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[1.]Jimit R. PATEL, [2.]G. M. DEHERI

ON THE PERFORMANCE OF A JENKINS MODEL BASED FERROFLUID SQUEEZE FILM IN CURVED ROUGH ANNULAR PLATES CONSIDERING THE SLIP EFFECT

[1-2.]Department of Mathematics, Sardar Patel University, Vallabh Vidyanagar, Anand, Gujarat, INDIA

Abstract: An endeavour has been made to study the effect of slip velocity on the Jenkins model based ferrofluid lubrication of a squeeze film in curved rough annular plates when the upper surface is described by a hyperbolic function while the lower surface is determined by an expression involving secant function. The roughness effect is analyzed by adopting the stochastic model of Christensen and Tonder while Beavers and Joseph's slip model is deployed to evaluate the influence of slip velocity. The pressure distribution is obtained after solving the associated stochastically averaged Reynolds type equation. Then the load carrying capacity is calculated. The results presented in graphical forms confirm that while the effect of transverse roughness is in general adverse, the magnetization results in sharply increased load carrying capacity. This investigation indicates that the effect of transverse roughness can be minimized to a large extent by the Jenkins model based ferrofluid lubrication. However, for any type of improvement in the performance characteristics the slip parameter is required to be reduced. Lastly, this paper also underlines the crucial role of the aspect ratio especially, when higher negative values of skewness and variance are involved, even if, the curvature parameters are chosen suitably.
Keywords: Annular plates, Roughness, Slip velocity, Jenkins model, Magnetic fluid

INTRODUCTION

Nowadays, many fascinating materials have been attracting the investigators and scientists due to their physical properties and technological usage. One of smart materials is magnetic fluid which is not available free state in nature, but are synthesized. One of the important properties of the magnetic fluid is that they can be retained at a desired location by an external magnetic field. Due to this main property, Ferrofluids have variety of applications in the field of sciences and engineering. Owing to the wide application and property of the magnetic fluids, many researchers have used ferrofluid as a lubricant in different physical geometry of bearing systems. Sinha et al. (1993) discussed the effect of ferrofluid lubrication on cylindrical rollers. Ram and Verma (1999) dealt with the performance of porous inclined slider bearing using ferrofluid lubrication. Osman et al. (2001) investigated the static and dynamic characteristics of magnetized journal bearings lubricated with ferrofluid. Shah and Bhat (2005) worked on the effect of ferrofluid lubrication on a squeeze film between curved annular plates considering rotation of magnetic particles. Deheri et al. (2006) examined the performance of circular step bearings under the presence of a magnetic fluid. Ahmed and Singh (2007) discussed the effect of porous-pivoted slider bearing with slip velocity using ferrofluid. Urreta et al. (2009) studied the effect of hydrodynamic bearing lubricated with magnetic fluids. Patel et al. (2010) investigated the performance of a short hydrodynamic slider bearing in the presence of magnetic fluids. Patel et al. (2012) analyzed the effect of hydrodynamic short journal bearings lubricated with magnetic fluids. All the above investigations have established that the performance of the bearing system gets enhanced due to magnetization.

In the above studies, most of the investigations dealt with no-slip boundary conditions. Beavers and Joseph (1967) obtained the interface between a porous medium and fluid layer in an experimental study and introduced a slip boundary condition at the interface. Salant and Fortier (2004) discussed the numerical analysis of a slider bearing with a heterogeneous slip/no-slip surface. Wu et al. (2006) analyzed the effect of low friction and high load support capacity of slider bearing with a mixed slip surface. Ahmed and Singh (2007) dealt with the performance of magnetic fluid lubrication of porous-pivoted slider bearing with slip velocity.

Wang et al. (2012) numerically analyzed the performance of the radial sleeve bearing with combined surface slip. In all of the above mentioned study, it was obtained that effect of slip remained vital for modifying the bearing performance.

The smoothness of the bearing surfaces were assumed in all the above investigations. But it is almost not possible because, the bearing surfaces could be rough through the manufacturing process and the impulsive damages. Sometimes, even the contamination of the lubricant causes roughness. Many methods have been proposed to find the effect of surface roughness on the performance characteristics of squeeze film bearings. Christensen and Tonder (1969a, 1969b, 1970) modified the stochastic theory of Tzeng and Saibel (1967) to study the effect of surface roughness in general. Quite a good number of research papers (Nanduvinamani et al. (2003), Chiang et al. (2004), Bujurke et al. (2007), Patel et al. (2009), Shimpi and Deheri (2010), Patel and Deheri (2011), Abhangi and Deheri (2012)) adopted the model of Christensen and Tonder (1969a, 1969b, 1970) to study the effect of roughness in different types of bearing systems.

Patel and Deheri (2013) analyzed the performance of a ferro fluid based squeeze film in rotating rough curved circular plates resorting to Shliomis model. It was found 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 parameters. Patel and Deheri (2014) investigated the effect of different porous structures on the performance of a Shliomis model-based magnetic squeeze film in rotating rough porous curved circular plates. It was concluded that the adverse effect of transverse roughness could be compensated by the positive effect of magnetization in the case of negatively skewed roughness, suitably choosing the rotation ratio and the curvature parameters. Patel and Deheri (2014) dealt with the combined effect of slip velocity and surface roughness on the performance of Jenkins model based magnetic squeeze film in curved rough circular plates. It was obtained that the Jenkins model modified the performance of the bearing system as compared to Neuringer-Rosensweig model, but this model provided little support to the negatively skewed roughness for overcoming the adverse effect of standard deviation and slip velocity, even if curvature parameters were suitably chosen.

This paper aims to analyze the effect of slip velocity and roughness on the Jenkins model based ferrofluid lubrication of a curved rough annular squeeze film bearing.

ANALYSIS

Figure 1 shows the squeeze film bearing configuration of two annular plates each of inside radius b and outside radius a. Here, the upper plate and lower plate are considered as in curved geometry and r is the radial coordinate.

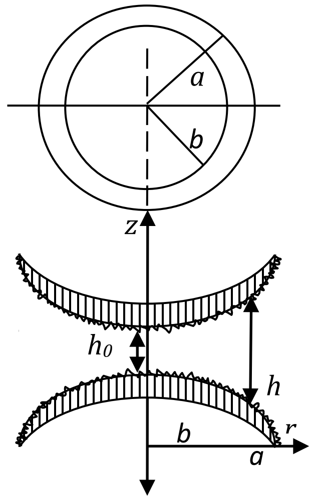

Figure 1. Configuration and geometry of the bearing system

It is assumed that bearing surfaces are transversely rough in the present study. In view of the model for stochastic theory of Christensen and Tonder (1969a, 1969b, 1970), the thickness h of the lubricant film is

$$h = \bar{h} + h_s \quad (1)$$

where $\bar{h}$ stand for the mean film thickness and h_s represents the deviation from the mean film thickness characterizing the random roughness of the bearing surfaces. h_s is decided by the probability density function

$$f(h_s) = \begin{cases} \frac{35}{32c^7}\left(c^2 - {h_s}^2\right)^3, & -c \le h_s \le c \\ 0 & , \text{elsewhere} \end{cases}.$$

wherein c denotes 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 adopted as in the theory of Christensen and Tonder (1969a, 1969b, 1970).

In 1972, Jenkins developed a simple model to state the flow of a magnetic fluid. Later on, it was found that Jenkins model was not only a generalization of the Neuringer- Rosensweig model but also modified both the pressure and the velocity of the magnetic fluid.

Using Maugin's theory, equations of the steady flow turns out to be (Jenkins (1972) and Ram and Verma (1999))

$$\rho(\bar{q}.\nabla)\bar{q} = -\nabla_p + \eta\nabla^2\bar{q} + \mu_0(\bar{M}.\nabla)\bar{H} + \frac{\rho A^2}{2}\nabla \times \left[\frac{\bar{M}}{M} \times \{(\nabla \times \bar{q}) \times \bar{M}\}\right] \quad (2)$$

together with

$$\nabla.\bar{q}=0, \nabla\times\bar{H}=0, \bar{M}=\bar{\mu}\bar{H}, \nabla.(\bar{H}+\bar{M})=0$$

(Bhat (2003). where ρ indicates the fluid density, $\bar{q}$ stand for the fluid velocity in the film region, $\bar{H}$ is external magnetic field, $\bar{\mu}$ denotes magnetic susceptibility of the magnetic fluid, p represents the film pressure, η denotes the fluid viscosity, μ_0 indicates the permeability of the free space, A indicates a material constant and $\bar{M}$ denotes magnetization vector. From equation (2) one establishes that Jenkins model is a generalization of Neuringer- Rosensweig model with the extra term

$$\frac{\rho A^2}{2}\nabla\times\left[\frac{\bar{M}}{M}\times\{(\nabla\times\bar{q})\times\bar{M}\}\right] = \frac{\rho A^2\bar{\mu}}{2}\nabla\times\left[\frac{\bar{H}}{H}\times\{(\nabla\times\bar{q})\times\bar{H}\}\right] \quad (3)$$

which modifies the velocity of the fluid.

Let (u, v, w) to be the velocity of the fluid at any point (r, θ, z) between two solid surfaces, with OZ as axis. With the usual assumptions of hydrodynamic lubrication theory and recalling that the flow is steady and axially symmetric, the equations of motion are

$$\left(1-\frac{\rho A^2\bar{\mu}H}{2\eta}\right)\frac{\partial^2 u}{\partial z^2}=\frac{1}{\eta}\frac{d}{dr}\left(p-\frac{\mu_0\bar{\mu}}{2}H^2\right) \quad (4)$$

$$\frac{1}{r}\frac{\partial}{\partial r}(ru)+\frac{\partial w}{\partial z}=0 \quad (5)$$

Solving the equation (4) under the boundary conditions, u = 0 when z = 0, h, one arrives at

$$u=\frac{z(z-h)}{2\eta\left(1-\frac{\rho A^2\bar{\mu}H}{2\eta}\right)}\frac{d}{dr}\left(p-\frac{\mu_0\bar{\mu}}{2}H^2\right) \quad (6)$$

Replacing the value of u in equation (5) and integrating it with respect to z over the interval (0, h), one finds the Reynolds type equation for film pressure, as

$$\frac{1}{r}\frac{d}{dr}\left(\frac{h^3}{\left(1-\frac{\rho A^2\bar{\mu}H}{2\eta}\right)}r\frac{d}{dr}\left(p-\frac{\mu_0\bar{\mu}}{2}H^2\right)\right)=12\eta\dot{h}_0 \quad (7)$$

It is considered that the upper plate lying along the surface determined by (Bhat (2003), Abhangi and Deheri (2012), Patel and Deheri (2013))

$$z_u=h_0\left[\frac{1}{1+\beta r}\right];\ b\le r\le a$$

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

$$z_l=h_0[\sec(\gamma r^2)-1];\ b\le r\le a$$

where β, γ and h_0 indicate the upper plate's curvature parameter, lower plate's curvature parameter and the centre film thickness respectively. Therefore, the mathematical expression for the film thickness h(r) is defined by (Bhat (2003), Abhangi and Deheri (2012), Patel and Deheri (2014))

$$h(r)=h_0\left[\frac{1}{1+\beta r}-\sec(\gamma r^2)+1\right];\ b\le r\le a$$

In view of the theory of Christensen and Tonder (1969a, 1969b, 1970), the stochastic averaging of the differential equation (7), under the usual hypothesises of hydro-magnetic lubrication yields (Bhat (2003), Prajapati (1995), Patel et al. (2009)), the modified Reynolds type equation,

$$\frac{1}{r}\frac{d}{dr}\left(\frac{g(h)}{\left(1-\frac{\rho A^2\bar{\mu}H}{2\eta}\right)}r\frac{d}{dr}\left(p-\frac{\mu_0\bar{\mu}}{2}H^2\right)\right)=12\eta\dot{h}_0 \quad (8)$$

where

$$H^2=K(r-b)(a-r), G=\left(\frac{4+sh}{2+sh}\right)$$

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

The following dimensionless quantities are introduced as

$$\bar{h}=\frac{h}{h_0}=\left[\frac{1}{1+BR}-\sec(CR^2)+1\right], R=\frac{r}{b}, \bar{\varepsilon}=\frac{\varepsilon}{h_0^3},$$

$$P=-\frac{h_0^3 p}{\eta b^2\dot{h}_0}, B=\beta b, C=\gamma b^2, k=\frac{a}{b}, \bar{s}=sh_0$$

$$\mu^*=-\frac{K\mu_0\bar{\mu}h_0^3}{\eta\dot{h}_0}, \bar{A}^2=\frac{\rho A^2\bar{\mu}b\sqrt{K}}{2\eta},$$

$$\bar{\sigma}=\frac{\sigma}{h_0}, \bar{\alpha}=\frac{\alpha}{h_0}, \quad (9)$$

Using equation (9), equation (8) reduces to,

$$\frac{1}{R}\frac{d}{dR}\left(\frac{g(\bar{h})}{\left(1-\bar{A}^2\sqrt{(R-1)(k-R)}\right)}R\frac{d}{dR}\left(P-\frac{1}{2}\mu^*(R-1)(k-R)\right)\right)=-12 \quad (10)$$

where

$$g(\bar{h})=(\bar{h}^3+3\bar{h}^2\bar{\alpha}+3(\bar{\sigma}^2+\bar{\alpha}^2)\bar{h}+3\bar{\sigma}^2\bar{\alpha}+\bar{\alpha}^3+\bar{\varepsilon})\bar{G},$$

$$\bar{G}=\left(\frac{4+\bar{s}\bar{h}}{2+\bar{s}\bar{h}}\right)$$

Under the boundary conditions

$$P(1)=P(k)=0 \quad (11)$$

one can derive the solution of equation (10), for the dimensionless pressure distribution, as

$$P=\frac{\mu^*}{2}(R-1)(k-R) - 6\int_1^R\frac{R}{g(\bar{h})}\left(1-\bar{A}^2\sqrt{(R-1)(k-R)}\right)dR$$

$$+6\frac{\int_1^k \frac{R}{g(\bar{h})}\left(1-\bar{A}^2\sqrt{(R-1)(k-R)}\right)dR}{\int_1^k \frac{1}{Rg(\bar{h})}\left(1-\bar{A}^2\sqrt{(R-1)(k-R)}\right)dR}\int_1^R \frac{1}{Rg(\bar{h})}\left(1-\bar{A}^2\sqrt{(R-1)(k-R)}\right)dR \quad (12)$$

The non-dimensional load carrying capacity of the bearing system then, is obtained as

$$W=-\frac{h_0^3 w}{2\pi\eta b^4 \dot{h}_0}=\frac{\mu^*}{24}(k+1)(k-1)^3$$
$$+3\int_1^k \frac{R^3}{g(\bar{h})}\left(1-\bar{A}^2\sqrt{(R-1)(k-R)}\right)dR$$
$$-3\frac{\left[\int_1^k \frac{R}{g(\bar{h})}\left(1-\bar{A}^2\sqrt{(R-1)(k-R)}\right)dR\right]^2}{\int_1^k \frac{1}{Rg(\bar{h})}\left(1-\bar{A}^2\sqrt{(R-1)(k-R)}\right)dR} \quad (13)$$

RESULTS AND DISCUSSION

A scrutiny of equation (13) suggests that the dimensionless load carrying capacity gets increased by

$$\frac{\mu^*}{24}(k+1)(k-1)^3$$

in comparison with the conventional lubricant based bearing system. The increased load is caused due to the fact that the magnetization enhances the viscosity of the lubricant. Further, one can easily notice that the expression found in equation (13) is linear with respect to the magnetization parameter. Consequently, an increase in the magnetization would result in increased load carrying capacity. This is reflected in figures 2-8. As can be seen the load carrying capacity rises sharply. The effect of standard deviation appears to be nominal while the effect of skewness is almost negligible on the distribution of load carrying capacity with respect to the magnetization.

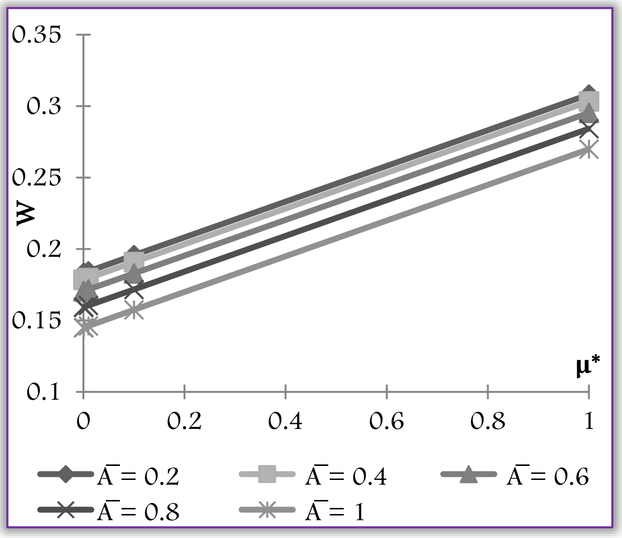

Figure 2. Variation of Load carrying capacity with respect to μ^* and $\bar{A}$.

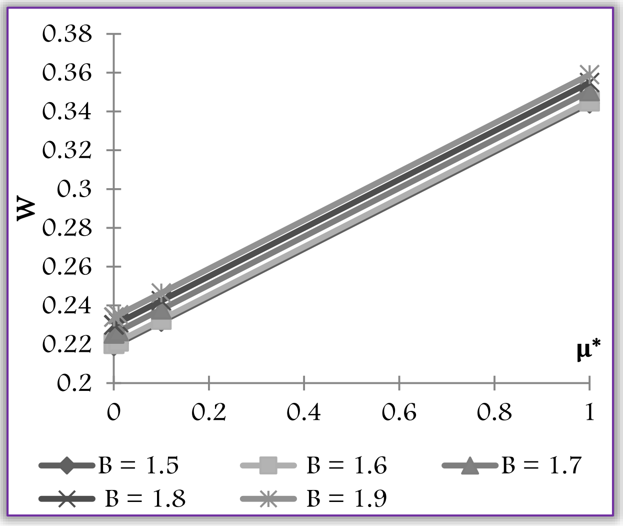

Figure 3. Variation of Load carrying capacity with respect to μ^* and B.

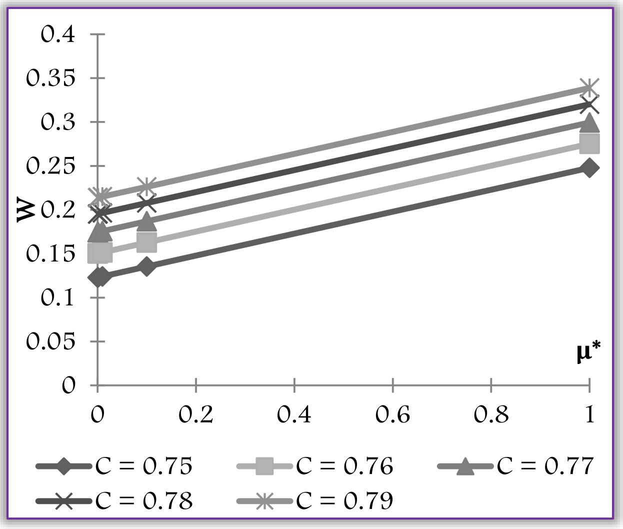

Figure 4. Variation of Load carrying capacity with respect to μ^* and C.

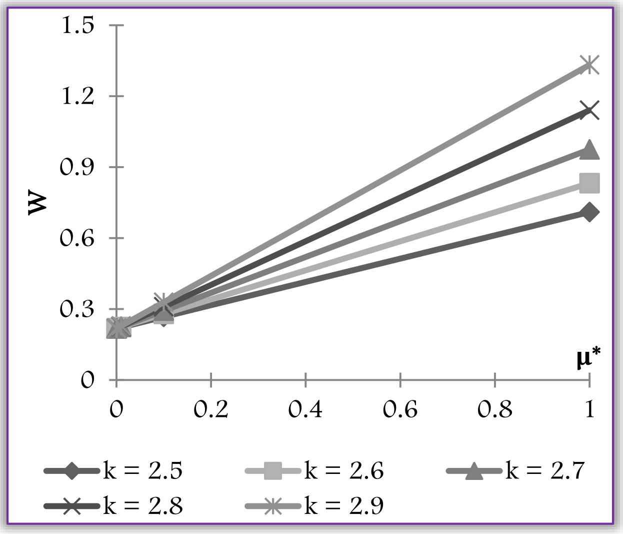

Figure 5. Variation of Load carrying capacity with respect to μ^* and k.

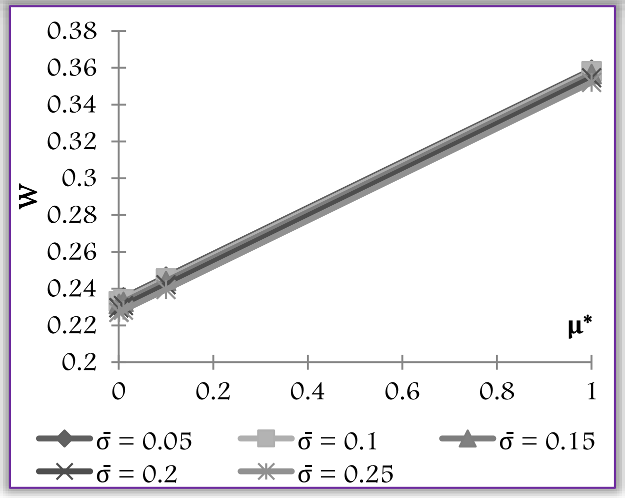

Figure 6. Variation of Load carrying capacity with respect to μ^* and $\bar{\sigma}$.

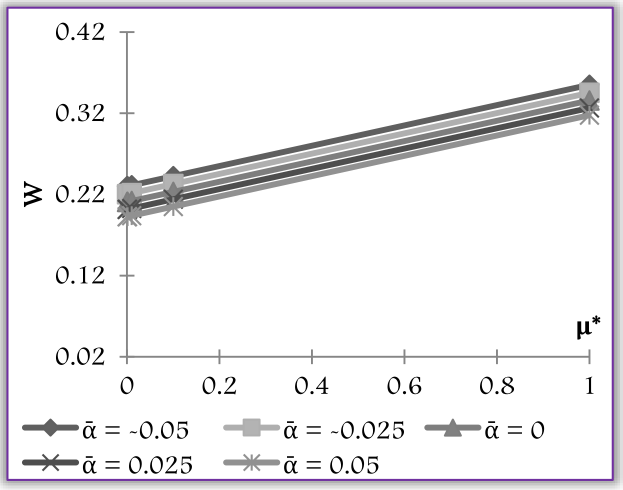

Figure 7. Variation of Load carrying capacity with respect to μ^* and $\bar{\alpha}$.

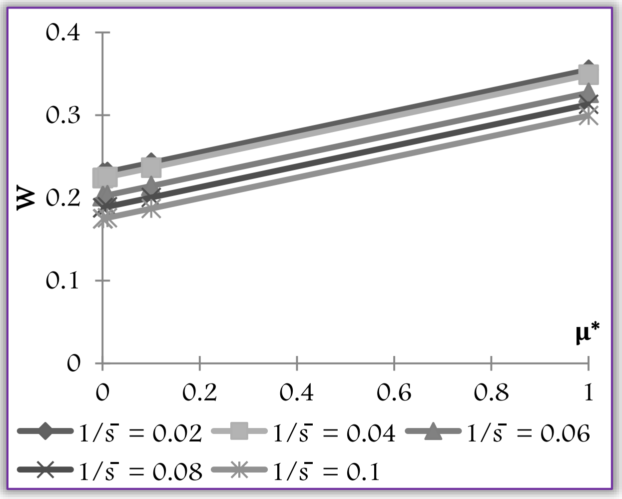

Figure 8. Variation of Load carrying capacity with respect to μ^* and $1/\bar{s}$.

The fact that the material constant parameter causes decreased load carrying capacity can be obtained from figures 9-14. Here the effect of standard deviation is negligible while the effect of skewness is nominal. Further, the effect of aspect ratio remains negligible up to the value of material constant parameter 0.4 (Figure 11).

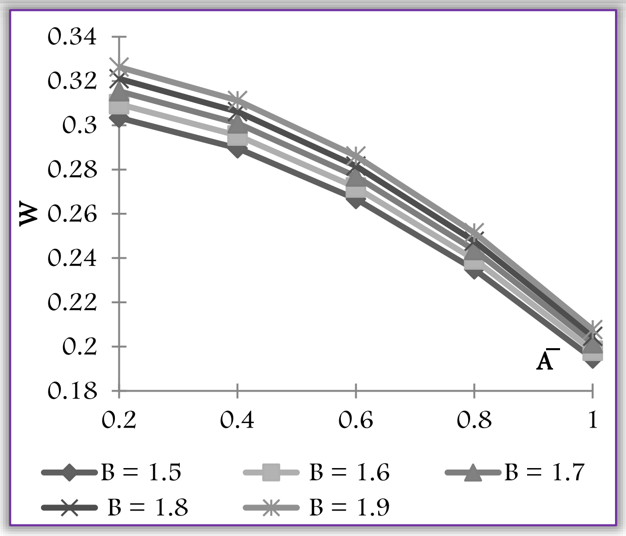

Figure 9. Variation of Load carrying capacity with respect to $\bar{A}$ and B.

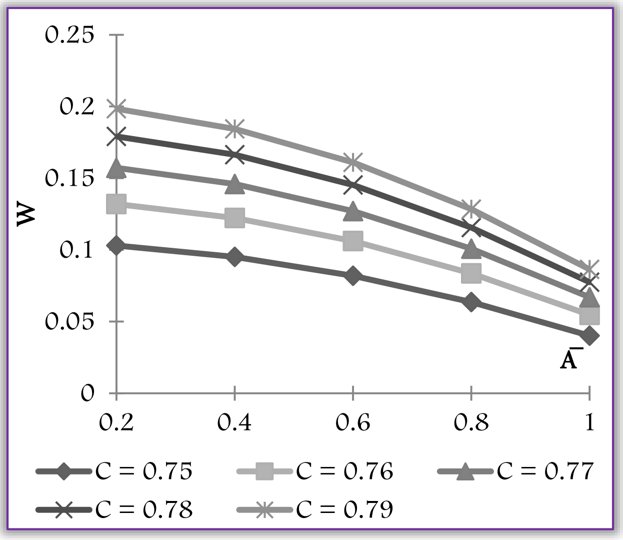

Figure 10. Variation of Load carrying capacity with respect to $\bar{A}$ and C.

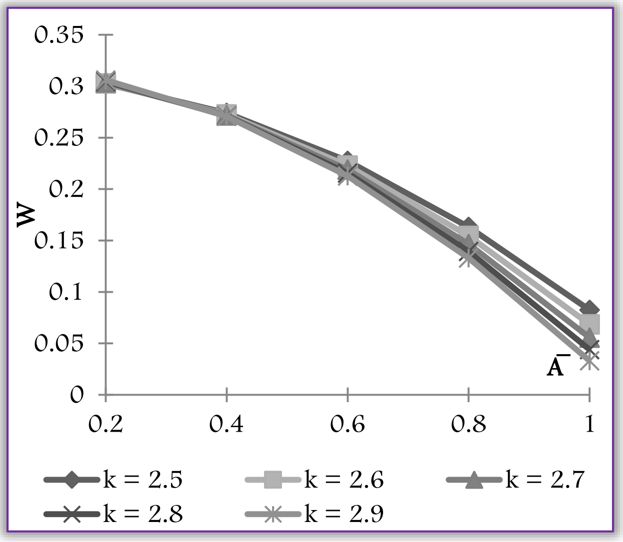

Figure 11. Variation of Load carrying capacity with respect to $\bar{A}$ and k.

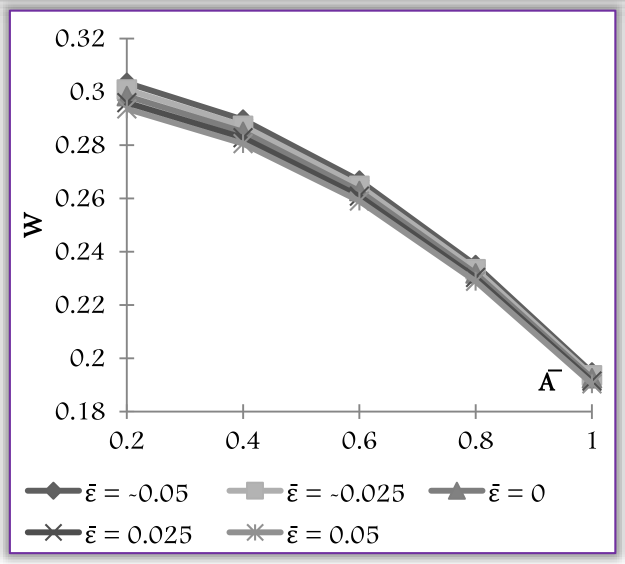

Figure 12. Variation of Load carrying capacity with respect to $\bar{A}$ and $\bar{\varepsilon}$.

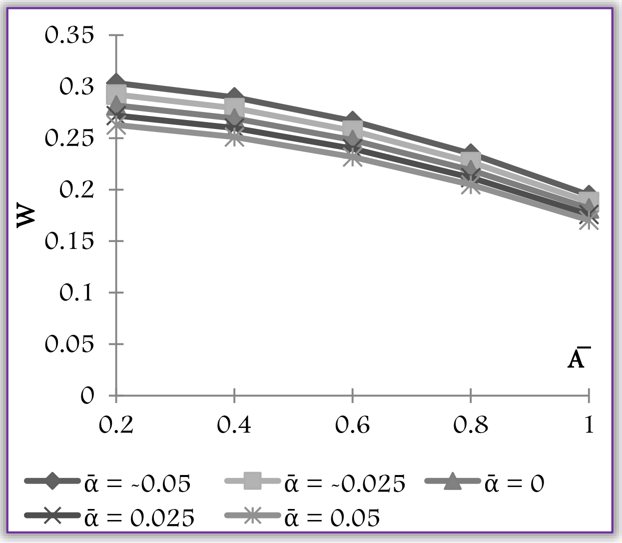

Figure 13. Variation of Load carrying capacity with respect to $\bar{A}$ and $\bar{\alpha}$.

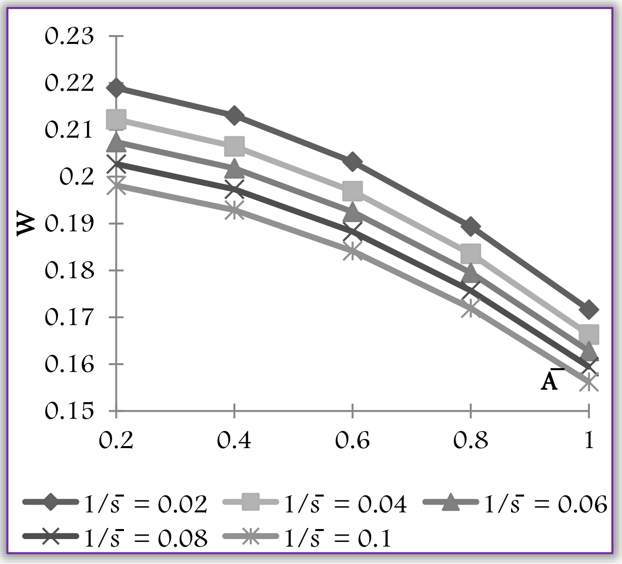

Figure 14. Variation of Load carrying capacity with respect to $\bar{A}$ and $1/\bar{s}$.

The effect of curvature parameters is described in figures 15-24. It is observed that the load carrying capacity increases with increasing values of the upper plate's curvature parameter. However, the lower plate's curvature parameter follows the path of the upper plate's curvature parameter which is mostly contrary to the other geometrical shapes of the curved surface (Patel and Deheri (2013,2014)). Therefore, for designing this type of bearing system the ratio of curvature parameters must be judiciously chosen to overcome the effect of slip velocity (Figures 20, 24). This can be explained mathematically as the trigonometric function sec is even in nature. Here also, the effect of aspect ratio turns out to be nominal. Further, the effect of skewness can be also regarded when the lower plate's curvature parameter is more then 0.78.

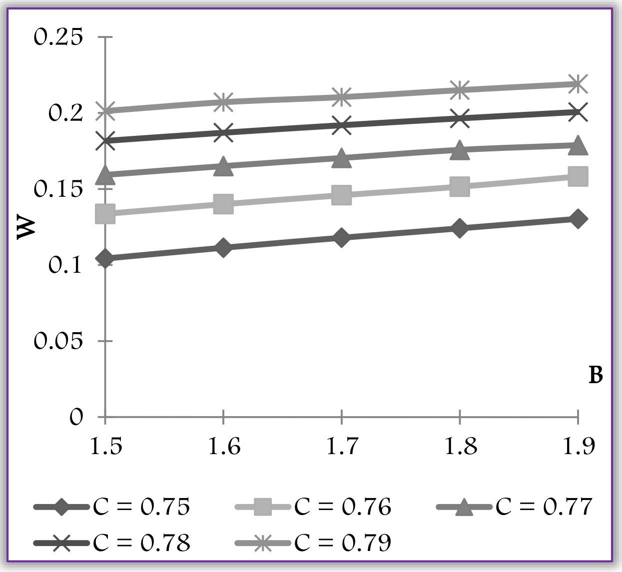

Figure 15. Variation of Load carrying capacity with respect to B and C.

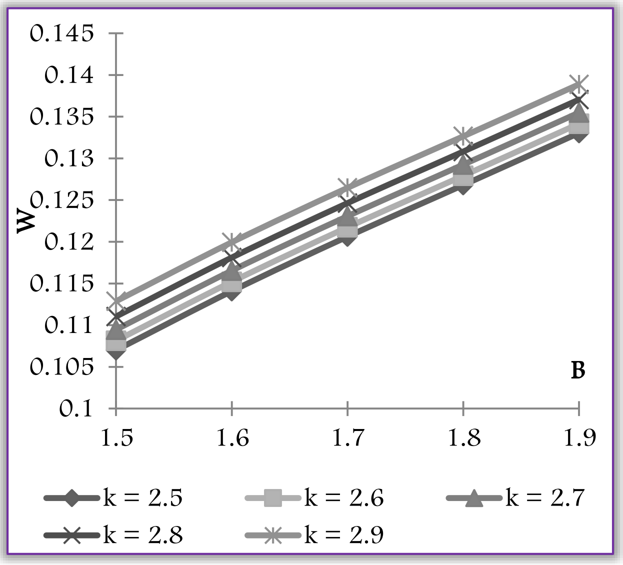

Figure 16. Variation of Load carrying capacity with respect to B and k.

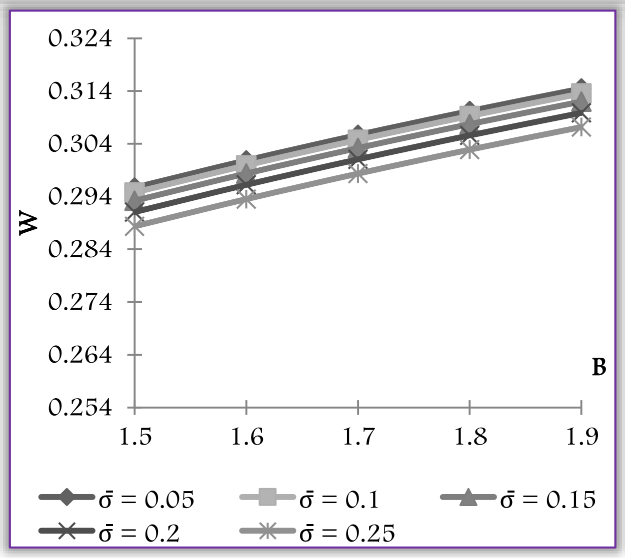

Figure 17. Variation of Load carrying capacity with respect to B and $\bar{\sigma}$.

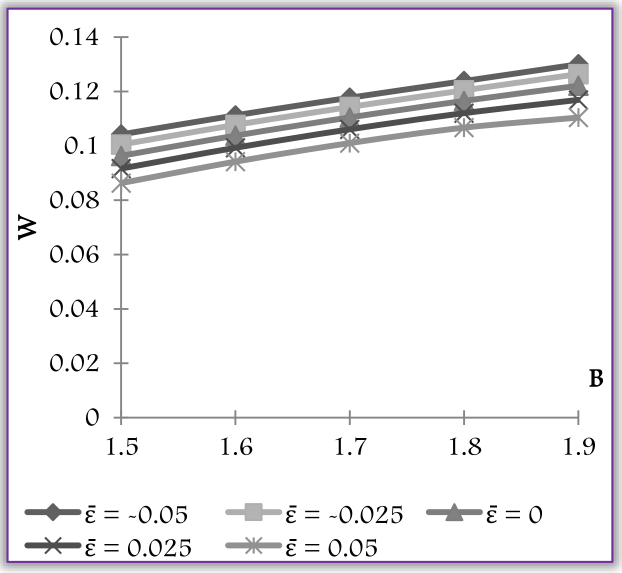

Figure 18. Variation of Load carrying capacity with respect to B and $\bar{\varepsilon}$.

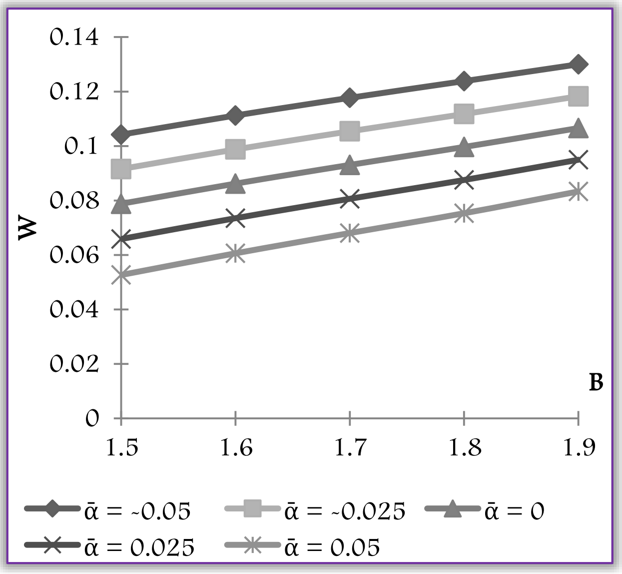

Figure 19.Variation of Load carrying capacity with respect to B and $\bar{\alpha}$.

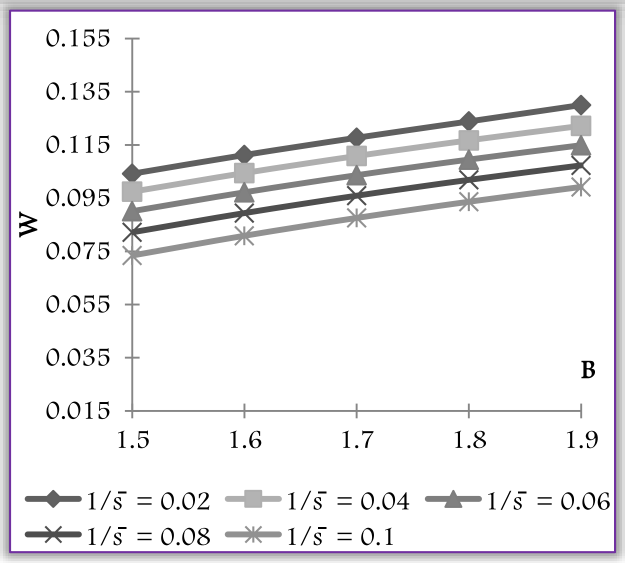

Figure 20. Variation of Load carrying capacity with respect to B and $1/\bar{s}$.

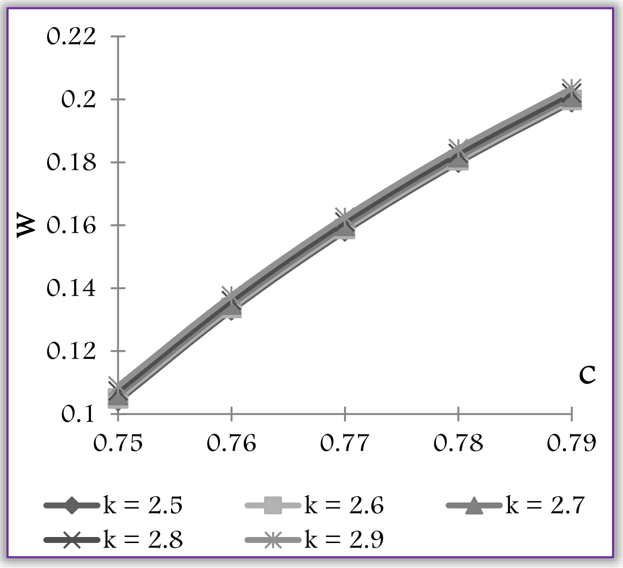

Figure 21. Variation of Load carrying capacity with respect to C and k.

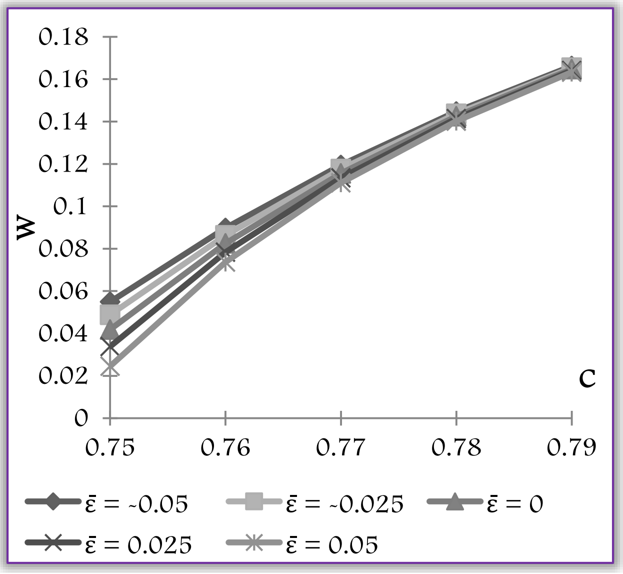

Figure 22. Variation of Load carrying capacity with respect to C and $\bar{\varepsilon}$.

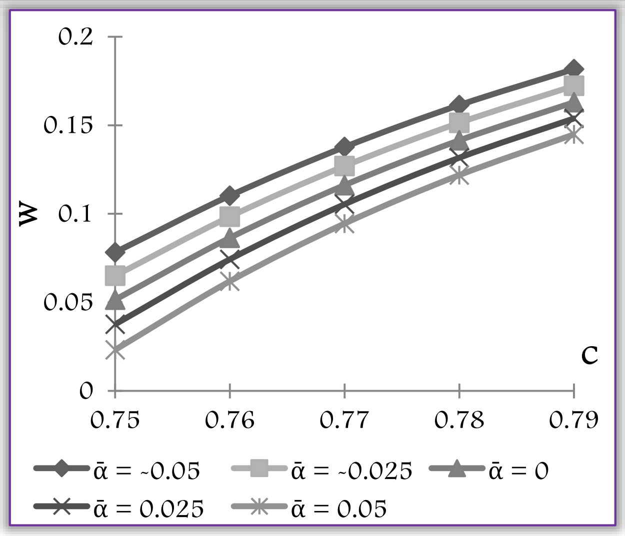

Figure 23. Variation of Load carrying capacity with respect to C and $\bar{\alpha}$.

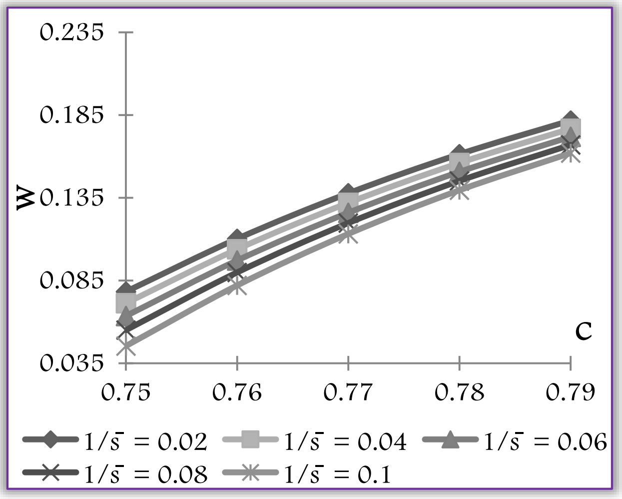

Figure 24. Variation of Load carrying capacity with respect to C and $1/\bar{s}$.

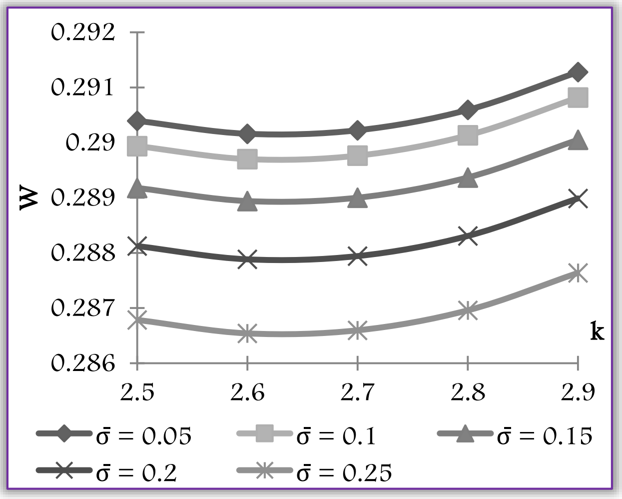

Figure 25. Variation of Load carrying capacity with respect to k and $\bar{\sigma}$.

The effect of aspect ratio k on the performance of this type of bearing system is found from Figures 25-28. It is manifest that the load carrying capacity increases owing to the increasing values of the aspect ratio. Therefore, this study underlines the crucial role of the aspect ratio to improve the performance characteristics of the bearing system.

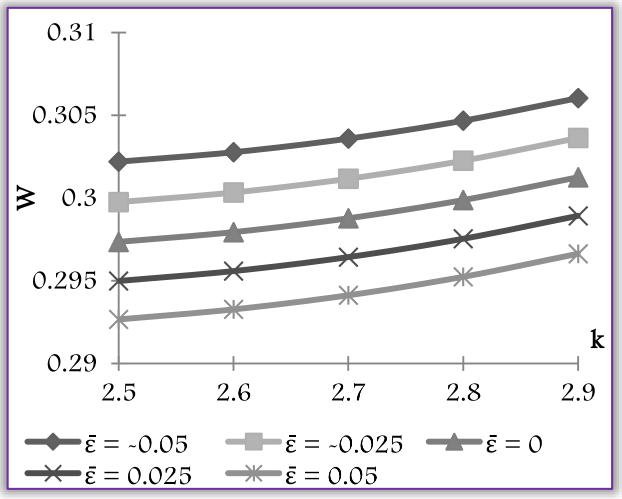

Figure 26. Variation of Load carrying capacity with respect to k and $\bar{\varepsilon}$.

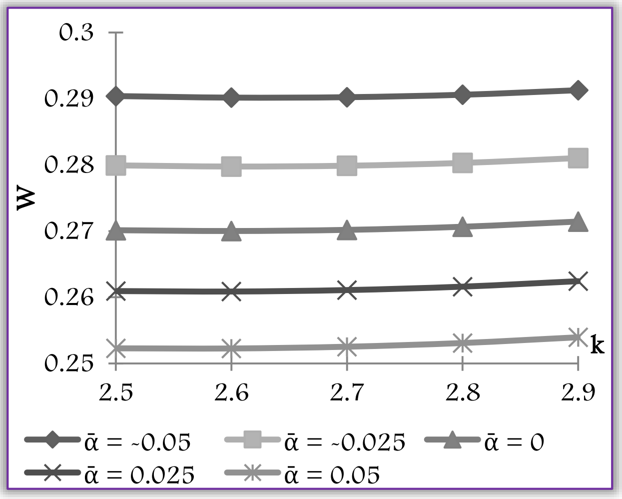

Figure 27. Variation of Load carrying capacity with respect to k and $\bar{\alpha}$.

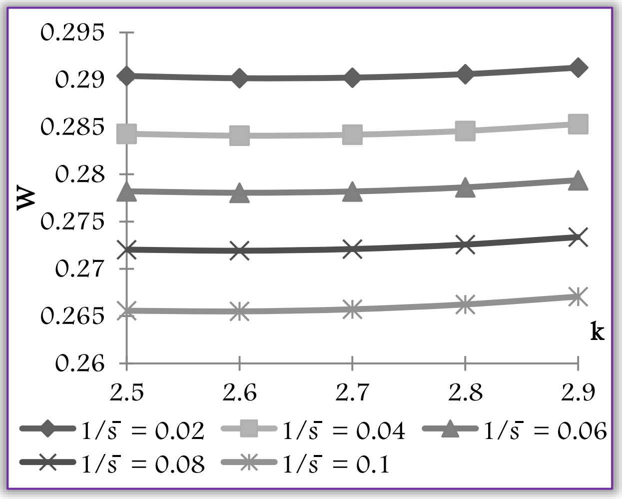

Figure 28. Variation of Load carrying capacity with respect to k and $1/\bar{s}$.

The effect of roughness is presented in figures 29-34. It is noticed that the standard deviation brings down the load carrying capacity while the negatively skewed roughness increases the load carrying capacity, same being case of variance (-ve). Therefore, this combined positive effect can be channelized to improve the performance of the bearing system. It is clear that slip has a good amount of effect. This effect gets compounded further in the case of standard deviation.

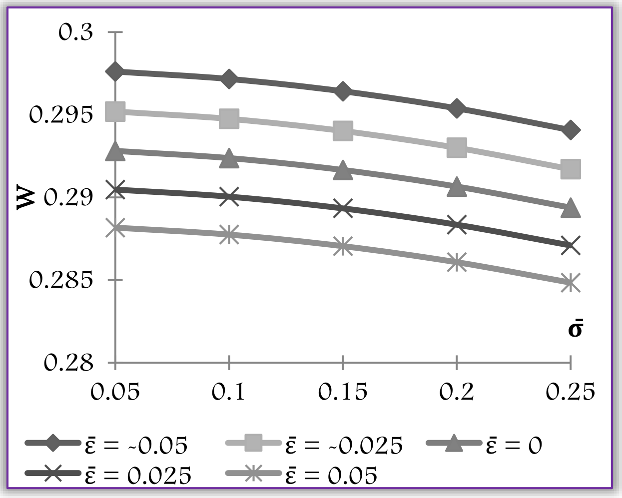

Figure 29. Variation of Load carrying capacity with respect to $\bar{\sigma}$ and $\bar{\varepsilon}$.

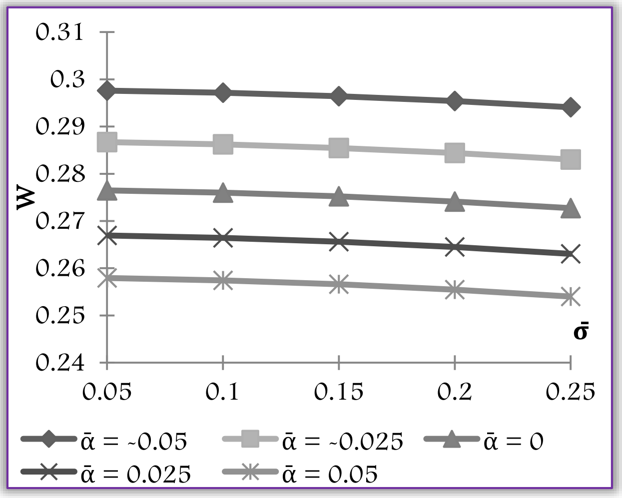

Figure 30. Variation of Load carrying capacity with respect to $\bar{\sigma}$ and $\bar{\alpha}$.

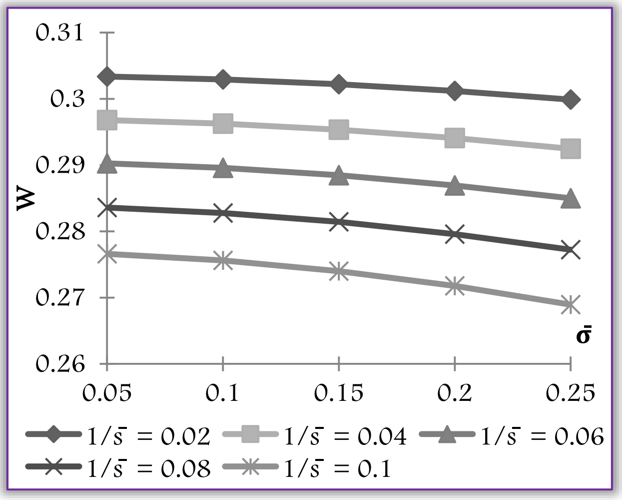

Figure 31. Variation of Load carrying capacity with respect to $\bar{\sigma}$ and $1/\bar{s}$.

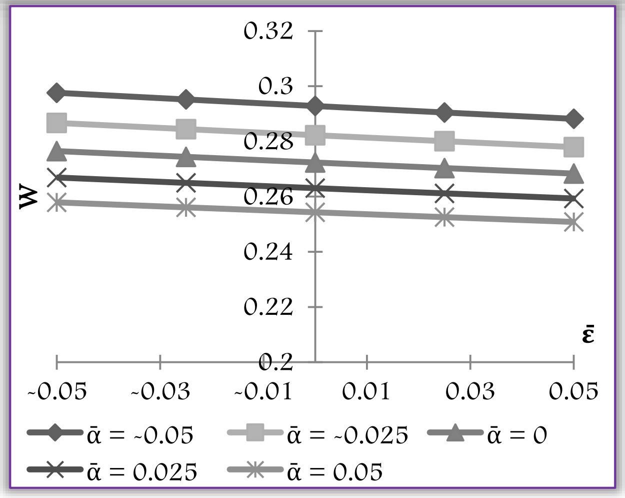

Figure 32. Variation of Load carrying capacity with respect to $\bar{\varepsilon}$ and $\bar{\alpha}$.

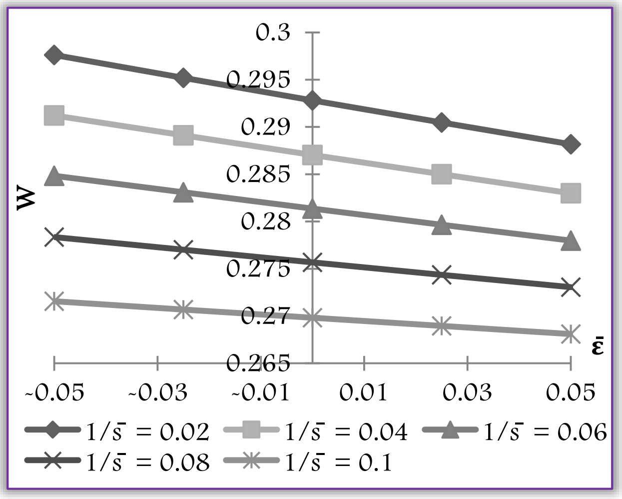

Figure 33. Variation of Load carrying capacity with respect to $\bar{\varepsilon}$ and $1/\bar{s}$.

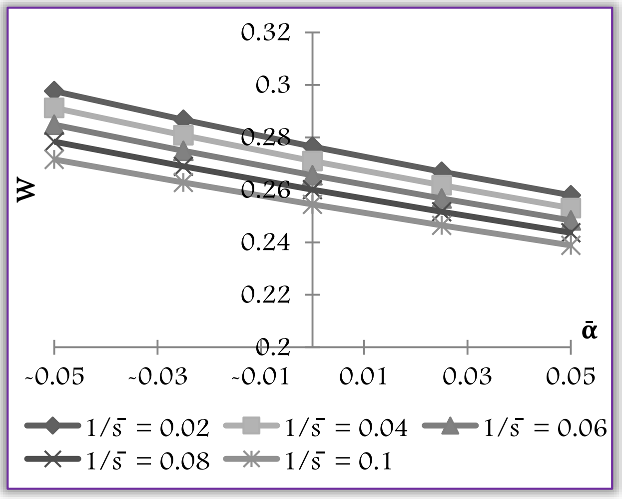

Figure 34. Variation of Load carrying capacity with respect to $\bar{\alpha}$ and $1/\bar{s}$.

There are ample proofs to certify that for any type of improvement in the bearing performance characteristics the slip parameter is required to be kept at reduced level. It is also established from this study that Jenkins model modifies the performance of the bearing system as compared to the case of

Neuringer-Rosensweig model based magnetic fluid flow.

A close glance at the graphs indicates that the Jenkins model based magnetic fluid flow goes to a large extent in minimizing the effect of standard deviation when the slip is at minimum at least in the case of variance (-ve) when negatively skewed roughness is involved.

CONCLUSION

It is visibly clear that the Jenkins model modifies and improves the performance of the bearing system in compensation with the case of Neuringer-Rosensweig model. As the slip parameter causes reduced load carrying capacity, this article makes it mandatory that the roughness must be accounted for while designing the bearing system even if the ratio of curvature parameters is suitably chosen. Further, this article confirms that Jenkins model based magnetic fluid flow may present a better option for the design aspects when the slip velocity is minimized. Lastly, this type of bearing system supports a good amount of load even when there is no flow, which is unlikely, in the case of conventional lubricants based bearing system.

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