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Effect of Longitudinal Surface Roughness on the Performance of a Shliomis Model Based Ferrofluid Circular Squeeze Film Considering Couple stresses Effect

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ABSTRACT: This paper aims to analyze the combined effect of couple stress and longitudinal surface roughness on the behavior of a Shliomis model based ferrofluid squeeze film in circular plates. The stochastic averaging model of Christensen and Tonder for longitudinally surface roughness has been adopted here. The generalized Reynolds' type equation obtained by the application of Shliomis ferro-hydrodynamic model incorporating the Stoke's micro- continuum theory has been stochastically averaged. The pressure distribution in the bearing system has been calculated taking recourse to appropriate boundary conditions. Then load carrying capacity of the bearing system is derived. The results presented in graphical forms, make it clear that the longitudinal surface roughness significantly modifies the squeeze film performance. It is clearly observed that the enhanced performance due to the combined effect of couple stress and magnetization gets further improved due to longitudinal surface roughness barring the case of negatively skewed roughness.

KEYWORDS: Circular plates, Squeeze film, roughness, Couple stress, Ferrofluid.

1 INTRODUCTION

The Squeeze film behavior is used in various fields of real life such as gears, hydraulic systems, rolling elements, engines, clutch plates etc.

In the last decade many theoretical and experimental investigations have been made on the bearing systems to increase the life period. One of the major investigations in the direction of lubrication of bearing system was the use of ferrofluid as a lubricant. Verma (1986) dealt with the effect of magnetic field on squeeze film bearing system, under an externally applied magnetic field oblique to the lower surface. It was found that the upper plate took longer time to come down in this case as compared to the conventional lubricant based squeeze film. The squeeze film for spherical and conical bearings using magnetic fluid as a lubricant with effect of rotation of particles and constant magnetic field in the transverse direction was studied for various bearing characteristics (Kumar et. al.(1992)). Bhat and Deheri (1993) investigated the curved porous circular squeeze film with the effect of magnetic fluid. It was noticed that the pressure, load carrying capacity and response time increased with the increase in magnetization. Prajapati (1995) discussed various squeeze films with magnetic field effect. Superior performance of the magnetic fluid lubricant as compared to the conventional lubricant was manifest. Patel and Deheri (2002) analyzed the effect of magnetic fluid based squeeze film between two curved plates lying along the surfaces determined by secant functions. Shah and Bhat (2005) presented the effects of ferrofluid on the curved squeeze film between two annular plates, when the upper plate approached the lower one normally, including the rotation of magnetic particles and their magnetic moments. Patel and Deheri (2007) discussed the performance of a magnetic fluid based squeeze film between porous conical plates. Lin (2013) investigated the influence of fluid inertia forces on the ferrofluid squeeze film between a sphere and a plate in the presence of external magnetic fields considering Shliomis model based magnetic fluid flow.

Many methods were proposed to improve the performance of the bearing system, one such method was the use of couple stress fluids. Bujurke and Jayaraman (1982) analyzed the influence of couple stresses in squeeze films. Bujurke and Naduvinamani (1991) investigated the performance of narrow porous journal bearing lubricated with couple stress fluid. Lin (1997) dealt with the effect of squeeze film characteristics of long partial journal bearings lubricated with couple stress fluids. Lin (2000) studied the performance of squeeze film characteristics between a sphere and a flat plate using couple stress fluid model. These studies have confirmed higher load carrying capacity, lower coefficient of friction, and delayed time of approach in comparison with the Newtonian case.

In all the above investigations bearing surfaces were considered to be smooth. After having some run-in and wear the bearing surfaces develop roughness, which appears to be random in character, hardly following any particular structural pattern. Many methods have been proposed to study and analyze the effect of surface roughness on the performance of the squeeze film bearing system. A quite a good number of investigations (Tzeng and Saible (1967), Christensen and Tonder (1969a, 1969b, 1970) discussed a stochastic approach to mathematically model the random nature of surface roughness. In view of the Christensen and Tonder's stochastic model of roughness, Prakash and Tiwari (1983), Prajapati (1991), Guha (1993), Deheri et al. (2004), Naduvinamani et al. (2005), Bujurke et al. (2008), dealt with the effect of surface roughness on the performance of different bearing systems considering transverse roughness as well as longitudinal roughness. It was established that the effect of roughness was quite significant. Andharia and Deheri (2010) analyzed the performance of a squeeze film formed by a magnetic fluid between longitudinally rough conical plates. An improvement in the performance of a squeeze film formed by a magnetic fluid between longitudinally rough conical plates was observed for, thereby, extending the life period of the bearing system. Patel et al. (2011) investigated the performance of a magnetic fluid based squeeze film between rough circular plates, while upper plate had a porous facing of variable porous matrix thickness. Many investigators (Patel et al. (2011), Abhangi and Deheri (2012), Patel and Deheri (2013)) dealt with the effect of transeverse roughness on different geometries of bearing systems. Patel and Deheri (2013) discussed the performance of a ferrofluid based squeeze film in rotating rough curved circular plates resorting to Shliomis model. It was noticed that the adverse effect of roughness could be reduced considerably at least in the case of negatively skewed roughness with a suitable choice of curvature parameter. Andharia and Deheri (2013) analyzed the performance of a magnetic fluid based squeeze film between longitudinally rough elliptical plates. Lin et al. (2014) studied the squeeze film performance between curved circular plates lubricated with an electrically conducting non-Newtonian fluid in the presence of external magnetic fields. Patel et al. (2015) presented the squeeze film behavior in annular disks with a non-Newtonian ferrofluid in the presence of transverse magnetic fields. It was suggested that the non-Newtinian ferrofluid lubricated squeeze film registered higher load carrying capacity in comparison with the Newtonian non ferrofluid cases. Patel et al. (2014) analyzed the effect of transverse surface roughness on the performance of a squeeze film in parallel circular disks with non -Newtonian ferrofluid under the presence of transverse magnetic field. The results established that the transverse surface roughness significantly affected the squeeze film performance. This article offered some measures to compensate the adverse effect of roughness under suitable conditions due to the positive effect of non-Newtonian ferrofluid.

Here, it has been proposed to deal with the effect of longitudinal surface roughness and couple stress on the behavior of a ferrofluid circular squeeze film taking Shliomis model of ferrofluid lubrication in to account.

2 ANALYSIS

The geometrical configuration of the bearing system is displayed in Figure 1.

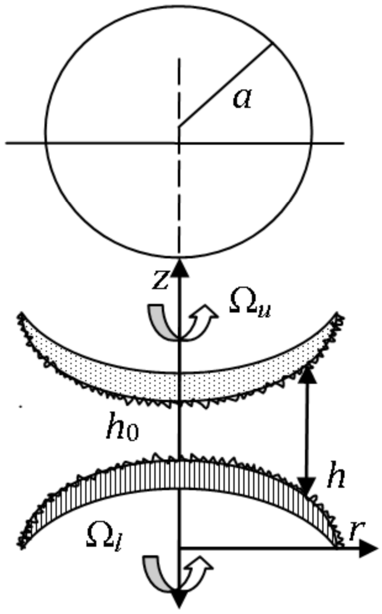

Fig. 1. The geometrical configuration of the bearing system

In view of the discussion of Christensen and Tonder (1969a, 1969b, 1970) for the stochastic modeling of roughness, the thickness $h(x)$ of the lubricant film is assumed to be;

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

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 surfaces. The deviation h_s is described by a generalized probability density function. The details of mean α, the standard deviation σ and the parameter ε, which is the measure of symmetry of the random variable h_s, are considered from the theory of Christensen and Tonder (1969a, 1969b, 1970).

The stochastically averaged [Christensen and Tonder (1969a, 1969b, 1970)] modified Reynolds' equation governing the pressure distribution, for the performance of a ferrofluid lubricated squeeze film in circular disk Bhat and Deheri (1993) with non-Newtonian couple stress [Lin et al. (2013)] is obtained as

$$f(h, l_c, \phi, \tau)\frac{1}{r}\frac{d}{dr}\left\{r\frac{dp}{dr}\right\} = 12\eta_0(1+\tau)(1+2.5\phi)\frac{dh}{dt} \tag{1}$$

Where

$$\begin{aligned} f(h, l_c, \phi, \tau) = {} & h^{-3}(1-4\alpha h^{-1}) - 12\frac{l_c^2}{(1+\tau)(1+2.5\phi)^{3/2}}.4h^{-2}(\sigma^2+\alpha^2) \\ & - 24\frac{l_c^3}{(1+\tau)^{3/2}(1+2.5\phi)^{3/2}}\left(\tanh\left[\frac{\sqrt{(1+\tau)(1+2.5\phi)}}{2l_c}.24h^{-3}(\varepsilon+3\sigma^2\alpha \right.\right. \\ & \left.\left. +\alpha^3)\right]\right) \end{aligned} \tag{2}$$

The associated boundary conditions are

$$r = 0, \frac{dp}{dr} = o \text{ and } r = R, p = 0 \tag{3}$$

Solving equation (1) with the boundary conditions (3), the expression for dimensionless pressure distribution is found to be :

$$P^* = \frac{3(1+\tau)(1+2.5\phi)}{f^*(h^*, l_c, \phi, \tau)}\left(1 - r^{*^2}\right) \qquad (4)$$

Where,

$$f^*(h^*, l_c, \phi, \tau) = h^{*-3}\left(1 - 4\alpha^* h^{*-1}\right) - 12\frac{C^2}{(1+\tau)(1+2.5\phi)^{3/2}}.4h^{*-2}\left(\sigma^{*2} + \alpha^{*2}\right) - 24\frac{C^3}{(1+\tau)^{3/2}(1+2.5\phi)^{3/2}}\left(\tanh\left[\frac{\sqrt{(1+\tau)(1+2.5\phi)}}{2C}.24h^{*-3}\left(\varepsilon^* + 3\sigma^{*2}\alpha^* + \alpha^{*3}\right)\right]\right)$$

Where, α^* is non dimensional variance, σ^* is non dimensional standard deviation and ε^* is non dimensional skewness.

Integrating the film pressure over the film region, one can get the non dimensional load carrying capacity in the form of

$$W^* = \frac{3\pi(1+\tau)(1+2.5\phi)}{f^*(h^*, l_c, \phi, \tau)} \qquad (5)$$

3 Results and Discussion

It is observed that in the absence of roughness this investigations turns to the discussions of Lin et al. (2013). Further, setting the couple stress parameter to be zero, one obtains the study of Bhat (2003).

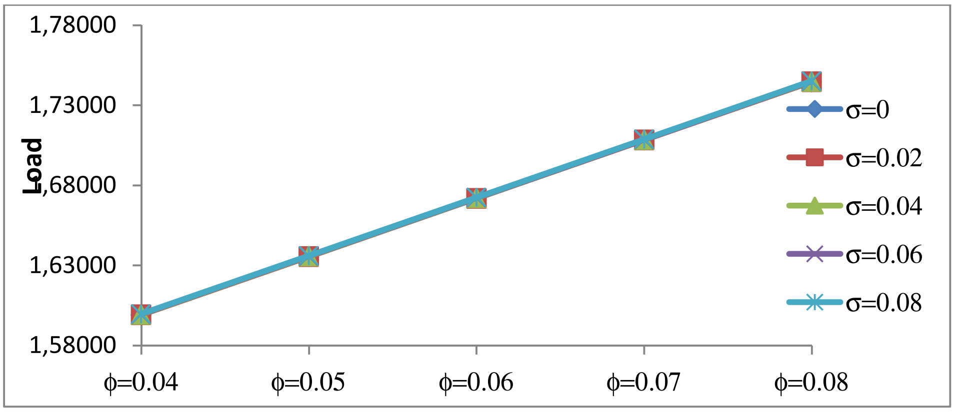

Fig. 2. Variation of Load carrying capacity with respect to ϕ and σ

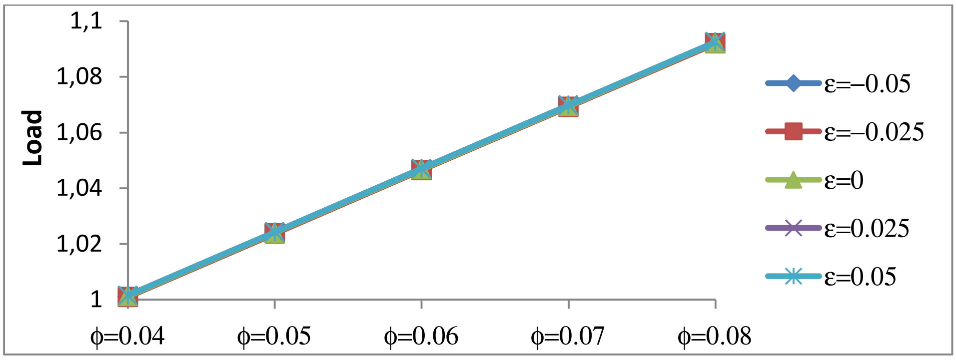

Figure- 3 Variation of Load carrying capacity with respect to ϕ and ε

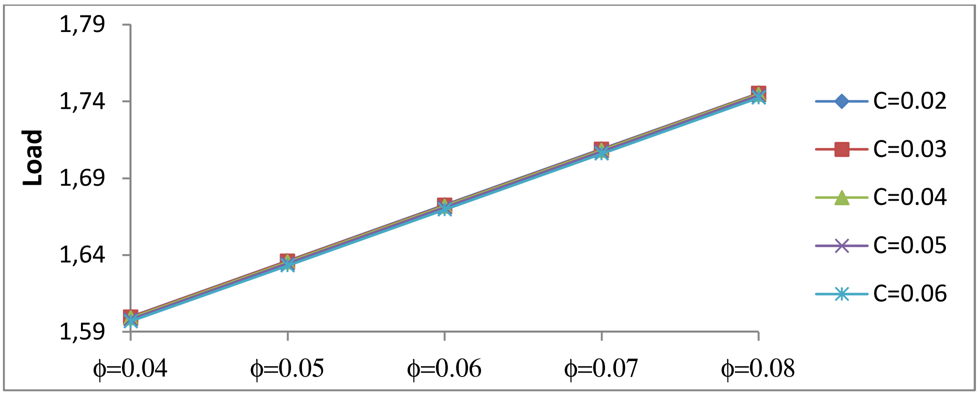

Figure- 4 Variation of Load carrying capacity with respect to ϕ andC

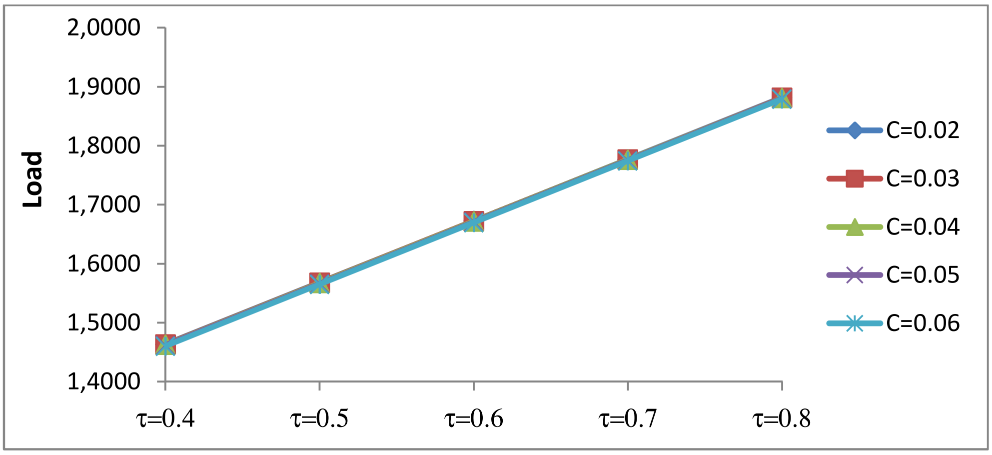

Figure- 5 Variation of Load carrying capacity with respect to τ andC

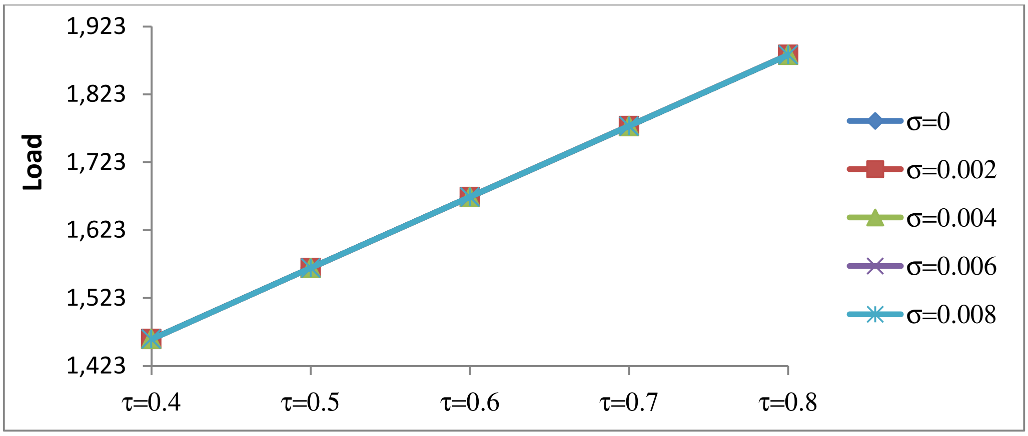

Figure- 6 Variation of Load carrying capacity with respect to τ andσ

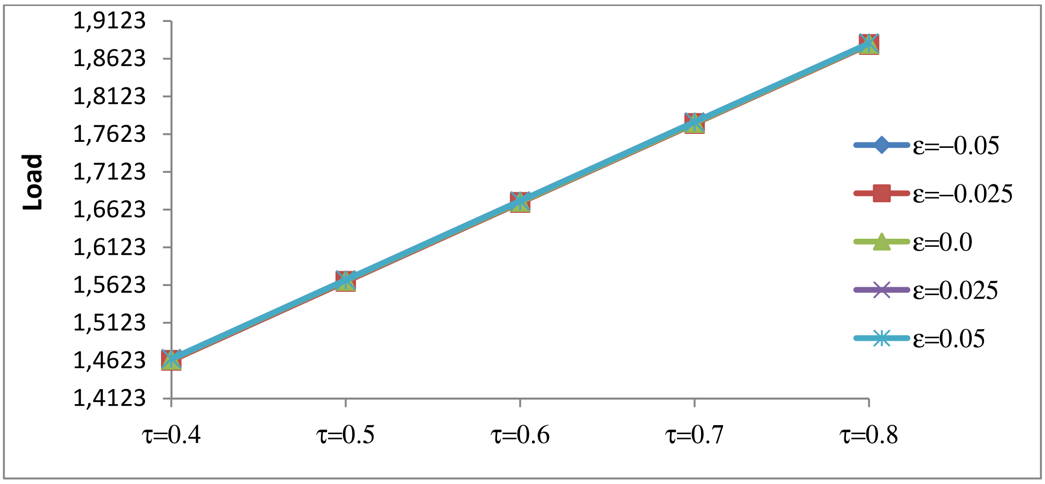

Figure- 7 Variation of Load carrying capacity with respect to τ andε

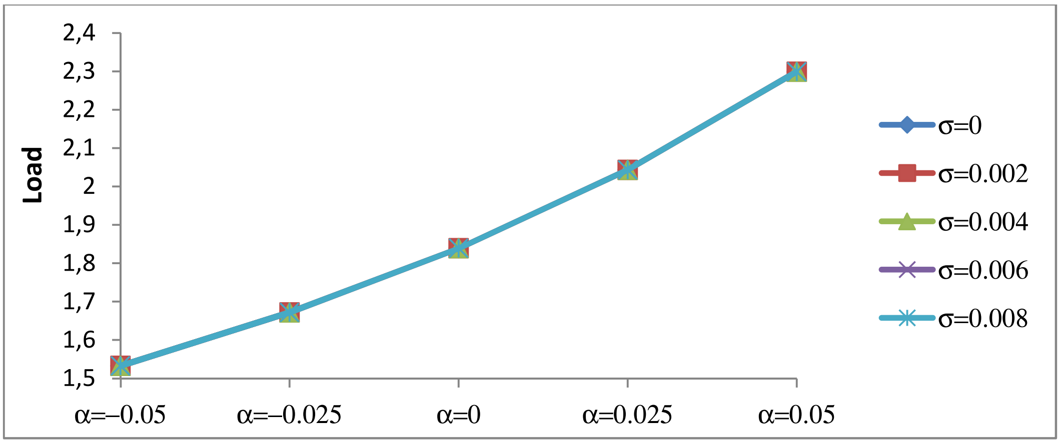

Figure- 8 Variation of Load carrying capacity with respect to α andσ

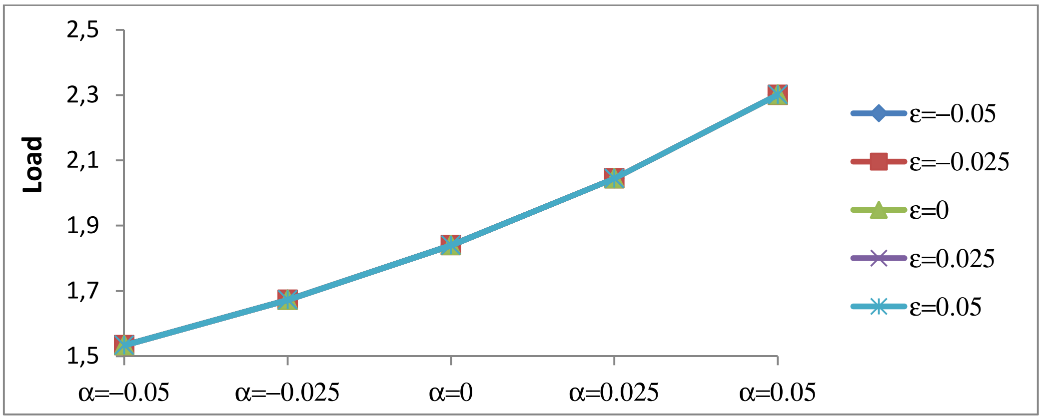

Figure- 9 Variation of Load carrying capacity with respect to α andε

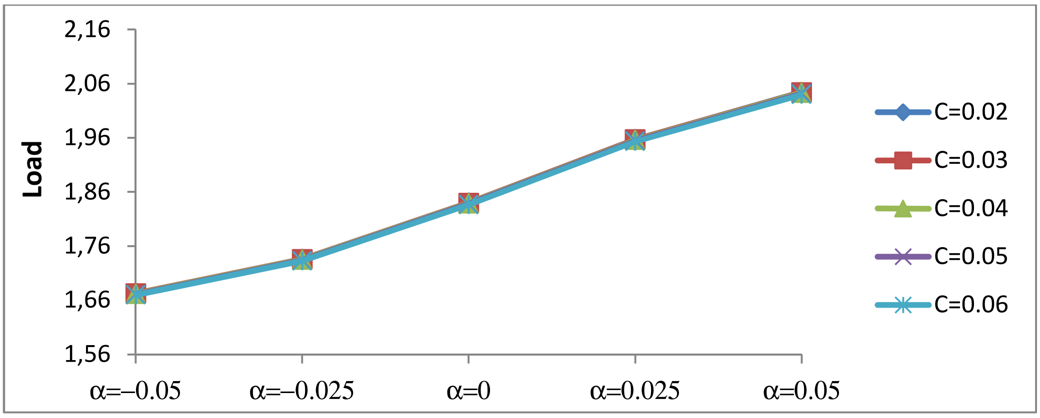

Figure- 10 Variation of Load carrying capacity with respect to α andC

It can be clearly observed that the combined effect of standard deviation, skewness and couple stress on the distribution of load carrying capacity with respect to volume concentration parameter, magnetization parameter and variance is almost negligible.

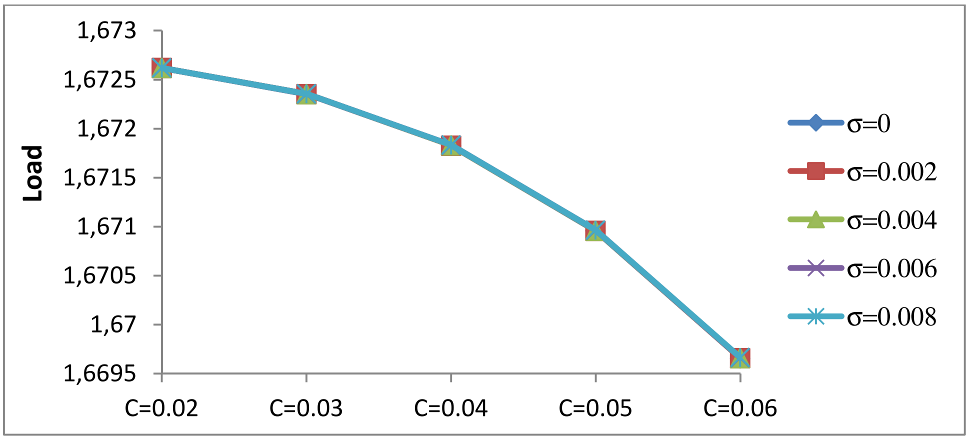

Figure- 11 Variation of Load carrying capacity with respect to C andσ.

Further, the effect of the standard deviation on the variation of the load carrying capacity with respect to the couple stress parameter is negligible.

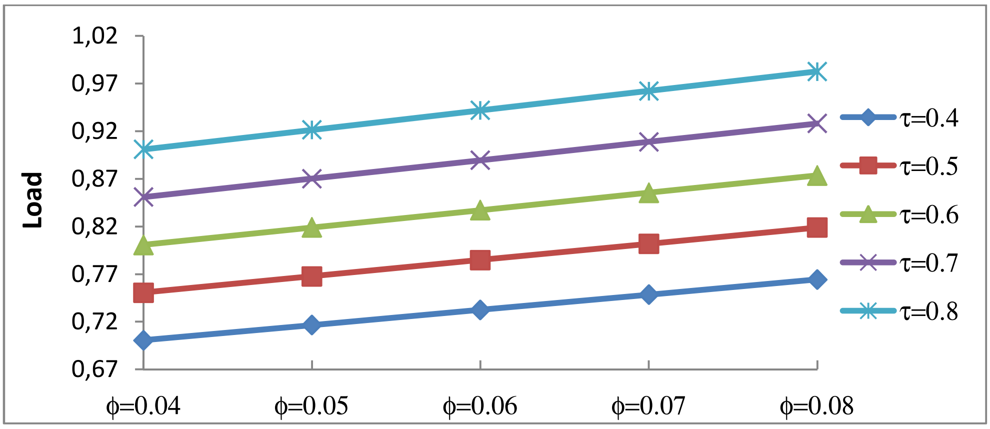

Figure- 12 Variation of Load carrying capacity with respect to ϕ andτ

Figure- 12 shows that the magnetization parameter and the volume concentration parameter increase the load carrying capacity.

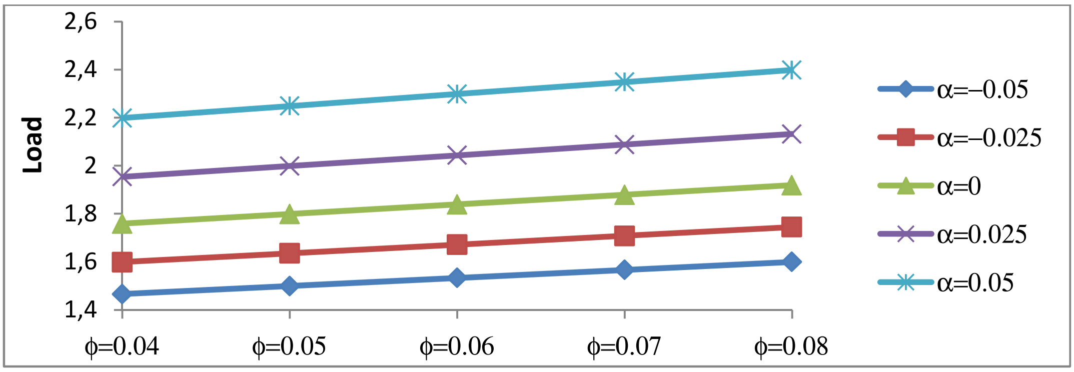

Figure- 13 Variation of Load carrying capacity with respect to ϕ and α

Figure- 13 indicates that positive variance increases the load carrying capacity while the reverse is true for variance (-ve). This does not happen in the case of transverse roughness. (Patel et al.(2014)).

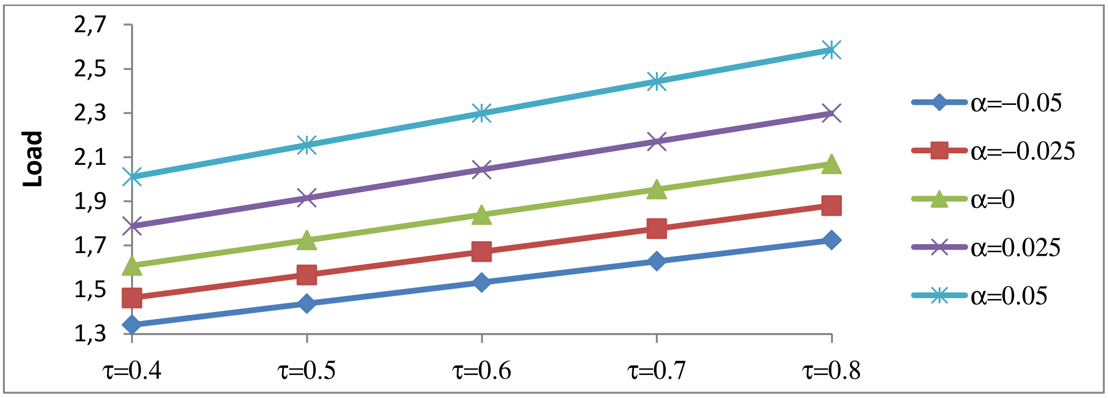

Figure- 14 Variation of Load carrying capacity with respect to τ and α

Figure- 14 makes it clear that the increased load carrying capacity due to magnetization. further, enhances because of variance (+ve).

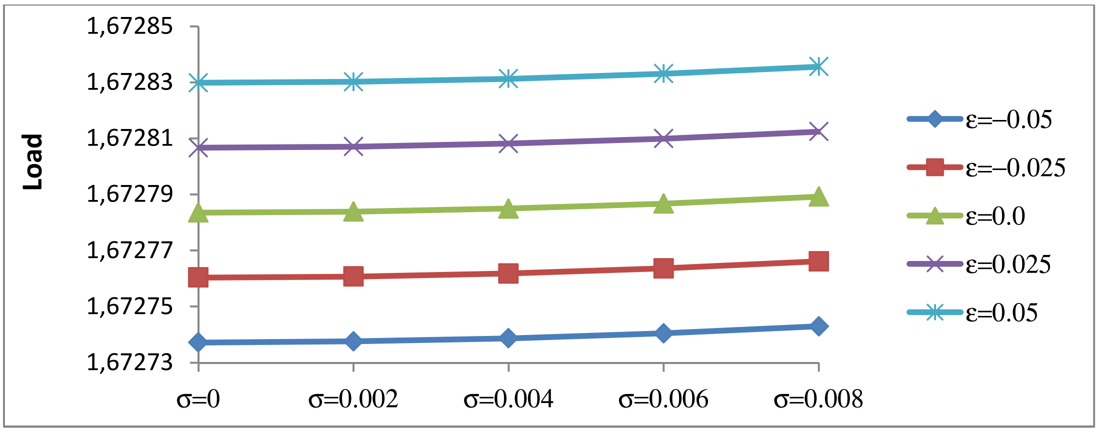

Figure- 15 Variation of Load carrying capacity with respect to σ and ε

The fact that the skewness follows the path of variance so far as the load carrying capacity is concerned is reflected in Figure- 15. Therefore, the combined effect of positively skewed roughness and variance (+ve) remains crucial from bearing design point of view. This is exactly opposite of what happens in the case of transverse roughness. (Patel et al.(2014)).

A close glance at the figures reveals that the combined effect of ferrofluid lubrication and couple stress may go a long way in reducing the adverse effect of roughness, at least in the case of positively skewed roughness when variance positive occurs. For an overall improved performance this study confirms that the use of Shliomis model may turn out to be more fruitful for this type of bearing systems, even if couple stress effect is in force.

4 Conclusion

Although, there are a number of factors improving the load carrying capacity, this study establishes that the roughness aspects must be considered carefully while designing this type of bearing system.

If suitably chosen then this type of bearing system may turn out to be favorable for industrial applications.

5 Nomenclature

C: couple stress parameter, $\mathrm{C} = {}^{l_c}/_{h_0}$.
h: film thickness
$\mathrm{h_0}$: initial film thickness
$\mathrm{H_0}$: transverse magnetic field
$\overline{\mathrm{H}}$: applied magnetic field vector
$\mathrm{k_B}$: Boltzmann constant
m: magnetic moment of a particle
$\mathrm{M_E}$: equilibrium magnetization, $\mathrm{M_E} = \mathrm{nm}\left(\coth\xi - {}^{1}/_{\xi}\right)$
$\vec{\mathrm{M}}$: magnetization vector
n: number of particles per unit volume
P: film pressure
P^*: dimension less film pressure
r, z: radial and vertical coordinates
u, w: velocity components in the r – and z – directions
$\mathrm{v_s}$: squeezing velocity, $\mathrm{v_s} = -{}^{\mathrm{dh}}/_{\mathrm{dt}}$
$\vec{\mathrm{V}}$: fluid velocity vector
W: load capacity
W^*: non-dimensional load capacity
η: viscosity of the suspension
η_0: viscosity of the main fluid
η_c: new material constant responsible for couple stress fluid property
μ_0: permeability of free space
ϕ: volume concentration of particles
ξ: Langevin parameter, $\xi = \mu_0 \mathrm{m} \mathrm{H_0}/\mathrm{k_B T}$
τ: rotational viscosity parameter, $\tau = {}^{(3\phi/2)(\xi - \tanh\xi)}/_{(\xi + \tanh\xi)}$

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