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RESEARCH

Magnetic Fluid-Based Squeeze Film Performance in Rotating Curved Porous Circular Plates: The Effect of Deformation and Surface Roughness

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A B S T R A C T

This investigation aims at analyzing the behaviour of a magnetic fluid based squeeze film between two rotating transversely rough porous circular plates taking bearing deformation into consideration. The results presented in graphical form inform that the transverse surface roughness introduces an adverse effect on the performance characteristics while the magnetic fluid lubricant turn in an improved performance. It is found that the combined effect of rotation and deformation causes significantly reduced load carrying capacity. However, this investigation establishes that the adverse effect of porosity, deformation and standard deviation can be compensated up to some extent by the positive effect of magnetic fluid lubricant in the case of negatively skewed roughness by choosing curvature parameters. To compensate, the rotational inertia needs to have smaller values.

1. INTRODUCTION

The squeeze film between various geometrical configurations is presented in a number of investigations (Archibald [1], Cameron [2], Prakash and Vij [3], Hamrock [4]). Wu [5,6] considered the effect of rotation on the squeeze film performance. Murti [7], Vora and Bhat [8], Ajwaliya [9] considered the effect of curvature of the plates on the squeeze film performance. According to their findings such situations might be found useful in the design of clutch plates and collar bearings.

Most of the theoretical studies dealing with the squeeze film behaviour are considered perfectly rigid bearing surfaces. But under high loads a bearing may deform producing wedge effects in the lubricant film thereby, changing the squeeze. From this point of view, Osterle and Seibel [10] and Ramanaiah and Sundarammal [11] analyzed the effect of the bearing deformation in slider bearings. In addition to this, Ramanaiah and Sundarammal [12] analyzed the effect of bearing deformation on the squeeze film characteristic between circular and rectangular plates. Here it was shown that the bearing deformation resulted in reduced load carrying capacity and increased squeeze.

Lately, considerable attention is being paid to the use of magnetic fluid as a lubricant to modify the performance of bearing system. Ferrofluids/magnetic fluids are a special category of smart nonmaterials, especially, magnetically controllable nano fluids. These types of nanofluids are colloids of magnetic nanoparticles such as Fe_3O_4, γ-Fe_3O_4, $CoFe_2O_4$, Co, Fe or Fe-C, dispersed stably in a carrier liquid. Consequently, these nonmaterials manifest simultaneously as magnetic fluids and magnetic properties. These fluids are found to be useful in engineering and biomedical applications. In fact, any discussion of the fluid which has magnetic properties can be divided into the following categories (1) Ferrofluids; (2) Magnetorehological fluids; (3) Dispersions of micron-sized particles, and (4) Fluids containing paramagnetic particles. The Magnetic fluid is a suspension of solids magnetic particles of subdomain size in a liquid carrier. The density of the particles is of the order of 1023 per cubic meter. Depending upon the Ferromagnetic material and the method of preparation, the mean diameter of a particle varies from 3 to 15 nm. These fluids which are properly stabilized, undergo practically no aging or separation. They remain liquid in a magnetic field and after removal of field recover their characteristics. Particles of this size, whether they are ferrite or metal, possess a single magnetic domain only, that is, the individual particles are in a permanent state of saturation magnetization. Therefore, a strong long-range magneto static attraction exists between individual particles. One of the most important properties of the magnetic fluid is that it can be retained at a desired location by an external magnetic field.

The method of Verma [13] was modified by Bhat and Deheri [14] to analyze the effect of magnetic fluid lubricant on the action of squeeze film in curved porous circular disks. Bhat [16] developed the analysis of Bhat and Deheri [15] regarding the magnetic fluid based squeeze film performance in porous annular disks to study the effect of magnetic fluid lubricant on the squeeze film performance between porous rotating circular plates in the presence of an external magnetic field oblique to the lower plate.

The squeeze film performance between curved circular plates lying along the surfaces determined by the secant function under the presence of a magnetic fluid lubricant was studied and analyzed by Patel and Deheri [17]. Hsu et al [18] discussed the squeeze film characteristics between rotating circular disks with an electrically conducting lubricant in the presence of transverse magnetic field. It was concluded that, on the whole the use of electrically conducting fluid in the presence of transversely magnetic field resulted in an improvement of performance characteristics in rotating circular disks. Lin et al [19] dealt with the performance between two curved circular plates in the presence of a transverse magnetic field making use of an electrically conducted fluid.

Sukla et al [20] investigated the performance characteristic of squeeze film bearings with power law lubricants [A type of generalized Newtonian lubricant for which the shear stress is given by:

$$\tau = K\left(\frac{\partial u}{\partial y}\right)^n$$

where K is the flow consistency index, $\partial u/\partial y$ is the shear rate or the velocity gradient perpendicular to the plane of shear, and n is the flow behaviour index (dimensionless)] considering the effects of consistency variations. Various bearing geometries were considered with rigid surfaces as well as the compliant layers.

Several methods have been discussed to study with the effect of surfaces roughness and the performance characteristic of squeeze film. Tzeng and Saibel [21] employed a stochastic approach to model the random roughness which in turn was extended by Christensen and Tonder [22-24] to study the effect of surface roughness in general. These studies recognized the random character of surface roughness. A number of investigations deployed the stochastic model of Christensen and Tonder to analyze the effect of surface roughness (Ting [25], Prakash and Tiwari [26], Guha [27], Gupta and Deheri [28]). Prajapati [29] analyzed the squeeze film performance in rotating porous rough circular plates with elastic deformation and proved that the combined effect of elastic deformation and roughness was comparatively adverse.

Saber and Gamel [30] analyzed the stability of the plane cylindrical journal bearing having an elastic shell. Here the effect of elastic deformation on its dynamic stability was evaluated. It was concluded from this work that increasing the elasticity of the bearing liner resulted in an increase in bearing stability.

Recently, Hsu et al [31] investigated the combined effect of surface roughness and rotational inertia on the squeeze film behaviour in parallel circular disks.

Although, the transverse surface roughness induced an adverse effect in general, the investigations carried out by Patel and Deheri [32], Deheri et al [33] and Shimpi and Deheri [34] suggested that the negatively skewed roughness resulted in a relatively better performance. Therefore, it was deemed appropriate to study the effect of surface roughness and bearing deformation on the squeeze film performance in curved porous rotating circular plates taking a magnetic fluid as a lubricant.

2. ANALYSIS

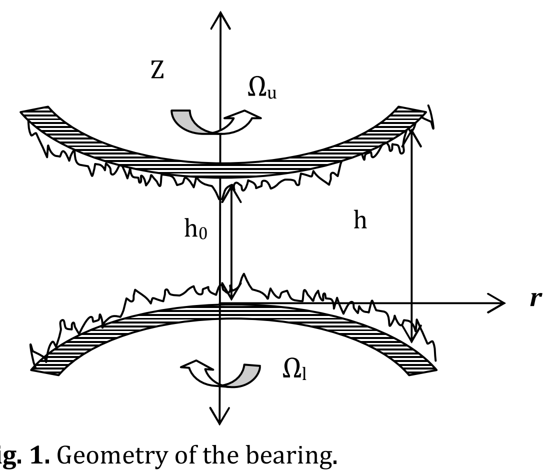

Fig. 1. Geometry of the bearing.

The geometry and configuration of the bearing system is shown in Fig. 1 which consists of the circular disks, where Ω_u and Ω_l are the rotational velocity of the upper plate and the lower plate respectively.

Both the disks are assumed to be elastically deformable and their contact surfaces are considered to be transversely rough (that causes a disturbance in the medium perpendicular to the direction it advances). The upper disk moves towards the lower disk normally with uniform velocity:

$$\dot{h} = \frac{dh}{dt}$$

In view of the discussions regarding stochastic modeling of the roughness adopted by Christensen and Tonder [22-24] and Prajapati [35] the film thickness is assumed to be:

$$h(r,t) + \delta(r,t) + h_s(r,\xi)$$

where in h denotes the smooth and unstressed part of the film thickness and h_s is the part due to surface roughness measured from the mean level $h + \delta$ and its random character is expressed by the variable ξ. h_s is governed by the probability density function:

$$f(h_s) = \begin{cases} \dfrac{35}{32c}\left(1 - \dfrac{h_s^{\,2}}{c^2}\right)^3, & -c \le h_s \le c \\ 0, & \text{otherwise.} \end{cases}$$

where c is the maximum deviation from the mean film thickness. The mean α, the standard deviation σ (It is the deviation from the mean surface level as explained in Christensen and Tonder [22]) and the parameter ε, which is the measure of symmetry of the random variable h_s, are defined by the relationships:

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

and

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

where E denotes the expected value defined by

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

It is assumed that the upper plate lying along the surface determined by:

$$Z_u = h_0\left[\sec\left(Br^2\right)\right]; \qquad 0 \le r \le a$$

approaches with normal velocity $\dot{h}_0$, to the lower plate lying along the surface:

$$Z_l = h_0\left[\sec\left(Cr^2\right) - 1\right]; \qquad 0 \le r \le a$$

where h_o is the central distance between the plates, B and C are the curvature parameters of the corresponding plates. The central film thickness h(r) (i.e., the thickness of the lubricant film at the central part) then is defined by (Hamrock [4], Bhat [16]):

$$h(r)=h_0\left[\sec\left(Br^2\right)-\sec\left(Cr^2\right)+1\right];\ \ 0\le r\le a$$

Assuming axially symmetric flow of the magnetic fluid between the annular plates under an oblique magnetic field:

$$\overline{H}=\left(H(r)\cos\theta(r,z),\,0,H(r)\sin\theta(r,z)\right)$$

whose magnitude H is a function of r vanishing at r = 0, a; the angle of inclination θ of the magnetic field as in (Bhat [16]), the modified Reynolds equation governing the film pressure p under the usual assumptions of hydromagnetic lubrication takes the form (Prajapati [35], Bhat and Deheri [15], Patel and Deheri [17]):

$$\frac{1}{r}\frac{\partial}{\partial r}\left[rg(h)\frac{\partial}{\partial r}\left\{p-0.5\mu_0\overline{\mu}H^2\right\}\right]=12\mu\dot{h}+4S\Phi(h) \tag{1}$$

with

$$g(h)=\left(h+p'p_a\delta\right)^3+3\alpha\left(h+p'p_a\delta\right)^2$$
$$+3\left(\alpha^2+\sigma^2\right)\left(h+p'p_a\delta\right)+3\sigma^2\alpha+\alpha^3+\varepsilon+12\phi H_0$$

$$\Phi(h)=\left(h+p'p_a\delta\right)^3+3\left(\alpha^2+\sigma^2\right)\left(h+p'p_a\delta\right)$$
$$+3\sigma^2\alpha+\alpha^3+\varepsilon,$$

where

$$H^2=r^2(a-r)/a$$

while μ_0 is permeability of free space, $\overline{\mu}$ is the magnetic susceptibility of particles and μ is the viscosity of the lubricant, ϕ is the permeability of porous facing, H_0 is the thickness of porous medium, δ is the local elastic deformation of the porous facing and p_a is the reference ambient pressure.

In view of the following non-dimensional quantities:

$$P=-\frac{h_0^3p}{\mu\dot{h}a^2},\ \ R=\frac{r}{a},\ \ \overline{\sigma}=\frac{\sigma}{h_0},\ \ \overline{\alpha}=\frac{\alpha}{h_0},$$

$$\overline{\varepsilon}=\frac{\varepsilon}{h_0^3},\ \psi=\frac{\phi H_0}{h_0^3},\ \ \kappa=\frac{12\mu\dot{h}}{h_0^3},\ \overline{B}=Ba^2$$

$$\overline{C}=Ca^2\qquad \mu^*=-\frac{\mu_0\overline{\mu}\,k\,h_0^3}{\mu\dot{h}},$$

$$\overline{p}=p'p_a,\ \overline{\delta}=\frac{\delta}{h},\quad S=\frac{3\rho\Omega^3}{p_a}$$

(p being density of lubricant and Ω the angular velocity) integrating the stochastically averaged Reynolds equation (1) under the boundary conditions:

$$\left(\frac{\partial P}{\partial R}\right)_{R=0}=-\frac{\mu^*}{2},\qquad P(1)=0 \tag{2}$$

one obtains the expression for the non-dimensional pressure distribution as:

$$P=0.5\mu^*R^2(R-1)+\frac{3}{\sqrt{A_1A_2}}+4\left(\frac{S}{\kappa}\right)\left(\frac{B_2}{A_2}\right)\left(1-R^2\right)$$
$$+X\left(\frac{S}{\kappa}\right)\left\{\frac{4\left(3B_1A_2-A_1B_2\right)}{A_2\sqrt{A_1A_2}}\right\} \tag{3}$$

where

$$X=\tan^{-1}\left(\sqrt{\frac{A_2}{A_1}}\right)-\tan^{-1}\left(\sqrt{\frac{A_2}{A_1}}\,R^2\right)$$

$$A_1=\left(1+\overline{p}\,\overline{\delta}\right)^3+3\overline{\alpha}\left(1+\overline{p}\,\overline{\delta}\right)^2$$
$$+3\left(\overline{\alpha}^2+\overline{\sigma}^2\right)\left(1+\overline{p}\,\overline{\delta}\right)+3\overline{\sigma}^2\overline{\alpha}+\overline{\alpha}^3+\overline{\varepsilon}+12\psi,$$

$$A_2=\frac{3}{2}\left(\overline{B}^2-\overline{C}^2\right)\left[\left(1+\overline{p}\,\overline{\delta}\right)^3\right.$$
$$\left.+2\overline{\alpha}\left(1+\overline{p}\,\overline{\delta}\right)^2+\left(\overline{\alpha}^2+\overline{\sigma}^2\right)\left(1+\overline{p}\,\overline{\delta}\right)\right]$$

$$B_1=\left(1+\overline{p}\,\overline{\delta}\right)^3+3\left(\overline{\alpha}^2+\overline{\sigma}^2\right)\left(1+\overline{p}\,\overline{\delta}\right)$$
$$+3\overline{\sigma}^2\overline{\alpha}+\overline{\alpha}^3+\overline{\varepsilon},$$

and

$$B_2=\frac{3}{2}\left(\overline{B}^2-\overline{C}^2\right)\left[\left(1+\overline{p}\,\overline{\delta}\right)^3+\left(\overline{\alpha}^2+\overline{\sigma}^2\right)\left(1+\overline{p}\,\overline{\delta}\right)\right].$$

The load carrying capacity in dimensionless form then, is calculated as:

$$W=-\frac{h_0^3w}{\mu\,\dot{h}_0\,a^4}=\int_0^1 RP\,dR$$

which leads to

$$W = \frac{\mu^*}{40} + \left(\frac{B_2}{A_2}\right)\left(\frac{S}{\kappa}\right) + \left[\frac{3}{4A_2} + \left(\frac{S}{\kappa}\right)\left(\frac{3B_1A_2 - A_1B_2}{A_2^2}\right)\right]\log\left(\frac{A_1 + A_2}{A_1}\right) \quad (4).$$

3. RESULTS AND DISCUSSION

It is noticed that the non-dimensional pressure distribution is obtained from Equation (3) while Equation (4) determines the dimensionless load carrying capacity. It is clearly seen that the non-dimensional pressure increases by:

$$0.5\mu^*\left[R^2(1-R)\right]$$

while the increase in the load carrying capacity turns out to be:

$$0.025\mu^*$$

as compared to the case of conventional lubricant.

It is observed that for a porous bearing with smooth surfaces, this investigation essentially, reduces to the study of Bhat and Deheri [14] concerning the performance of a magnetic fluid based squeeze film between circular plates in the absence of rotation and deformation. Moreover, setting the magnetization parameter (μ^*) to be equal to zero for a nonporous bearing with smooth surfaces, one can find the results of Prakash and Vij [3] when no rotation and deformation occur. Besides, considering the magnetization parameter to be equal to zero for a porous bearing with smooth surfaces one can obtain the results of Bhat [16] in case there is no deformation.

Further, a close observation of the expression of pressure distribution indicates that the profile of the pressure distributions gets altered significantly due to deformation. It is noticed that the transverse surface roughness in general, induces an adverse effect and the elastic deformation makes this negative effect more significant. Possibly, this may be due to the fact that the roughness of the bearing surfaces retards the motion of the lubricant which results in reduced load carrying capacity.

The fixed values of the roughness parameters are taken from Christensen and Tonder [22-24] while the values of magnetization parameters are adopted from Deheri et al [33] and the values of deformation and rotation are selected from Shimpi and Deheri [34].

The effect of magnetization on the distribution of the load carrying capacity is presented in Figs. 2-9. It is observed that the load carrying capacity increases sharply with increase in magnetization parameter. However, there is a nominal increase in load carrying capacity in the case of skewness.

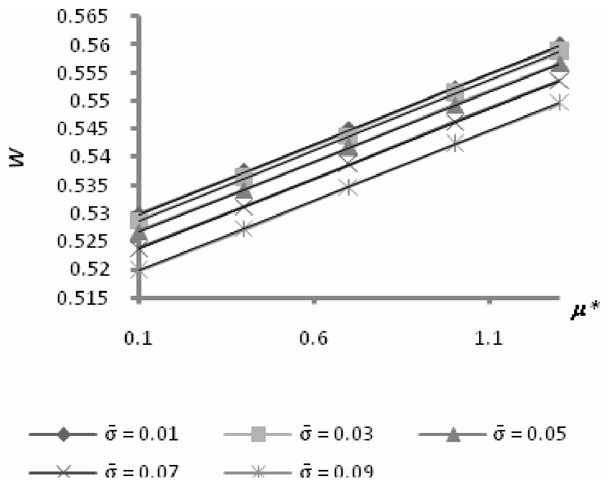

Fig. 2. Variation of Load carrying capacity with respect to μ^* and σ.

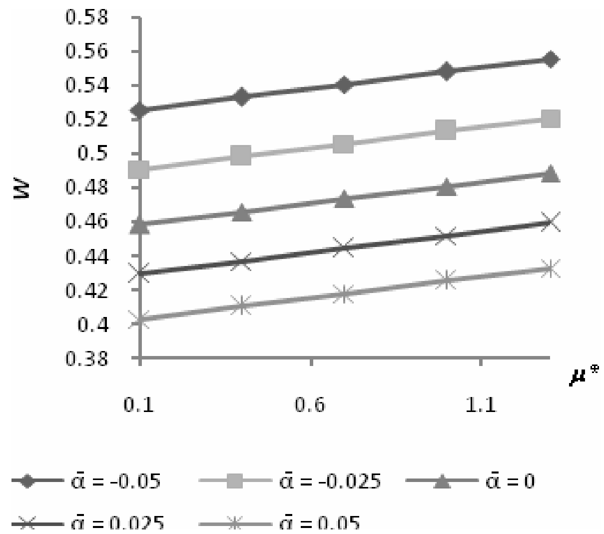

Fig. 3. Variation of Load carrying capacity with respect to μ^* and α.

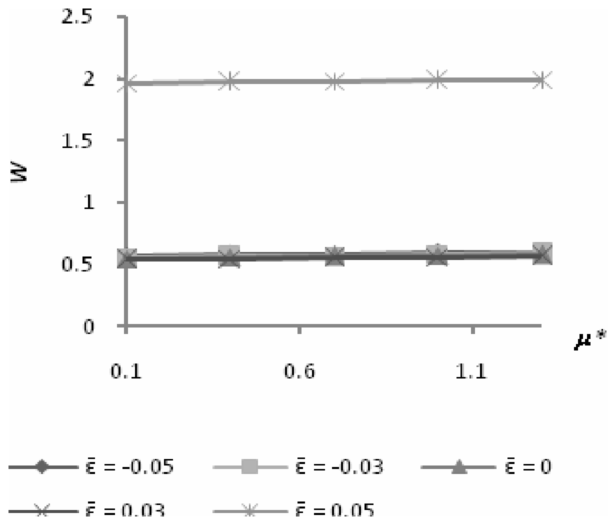

Fig. 4. Variation of Load carrying capacity with respect to μ^* and ε.

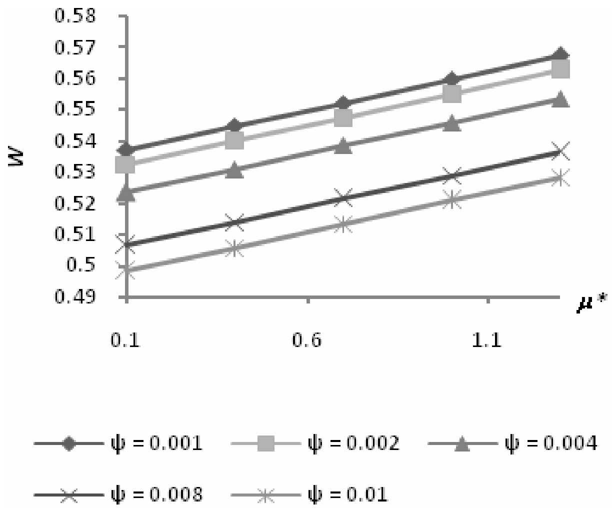

Fig. 5. Variation of Load carrying capacity with respect to μ^* and Ψ.

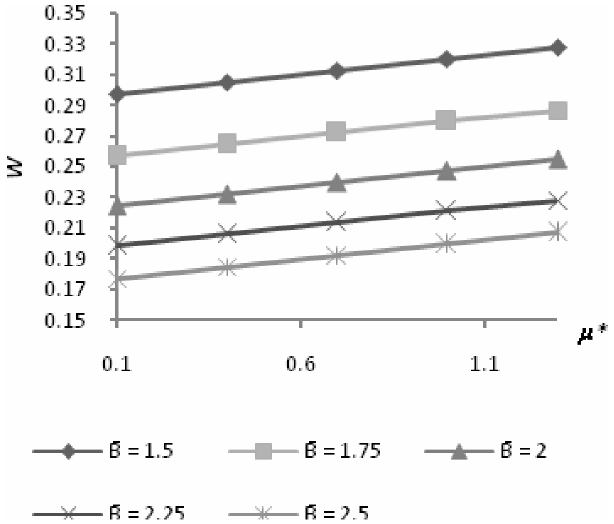

Fig. 6. Variation of Load carrying capacity with respect to μ^* and $\overline{B}$.

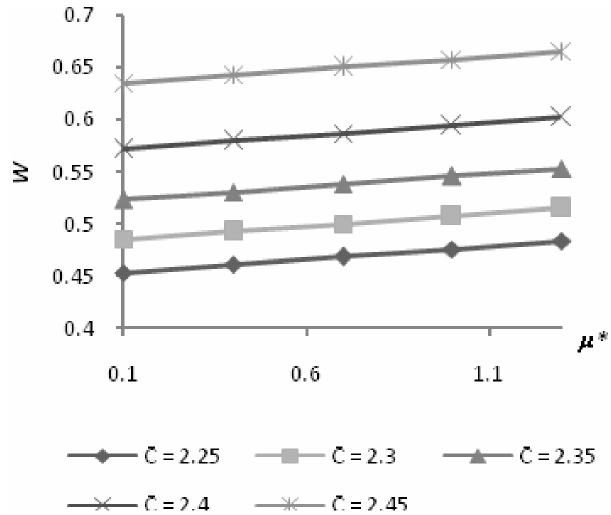

Fig. 7. Variation of Load carrying capacity with respect to μ^* and $\overline{C}$.

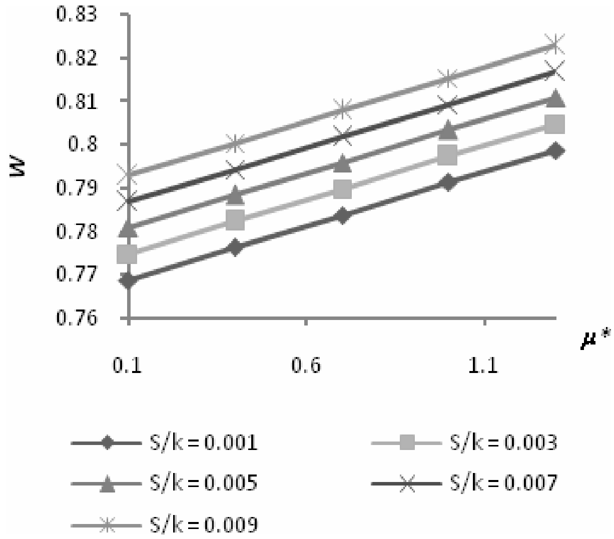

Fig. 8. Variation of Load carrying capacity with respect to μ^* and S/k.

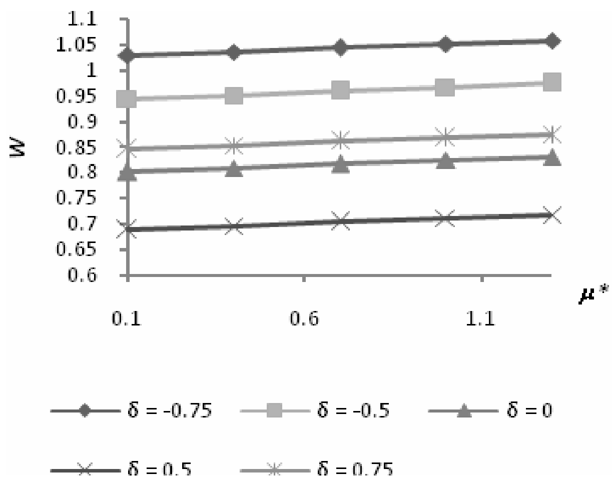

Fig. 9. Variation of Load carrying capacity with respect to μ^* and $\overline{\delta}$.

Figs. 10-14 describe the effect of standard deviation on the variation of the load carrying capacity. It is clearly seen that the load carrying capacity decreases significantly with increasing values of the standard deviation. Further, the effect of deformation with respect to the standard deviation on the load carrying capacity is comparatively not that significant.

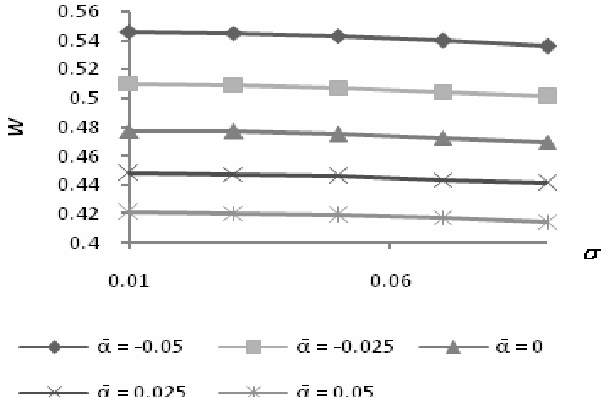

Fig. 10. Variation of Load carrying capacity with respect to σ and α.

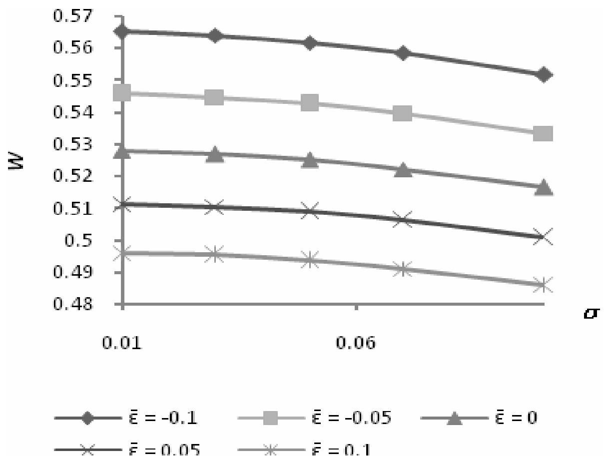

Fig. 11. Variation of Load carrying capacity with respect to σ and ε.

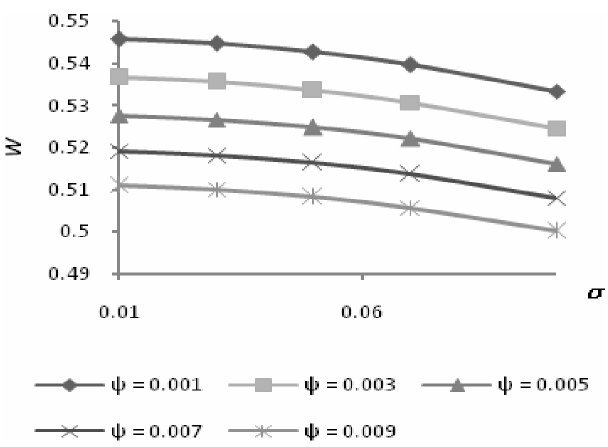

Fig. 12. Variation of Load carrying capacity with respect to σ and Ψ.

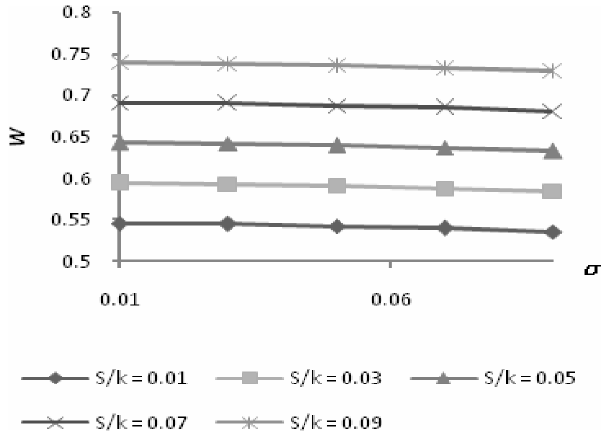

Fig. 13. Variation of Load carrying capacity with respect to σ and S/k

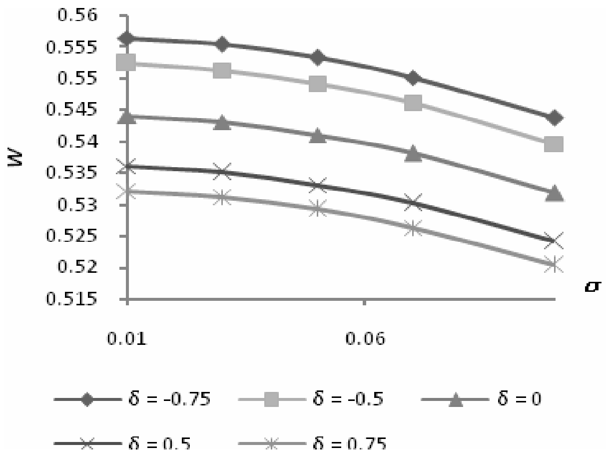

Fig. 14. Variation of Load carrying capacity with respect to σ and δ.

The effect of the variance presented in the Figs. 15-18 makes it clear that the variance (positive) decreases the load carrying capacity while the variance (negative) increases the load carrying capacity. Moreover, the effect of $\bar{\delta}$ and Ψ on the variance of load carrying capacity with respect to variance is nominal.

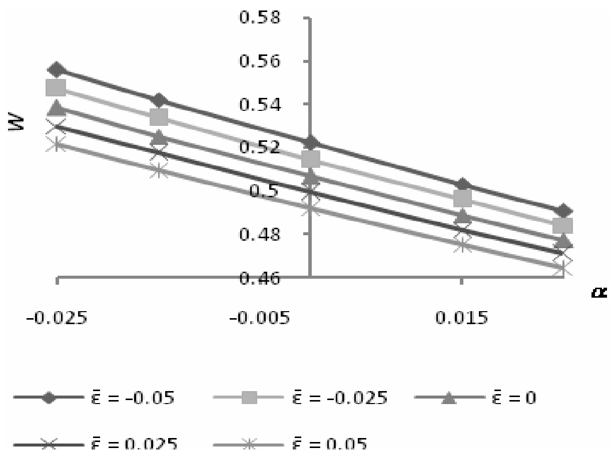

Fig. 15. Variation of Load carrying capacity with respect to α and ε.

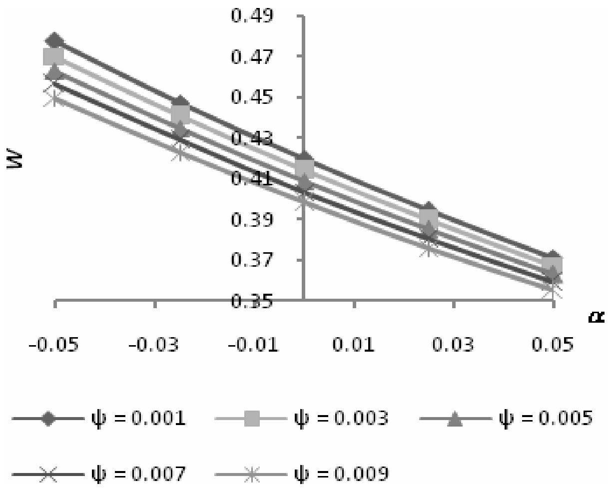

Fig. 16. Variation of Load carrying capacity with respect to α and Ψ.

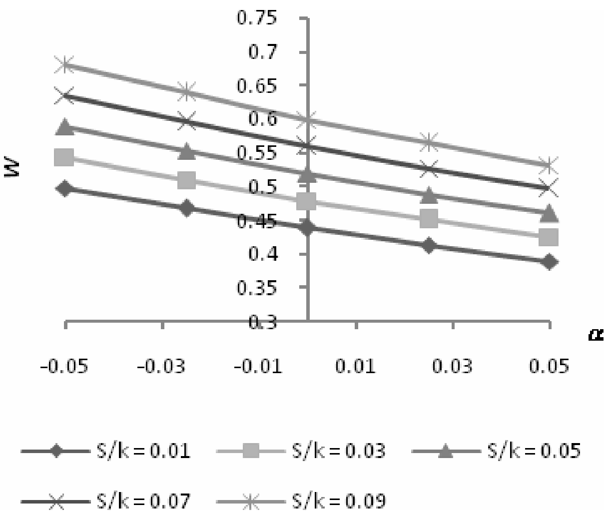

Fig. 17. Variation of Load carrying capacity with respect to α and S/k.

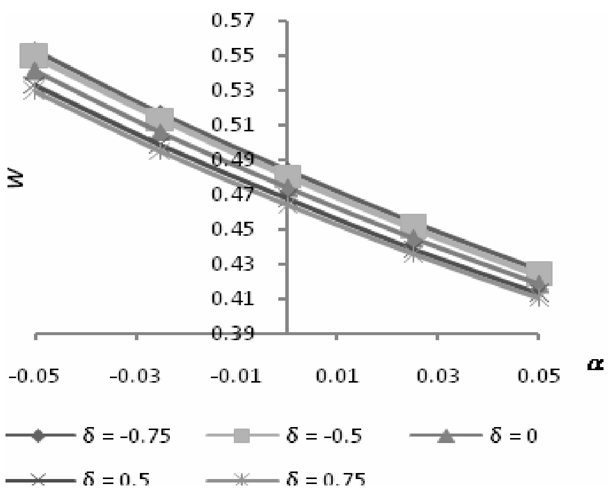

Fig. 18. Variation of Load carrying capacity with respect to α and δ.

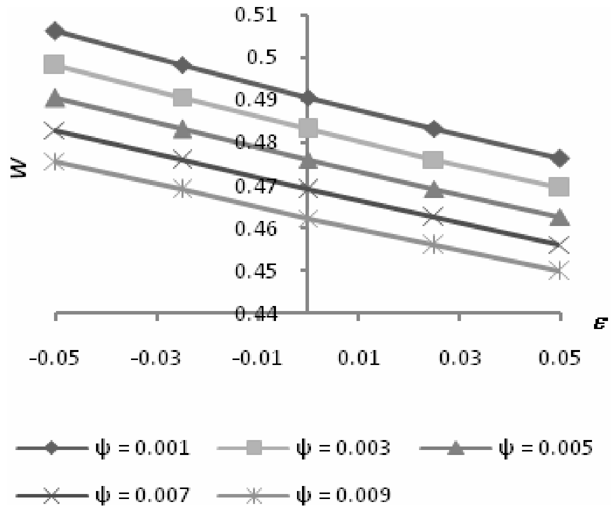

Fig. 19. Variation of Load carrying capacity with respect to ε and Ψ.

The fact that the negatively skewed roughness increases the load carrying capacity significantly can be seen from Figs. 19-21. Unlike, the earlier investigations of Shimpi and Deheri [34], here the effect of rotational inertia with respect to is opposite. Also, the load carrying capacity decreases due to positively skewed roughness. Notably, it is observed that the effect of rotational inertia on the distribution of load carrying capacity with respect to the porosity is almost negligible. It is found that the effect of S/*k* on the variance of load carrying capacity with respect to skewness is quite significant contrary to the findings of Shimpi and Deheri [34].

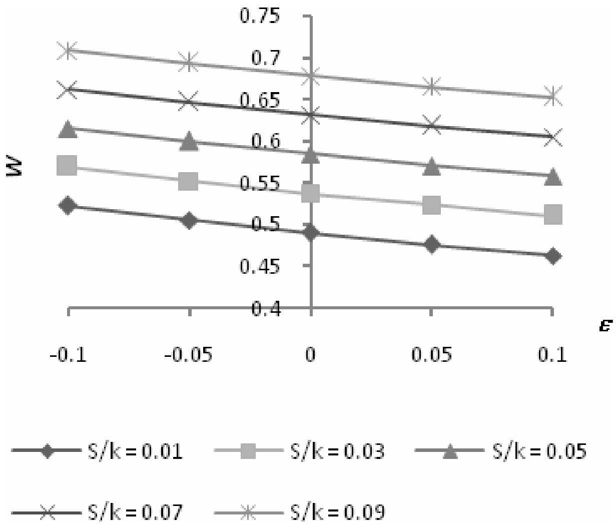

Fig. 20. Variation of Load carrying capacity with respect to ε and S/k.

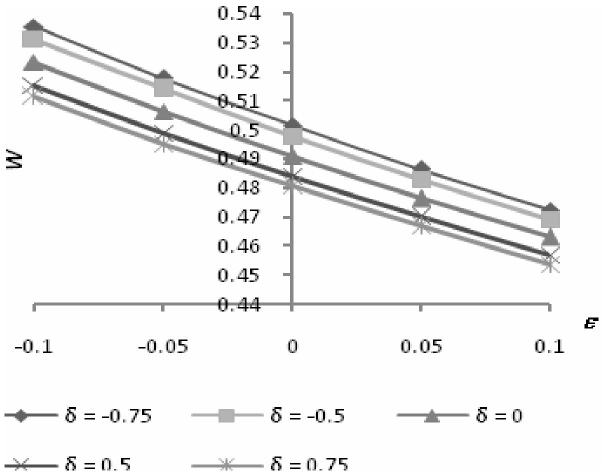

Fig. 21. Variation of Load carrying capacity with respect to ε and $\bar{\delta}$.

The deformation has a considerable adverse effect as can be seen from Figs. 22-23. Interestingly, a symmetric nature of load profile is observed with respect to $\bar{\delta}$ as can be seen from Fig. 23.

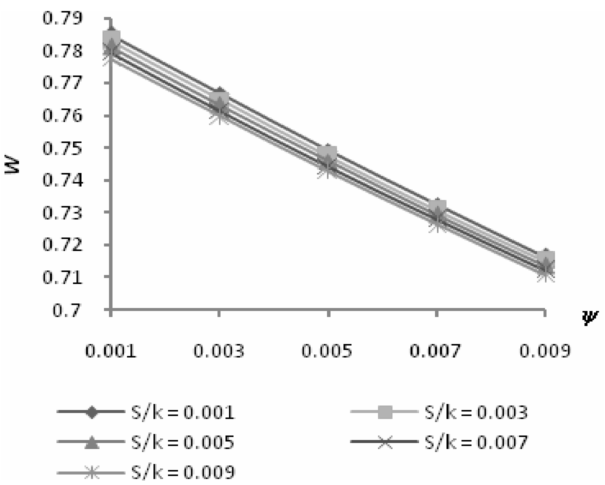

Fig. 22. Variation of Load carrying capacity with respect to Ψ and S/k.

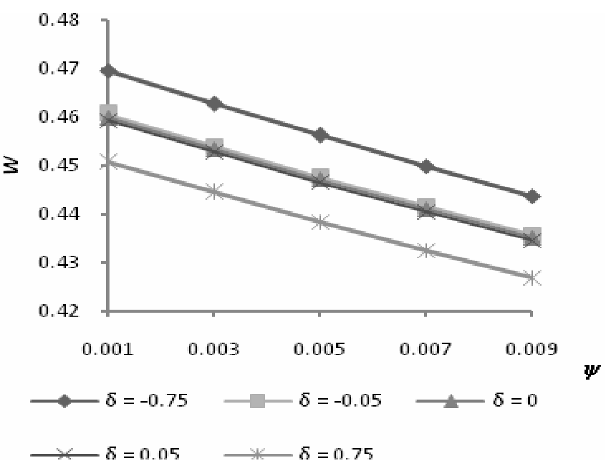

Fig. 23. Variation of Load carrying capacity with respect to Ψ and $\bar{\delta}$.

Besides, Fig. 24 reveals that the rotational inertia S/k negligibly increases the load carrying capacity which is in compatible with the findings of Shimpi and Deheri [34].

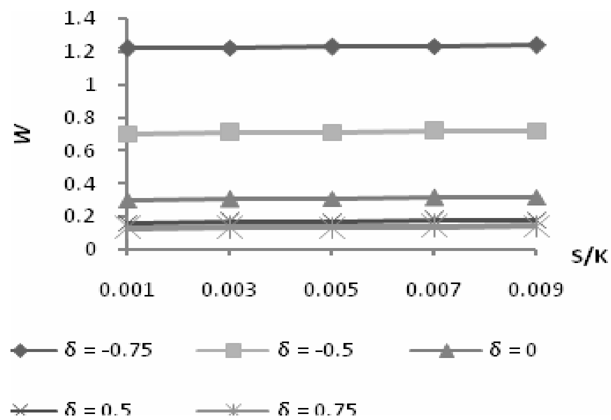

Fig. 24. Variation of Load carrying capacity with respect to S/k and $\bar{\delta}$.

Lastly, the opposite nature of the curvature parameters appears to be reflected in Figs. 25-26, so far as variation of load carrying capacity in concerned.

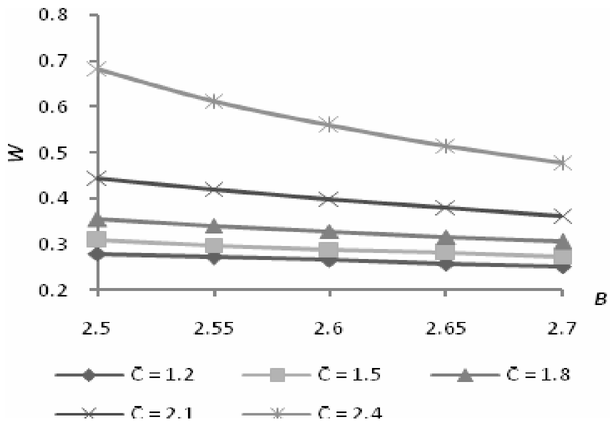

Fig. 25. Variation of Load carrying capacity with respect to $\bar{B}$ and $\bar{C}$.

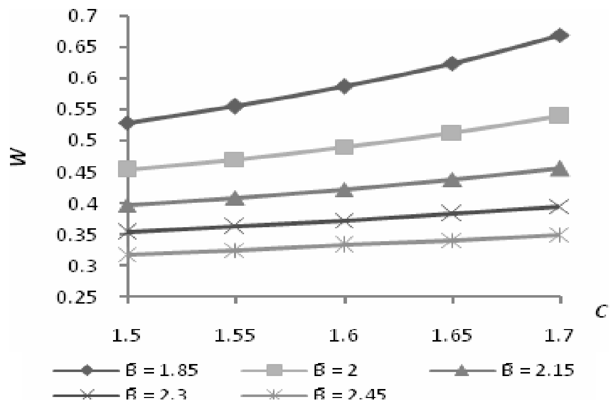

Fig. 26. Variation of Load carrying capacity with respect to $\bar{C}$ and $\bar{B}$.

Some of the figures witnessed here indicate that the combined adverse effect of standard deviation $\bar{\sigma}$, rotational inertia S/k and porosity Ψ is a fraction sharp when large values of deformation are involved. It is found that the adverse effect of porosity and standard deviation can be reduced to some extent by the magnetic fluid lubricant, at least in the case of negatively skewed roughness and this reduction becomes more with the involvement of negative variance. Besides, it is noticed that the bearing deformation tends to be more and more significant when large values of rotational inertia is involved. In addition, the combined effect of porosity and standard deviation remains considerably adverse for a large range of deformation parameter.

A close scrutiny of this study in comparison with the annular configuration of Shimpi and Deheri [34] signifies that the overall performance is relatively better in the present case. Unlike the annular geometry here the combined effect of rotational inertia and deformation is not that sharp. Lastly, it is observed that the curvature parameters induce a relatively better performance as compared to that of Shimpi and Deheri [34].

4. CONCLUSIONS

The occurrence of bearing deformation tends to suggest strongly that the roughness must be given due consideration while designing this type of bearing system, even if, magnetization parameter and curvature parameters are suitably chosen. Moreover, although there are several factors affecting the system adversely, it is important to note that the bearing can support a load even in the absence of flow.

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