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EFFECT OF SURFACE ROUGHNESS ON CHARACTERISTICS OF SQUEEZE FILM BETWEEN POROUS RECTANGULAR PLATES

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ABSTRACT

In investigation aims to analyse the effect of transverse surface roughness on the squeeze film performance between porous rectangular plates. The associated differential equation is stochastically averaged making use of stochastic averaging method of Christensen and Tonder for transverse surface roughness. The equation is solved with appropriate boundary conditions to obtain the pressure and consequently the load bearing. The graphical results suggest that the bearing suffers because of transverse surface roughness. However the situation is slightly better in the case of negatively skew roughness. Further variance (-ve) makes the situation further improved even if moderate values of porosity are involved.

Key words: Load carrying capacity, Porosity, Pressure distribution, Reynolds equation,Surface roughness.

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1. INTRODUCTION

Squeeze film behaviour is found to occur in engineering applications such as machine tools, gears, bearings, rolling elements, automotive engines and hydraulic systems and also in skeletal joints. The squeeze film arises when one of the lubricating surfaces approaches the other with a nominal velocity. It does take a certain period for the two surfaces to come in contact, because of the pressure built up during the squeezing action due to viscous resistance to extrusion of the lubricant.

Hays [1] made a theoretical analysis for the normal approach of flat and curved rectangular plates which were separated by a thin film of lubricant. Load capacity curves were presented and some typical pressure distributions were shown. The effect of surface curvature on the squeeze film generation was investigated. Wu [2] theoretically observed the squeeze film between two rectangular plates when one had a porous facing. The problem was described by the modified Reynolds equation in the film region and the Laplace equation in the porous region. Results were presented for pressure distribution, load-carrying capacity and film thickness as functions of time in series form. The effect of the porous facing on the squeeze film behaviour was discussed. Prakash and Tiwari [6] launched a theoretical analysis on the effects of unidirectional surface roughness on the response of a squeeze film between two porous rectangular plates of finite dimensions. The problem was solved analytically using Fourier series expansions. It was shown that the nominal geometry as characterized by the aspect ratio of the plates had a profound effect on the system.

On the basis of the Stokes micro continuum theory and the Christensen stochastic model, a theoretical study of squeeze film performance for isotropic rough rectangular plates with couple stress fluids as lubricants was presented by Lin [7].

A stochastic non-Newtonian Reynolds-type equation was derived and solved analytically for the mean film pressure distribution. According to the results, bearing surfaces with isotropic roughness pattern resulted in poor bearing characteristics as compared to the smooth-surface case. Naduvinamani et al. [8] observered the effect of surface roughness on the couple stress squeeze film behaviour between two rectangular plates, when one plate had a porous facing with anisotropic permeability by taking into account the slip velocity at the fluid and porous material interface. The stochastic Reynolds equation accounting for the couple stresses and randomized surface roughness structure was mathematically derived. Bujurke and Kudenatti[9] discussed the effects of surface roughness on the squeeze film behaviour between two rectangular plates with an electrically conducting fluid in the presence of a transverse magnetic field. The modified Reynolds equation which incorporates randomized roughness structure with magnetic field effect was derived by assuming the roughness asperity heights to be small compared to the film thickness. The finite difference based multigrid method was used for the solution of the modified Reynolds equation On the basis of Stokes couple-stress fluid model together with the hydromagnetic flow equations, Lin et al.[10] discussed concerned Reynolds equation. According to the results, the effects of couple stresses and external magnetic fields provided an increase in the load capacity and the response time as compared to the classical Newtonian hydrodynamic rectangular squeeze-film lubrication.

Shimpi et al.[11] observed squeeze film performance in porous rough rectangular plates under the presence of magnetic fluid lubricant. This investigation indicated that the bearing system registered an improved performance as compared to that of bearing system working with a conventional lubricant. Prakash and vij [12] analysed squeeze film between porous plates of various plates. The effect of the shape of plate and porosity on the bearing performance was calculated.

Most of the above studies neglected the effect of surface roughness by considering the bearing su*rface* to be smooth. However, it is well known that the bearing surfaces are rough to certain extent. Hence it has been proposed to study the effect of surface roughness on the characteristics of squeeze film between porous rectangular plates.

2. ANALYSIS

The bearing configuration is presented below

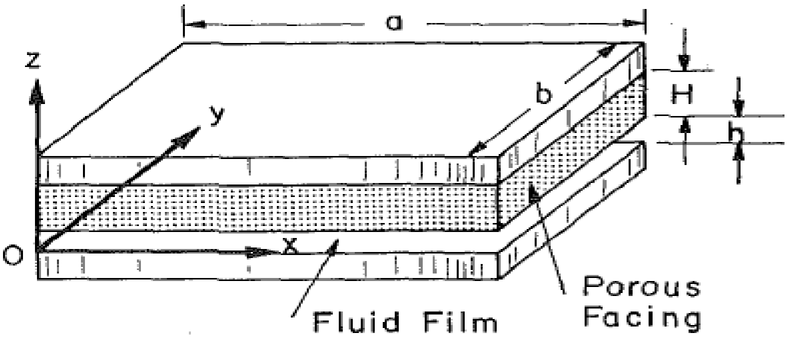

Figure 1 Squeeze film geometry

Following the method of Christensen and Tonder [3-5] the thickness h(x) is considered as

$$h(x) = \bar{h}(x) + h_s$$

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 roughness.Here h_s is assumed to be stochastic in nature and governed by the probability density function $F(h_s)$, which is defined by the relationship

$$F(h_s) = \begin{cases} \dfrac{32}{35b}\left(1 - \dfrac{h_s^2}{b^2}\right)^3 & -b \le h_s \le b \\ 0 & elsewhere \end{cases}$$

The mean α ,the standard devition σ and the parameter ε which is the measure of symmetry associated with the random vaivable h_s are determined by the relations

$$\alpha = E(h_s)$$

$$\sigma^2 = E\left[\left(h_s - \alpha\right)^2\right]$$

and

$$\varepsilon = E\left[\left(h_s - \alpha\right)^3\right]$$

where E denotes the expected value defined by

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

The details regarding the charecterization of the roughness aspects can be had from Christensen and Tonder[3-5]

In view of Prakash and vij [12], in the case of smooth surfaces the associated Reynolds equation for the porous facing is given by

$$\frac{\partial^2 p}{\partial x^2} + \frac{\partial^2 p}{\partial z^2} = \frac{12\mu\left(dh/dt\right)}{h^3 + 12\phi H} \tag{1}$$

Where

p = pressure in the film region, μ = absolute viscosity of the lubricant

ϕ = permeability of porous facing

In view of stochastical avereging method of Christensen and Tonder[3-5], equation (1) is transformed to

$$\frac{\partial^2 p}{\partial x^2}+\frac{\partial^2 p}{\partial z^2}=\frac{12\mu\left(dh/dt\right)}{g(h)} \qquad (2)$$

Where

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

Using following boundary conditions solving equation (2)

$$p\left(\pm a/2, z\right)=0$$

$$p\left(x, \pm b/2\right)=0$$

one gets the pressure distribution as

$$p=-\frac{\mu\dfrac{dh}{dt}ab}{g(h)}\times\frac{6}{\pi^3\left(a/b\right)\left(1+12\psi\right)}$$
$$\times\left[\left\{\frac{1}{4}-\left(\frac{z}{b}\right)^2\right\}\pi^3\times 8\sum_{n=1,3,..}^{\infty}\frac{(-1)^{\frac{1}{2}(n+1)}\cos\left(\dfrac{n\pi z}{b}\right)\cosh\left(\dfrac{n\pi x}{b}\right)}{n^3\cosh\left(\dfrac{n\pi a}{2b}\right)}\right] \qquad (3)$$

So the non dimensional pressure distribution comes out to be

$$P=\frac{h^3 p}{\mu\dfrac{dh}{dt}ab}$$
$$=-\frac{1}{(1+3\sigma^{*2}+3\alpha^{*2}+3\alpha^*+3\sigma^{*2}\alpha^*+\alpha^{*3}+\varepsilon^*)}\times\frac{6}{\pi^3\left(a/b\right)\left(1+12\psi\right)}$$
$$\times\left[\left\{\frac{1}{4}-z^{*2}\right\}\pi^3\times 8\sum_{n=1,3,..}^{\infty}\frac{(-1)^{\frac{1}{2}(n+1)}\cos\left(n\pi z^*\right)\cosh\left(n\pi x^*\right)}{n^3\cosh\left(\dfrac{n\pi}{2\beta}\right)}\right] \qquad (4)$$

where

$$\sigma^*=\frac{\sigma}{h} \quad \alpha^*=\frac{\alpha}{h} \quad z^*=\frac{z}{b} \quad \varepsilon^*=\frac{\varepsilon}{h^3} \quad x^*=\frac{x}{b} \quad and\ \beta=\frac{b}{a}$$

The load carrying capacity of squeeze film is found by integrating the pressure over the plate, as

$$w=\int_0^b\int_0^a p\left(x,y\right)dxdy \qquad (5)$$

$$= -\frac{\mu \frac{dh}{dt} a^2 b^2}{g(h)} \times \frac{192}{\pi^4 \left(a^2/b^2\right)(1+12\psi)} \times \left[\frac{\pi^4}{192}(a/b) - \frac{1}{\pi} \cdot \sum_{n=1,3,\ldots}^{\infty} \frac{1}{n^5} \tanh\left(\frac{n\pi a}{2b}\right)\right] \tag{6}$$

The non-dimensional load carrying capacity turns out to be

$$W = -\frac{h^3 w}{\mu \frac{dh}{dt} a^2 b^2}$$

$$= \frac{1}{(1+3\sigma^{*2} + 3\alpha^{*2} + 3\alpha^* + 3\sigma^{*2}\alpha^* + \alpha^{*3} + \varepsilon^*)} \times \frac{192\beta^2}{\pi^4 (1+12\psi)} \times \left[\frac{\pi^4}{192\beta} - \frac{1}{\pi} \cdot \sum_{n=1,3,\ldots}^{\infty} \frac{1}{n^5} \tanh\left(\frac{n\pi}{2\beta}\right)\right] \tag{7}$$

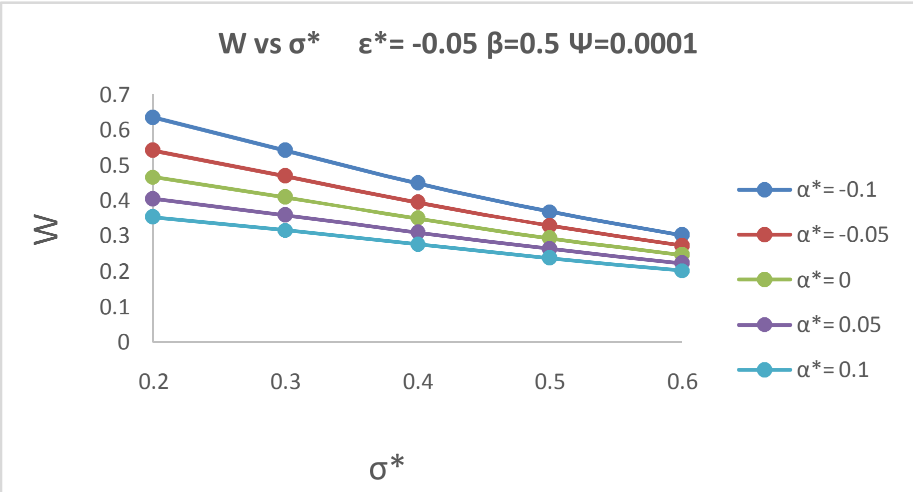

Figure 2 Non dimensional Load Carrying Capacity W versus σ* for different values of α*

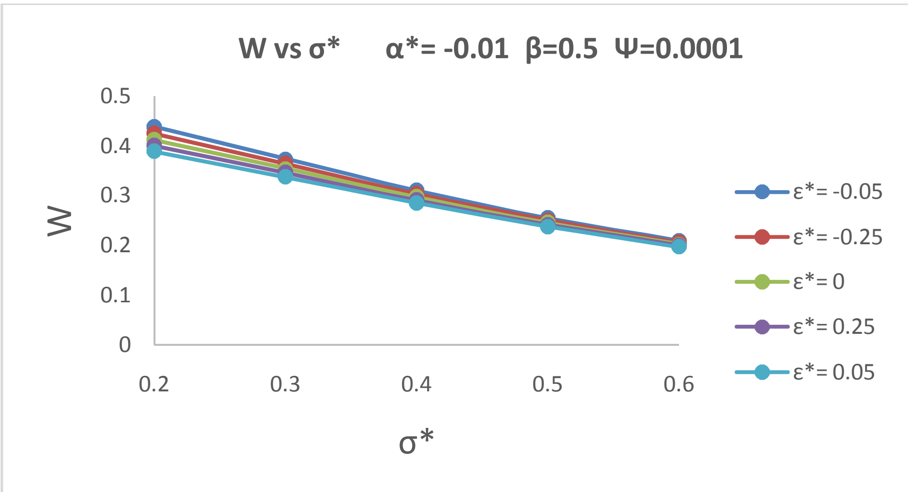

Figure 3 Non dimensional Load Carrying Capacity W versus σ* for different values of ε*

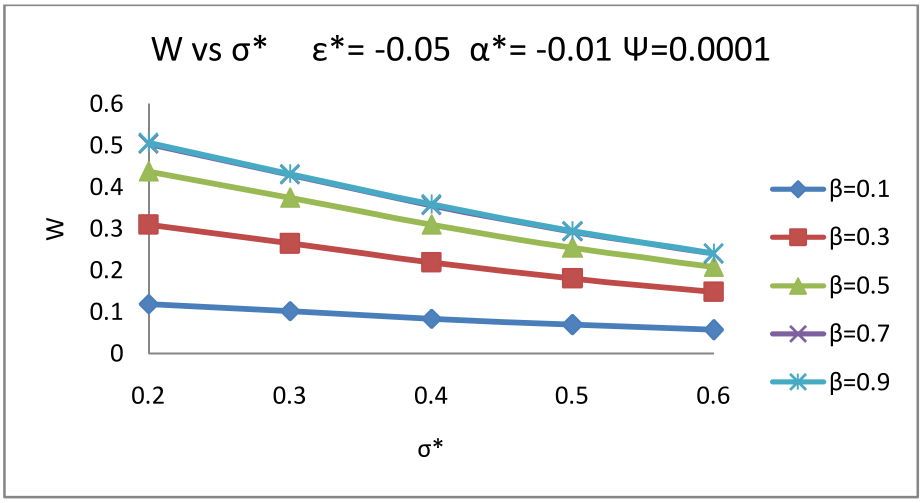

Figure 4 Non dimensional Load Carrying Capacity W versus σ* for different values of β

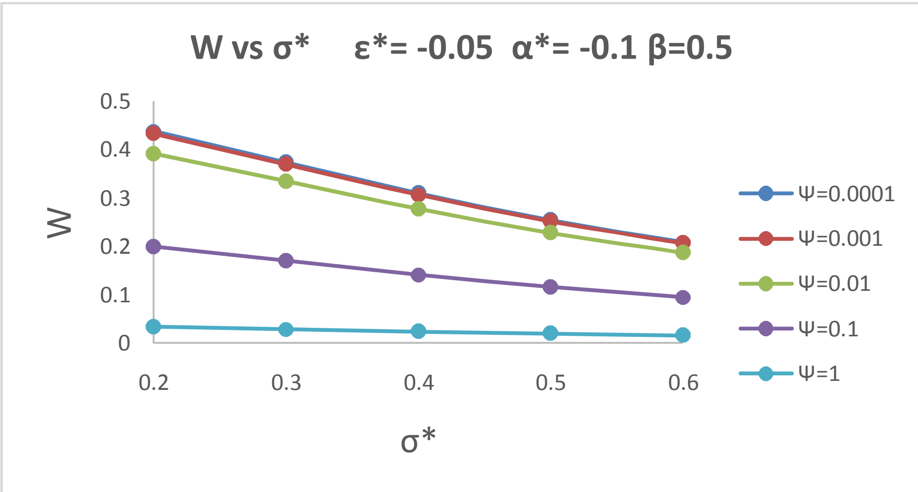

Figure 5 Non dimensional Load Carrying Capacity W versus σ* for different values of Ψ

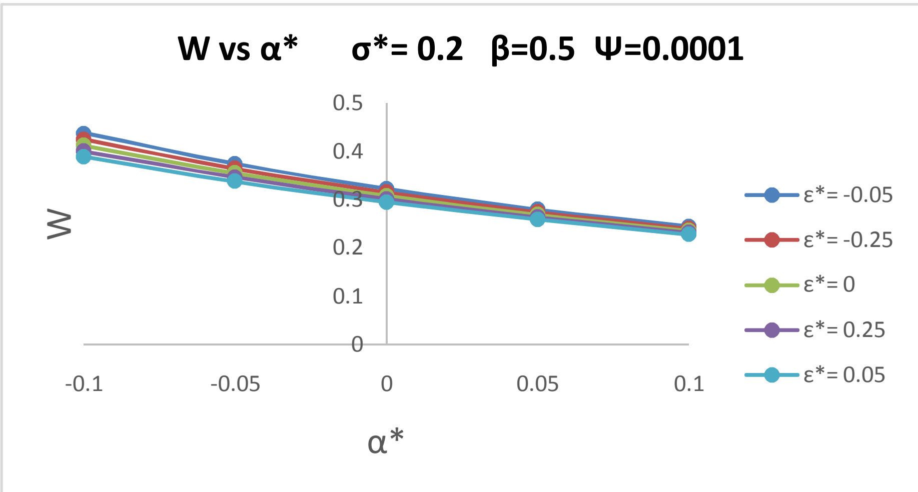

Figure 6 Non dimensional Load Carrying Capacity W versus α* for different values of ε*

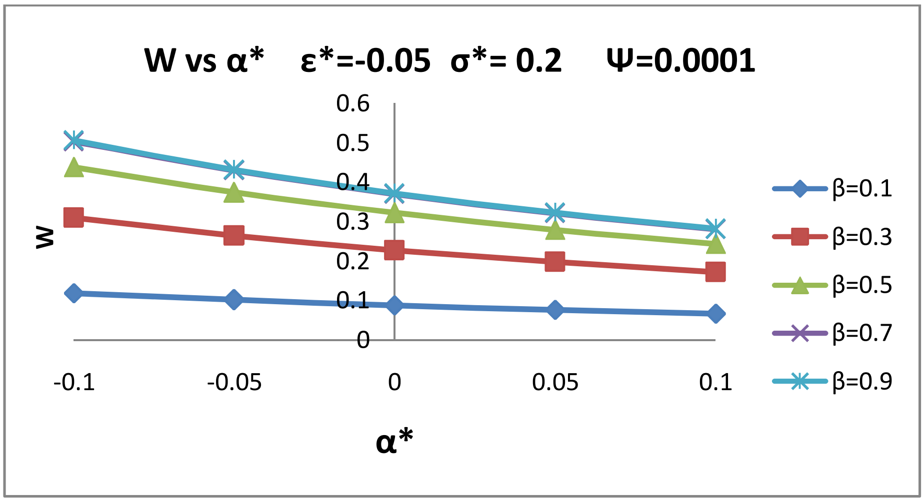

Figure 7 Non dimensional Load Carrying Capacity W versus α* for different values of β

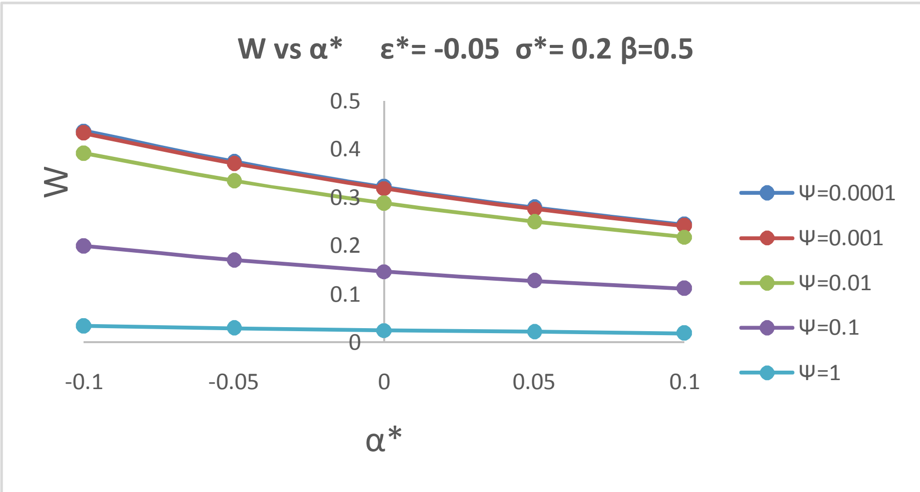

Figure 8 Non dimensional Load Carrying Capacity W versus α* for different values of Ψ

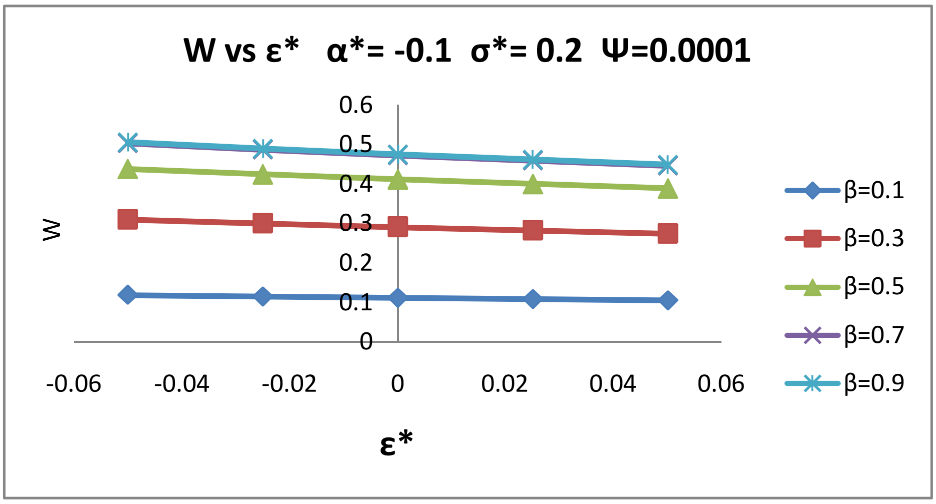

Figure 9 Non dimensional Load Carrying Capacity W versus ε* for different values of β

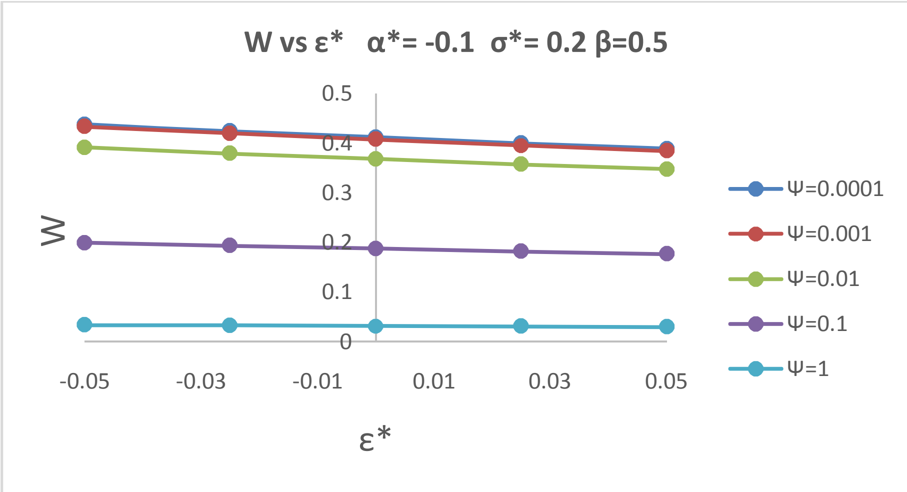

Figure 10 Non dimensional Load Carrying Capacity W versus ε* for different values of Ψ

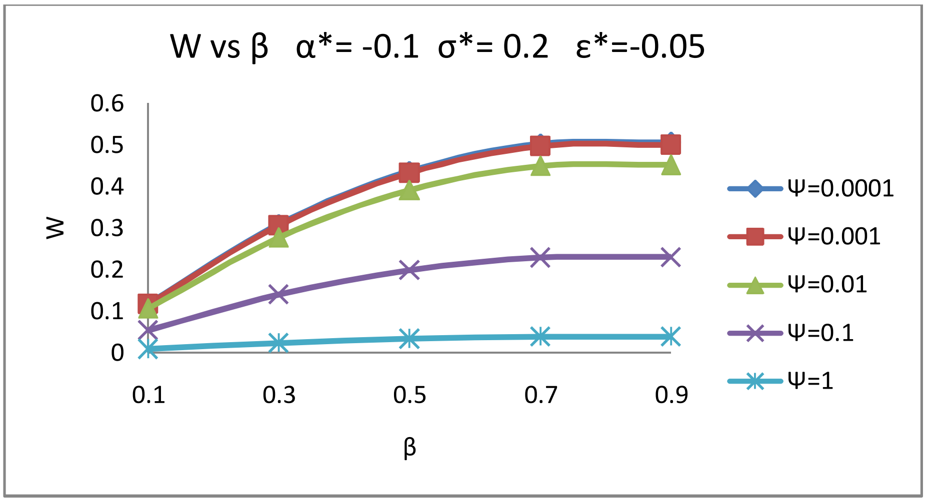

Figure 11 Non dimensional Load Carrying Capacity W versus ε* for different values of β

3. RESULT AND DISCUSSION

The non-dimensional pressure distribution is obtained from equation (4) while equation (7) determines the non-dimensional load carrying capacity. In the absence of roughness the investigation reduces to the discussion of Prakash and Vij [12].

The non-dimensional load carrying capacity is presented in fig (2-11). From figure (3) one can observe that the effect of skewness on load carrying capacity with respect to $\sigma*$ is nominal when $\sigma*$ exceeds o.4.From figure (4) one can see that the effect of aspect ratio on load carrying capacity with respect to $\sigma*$ is negligible. From figure (5) one can find that the load carrying capacity decreases as values of porosity increases and porosity effects are negligible up to $\psi \approx 0.001$.

Figure (6-8) represents non-dimensional load carrying capacity with respect to α* for different values of skewness, aspect ratio and porosity respectively. The negative variance (-ve) makes the situation batter. From figure (10) one can find that the load carrying capacity decreases as values of porosity increases and porosity effects are negligible up to $\psi \approx 0.001$.

4. CONCLUSION

This investigation establishes that from the bearing`s life period point of view the roughness is required to be carefully evaluated while designing the bearing system. The situation is slightly better in the case of negatively skew roughness. Further variance (-ve) makes the situation further improved even if moderate values of porosity are involved.

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NOMENCLATURE

α : Variance
σ : Standard deviation
ε : Skewness
α^* : Dimensionless variance
σ^* : Dimensionless standard deviation
ε^* : Dimensionless skewness
p : Lubricant Pressure (N/mm2)
P : Dimensionless Pressure
w : Load carrying capacity (N)
W : Dimensionless load carrying capacity
e : Eccentricity (mm)