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Surface Roughness and Deformation Effects on the Behaviour of a Magnetic Fluid Based Squeeze Film in Rotating Curved Porous Circular Plates

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Abstract The combined effect of surface roughness and bearing deformation on the magnetic fluid lubrication of a squeeze film between two rotating transversely rough porous circular plates has been investigated. The results indicate that the bearing performance gets adversely affected by the surface roughness, bearing deformation combination, even if, a magnetic fluid has been considered as the lubricant. However, the negatively skewed roughness introduces a better performance for a good range of deformation by suitably choosing the curvature parameters. It is appealing to note that, although there are several factors bringing down the load carrying capacity, still the bearing can support a good amount of load even when there is no flow unlike, the case of conventional lubricants. In addition, this article also emphasizes the role of rotation for improving the bearing performance.

Keywords: *circular plates, magnetic fluid, roughness, deformation, rotational inertia, load carrying capacity*

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1. Introduction

It was Archibald [1] who studied the squeeze film performance between various geometrical configurations of the flat surfaces. In [2], Prakash and Vij used the well known Morgan-Cameron approximation for a squeeze film between porous plates considering the porous facing thickness to be small enough, while the inertia effects induced by rotation were significant with higher rotational speed and small squeeze velocity. Wu [3] modified the analysis of Wu [4] to develop the Reynolds equation for the effect of rotation.

Murti [5] presented the squeeze film performance in curved circular plates describing the film thickness by an exponential expression. The investigation was based on the assumption that the central film thickness was constant unlike the minimum film thickness as considered by Hays [6]. Vora and Bhat [7] studied squeeze film behaviour between a curved upper plate with a porous facing and an impermeable flat plate taking rotation in to account. This investigation suggested that rotation of plates turned in an adverse effect on the performance of the bearing system. Ajwaliya [8] modified the above analysis by considering the lower plate also to be curved.

Normally, most of the theoretical investigations deal with squeeze film behaviour assuming that the bearing surfaces are perfectly rigid. But, under high loads the bearing may deform producing wedge effects in the lubricant film there by, altering the squeeze. Ramanaiah and Sundarammal [9] studied the effect of bearing deformation on the squeeze film character between circular and rectangular plates. It was established that the bearing deformation reduced the load carrying capacity and increased the squeeze. Muhsin [10] discussed the effect of elastic deformation of bearing surfaces and analyzed the steady state performance of offset halves bearing. It was shown that elastic deformation had a considerable influence on the performance especially, at higher eccentricity values.

Verma [11] studied the magnetic fluid lubrication of squeeze films and established its superiorely over conventional lubricant based squeeze films. Bhat and Deheri [12] investigated the effects of magnetic fluid lubricant on the action of squeeze film in curved porous circular disks, considering the method of Verma [11]. Chandra and Sinha [13] considered Ferrofluid lubrication of externally pressurized circular plates and conical bearings. It was observed that the effect of applied transverse magnetic field as well as Brownian time relaxation parameter was negligible on load carrying capacity.

Bhat and Deheri [14] investigated the performance of magnetic fluid based squeeze film in porous annular disks which was developed by Bhat [15] 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. Patel and Deheri [16] studied the squeeze film behaviour between curved circular plates lying along the surfaces determined by the secant function under the presence of a magnetic fluid lubricant. Lin et al [17] studied the effects of couple stresses resulting from the lubricant blended with various additives upon the squeeze film behavior between a long cylinder and a plane surface were analyzed. It was concluded that compared to the conventional Newtonian lubricant case, the couple stress effects characterized by the couple stress parameter signify an improvement in the squeeze film characteristics. Hsu et al [18] investigated 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 a transversely magnetic field resulted in an improvement of performance characteristics in rotating circular disks. Shah and Bhat [19] dealt with the Ferrofluid squeeze film between curved annular plates considering rotation of magnetic particles. It was found that load carrying capacity increased when volume concentration of the solid phase, Langrin's parameter or the curvature of the upper plate were increased. Lu et al [20] discussed the fluid inertia effect in magneto-hydrodynamic annular squeeze films. It was shown that the inertia correction factor in the magneto-hydrodynamic load carrying capacity was more pronounced with large Hartmann numbers.

Many methods were proposed to deal with the effect of surface roughness on the squeeze film performance characteristics. Tzeng and Saibel [21] adopted a stochastic approach to model the random roughness which in turn, was modified and developed by Christensen and Tonder [22,23,24] to study the effect of surface roughness in general. A number of investigations deployed the stochastic model of Christensen and Tonder to study effect of surface roughness [Ting [25], Prakash and Tiwari [26], Gupta and Deheri [27]]. Prajapati [28] investigated the squeeze film performance in rotating porous rough circular plates with elastic deformation modifying slightly the approach of Christensen and Tonder [22,23,24]. Hsu et al [29] observed the combined effect of surface roughness and rotational inertia on squeeze film performance between parallel circular disks. Davim [30] discussed about the performance of the hydrodynamic rough bearing which also examined tribology aspects with special emphasis on surface topography, wear of materials and lubrication.

Deheri et al [31] analyzed the behaviour of magnetic fluid lubrication of squeeze film between transversely rough curved plates. It was shown that the load carrying capacity increased with increasing magnetization. Although, the effect of transverse roughness was adverse in general, negatively skewed roughness resulted in a little better performance.

Ram et al [32] analyzed the effect of porosity on revolving Ferrofluid flow in rotating disk using Neuringer-Rosenwig model. It was found that the influence of rotation of fluid flow was not that significant on the displacement thickness of fluid layer. Huang et al [33] studied the effect of Ferrofluid lubrication under the presence of an external magnetic field. It was proved that the Ferrofluid had a good friction reduction performance in the presence of an external magnetic field compared with the carrier liquid and that its life period could be greatly improved.

Shimpi and Deheri [34] discussed the combined effect of surface roughness and deformation on the performance of a magnetic fluid based squeeze film between rotating curved porous circular plates. It was shown that the negatively skewed roughness resulted in a improved performance in spite of the adverse effect of porosity and deformation. However, for an overall improved performance, deformation deserved to be minimized even if a suitable magnetic strength was in force.

Lu and Deng [35] considered squeeze flow analysis of magnetorheological Fluids in parallel circular disks. The effect of slip coefficient and power law inertia on the flow field and squeeze force were discussed. Raoa et al [36] obtained a generalized form of Reynolds' equation for two symmetrical surfaces considering velocity slip and viscosity variation for the lubrication of the squeeze film between the circular plates. The load carrying capacity and squeezing time were found to be decreased due to the slip.

In spite of the fact that the transverse surface roughness introduced an adverse affect in general, the studies carried out by Vadher et al [37], Shimpi and Deheri [38] and Deheri et al [39] revealed that the negatively skewed roughness registered a relatively better performance. Therefore, it was thought appropriate to deal with the effect of surface roughness and bearing deformation on the squeeze film performance in curved porous rotating circular plates under the presence of a magnetic fluid lubricant.

2. Analysis

The geometry and configuration of the bearing system is shown in Figure 1, which consists of the circular disks.

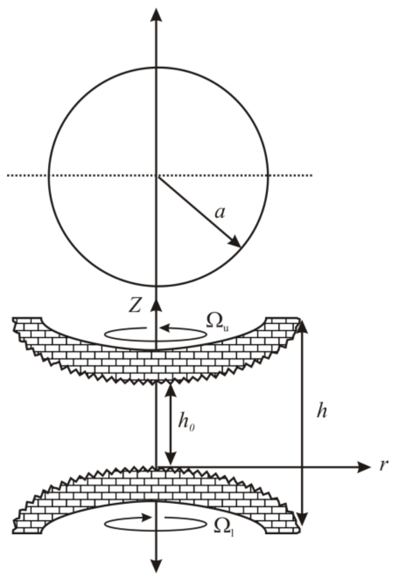

Figure 1.

Both the disks are assumed to be elastically deformable and their contact surfaces are considered to be transversely rough. The upper disk moves towards the lower disk normally with uniform velocity $\dot{h} = dh/dt$. Both the disks are assumed to have transversely rough surfaces.

Normally the following assumptions are made:

1) The viscosity remains constant and the flow is laminar.
2) There are no external forces acting on the lubricant.
3) The lubricant film thickness is very small in comparison with the dimension of the bearing.
4) Fluid inertia is small compared to viscous forces.
5) Velocity gradients across the film predominate.
6) The porous region is homogeneous and isotropic while the flow in the porous region is governed by Darcy's law.
7) Pressure and normal velocity component are continuous at the interface.
8) Zero leakage occurs within the operating specification.
9) Particles are not generated by wear.
10) Heat generated is almost negligible.

In view of the discussions regarding stochastic modeling of the roughness adopted by Christensen and Tonder [21,22,23] and Prajapati [28] the film thickness is assumed to be

$$h(r,t)+\delta(r,t)+h_s(r,\xi) \tag{1}$$

wherein 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

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

where c is the maximum deviation from the mean film thickness. The mean α, the standard deviation σ and the parameter ε, which is the measure of symmetry of the random variable h_S, are defined by the relationships

$$\alpha=E(h_s),\ \ \sigma^2=E\left[(h_s-\alpha)^2\right] \tag{3}$$

and

$$\varepsilon=E\left[(h_s-\alpha)^3\right] \tag{4}$$

where E denotes the expected value defined by

$$E(R)=\int_{-c}^{c} Rf(h_s)\,ds \tag{5}$$

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

$$Z_u=h_0\left[e^{-Br}\right];\qquad 0\le r\le a \tag{6}$$

approaches with normal velocity $\dot{h}_0=dh_0/dt$, to the lower plate lying along the surface

$$Z_l=h_0\left[\frac{1}{1+Cr}-1\right];\qquad 0\le r\le a \tag{7}$$

where h_0 is the central distance between the plates, B and C are the curvature parameters of the corresponding plates. The central film thickness $h(r)$ then is defined by [Bhat [15]]

$$h(r)=h_0\left[e^{-Br}-\frac{1}{1+Cr}+1\right];\qquad 0\le r\le a \tag{8}$$

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

$$\overline{H}=\left(H(r)\cos(\theta,z),\ 0,\ H(r)\sin(\theta,z)\right) \tag{9}$$

whose magnitude H is a function of r vanishing at $r=0$, the angle of inclination θ of the magnetic field as in Bhat [15]. Now the modified Reynolds equation governing the film pressure p takes the form [Prajapati [40], Bhat and Deheri [14], Gupta and Deheri [27], Vadher et al [37]]

$$\frac{1}{r}\frac{\partial}{\partial r}\left[rg(h)\frac{\partial}{\partial r}\left\{p-\frac{0.5\mu_0\overline{\mu}H^2}{a}\right\}\right]=12\mu\dot{h}_0+4S\Phi(h) \tag{10}$$

with

$$H^2=kr^2(a-r) \tag{11}$$

$$\begin{aligned}\Phi(h)=&\left(h+p'p_a\delta\right)^3+3\left(\sigma^2+\alpha^2\right)\left(h+p'p_a\delta\right)\\&+3\sigma^2\alpha+\alpha^3+\varepsilon\end{aligned} \tag{12}$$

and

$$\begin{aligned}g(h)=&\left(h+p'p_a\delta\right)^3+3\alpha\left(h+p'p_a\delta\right)^2\\&+3\left(\sigma^2+\alpha^2\right)\left(h+p'p_a\delta\right)+3\sigma^2\alpha+\alpha^3+\varepsilon+12\phi H_0\end{aligned} \tag{13}$$

where 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 and k is a suitably chosen constant to suit the dimension and strength of the magnetic field.

In view of the following non-dimensional quantities,

$$\begin{aligned}&P=-\frac{h_0^{3}p}{\mu\dot{h}a^2},\ R=\frac{r}{a},\ \overline{\sigma}=\frac{\sigma}{h_0},\ \overline{\alpha}=\frac{\alpha}{h_0},\ \overline{\varepsilon}=\frac{\varepsilon}{h_0^{3}}\\&\psi=\frac{\phi H_0}{h_0^{3}},\ \kappa=\frac{12\mu\dot{h}}{h_0^{3}},\ \overline{B}=Ba^2,\ \overline{C}=Ca^2,\\&\mu^*=-\frac{\mu_0\overline{\mu}kh_0^{3}}{\mu\dot{h}_0},\ \overline{p}=p'p_a,\ \overline{\delta}=\frac{\delta}{h},\ S=\frac{3\rho\Omega^3}{P_a}\end{aligned} \tag{14}$$

(ρ being density of lubricant and Ω the angular velocity)

integrating the stochastically averaged Reynolds Equation (11) under the boundary conditions

$$\left[\frac{\partial P}{\partial R}\right]_{R=0}=0,\qquad P(1)=0 \tag{15}$$

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

$$
\begin{aligned}
P &= 0.5\mu^* R^2 (R-1) \\
&-\left\{\frac{3+2D_1(S/\kappa)}{A_3}\right\}\ln\left(\frac{A_1+A_2R+A_3R^2}{A_1+A_2+A_3}\right) \\
&+\left\{\begin{array}{l}\dfrac{6A_2}{A_3\sqrt{4A_1A_3-A_2^2}} \\ +4\left(\dfrac{S}{\kappa}\right)\left(\dfrac{2A_3D_2-A_2D_1}{A_3\sqrt{4A_1A_3-A_2^2}}\right)\end{array}\right\} \\
&\times\left\{\begin{array}{l}\tan^{-1}\left[\dfrac{2A_3R+A_2}{\sqrt{4A_1A_3-A_2^2}}\right] \\ -\tan^{-1}\left[\dfrac{2A_3+A_2}{\sqrt{4A_1A_3-A_2^2}}\right]\end{array}\right\}
\end{aligned} \tag{16}
$$

where

$$
\begin{aligned}
A_1 &= \left(1+\overline{p\delta}\right)^3 + 3\overline{\alpha}\left(1+\overline{p\delta}\right)^2 \\
&+3\left(\overline{\sigma}^2+\overline{\alpha}^2\right)\left(1+\overline{p\delta}\right)+3\overline{\sigma}^2\overline{\alpha}+\overline{\alpha}^3+\overline{\varepsilon}+12\psi;
\end{aligned} \tag{17}
$$

$$
\begin{aligned}
A_2 &= \left(\overline{C}-\overline{B}\right)\Big[\left(1+\overline{p\delta}\right)^3 \\
&+2\overline{\alpha}\left(1+\overline{p\delta}\right)^2+\left(\overline{\sigma}^2+\overline{\alpha}^2\right)\left(1+\overline{p\delta}\right)\Big];
\end{aligned} \tag{18}
$$

$$
\begin{aligned}
A_3 &= 1.5\Big[\left(3\overline{B}^2-4\overline{BC}\right)\left(1+\overline{p\delta}\right)^3 \\
&+2\overline{\alpha}\left(2\overline{B}^2-2\overline{BC}-\overline{C}^2\right)\left(1+\overline{p\delta}\right)^2 \\
&+\left(\overline{\alpha}^2+\overline{\sigma}^2\right)\left(\overline{B}^2-2\overline{C}^2\right)\left(1+\overline{p\delta}\right)\Big]
\end{aligned} \tag{19}
$$

$$
\begin{aligned}
B_1 &= \left(1+\overline{p\delta}\right)^3+3\left(\overline{\sigma}^2+\overline{\alpha}^2\right)\left(1+\overline{p\delta}\right) \\
&+3\overline{\sigma}^2\overline{\alpha}+\overline{\alpha}^3+\overline{\varepsilon};
\end{aligned} \tag{20}
$$

$$
B_2 = 3\left(\overline{C}-\overline{B}\right)\left[\left(1+\overline{p\delta}\right)^3+\left(\overline{\sigma}^2+\overline{\alpha}^2\right)\left(1+\overline{p\delta}\right)\right]; \tag{21}
$$

and

$$
\begin{aligned}
B_3 &= 1.5\Big[\left(3\overline{B}^2-4\overline{BC}\right)\left(1+\overline{p\delta}\right)^3 \\
&+\left(\overline{\alpha}^2+\overline{\sigma}^2\right)\left(\overline{B}^2-2\overline{C}^2\right)\left(1+\overline{p\delta}\right)\Big];
\end{aligned} \tag{22}
$$

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

$$
W = -\frac{h_0^3 w}{\mu \dot{h}_0 a^4} = \int_0^1 RP\, dR \tag{23}
$$

which leads to

$$
\begin{aligned}
W &= \frac{\mu^*}{40}-\left\{\frac{X_1(A_3-A_2)-X_2\sqrt{4A_1A_3-A_2^2}}{2A_3}\right\} \\
&+\left\{\frac{X_1\left(A_2^2-2A_1A_3\right)+A_2X_2\sqrt{4A_1A_3-A_2^2}}{4A_3^2}\right\} \\
&\times\ln\left(\frac{A_1+A_2+A_3}{A_1}\right) \\
&+\left\{\frac{A_2X_1\sqrt{4A_1A_3-A_2^2}-X_2\left(A_2^2-2A_1A_3\right)}{2A_3^2}\right\} \\
&\times\left\{\tan^{-1}\left[\frac{2A_3+A_2}{\sqrt{4A_1A_3-A_2^2}}\right]-\tan^{-1}\left[\frac{A_2}{\sqrt{4A_1A_3-A_2^2}}\right]\right\}.
\end{aligned} \tag{24}
$$

where,

$$
X_1 = \frac{3+(S/\kappa)D_1}{A_3}; \tag{25}
$$

$$
X_2 = \frac{3A_2-2(S/\kappa)(2A_1D_2-A_2D_1)}{A_3\sqrt{4A_1A_3-A_2^2}}; \tag{26}
$$

$$
D_1 = \frac{6B_1A_3-3A_1B_3-4A_2B_2A_3+3B_3A_2^2}{A_3}; \tag{27}
$$

$$
D_2 = \frac{4B_2A_3-3A_2B_3}{A_3^2} \tag{28}
$$

3. Results and Discussion

It is observed that the pressure distribution and the load carrying capacity in dimensionless form are obtained from Equation (16) and Equation (24) respectively. Also, It can be seen from these equations that the non-dimensional pressure distribution and the load carrying capacity are decided by several parameters such as μ^*, $\overline{\sigma}$, $\overline{\alpha}$, $\overline{\varepsilon}$, ψ, $\overline{B}$, $\overline{C}$, S/κ and $\overline{\delta}$. It is noticed that for porous bearing with smooth surfaces this investigation gets reduced to the study of Bhat and Deheri [12] considering with the magnetic fluid based squeeze film between circular plates in the absence of rotation and deformation.

Further, taking 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 [2] when no rotation and deformation are involved. Moreover, setting the magnetization parameter to be equal to zero for a porous bearing with smooth surfaces one can obtain the results of Bhat [15] in the absence of deformation.

It is noticed from Equation (16) that the non-dimensional pressure increases by

$$
0.5\mu^*\left[R^2(1-R)\right] \tag{30}
$$

while the dimensionless load carrying capacity gets enhanced by

$$\mu^* / 40 \qquad (31)$$

as compared to the case of conventional lubricant. It is easily observed that W is linear with respect to μ^* as a result, higher values of μ^* may provide increased load carrying capacity. Also, it is noticed that bearing deformation significantly alters the profile of the pressure distribution. It is seen that, the effect of the transverse surface roughness is, in general, adverse. Probably, this may be due to the fact that the roughness of the bearing surfaces retards the motion of the lubricant resulting in reduced load carrying capacity. The elastic deformation further makes this negative effect more significant.

For the graphical representation the following fixed values are taken it to consideration

$$\mu^* = 0.7,\ \overline{\alpha} = -0.05,\ \overline{\varepsilon} = -0.05,\ \psi = 0.01,$$
$$S/\kappa = 0.01,\ \overline{C} = 2.3,\ \overline{B} = 2.4,\ \overline{\delta} = 0.01.$$

Figure 2 - Figure 9 giving the variation of load carrying capacity with respect to the magnetization parameter indicate that load carrying capacity increases due to the magnetic fluid lubricant, and this increase is more in the case of standard deviation and rotational inertia. The effect of skewness on the load carrying capacity with respect to μ^* appears not that significant.

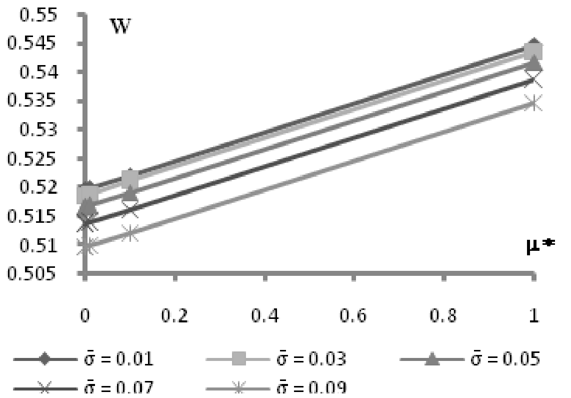

Figure 2. Variation of Load carrying capacity with respect to μ^* and σ

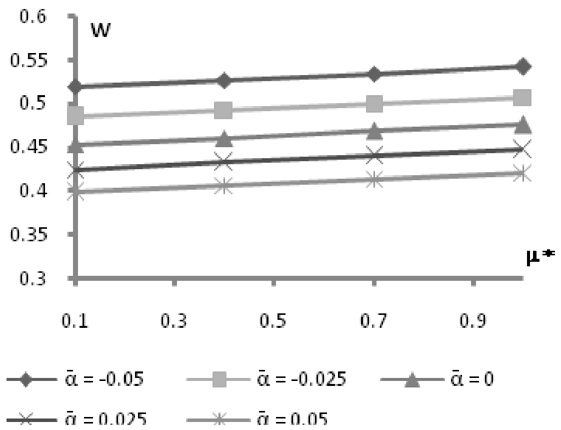

Figure 3. Variation of Load carrying capacity with respect to μ^* and α

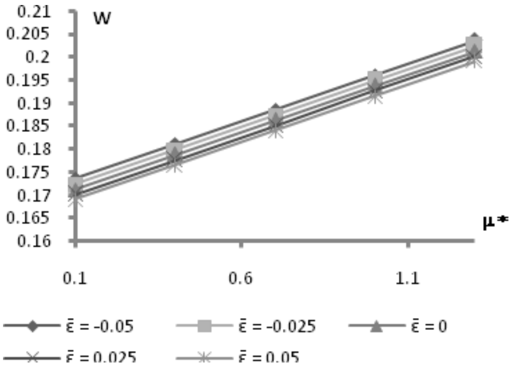

Figure 4. Variation of Load carrying capacity with respect to μ^* and $\overline{\varepsilon}$

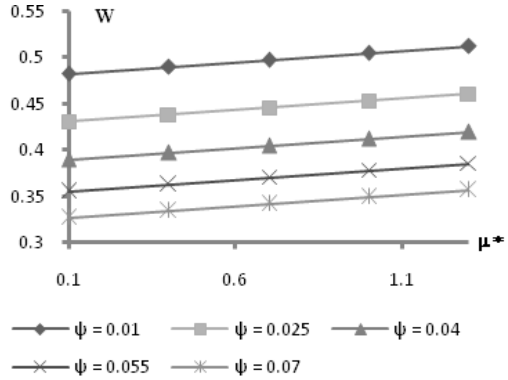

Figure 5. Variation of Load carrying capacity with respect to μ^* and ψ

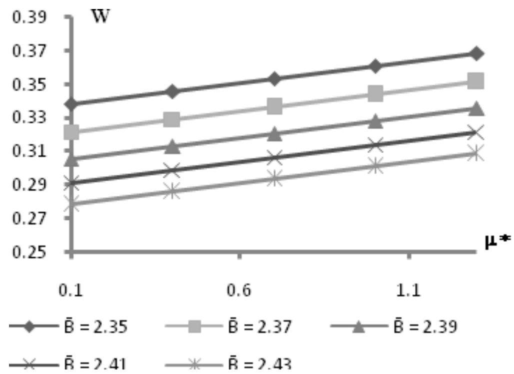

Figure 6. Variation of Load carrying capacity with respect to μ^* and $\overline{B}$

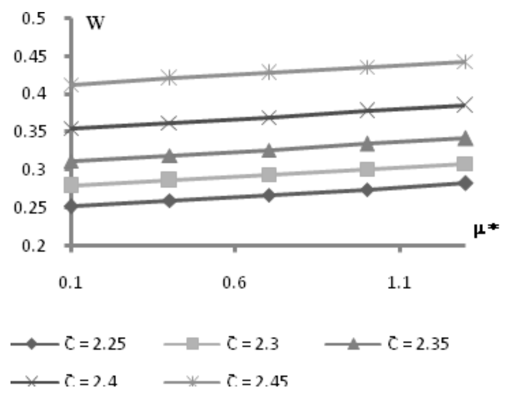

Figure 7. Variation of Load carrying capacity with respect to μ^* and $\overline{C}$

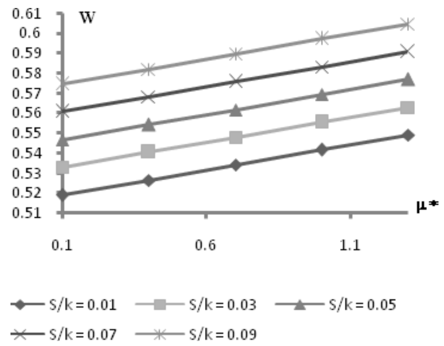

Figure 8. Variation of Load carrying capacity with respect to μ^* and S/κ

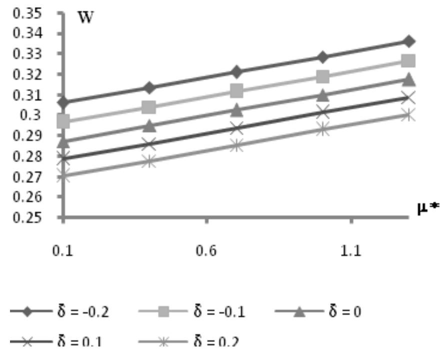

Figure 9. Variation of Load carrying capacity with respect to μ^* and $\overline{\delta}$

A closed scrutiny of these Figures reveals that the performance of the bearing system registers an improved performance. However, variance (negative) induces an increase in the load carrying capacity, while negatively skewed roughness marginally increases the load carrying capacity. Further, the curvature parameters increase the load carrying capacity similar to the case of circular plates as discussed in Shimpi and Deheri [39].

The profile of the distribution of dimensionless load carrying capacity with respect to the standard deviation associated with roughness is depicted in Figures 10-14. It is evident that the standard deviation adversely affects the performance of the bearing system in the sense that the load carrying capacity decreases considerably due to the standard deviation. However, this decrease is relatively more in the cases of rotational inertia. In other words, the combined effect of standard deviation, rotational inertia and deformation turns in significantly reduced load carrying capacity there by, adversely affecting the bearing system.

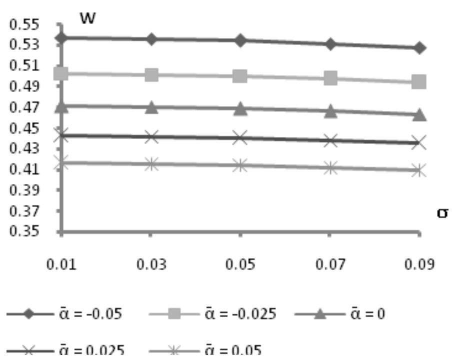

Figure 10. Variation of Load carrying capacity with respect to $\overline{\sigma}$ and α

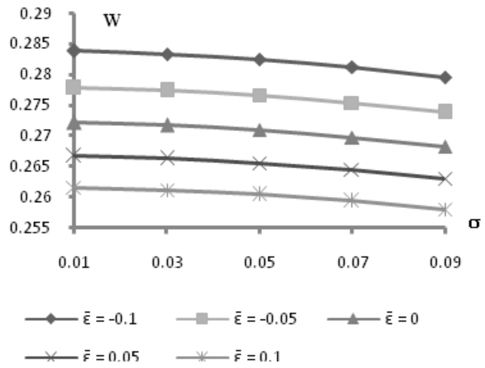

Figure 11. Variation of Load carrying capacity with respect to $\overline{\sigma}$ and ε

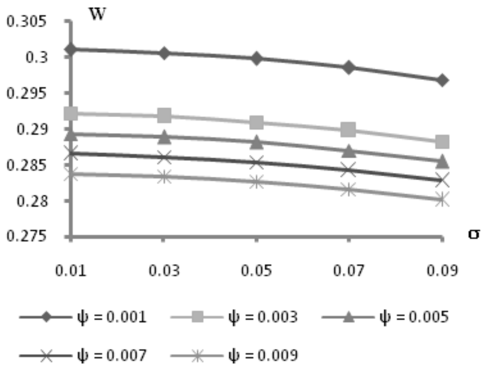

Figure 12. Variation of Load carrying capacity with respect to $\overline{\sigma}$ and ψ

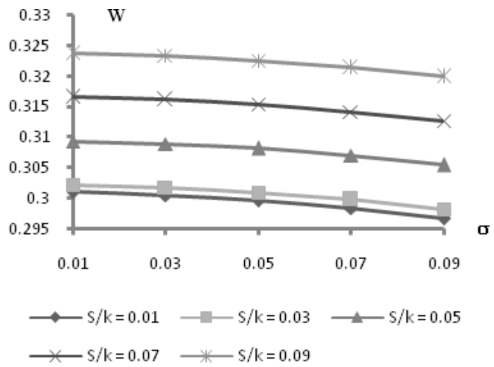

Figure 13. Variation of Load carrying capacity with respect to $\overline{\sigma}$ and S/κ

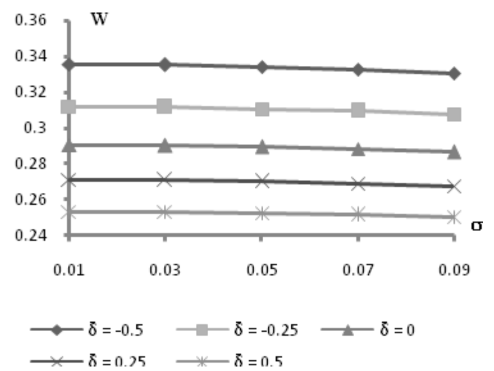

Figure 14. Variation of Load carrying capacity with respect to $\overline{\sigma}$ and δ

It is also observed that the decreased load carrying capacity due to standard deviation gets further decreased owing to porosity. It is revealed that for larger values of deformation, this negative effect is more crucial.

The distribution of non- dimensional load carrying capacity with respect $\bar{\alpha}$ can be obtained from form Figure 15 – Figure 18. It is clearly seen that $\bar{\alpha}$ (positive) decreases the load carrying capacity while $\bar{\alpha}$ (negative) introduces an increase in the load carrying capacity. The effect of deformation on distribution of the load carrying capacity with respect to $\bar{\alpha}$ is quite significant. In addition, the effect of rotational inertia on the distribution of load carrying capacity with respect to variance (positive) is equally significant.

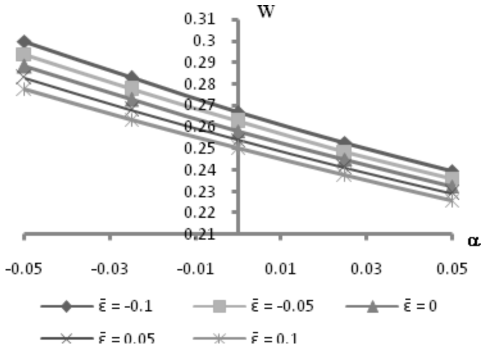

Figure 15. Variation of Load carrying capacity with respect to $\bar{\alpha}$ and ε

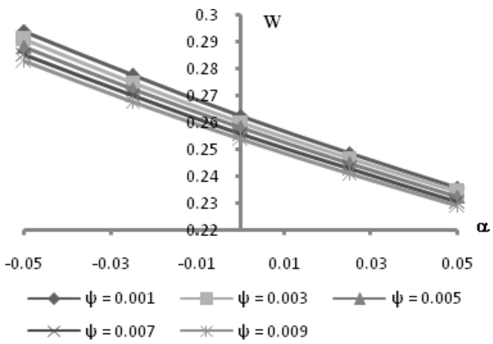

Figure 16. Variation of Load carrying capacity with respect to α and ψ

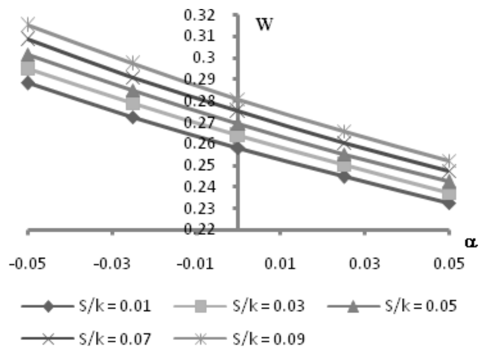

Figure 17. Variation of Load carrying capacity with respect to $\bar{\alpha}$ and S/κ

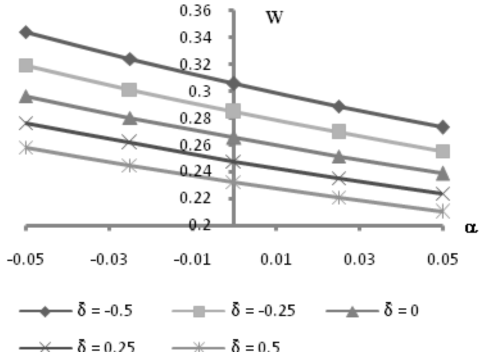

Figure 18. Variation of Load carrying capacity with respect to $\bar{\alpha}$ and $\bar{\delta}$

The effect of skewness on the profile of the load carrying capacity given in Figure 19 – Figure 21 indicates that the skewness follows almost the trends of variance. This also proves that the combined effect of negatively skewed roughness and negative variance is quite significant from the bearing design point of view. Further more, it is interesting to note that the effect of the deformation with respect to skewness is significant. The fact that, the porosity effect on the distribution of load carrying capacity is considerably adverse that can be obtained from the Figure 22 – Figure 23. One can easily infer that the decrease in the load carrying capacity due to porosity is crucial when considered with the case of bearing deformation. In addition, larger values of deformation make this negative effect of porosity quite crucial for the bearing design. Figure 24 underlines that the rotational inertia decreases the load carrying capacity.

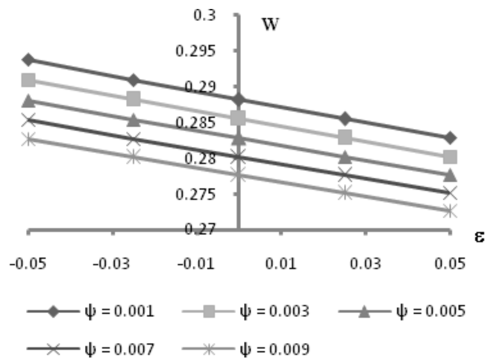

Figure 19. Variation of Load carrying capacity with respect to $\bar{\varepsilon}$ and ψ

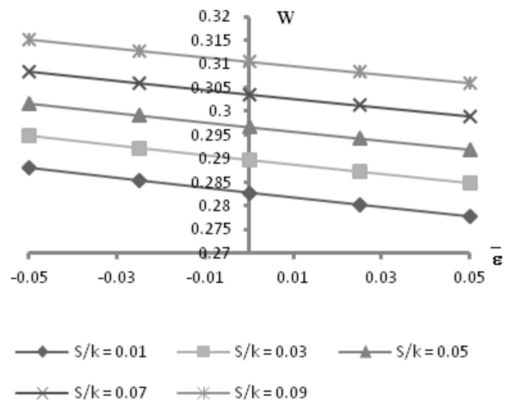

Figure 20. Variation of Load carrying capacity with respect to ε and S/κ

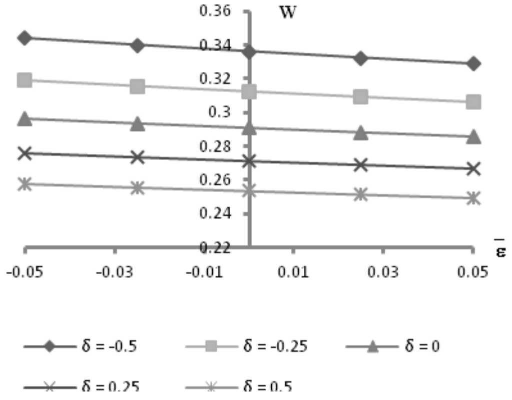

Figure 21. Variation of Load carrying capacity with respect to $\overline{\varepsilon}$ and $\overline{\delta}$

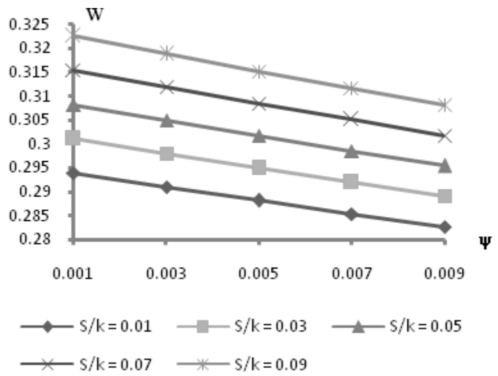

Figure 22. Variation of Load carrying capacity with respect to ψ and S/κ

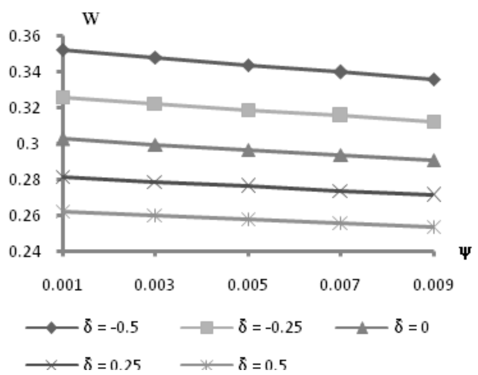

Figure 23. Variation of Load carrying capacity with respect to ψ and δ

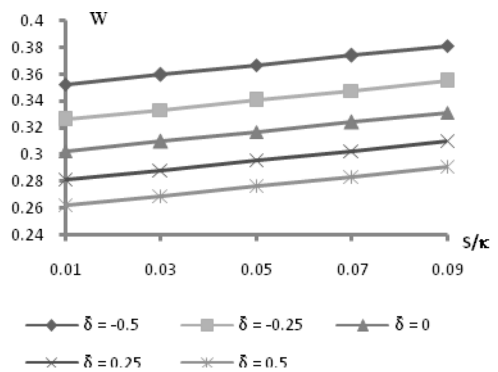

Figure 24. Variation of Load carrying capacity with respect to S/κ and δ

Lastly, Figure 25- Figure 26 presenting the variation of the load carrying capacity with respect to the curvature parameters make an interesting scenario in the sense that the load carrying capacity increases sharply with increase in $\overline{C}$ when lower values of $\overline{B}$ are involved. Besides, the trends of the load carrying capacity with respect to the curvature parameters are opposite to that of the investigation of Shimpi and Deheri [33].

Also, the trends of load carrying with respect to $\overline{C}$ is almost opposite to that of $\overline{B}$. Accordingly, the difference $\overline{C}-\overline{B}$ encountered in Equation (15) plays a prominent role in bearing design.

Some of the Figures witnessed here tend to indicate that the combined adverse effect of $S/\kappa, \psi$ and $\overline{\sigma}$ is more pronounced especially when larger values of deformation are involved. It is noticed that the negative effect of porosity, standard deviation and rotational inertia can be minimized up to some extent by the positive effect of the magnetic fluid lubricant in the case of negatively skewed roughness, particularly, when negative variance occurs. It is observed that the effect of the bearing deformation becomes more and more significant when large values of rotational inertia are involved.

Lastly, the combined effect of standard deviation and porosity remains considerably adverse for a good amount of range of deformation parameter.

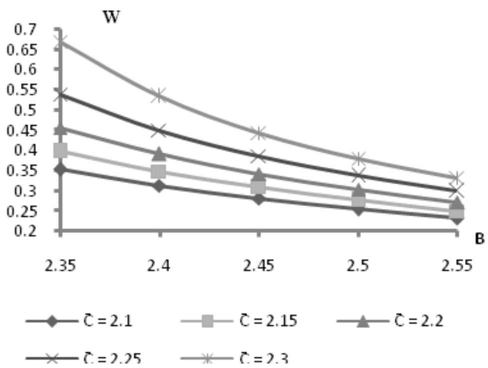

Figure 25. Variation of Load carrying capacity with respect to $\overline{B}$ and $\overline{C}$

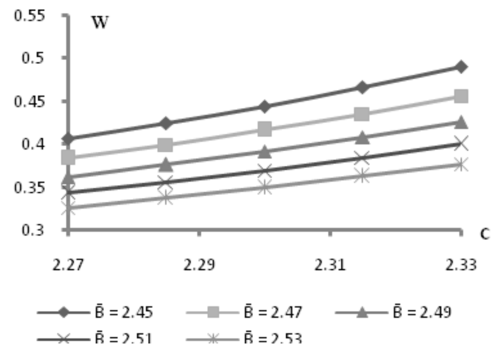

Figure 26. Variation of Load carrying capacity with respect to $\overline{C}$ and $\overline{B}$

In this type of bearing system even the lower to moderate values of rotational inertia may provide a enhanced performance for a large range of deformation.

A critical review of the graphs presented in Vadher et al [37], in comparison with the graphs of the present investigation suggests that the load carrying capacity is relatively less increased here so far as the combined effect of skewness and variance is concerned [cf. Figure 27].

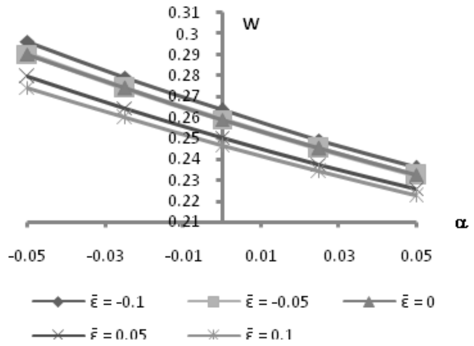

Figure 27. Variation of Load carrying capacity with respect to $\bar{\alpha}$ and $\bar{\varepsilon}$

Furthermore, a comparison of present investigation with the results of Deheri et al [38] tells that even the load carrying capacity is less increased here so far as the effect of porosity and standard deviation are concerned [cf. Figure 28].

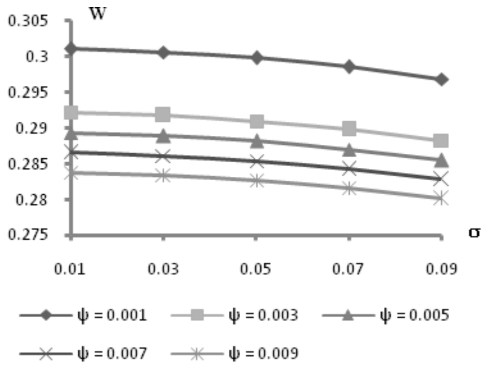

Figure 28. Variation of Load carrying capacity with respect to σ and ψ

Also, a critical study of the results of Shimpi and Deheri [39] together with the conclusions of the current study reveals that the combined effect of rotation and deformation leads to a decrease in the load carrying capacity here. It is seen that the curvature parameters present a relatively better performance here as compared to the squeeze film behaviour discussed in Patel and Deheri [16] in the absence of rotation and deformation.

A close look at the current investigation indicates that the load carrying capacity is comparatively more here with regard to the discussion of Hsu at el [18] while the effect of deformation is also significant. It can be observed that the effect of the roughness is almost identical with that of Hsu at el [29] for small values of rotational inertia.

4. Conclusion

It is seen that in this type of bearing system, the effect of bearing deformation is quite significant unlike the study of quite Shimpi and Deheri [34]. It is studied that the positive effect of magnetization can be obtained by considering the following combination of the parameters:

$$\bar{B} = 2.401,\ \bar{C} = 2.304,\quad S/\kappa = 0.097,\ \psi = 0.0098,$$
$$\bar{\alpha} = -0.0510,\ \bar{\varepsilon} = -0.0514,\ \bar{\sigma} = 0.0487,\ \bar{\delta} = 0.012$$

It is found that hydrodynamic effect tends to reduce the adverse effect of roughness and porosity. Due to the positive effect of negatively skewed roughness, the presence of bearing deformation makes it mandatory that the roughness must be accounted while designing the bearing system even if curvature parameters, rotational parameter and magnetic strength are appropriately considered. It is needless to say that this article suggests an additional degree of freedom from design point of view.

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