BIOINFO Mechanical Engineering
ISSN: 2277-3738 & E-ISSN: 2277-3746, Volume 4, Issue 1, 2016, pp.-29-34.
Available online at http://bioinfopublication.org/jouarchive.php?opt=&jouid=BPJ0000202

Research Article

JENKINS MODEL BASED FERRO-FLUID LUBRICATION OF A ROUGH, POROUS CONVEX PAD SLIDER BEARING WITH SLIP VELOCITY

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Received: June 21, 2016; Revised: July 19, 2016; Accepted: July 20, 2016; Published: July 28, 2016

Abstract- This investigation deals with the Ferro-fluid lubrication of a rough, porous convex pad slider bearing considering slip velocity. The magnetic fluid flow is governed by Jenkins model. The roughness has been characterized by a random variable with non zero mean, variance and skewness. The associated stochastically averaged Reynolds type equation is solved to obtain the pressure distribution, which gives the load carrying capacity. Further, the friction is evaluated. The graphical results suggest that the magnetization may not go a long way for reducing the adverse effect of roughness, even if the slip parameter is minimum, However the situations improves when negatively skewed roughness occurs. Besides, the magnetization fails to have any impact on friction.

Keywords- Convex pad slider bearing, Porosity, Roughness, Magnetic fluid, Slip velocity.

Citation: Patel Paresh A., et al., (2016) Jenkins Model Based Ferro-Fluid Lubrication of a Rough, Porous Convex Pad Slider Bearing With Slip Velocity. BIOINFO Mechanical Engineering, ISSN: 2277-3738 & E-ISSN: 2277-3746, Volume 4, Issue 1, pp.-29-34.

Academic Editor / Reviewer: Dr Jimit R Patel, Dr Nitin D Patel, Prof Himanshu C Patel

Introduction

The slider bearing is the simplest and most often encountered bearing. These bearings are frequently associated with reciprocating motion. In cross section, they may be flat, convex, concave. The use of squeeze film slider bearings in clutch plates, automobile, transmission and domestic appliances are well known. Jenkins [1] used a simple continuum model for a paramagnetic fluid to analyze a simple shearing flow and parallel flow through a pipe and examined the possibility of maintaining a steady circular flow in a circular cylinder by rotating a magnetic field, Verma and Singh [2] analyzed the interaction between magnetic and mechanical forces in the case of flow of an incompressible paramagnetic fluid through a porous annulus subjected to the external magnetic field. Shah and Bhat [3] investigated the behavior of a porous exponential slider bearing with a Ferro-fluid lubricant whose flow was governed by Jenkins model taking velocity slip into account. It was observed that the load carrying capacity as well as the friction decreased when the slip parameter increased. However, increase in the material parameter caused decreased load carrying capacity and increased friction. The position of centre of pressure was not affected significantly by the slip parameter but the position of centre of pressure shifted towards the bearing inlet for large values of material parameter. Performance of a slider bearing with its stator having circular convex pad surface was subjected to studied by Shah and Bhat [4] under the presence of Ferro-fluid lubrication when Jenkins model described the flow. It was found that the load carrying capacity increased with the increasing values of the film thickness ratio and decreasing value of the material parameter. The friction force on the slider decreased with increasing film thickness ratio while the position of centre of pressure shifted towards the outlet. Chaves et al. [5] derived direct measurements of the bulk flow of a ferro-fluid in a uniform rotating magnetic field using the ultrasonic velocity profile method. Zakaria et al. [6] proved the static and dynamic performance characteristics of finite journal bearing lubricated with a non Newtonian Ferro-fluid in the presence of an external magnetic field using the spectral method technique. Here it was shown that the magnetic parameter played an important role in administrating the stability behavior.

Slider bearings mainly, are designed for supporting the transverse load in engineering systems. The performance characteristic of a bearing system has been analyzed taking various film shapes into consideration [7-10]. Lin [11] discussed the couple stress effect on the steady state performance of a wide parabolic shaped slider bearing and found that the couple stress effect resulted in an improvement in the steady state performance.

Fluids with strong magnetic properties have drawn considerable attentions in recent years. For lubricating the bearing system in technical applications in the domain of nano scale science and technology, significant progress has been made. Therefore, the use of magnetic fluid lubrication adds an additional importance from nano science point of view. Agrawal [8] studied the performance of a Ferro-fluid lubricated plane inclined slider bearing and analyzed that the performance was relatively better than the corresponding bearing system with a conventional lubricant. Bhat and Patel [12] analyzed the exponential slider bearing with a Ferro-fluid lubricant suggested that the magnetic fluid lubricant caused increased load carrying capacity while the friction remained almost unchanged. The discussion of Bhat and Deheri [13] revealed that the magnetic fluid sharply increased the load carrying capacity for a squeeze film performance between porous annular disks. The contribution of Bhat and Deheri [14] confirmed the positive impact of magnetic fluid lubrication on the steady state performance of a porous composite slider bearing.

It is well known that the roughness of the bearing surfaces retards the motion of the lubricant thereby affecting adversely the bearing system. Tzeng and Saibel [15] observed the random character of the roughness and adopted a stochastic approach to study the effect of surface roughness. This modeling of Tzeng and

Saibel [15] was modified by Christensen and Tonder [16-18] to deal with the effect of surface roughness in general, on the performance of the bearing system. This method was adopted by many investigators [19-23]. Patel et. al. [24] studied that the performance of an idealized rough porous hydrodynamic plane slider bearing, It was proved that the transverse surface roughness induced an adverse effect on the performance of bearing due to the negative effect of porosity. However, the situation was relatively better in the case of negatively skewed roughness The magnetic fluid lubrication of a transversely rough slider bearing was analyzed by Deheri et al. [25] by taking various film shapes in to consideration. The effect of transverse surface roughness was adverse in general while magnetic fluid lubricant increased the load carrying capacity. This squeeze film performance in the case of longitudinal surface roughness was observed to be relatively better than the corresponding one with transverse surface roughness as established by Deheri et al. [26].The hydrodynamic lubrication of rough slider bearings was discussed by Nanduvinamani et al. [27] taking couple stress effect in to account. Patel and Deheri [28] studied the Ferro-fluid lubrication of a rough, porous inclined slider bearing with slip velocity The flow of the Ferro-fluid was based on Jenkins model. Sukla and Deheri [29] discussed regarding the performance of a transversely rough porous circular convex pad slider bearing in the presence of a magnetic fluid lubricant using Jenkins model. Deheri et al. [30] obtained the performance characteristics of a Shliomis model based Ferro-fluid lubrication of a rough porous convex pad slider bearing and proved that the adverse effect of surface roughness could be reduced to certain extent by the positive effect of Shliomis model based Ferro-fluid lubrication. It was shown that the magnetization could reduce the adverse effect of surface roughness up to some extent when suitable values of slip parameter were in place.

Here, it has been sought to study the hydrodynamic Ferro-fluid lubrication of a rough, porous, convex pad slider bearing with slip velocity.

Analysis

The geometry and configuration of the Ferro-fluid lubricant based a rough, porous convex pad slider bearing is presented below in [Fig-1].

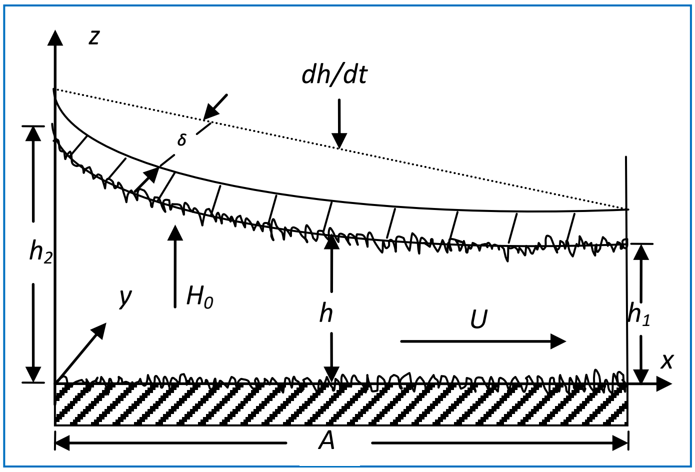

Fig-1 Configuration of bearing system

Following the stochastic modelling of surface roughness by Christensen and Tonder [16-18] the thickness $h(x)$ is considered as:

$$h(x) = \bar{h}(x) + h_s \qquad [1]$$

$$f(h_s) = \begin{cases} \dfrac{15}{16c^7}\left(c^2 - h_s^2\right)^3, & -c \le h_s \le c \\ 0 & ,Otherwise \end{cases} \qquad [2]$$

where c is the maximum deviation from the mean film thickness. The mean $\bar{\alpha}$, the standard deviation $\bar{\sigma}$ and the parameter $\bar{\varepsilon}$ which is the, measure of symmetry of the random variable h_s are defined by the relationships:

$$\alpha = E(h_s),\ \sigma^2 = E\left\{(h_s - \alpha)^2\right\} \text{ and } \varepsilon = E\left\{(h_s - \alpha)^3\right\}$$

where, E denotes the expected value defined by

$$E(R) = \int_{-c}^{c} R f(h_s)\, dh_s \qquad [3]$$

Following Bhat (2003),
for a convex pad slider bearing, The film thickness is taken as

$$\bar{h} = \frac{h}{h_0} = 4\bar{\delta}X^2 - \left(a - 1 + 4\bar{\delta}\right)X + a,$$

$$a = \frac{h_1}{h_0},\ X = \frac{x}{A}$$

Thus, incorporating the effects of porosity and slip, the associated Reynolds type equation turns out to be

$$\frac{d}{dx}\left[\frac{g(h)}{1 - \dfrac{\rho\alpha'^2\bar{\mu}H}{2\eta}}\frac{d}{dx}\left(p - \frac{\mu_0\bar{\mu}}{2}H^2\right)\right] = 6\eta U\frac{dh}{dx} + 12\eta\dot{h}_0 \qquad [4]$$

Where

$$g(h) = \left\{h^3 + 3\alpha h^2 + 3\left(\alpha^2 + \sigma^2\right)h + 3\sigma^2\alpha + \alpha^3 + \varepsilon + 12\phi h\right\}\frac{(4 + sh)}{(2 + sh)}$$

[5]

Taking $H^2 = KA^2\sin(\pi X)$ and introducing the dimensionless quantities

$$X = \frac{x}{A},\ \bar{h} = \frac{h}{h_0},\ \mu^* = \frac{K\mu_0\bar{\mu}h_0^2 A}{\eta U},\ \bar{\alpha}'^2 = \frac{\rho\alpha'^2\bar{\mu}A\sqrt{K}}{2\eta},$$

$$P = \frac{h_0^3 p}{\eta A^2\dot{h}_0},\ \beta_1 = \frac{h_0^3}{2\dot{h}_0 A},\ \bar{s} = sh_0,\ \bar{\psi} = \frac{\phi h}{h_0^3}$$

$$W = \frac{h_0^3 w}{\eta A^4\dot{h}_0},\ F = \frac{-h_0 f}{\eta L^2\dot{h}_0},\ Y = \frac{\bar{X}}{A},\ \bar{\alpha} = \frac{\alpha}{h_0}, \bar{\sigma} = \frac{\sigma}{h_0}, \bar{\varepsilon} = \frac{\varepsilon}{h_0^3},$$

[Eq-4] yields on integration,

$$\frac{d}{dx}\left[P - \frac{1}{2}\mu^* X(1 - X)\right] = \frac{6}{g(\bar{h})}\left(A_1\frac{(2 + \bar{s}\bar{h})}{(1 + \bar{s}\bar{h})}\bar{h} - \beta^{-1}X + A_2 Q\right)\left(1 - \bar{\alpha}'^2\sqrt{X(1 - X)}\right) \qquad [6]$$

where

$$A_1 = 3\left(\bar{\alpha}^2 + \bar{\sigma}^2\right),\ A_2 = \left(3\bar{\sigma}^2\bar{\alpha} + \bar{\alpha}^3 + \bar{\varepsilon} + 12\bar{\psi}\right)$$

and

$$g(\bar{h}) = \left\{\bar{h}^3 + 3\bar{\alpha}\bar{h}^2 + A_1 + A_2\right\}\frac{\left(4 + \bar{s}\bar{h}\right)}{\left(2 + \bar{s}\bar{h}\right)} \qquad [7]$$

Solving [Eq-6] under the boundary conditions

P (0) =P (1) =0

yields

$$P=\frac{1}{2}\mu^{*}\sin(\pi X)+\left\{6\int_{0}^{1}\frac{1}{g(\bar h)}\left(A_1\frac{(2+\bar s\bar h)}{(1+\bar s\bar h)}\bar h-\beta^{-1}X+A_2Q\right)\right\}\left(1-\bar\alpha'^{2}\sqrt{X(1-X)}\right)dX \qquad [8]$$

where

$$Q=-\int_{0}^{1}\frac{\frac{1}{g(\bar h)}\left(A_1\frac{(2+\bar s\bar h)}{(1+\bar s\bar h)}\bar h-\beta^{-1}X\right)\left(1-\bar\alpha'^{2}\sqrt{X(1-X)}\right)dX}{\frac{1}{g(\bar h)}\left(1-\bar\alpha'^{2}\sqrt{X(1-X)}\right)dX} \qquad [9]$$

The load capacity W, frictional force F and Position of centre of pressure Y are respectively expressed in dimensionless form as

$$W=\frac{2\mu^{*}}{\pi}-\left\{6\int_{0}^{1}\frac{X}{g(\bar h)}\left(A_1\frac{(2+\bar s\bar h)}{(1+\bar s\bar h)}\bar h-\beta^{-1}X+A_2Q\right)\left(1-\bar\alpha'^{2}\sqrt{X(1-X)}\right)\right\}dx \qquad [10]$$

$$F=-\int_{0}^{1}\left[\frac{1}{A_1\frac{(2+\bar s\bar h)}{(1+\bar s\bar h)}\bar h}+\frac{1}{\alpha\bar h^{2}}\left(A_1\frac{(2+\bar s\bar h)}{(1+\bar s\bar h)}\bar h-\beta^{-1}X+A_2Q\right)\right]dX \qquad [11]$$

and

$$Y=\frac{1}{W}\left[\frac{\mu^{*}}{\pi}-\left\{3\int_{0}^{1}\frac{X^{2}}{g(\bar h)}\left(A_1\frac{(2+\bar s\bar h)}{(1+\bar s\bar h)}\bar h-\beta^{-1}X+A_2Q\right)\left(1-\bar\alpha'^{2}\sqrt{X(1-X)}\right)dX\right\}\right] \qquad [12]$$

Result and Discussion:

It is easily seen that the pressure distribution is determined from the [Eq-8] while the [Eq-10] gives the load caring capacity. Further. From [Eq-8] and [Eq-10] it is observed that the dimensionless pressure increases by

$$\frac{1}{2}\mu^{*}\sin(\pi X)$$

While the non-dimensional load carrying capacity gets enhanced by

$$\frac{2\mu^{*}}{\pi}$$

as compared to the case of conventional lubricant based bearing system. For a smooth bearing system. This investigation reduces to the discussion of Patel and Deheri [28].

It is well known fact that magnetization accelerated the viscosity of the lubricant accordingly pressure gets increased and hence the load increase. These can be seen from [Fig-2 to 3].

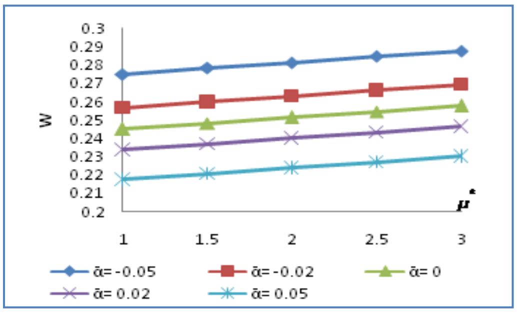

Fig-2 Variation of load carrying capacity with respect to μ^{*} and $\bar\alpha$

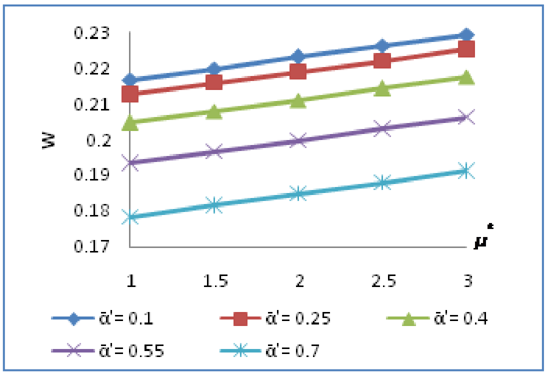

Fig-3 Variation of load carrying capacity with respect to μ^{*} and $\bar\alpha'$

The fact that the slip effect induces adverse effect on the performance of the bearing system by reducing the load carrying capacity. These can be a observed from the [Fig-4-7].

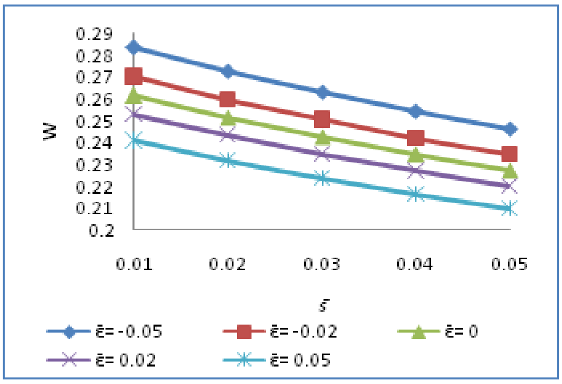

Fig-4 Variation of load carrying capacity with respect to $\bar s$ and $\bar\varepsilon$

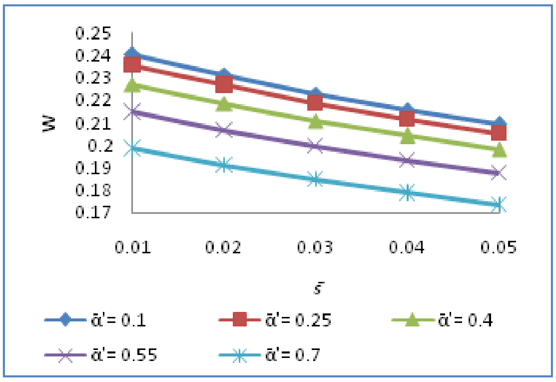

Fig-5 Variation of load carrying capacity with respect to $\bar s$ and $\bar\alpha'$

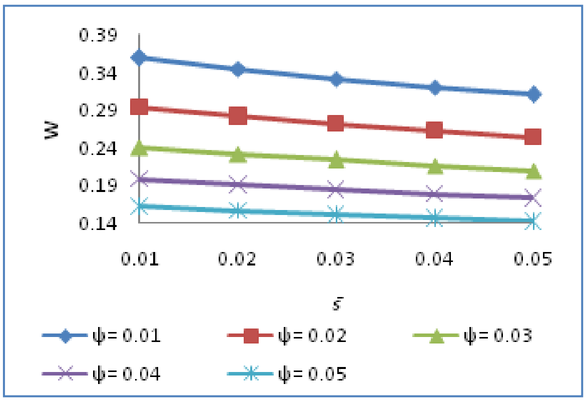

Fig-6 Variation of load carrying capacity with respect to $\bar s$ and $\bar\psi$

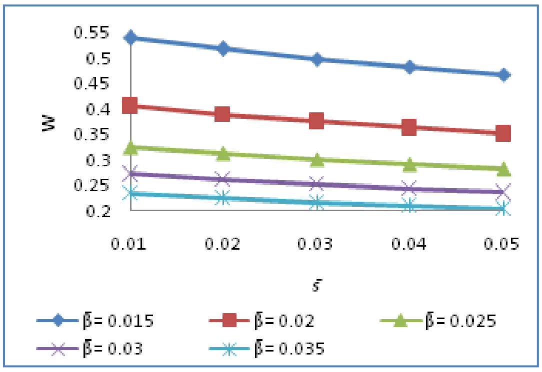

Fig-7 Variation of load carrying capacity with respect to $\bar{s}$ and $\bar{\beta}$

[Fig-8-10] describe the effect of variance on the performance characteristic it is seen that the load carrying capacity is a decrease, with increasing variance (+ve), while variances (-ve) causes increases load carrying capacity.

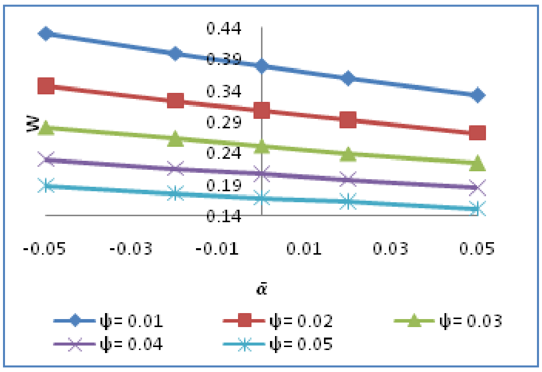

Fig-8 Variation of load carrying capacity with respect to $\bar{\alpha}$ and ψ

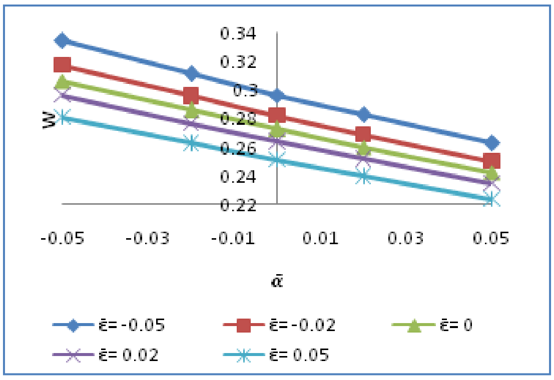

Fig-9 Variation of load carrying capacity with respect to $\bar{\alpha}$ and $\bar{\varepsilon}$

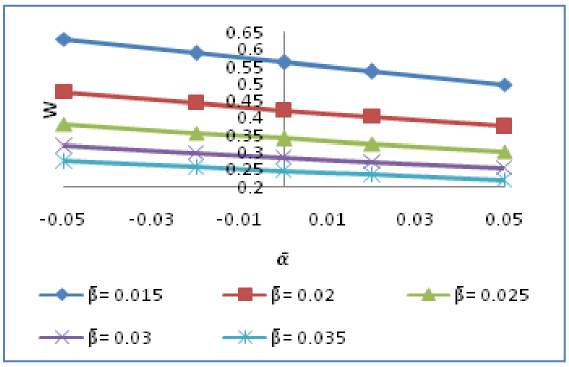

Fig-10 Variation of load carrying capacity with respect to $\bar{\alpha}$ and $\bar{\beta}$

Considerable load reduction occurs, when the standard deviation increase, which can be observed from the [Fig-11-12].

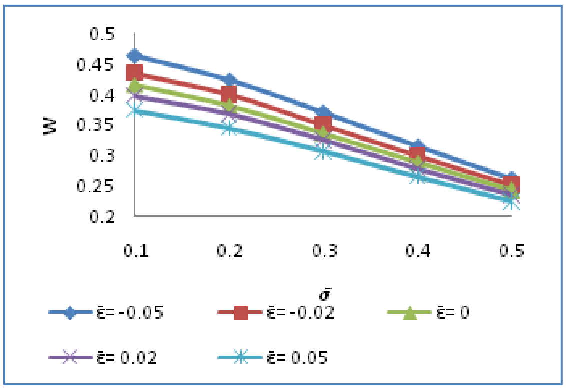

Fig-11 Variation of load carrying capacity with respect to $\bar{\sigma}$ and $\bar{\varepsilon}$

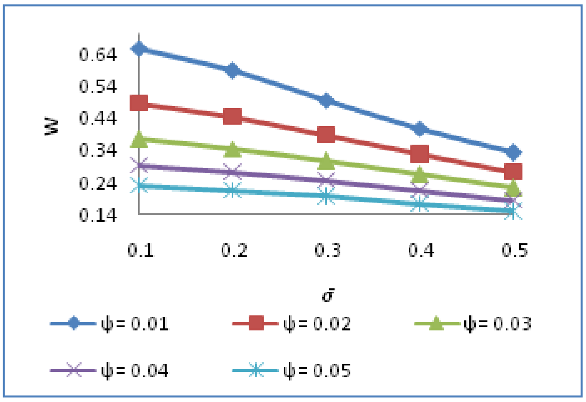

Fig-12 Variation of load carrying capacity with respect to $\bar{\sigma}$ and ψ

It is clearly observed from [Fig-13-14]. that the skewness follows the path of variance so far as the trends of load carrying capacity is concerned

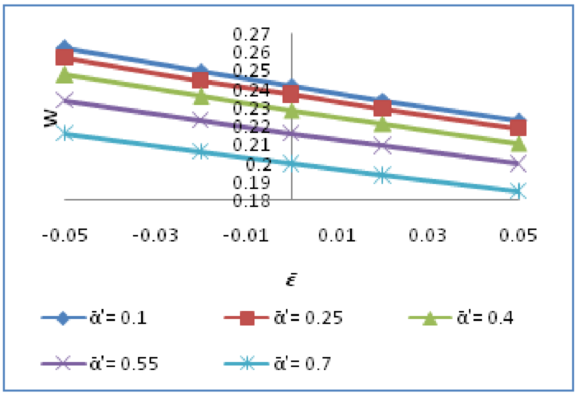

Fig-13 Variation of load carrying capacity with respect to $\bar{\varepsilon}$ and $\bar{\alpha}'$

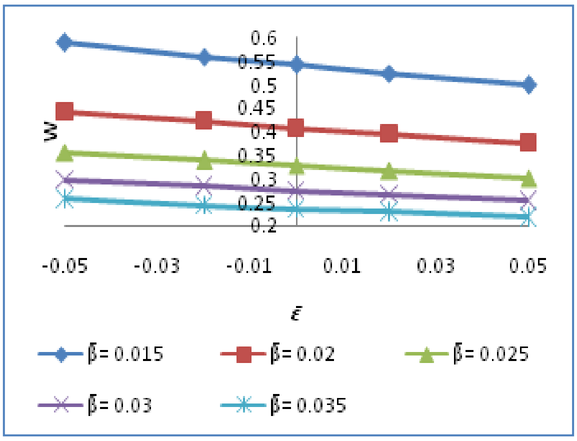

Fig-14 Variation of load carrying capacity with respect to $\bar{\varepsilon}$ and $\bar{\beta}$

The material constant parameter decreases the load carrying capacity and this effect gets compounded when porosity is increased. Porosity mostly induces a reduction in load. It can be seen in [Fig-15].

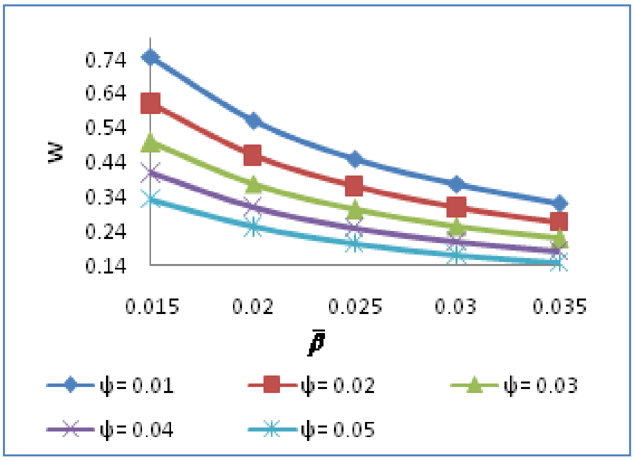

Fig-15 Variation of load carrying capacity with respect to $\bar{\beta}$ and $\bar{\psi}$

It is appealing to not that the magnetization fails to induce a change in the friction, this can be observed from the [Eq-11]

The profit a friction presented in [Fig-16- 24] makes it clear that either the friction is reduced or marginally increased some of the graphs presented here suggest that the combined positive effect of negatively skewed roughness and variance(-ve) may be channelized to improve the bearing performance. Closed scrutiny of the result found here indicates that as compare to Neuringer–Rosensweig model, the load carrying capacity is found to be more for Jenkins model.

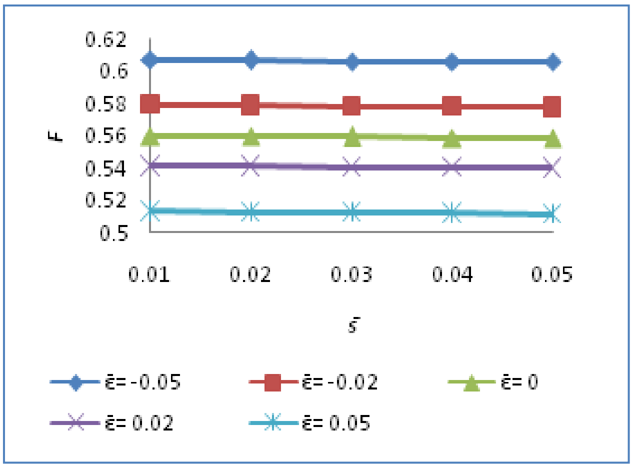

Fig-16 Variation of friction with respect to $\bar{s}$ and $\bar{\varepsilon}$

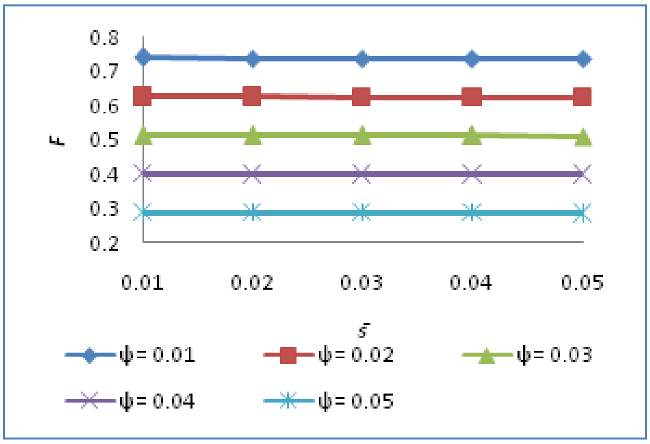

Fig-17 Variation of friction with respect to $\bar{s}$ and $\bar{\psi}$

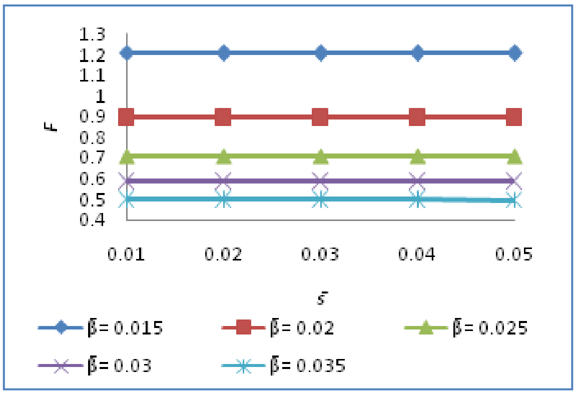

Fig-18 Variation of friction with respect to $\bar{s}$ and $\bar{\beta}$

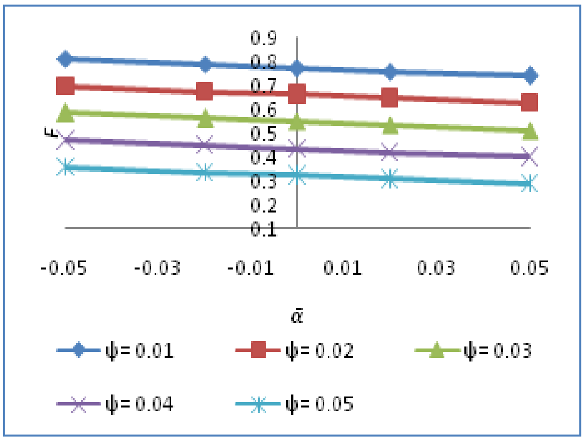

Fig-19 Variation of friction with respect to $\bar{\alpha}$ and $\bar{\psi}$

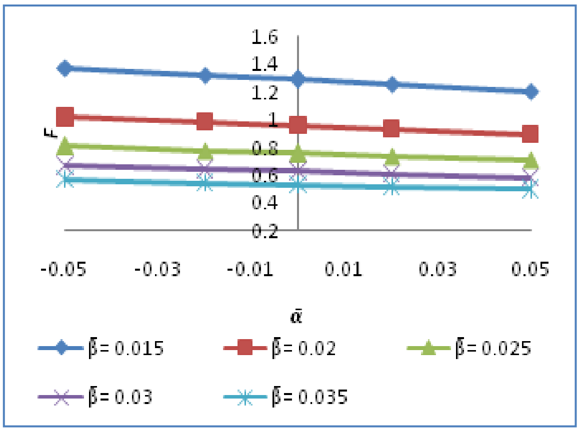

Fig-20 Variation of friction with respect to $\bar{\alpha}$ and $\bar{\beta}$

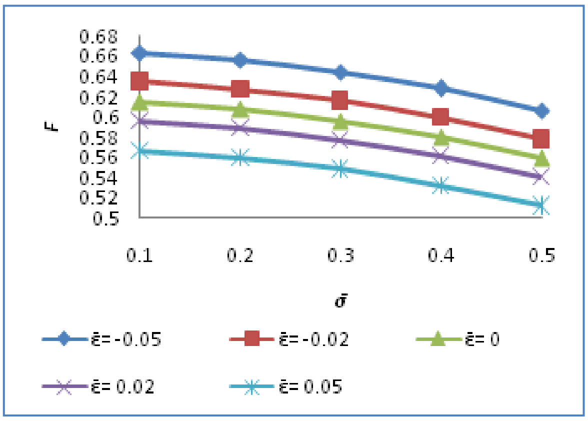

Fig- 21 Variation of friction with respect to $\bar{\sigma}$ and $\bar{\varepsilon}$

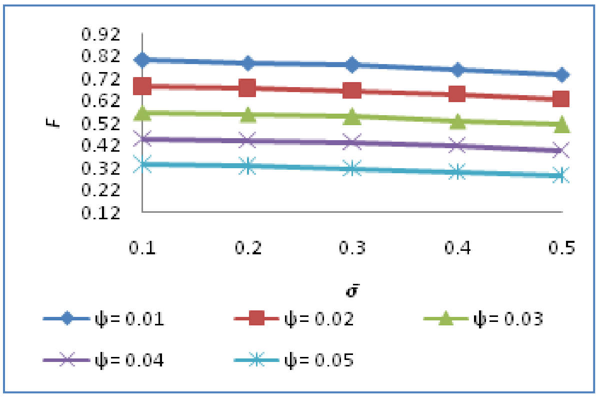

Fig-22 Variation of friction with respect to $\bar{\sigma}$ and ψ

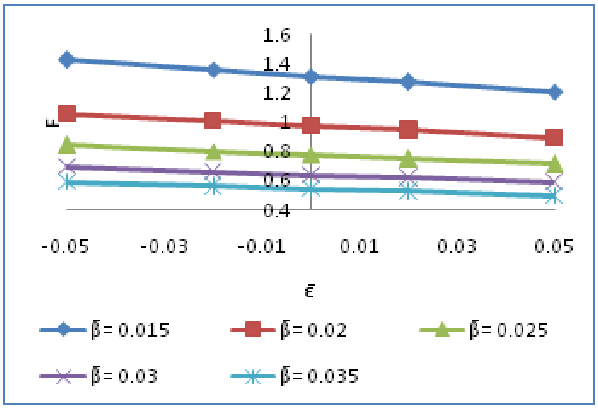

Fig-23 Variation of friction with respect to $\bar{\varepsilon}$ and $\bar{\beta}$

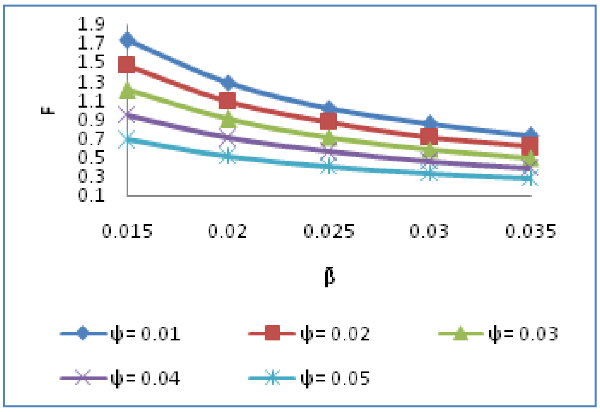

Fig-24 Variation of friction with respect to $\bar{\beta}$ and ψ

Conclusion

The magnetization has a limited option for neutralizing the adverse effect of roughness, even if the slip parameter is minimum. However, the situation remains better in case of negatively skewed roughness. Undoubtedly, our investigations established that Jenkins model is more suitable as compared to Neuringer Rosensweig model for improving the bearing performance. Hence, while designing the bearing system one needs to accord priority to roughness aspects.

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