EFFECT OF MAGNETIC FLUID THROUGH A SERIES OF FLOWFACTORS ON THE BEHAVIOR OF A LONGITUDEALLY ROUGH EXPONANTIAL SLIDER BEARING

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Abstract
This study aims to analyze the effect of magnetic fluid through a series of flowfactors, which is strongly dependent on the surface pattern parameter on the behavior of a longitudealy rough exponantial slider bearing. The roughness of the bearing surfaces is characterized by a stochastic random variable with non-zero mean variance and skewness. The associated Reynolds' equation is stochastically averaged with respect to the random roughness parameter, and is solved with appropriate boundary conditions to obtain the pressure distribution. From this, the expression for load carrying capacity is derived. This results in the calculation of load carrying capacity presented in graphical forms suggest that the magnetization parameter increases the load carrying capacity while the load carrying capacity gets decreased due to the standard deviation. . It is seen that the negatively skewed roughness increases the load carrying capacity substantially especially, when the negative variance is involved. In addition, it is easily seen that the increment in the positivly skewed roughnees longitudinally causes the decrease in the load carrying capacity of the bearing.

Keywords: Exponantial slider bearing, Flow factor, Longitudinal roughness, Magnetic fluid, Pressure, Load carrying capacity Friction.

Introduction

The slider bearing is the simplest and frequently encountered among the hydrodynamic bearings. The slider bearings are used in frictional devices such as clutch plates, automobile transmissions etc. Slider bearings have been studied for several film shapes [Lord Rayleigh (1), Pinkus and Sternlicht (2), Cameron (3)]. It is well known that the bearing surfaces after having some run-in and wear develop roughness. Tzeng and Saibel (4) have introduced stochastic concepts to analyze a two dimensional inclined slider bearing with one-dimensional roughness in the direction transverse to the sliding direction. Christensen and Tonder (5-7) analyzed three different models of hydrodynamic lubrication of an inclined slider bearing with rough surfaces. The first model is associated with longitudinal one-dimensional roughness, the second is related to one-dimensional transverse roughness and the third deals with the case of uniform, isotropic roughness. This approach of Christensen and Tonder (5-7) was used to study the effect of surface roughness on the performance of bearing systems in a good number of investigations [Prakash and Tiwari (8), Guha (9) and Gupta and Deheri (10)). Patir and Cheng (11) modified the averaged Reynolds equation for rough surfaces. Infacts they defined pressure and shear flow factors, which were obtained independently by numerical flow simulation using randomly generated or measured surface roughness profiles.

By now, it is well know that the magnetic fluid as a lubricant improves the performance of the bearing system as compared to the conventional lubricant. Agrawal (12) studied the performance of an inclined plane slider bearing with a ferrofluid lubricant and found that its performance was comparatively better than the corresponding bearing system with a conventional lubricant. The study of Bhat and Patel (13) regarding the performance of an exponential slider bearing with a ferrofluid lubricant concluded that the magnetic fluid lubrication caused increased load carrying capacity slightly altering the friction on the slider. The analysis of Shah et al. (14) suggested the positive effect of magnetic fluid lubrication over the conventional fluids. Andharia *et al.*(15) and Deheri *et al.*(17)observed that in case of longitudinal roughness by suitably choosing of the magnetization parameter, the performance of the bearing could be improved. Shukla and Deheri (22) studied the performance of a rough exponential slider bearing under the presence of a magnetic fluid lubricant. It is established that the load carrying capacity in addition, to friction increase with increasing magnetization. The negatively skewed roughness induced to increase load carrying capacity goes a long way in mitigating the adverse effect of the standard deviation taking recourse to suitable values of magnetization parameter. Patel and Deheri [23] analyzed the comparison of various porous structures on the performance of a magnetic fluid based transversely rough short bearing. It was found that the magnetization affected the bearing system positively while the bearing suffered owing to the transverse roughness.

Deheri et al. (21) analyzed the influence of roughness parameters on the pressure and load carrying capacity in a rough finite plane slider bearing for longitudinally rough surfaces by taking account of the influence of surface roughness through a series of flow factors.

Here, it has been sought to study and analyze the effect of magnetic fluid through a series of flowfactors, which is strongly dependent on the surface pattern parameter (γ>1) on the behavior of a longitudeally rough exponantial slider bearing

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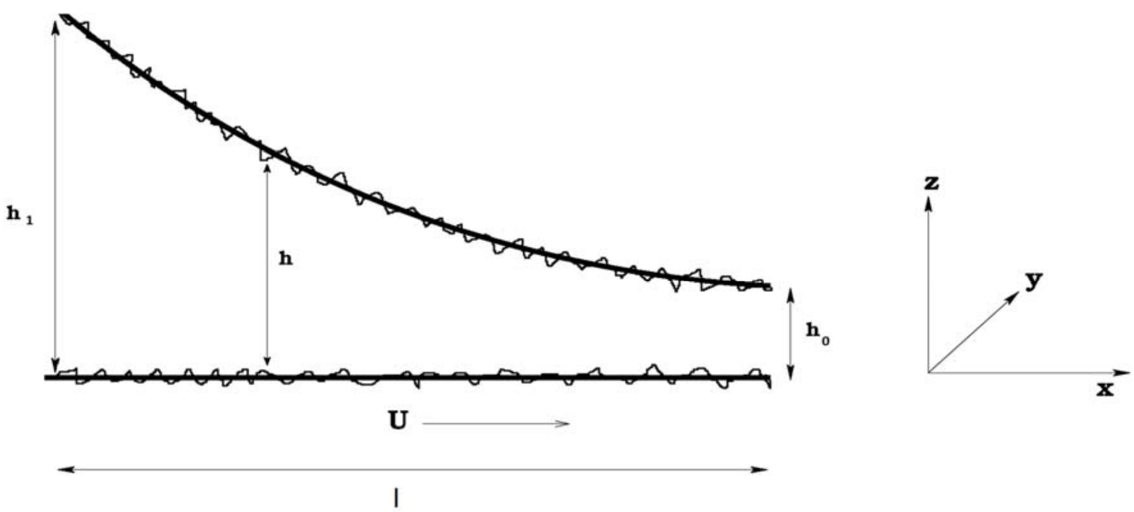
Analysis



The bearing configuration is presented in figure-1. Assuming the slider moves with the uniform velocity U in the X direction. The length of the bearing is l and breath of bearing is b with l<<b while h_0 and h_1 minimum and maximum film thickness respectively. The bearing surfaces are assumed to be the longitudinally rough. The film thickness h is defined as

$$h = h_0 \ e^{\left(\frac{l-x}{l}\right)} \tag{1}$$

Following the investigations of Agrawal [25] the magnitude M of the magnetic field $\overline{H}$ is considered to be

$$M^2 = Kx(1-x) \tag{2}$$

K being a quantity chosen to suit the dimensions so as to manufacture a magnetic field of required strength.[Bhat(2003)]

Now, under the usual assumptions of the hydrodynamic lubrication, the modified Reynolds' equation [Agrawal(12) , Bhat(18) , Bhat & Patel(19),Deheri(21)]turns out to be

$$\frac{d}{dx}\left[\varphi_x \frac{h^3}{12\mu}\frac{d}{dx}\left(p-0.5\mu_0\mu_f M^2\right)\right]=\frac{U}{2}\frac{dh}{dx} \quad (3)$$

where φ_x is pressure flow factor in x direction, μ is the viscocity of the lubricunt, μ_0 is magnetic susceptibility, which is a dimensionless proportionality constant that includes the magnetization of a lubricant in response to an applied magnetic field. μ_f is free space permeability . The free space permeability (permeability in vacuum) of a material characterizes the response of that material to electric or magnetic field. In simplified models, it is often regarded to be constant ($4\pi \times 10^{-7}\ N/A^2$) for a given material.

Following the stochastic modeling of Christensen and Tonder (1969a, 1969b, 1970) the thickness $h(x)$ of the lubricant film is considered as

$$h(x)=\bar{h}(x)+\delta \quad (4)$$

where, $\bar{h}(x)$ is the mean film thickness and δ is the deviation from the mean film thickness characterizing the random roughness of the bearing surfaces. δ is assumed to be stochastic in nature and governed by the probability density function $f(\delta)$, –c< δ <c, where c is the maximum deviation from the mean film thickness. The mean α standard deviation σ and skewness ε, which is the measure of symmetry of the random variable δ, are determined by the relationships,

$$\alpha = E(\delta) \quad (5)$$

$$\sigma^2 = E[(\delta-\alpha)^2] \quad (6)$$

and

$$\varepsilon = E[(\delta-\alpha)^3] \quad (7)$$

where E denotes the expected value defined by

$$E(R)=\int_{-c}^{c} R\, f(\delta)d\delta \quad (9)$$

while the probability density function is associated as

$$f(\delta)=\begin{cases}\frac{32}{35c}\left(1-\frac{\delta^2}{c^2}\right)^3 & -c\le\delta\le c\\ 0 & elsewhere\end{cases} \quad (10)$$

Following the averaging process discussed by Andharia et al. (16), equation (3) reduces to

$$\frac{d}{dx}\left[\varphi_x \frac{a(h)^{-1}}{12\mu}\frac{d}{dx}\left(p-0.5\mu_0\mu_f M^2\right)\right]=\frac{U}{2}\frac{d}{dx}[b(h)^{-1}] \quad (11)$$

where

$$a(h)=h^{-3}[1-3\alpha h^{-1}+6h^{-2}(\sigma^2+\alpha^2)-20h^{-3}(\varepsilon+3\sigma^2\alpha+\alpha^3)] \quad (12)$$

and

$$b(h)=h^{-1}[1-\alpha h^{-1}+h^{-2}(\sigma^2+\alpha^2)-h^{-3}(\varepsilon+3\sigma^2\alpha+\alpha^3)] \quad (13)$$

Making use of equations (1) and (2) and dimensionless quantities:

$$h^*=\frac{h}{h_0}, X=\frac{x}{l}, , P=\frac{h_m^2 p}{\mu U l}, \mu^*=\frac{\mu_0\mu_f h_m^2 l}{2\mu U}, M^2 = X(1-X) \quad (14)$$

$$A(h^*)=h^{*-3}\left[1-3\alpha^* h^{*-1}+6h^{*-2}\left(\sigma^{*2}+\alpha^{*2}\right)-20h^{*-3}(\varepsilon^*+3\sigma^{*2}\alpha^*+\alpha^{*3})\right] \quad (15)$$

and

$$B(h^*)=h^{*-1}\left[1-\alpha^* h^{*-1}+h^{*-2}\left(\sigma^{*2}+\alpha^{*2}\right)-h^{*-3}(\varepsilon^*+3\sigma^{*2}\alpha^*+\alpha^{*3})\right] \quad (16)$$

equation (11) transforms to,

$$\frac{d}{dX}\left[\varphi_X A(h^*)^{-1}\frac{d}{dX}(P-\mu^* X(1-X))\right]=6\frac{d}{dX}[B(h^*)^{-1}] \quad (17)$$

An experimental relation for φ_x obtained by Patir (21) is as under,

$$\varphi_x = 1 + C\,H^{-r} \qquad (for\ \gamma > 1) \quad (18)$$

$$\therefore \quad \varphi_X = 1 + C\,(h^* H_m)^{-r} \qquad (for\ \gamma > 1) \quad (19)$$

where

$$H=\frac{h}{\sigma},\ \ H_m=\frac{h_m}{\sigma}. \quad (20)$$

and the constants C and r are given as functions of γ in following table,

γ	C	r	H
3	0.225	1.5	H > 0.5
6	0.520	1.5	H > 0.5
9	0.870	1.5	H > 0.5

Table-1. Relation between , C , r and H

Solving equation (7) under the boundary conditions:

$$P=0, at\ X=0 \quad (21)$$

$$P=0, at\ X=1 \quad (22)$$

one obtains the expression for non dimensional pressure distribution as:

$$P(X) = \mu^* X(1-X) + \int_0^X \frac{1}{\varphi_X} \frac{1}{A(h^*)^{-1}} [\, 6\, B(h^*)^{-1} - Q^*] dX \qquad (23)$$

where,

$$Q^* = \frac{\int_0^1 \frac{6\,A(h^*)}{\varphi_X\, B(h^*)}\, dX}{\int_0^1 \frac{M(h^*)}{\varphi_X} dX} \qquad (24)$$

Then, the non dimensional form of the load carrying capacity W^* is expressed as:

$$W^* = \frac{W h_m^2}{\mu U l} = \int_0^1 P\ dX \qquad (25)$$

Results and discussion

It is easily seen from equation (23) that the pressure increases by

$$\mu^* X(1-X)$$

as compared to conventional lubricant based bearing system. therefore, the load carrying capacity enhances. The variation of dimensionless load carrying capacity W^* with respect to magnetization parameter is presented in Figures (1 - 4) for various values of σ^*, ε^* α^* and γ respectively. It is seen that the load carrying capacity increases sharply due to the magnetization parameter. It is noticed that for smaller values of σ^* the effect of magnetization parameter on the variation of dimensionless load carrying capacity $W^*$is marginal. (Figure-1) It is observed that skewness (+ve) decreases the load carrying capacity while (-ve) skewness increases the load carrying capacity. (Figure-2) The variance follows the same trends of skewness. (Figure-3)

The variation of the dimensionless load carrying capacity W^*versus the standard deviation σ^* for different values of the ε^* and α^* is presented in figures (5) and (6). The combined effect of variance and skewness is illustrated in figure (7). It is clear from these graphs that increased the load carrying capacity due to σ^*, gets further increased owing to the negatively skewed roughness and variance(-ve).

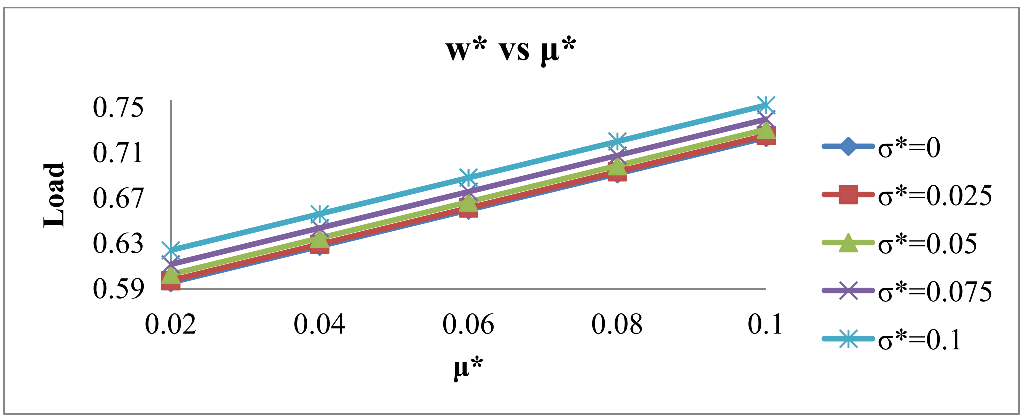


Fig.1. Variation of load carrying capacity with respect to μ*

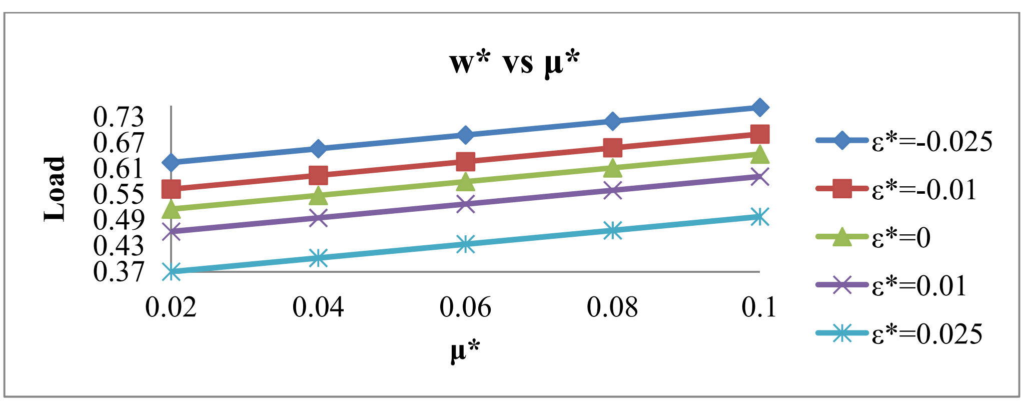


Fig.2. Variation of load carrying capacity with respect to μ*

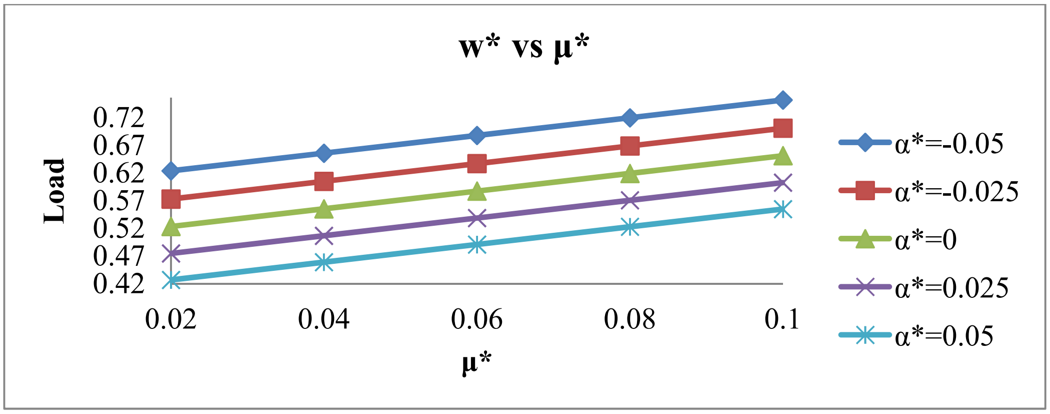


Fig.3. Variation of load carrying capacity with respect to μ*

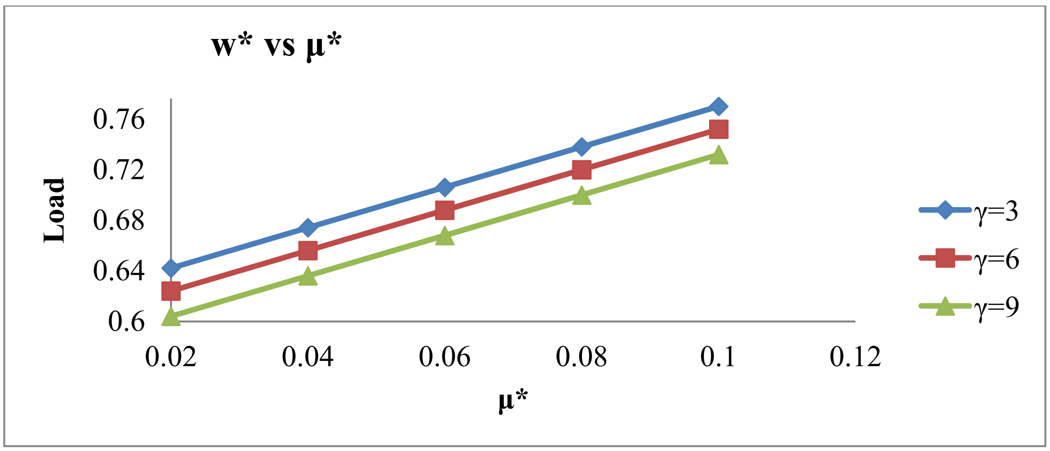


Fig.4. Variation of load carrying capacity with respect to μ*

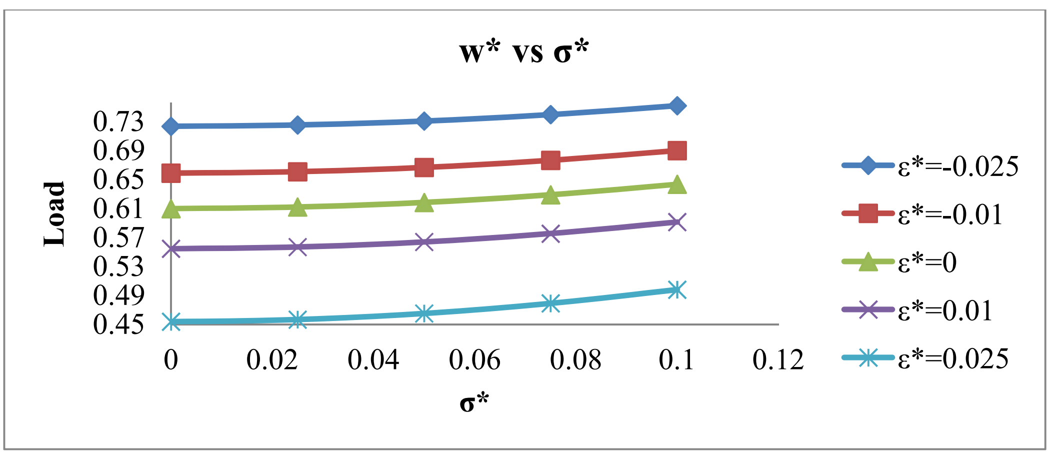


Fig.5. Variation of load carrying capacity with respect to σ*

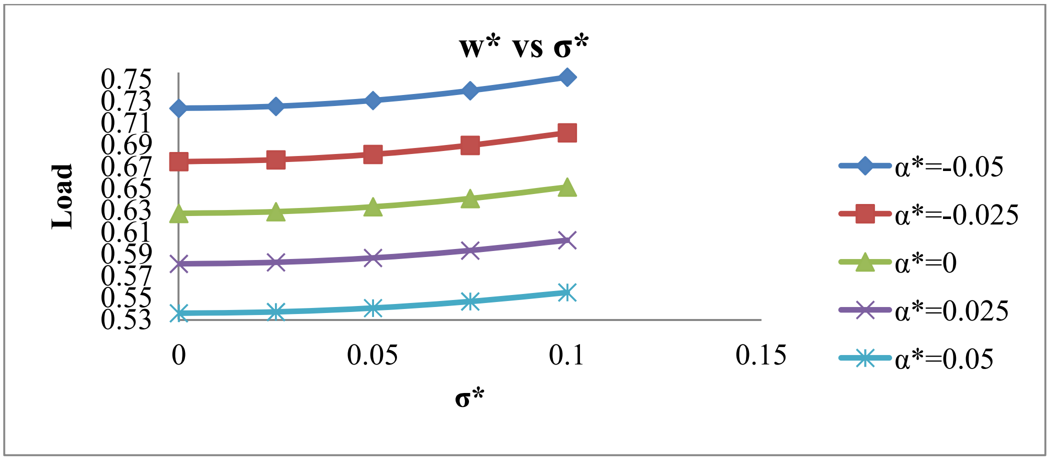


Fig.6. Variation of load carrying capacity with respect to σ*

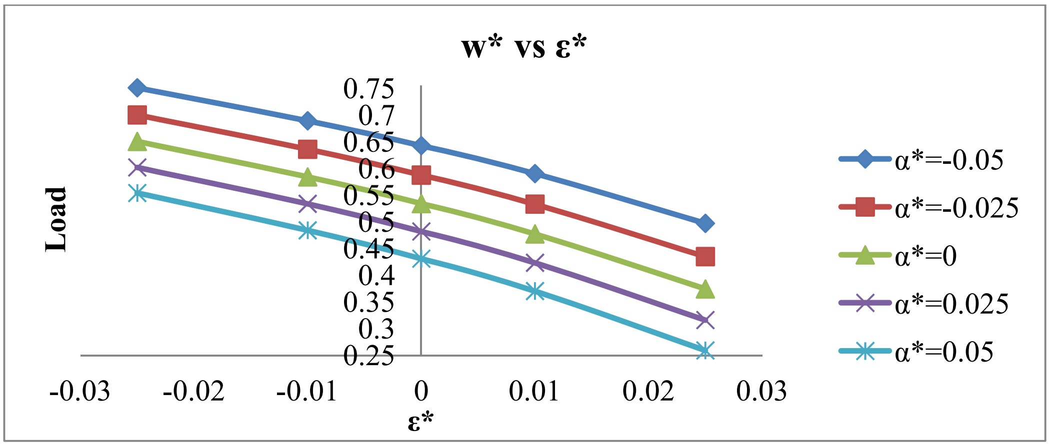


Fig.6. Variation of load carrying capacity with respect to ε^*

Conclusion

The investigation reveals that the magnetization may go a long way in overcoming the adverse effect of rougness. However, the study strongly indicates that the roughness aspest must be duly addressed while designing the bearing system,even if a suitable magnetic strenth is considered.

References

1. Rayleigh, Lord. "I. Notes on the theory of lubrication." The London, Edinburgh, and Dublin Philosophical Magazine and Journal of Science 35.205 (1918): 1-12.
2. Pinkus, Oscar, and Beno Sternlicht. Theory of hydrodynamic lubrication. McGraw-Hill, 1961.
3. A. Cameron: Basic Lubrication Theory, Wiley Eastern Ltd., pp. 62, 1987
4. Tzeng, S. T., and E. Saibel. "On the effects of surface roughness in the hydrodynamic lubrication theory of a short journal bearing." Wear 10.3 (1967): 179-184.
5. H. Christensen, K.C. Tonder: Tribology of rough surfaces: Stochastic models of hydrodynamic lubrication, SINTEF Report No.10/69-18, 1969.
6. H. Christensen, K.C. Tonder: Tribology of rough surfaces: Parametric study and comparison of lubrication models, SINTEF Report No. 22/69-18, 1969.
7. H. Christensen, K.C. Tonder: The hydrodynamic lubrication of rough bearing surfaces of finite width, ASME-ASLE lubrication conference, paper No.70-Lub-7, 1970.
8. Prakash, J., and K. Tiwari. "Lubrication of a porous bearing with surface corrugations." Journal of Tribology 104.1 (1982): 127-134.
9. Guha, S. K. "Analysis of dynamic characteristics of hydrodynamic journal bearings with isotropic roughness effects." Wear 167.2 (1993): 173-179.

10. Gupta, J. L., and G. M. Deheri. "Effect of roughness on the behavior of squeeze film in a spherical bearing." Tribology Transactions 39.1 (1996): 99-102.
11. Patir, Nadir, and H. S. Cheng. "An average flow model for determining effects of three-dimensional roughness on partial hydrodynamic lubrication." Journal of Tribology 100.1 (1978): 12-17.
12. Agrawal, V. K. "Magnetic-fluid-based porous inclined slider bearing." Wear 107.2 (1986): 133-139.
13. Bhat, M. V., and G. M. Deheri. "Porous composite slider bearing lubricated with magnetic fluid." Japanese journal of applied physics 30.10R (1991): 2513.
14. Shah, Rajesh C., S. R. Tripathi, and M. V. Bhat. "Magnetic fluid based squeeze film between porous annular curved plates with the effect of rotational inertia." Pramana 58.3 (2002): 545-550.
15. Andharia, P. I., J. L. Gupta, and G. M. Deheri. "Effect of longitudinal surface roughness on hydrodynamic lubrication of slider bearings." BOOK-INSTITUTE OF MATERIALS 668 (1997): 872-880.
16. Andharia, P. I., Gupta, J. L., &Deheri, G. M. (2001). Effect of surface roughness on hydrodynamic lubrication of slider bearings. Tribology transactions, 44(2), 291-297.
17. Deheri, G. M., P. I. Andharia, and R. M. Patel. "Longitudinally rough slider bearings with squeeze film formed by a magnetic fluid." Industrial Lubrication and Tribology 56.3 (2004): 177-187.
18. Bhat, M. V. "Hydrodynamic lubrication of a porous composite slider bearing." Japanese Journal of Applied Physics 17.3 (1978): 479.
19. Bhat, M. V., and C. M. Patel. "The squeeze film in an inclined porous slider bearing." Wear 66.2 (1981): 189-193.
20. Patir N, 1978, "Effects of surface roughness on Partial film lubrication using an average flow model based on numerical simulation", Northwestern University, Ph.D. Chapter-2.
21. Panchal, Girishkumar C., Himanshu C. Patel, and G. M. Deheri. "INFLUENCE OF SURFACE ROUGHNESS THROUGH A SERIES OF FLOW FACTORS ON THE PERFORMANCE OF A LONGITUDINALLY ROUGH FINITE SLIDER BEARING." Annals of the Faculty of Engineering Hunedoara-International Journal of Engineering 14.2 (2016).
22. Shukla, S. D., and G. M. Deheri. "Surface Roughness Effect on the Performance of a Magnetic Fluid Based Hyperbolic Slider Bearing." International Journal of Engineering Research and Applications 1: 948-962.
23. Jimit R. Patel, G.M. Deheri: A comparison of porous structures on the performance of a magnetic fluid based rough short bearing, Tribology in Industry, Vol. 35, No. 3, pp. 177-189, 2013.