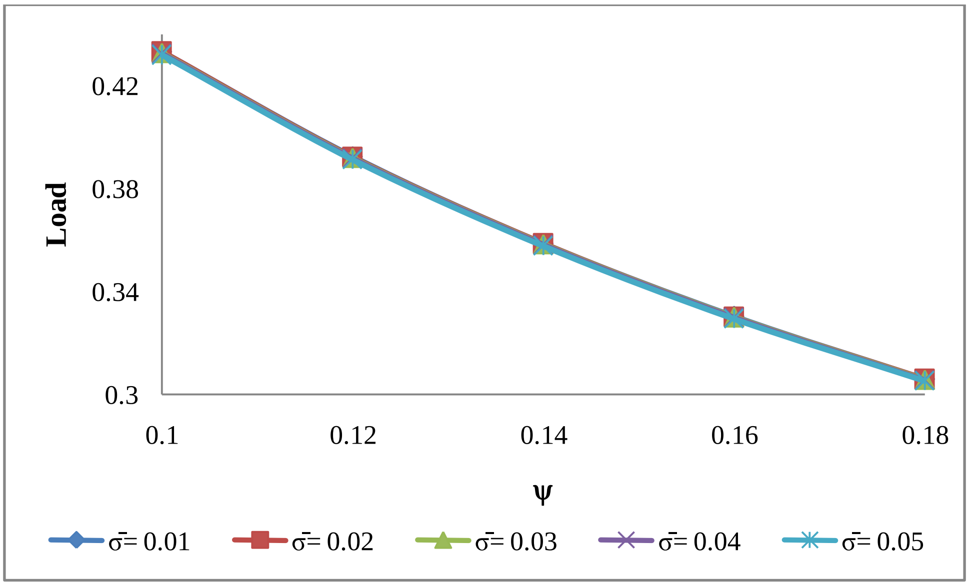

Fig. 5.9 : Variation of Load carrying capacity with respect to ψ and $\bar{\sigma}$

We can observe from figures 5.6 – 5.9 that, load carrying capacity decreased due to porosity. Further, the nominally decrease effect of standard deviation is seen.

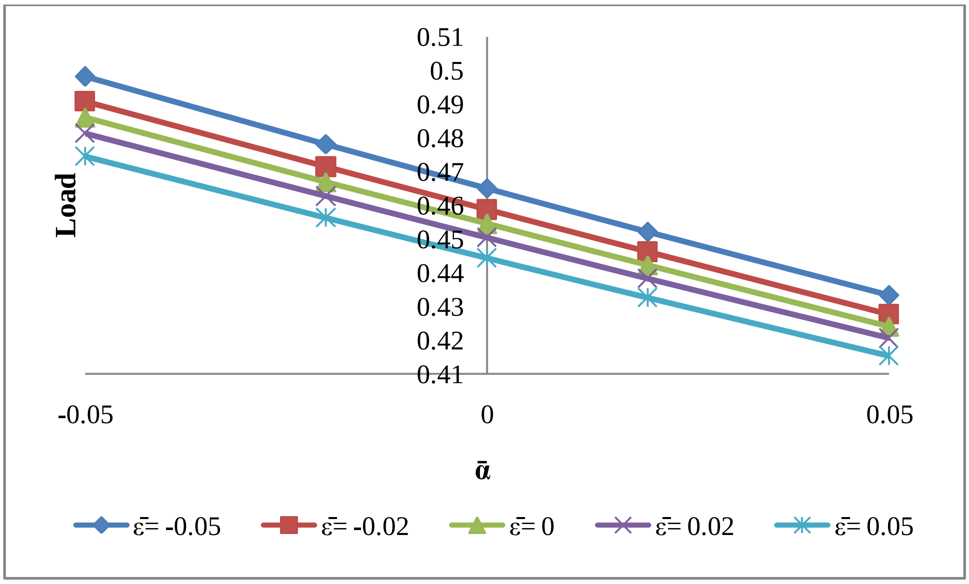

Fig. 5.10 : Variation of Load carrying capacity with respect to $\bar{\alpha}$ and $\bar{\varepsilon}$

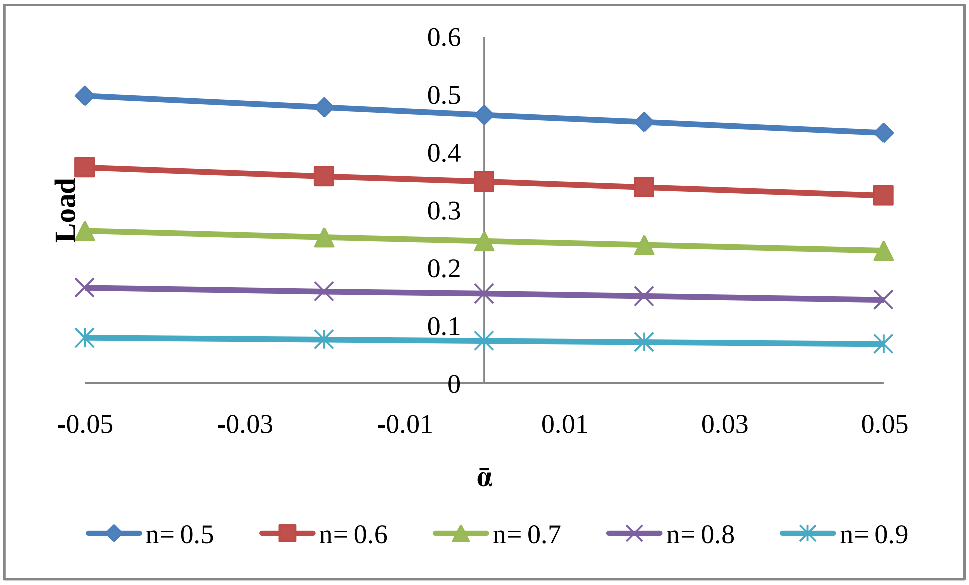

Fig. 5.11 : Variation of Load carrying capacity with respect to $\bar{\alpha}$ and *n*

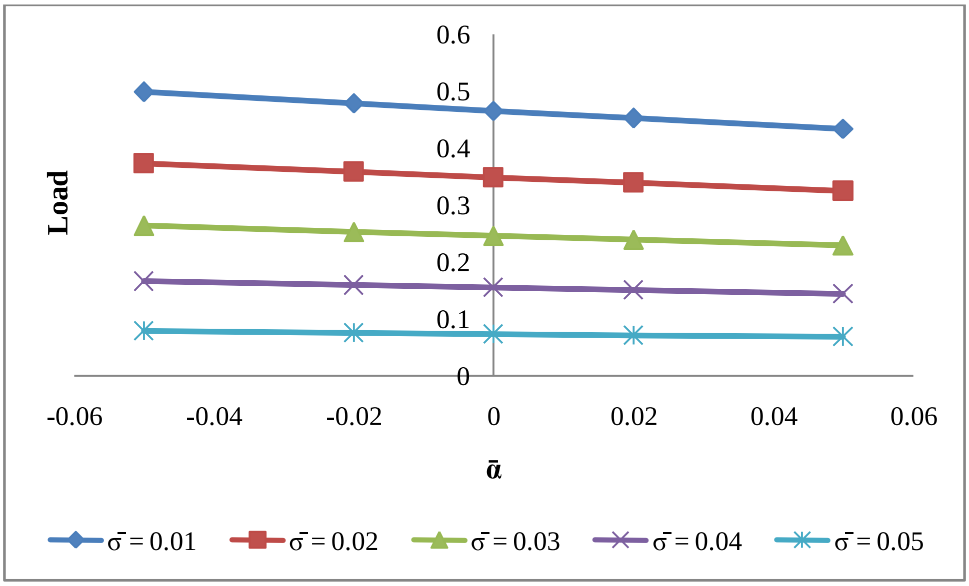

Fig. 5.12 : Variation of Load carrying capacity with respect to $\bar{\alpha}$ and $\bar{\sigma}$

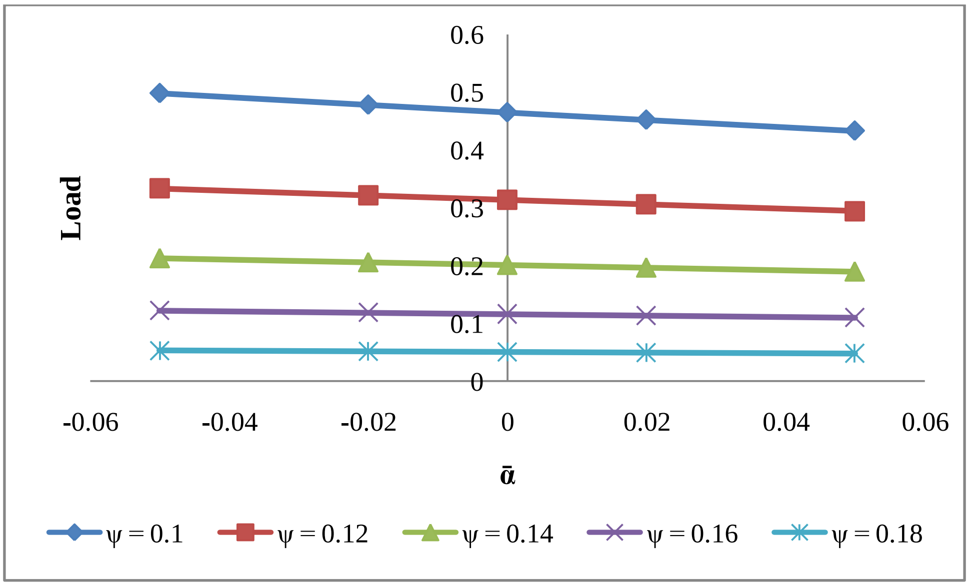

Fig. 5.13 : Variation of Load carrying capacity with respect to $\bar{\alpha}$ and ψ

In figures 5.10 – 5.13, The load carrying capacity gets decreased due to the positive skewness while negatively skewed roughness increases the load carrying capacity.

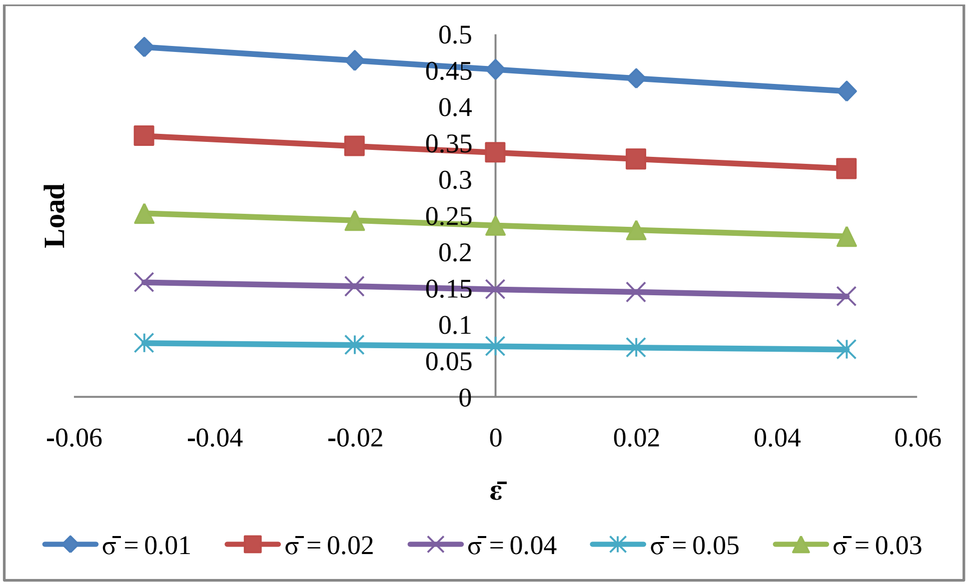

Fig. 5.14 : Variation of Load carrying capacity with respect to $\bar{\varepsilon}$ and $\bar{\sigma}$

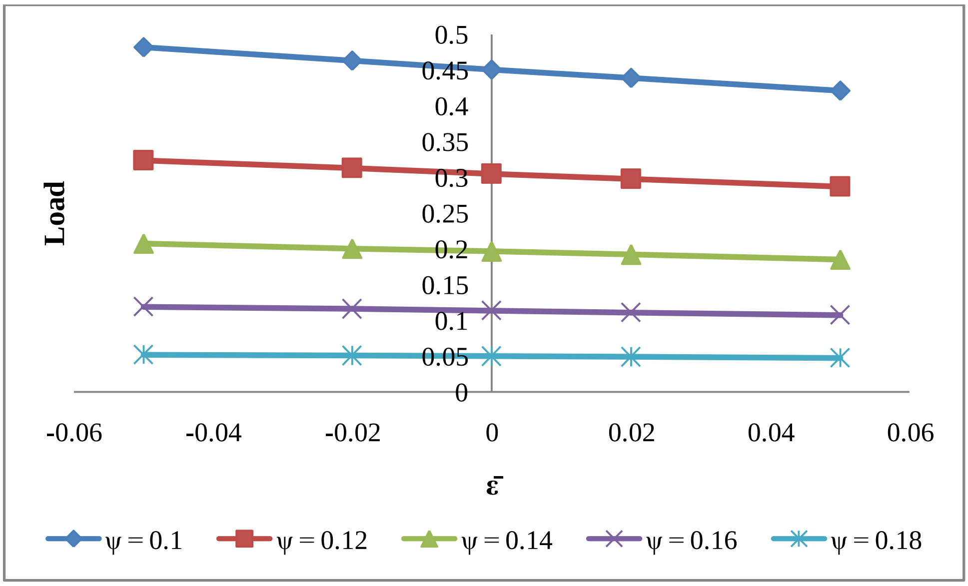

Fig. 5.15 : Variation of Load carrying capacity with respect to $\bar{\varepsilon}$ and ψ

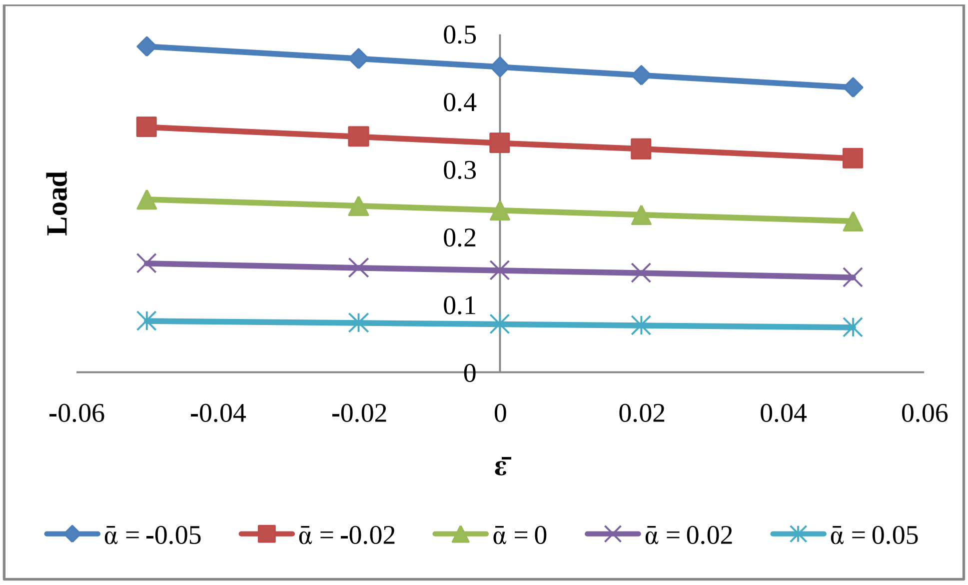

Fig. 5.16 : Variation of Load carrying capacity with respect to $\bar{\varepsilon}$ and $\bar{\alpha}$

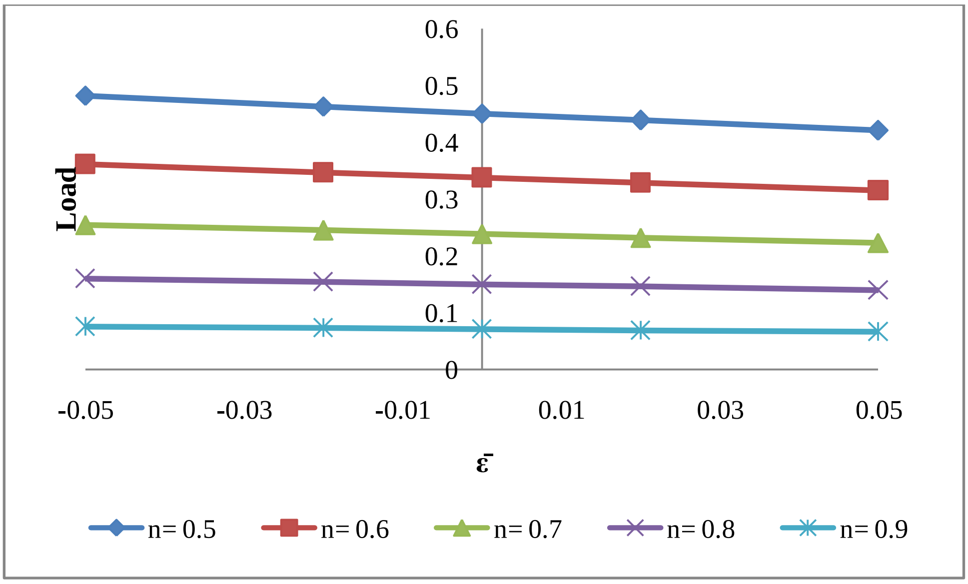

Fig. 5.17 : Variation of Load carrying capacity with respect to $\bar{\varepsilon}$ and *n*

That the variance follows the path of skewness is reflected in figures 5.14 – 5.17.

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 5.7.

5.4 Validation

Majumdar (2008) has discussed the same bearing system with smooth surface. The present study is done with roughness. It is found that load carrying capacity considerably increases due to roughness.

Table 5.1

	Load Carrying Capacity (calculated for $\bar{\sigma}$=0.01, $\bar{\alpha}$= -0.05, $\bar{\varepsilon}$= -0.05)		
ψ =	**Bearing system with Rough Surface as in this chapter**	**Bearing system with Smooth Surface by Majumdar, B. C. (2008)***	**increase in %**
0.01	1.012494178	0.892857143	13.40
0.02	0.902804110	0.806451613	11.95
0.03	0.814557777	0.735294118	10.78
0.04	0.742026921	0.675675676	9.82
0.05	0.681356717	0.625000000	9.02

5.5 Conclusion

Although, the magnetization brings in positive effect, 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.