Surface Roughness Effect on the Performance of a Magnetic Fluid Based Hyperbolic Slider Bearing

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Abstract:

An attempt has been made to study and analyze the performance of a rough hyperbolic slider bearing under the presence of a magnetic fluid lubricant. The magnetic field is oblique to the stator and the bearing surfaces are considered to be transversely rough. The roughness of the bearing surface is characterized by a stochastically random variable with non-zero mean, variance and skewness. The concerned Reynolds equation is stochastically averaged with respect to the random roughness parameters. This stochastically averaged equation is solved with suitable boundary conditions to obtain the expression of pressure distribution, which is then used to calculate the load carrying capacity. Besides, friction is also computed. The results show that the transverse surface roughness affects the performance of the bearing system adversely. It is seen that the load carrying capacity increases while friction decreases due to the magnetic fluid lubricant. It is revealed that the negative effect of transverse roughness can be minimized to certain extent by the positive effect of the magnetization in the case of negatively skew roughness. It is noticed that positively skewed roughness and variance (+ve) decrease the friction. Thus, this article offers some scopes for reducing the friction, increasing the load carrying capacity at the same time. This article may also be prevailed some measures for the extending the life period of the bearing system.

Keywords: hyperbolic slider bearing, magnetic fluid, roughness, load carrying capacity, friction

1. Introduction

Amongst the hydrodynamic bearings, slider bearing is the simplest and frequently encountered because the expression of film thickness is simple and boundary conditions to be required zero at the bearing ends are not that complicated. The fundamental aspect in a hydrodynamic slider bearing is the formation of a converging wedge of the lubricant. The hydrodynamic slider may be constructed to provide this converging wedge in a number of ways. [1] established that the shape of the wedge is not that important and all that mattered is in the aspect ratio.

The analysis of hydrodynamic lubrication of a non-porous slider is classical one, for instance, one can turn to [2] and [8]. In fact, the infinite long slider bearing is the idealization of a single sector shaped pad of a hydrodynamic thrust bearing. Such a bearing consists of a fixed or pivoted pad and a moving pad which may be plane, stepped, curved or composite shaped (such bearings are widely used in hydrodynamic generators and turbines). Because of the use of squeeze film slider bearing in clutch plates, automobile, transmissions and domestic appliances many investigations ([4] and [5]) dealt with the problem of a squeeze film slider bearing. Slider bearing has been studied for various film shapes. ([2], [6] and [8]) as slider bearing is often used for supporting transverse loads.

By now, it is a well know fact that the bearing surfaces, as particularly after having some run- in and wear develop roughness. Even, the contamination of the lubricant contributes to the roughness through chemical degradation of the surface as contribute the roughness. The roughness appears to be random in character which does not seem to follow any particular structural pattern. The randomness of the roughness was recognized by several investigators to analyze the effect of surface roughness ([9] - [17]). [13]- [15] proposed a comprehensive general analysis for surface roughness (both transverse as well as longitudinal) based on a general probability density function by developing the approach of [12].

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The method of [13] - [15] was deployed in many investigations to discuss the effect of surface roughness ([18] - [26]).

All the above studies dealt with a conventional lubricant. The magnetic fluid is prepared by suspending fine magnetic grains coated with a surfactant and dispersing it in a non-conducting and magnetically passive solvent, such as benzene, hydrocarbons and fluorocarbons. These magnetic fluids remain liquid in a magnetic field and after removal of a field recover their characteristics. Particles within the liquid experience a force due to the field gradient and move through the liquid imparting drag to it causing it to flow. The advantage of magnetic fluid lubricant over the conventional ones is that the former can be retained at the desired location by an external magnetic field [32].

[27] and [28] presented the squeeze film performance by taking a magnetic fluid as a lubricant. [29] extended the analysis of ([27]-[28]) to analyze the squeeze film behavior between porous annular disks using a magnetic fluid lubricant with the external magnetic field oblique to the lower disk. Here, it was concluded that the application of magnetic fluid lubricant enhanced the performance of squeeze film bearing system. Subsequently, [30] modified the analysis of [28] by considering a magnetic fluid based porous composite slider bearing with its slider consisting of an inclined pad and a flat pad. These discussions suggested that the magnetic fluid lubricant unaltered the friction and shifted the centre of pressure towards the inlet. Moreover, the load carrying capacity registered a sharpe rise due to the magnetic fluid lubricant. Also, [31] considered the hydrodynamic lubrication of a porous slider bearing and compared the performance by taking the various geometrical shapes. Here, it was observed that mostly, the magnetic fluid lubricant resulted was increased load carrying capacity and shifted the centre of pressure towards the outlet edge. [33] extended the analysis of [7] to develop a mathematical model, which study the effect of slip velocity on a porous secant shaped slider bearing with ferro fluid lubricant using Jenkins model. [34] analyzed the performance of transversely rough slider bearing with squeeze film formed by a magnetic fluid. Here, it was shown that the increase in the load carrying capacity due to the magnetic fluid lubricant enhanced the performance of the squeeze film bearing system. This performance was more pronounced, when suitable values of magnetization parameters were taken into the consideration. This study included the performance of a magnetic fluid based hyperbolic slider bearing.

Here, it has been proposed to study and analyze the performance of a magnetic fluid based rough hyperbolic slider bearing.

2. Analysis

The geometry and configuration of the bearing system is given in Fig.1

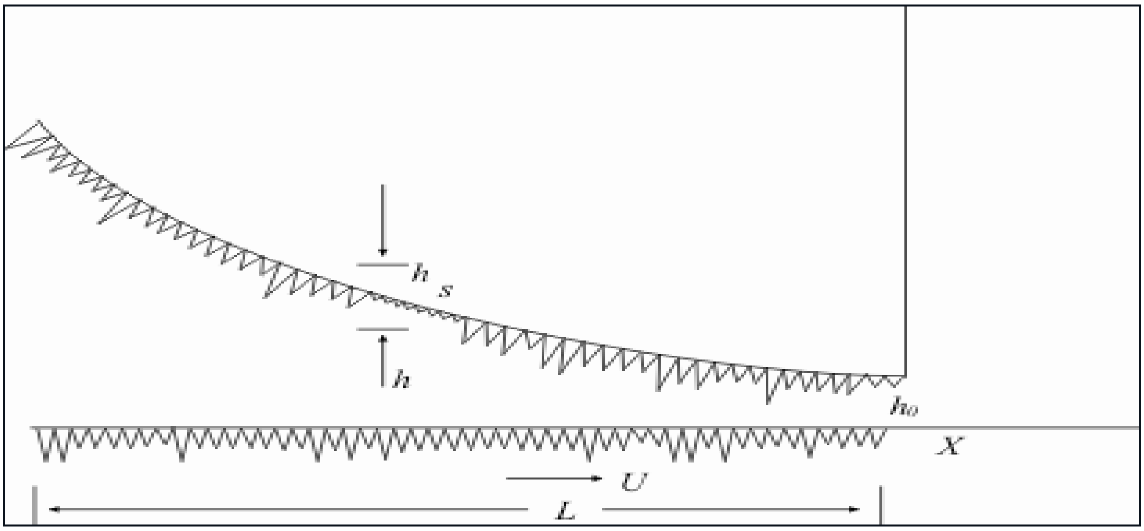

Fig.1 The configuration of the bearing system

The bearing surfaces are assumed to be transversely rough. The thickness $h(x)$ of the lubricant film is taken as ([13-15]).

$$h(x) = \bar{h}(x) + h_s$$

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. Here h_s is assumed to be stochastic in nature and governed by the probability density function $F(h_s)$, which is defined by

$$F(h_s) = \begin{cases} \frac{32}{35b}\left(1 - \frac{h_s^2}{b^2}\right)^3 & -b \le h_s \le b, \\ 0, & \text{elsewhere;} \end{cases}$$

where b is the maximum deviation from the mean film thickness. The mean α, standard deviation σ and the parameter ε which is the measure of symmetry associated with the random variable h_s, are determined by the relations

$$\alpha = E(h_s);$$

$$\sigma^2 = E[(h_s - \alpha)^2];$$

$$\varepsilon = E[(h_s - \alpha)^3]$$

Where E denotes the expected value defined by

$$E(R) = \int_{-b}^{b} RF(h_s)dh_s$$

The magnetic field is oblique to the stator and its magnitude is given by,

$$H^2 = kx(x + L)$$

Where, $k = 10^{14}A^2m^{-4}$ chosen so as to have a magnetic field of strength over 10^5 ([26], [28] and [31]).

Taking in to account the usual assumptions of hydro magnetic lubrication with the aid of the modeling resorted to by ([25] - [26]),

$$\frac{d}{dx}\left(p - \frac{\mu_0\bar{\mu}H^2}{2}\right) = 6U\eta\frac{h - \bar{h}}{A(h)} \qquad (1)$$

$$A(h) = h^3 + 3\alpha h^2 + 3(\sigma^2 + \alpha^2)h + \varepsilon + 3\sigma^2\alpha + \alpha^3$$

The concerned boundary conditions are,

$$P = 0, \qquad x = -L \text{ and } x = 0$$

Introducing the dimensionless quantities

$$x^* = \frac{x}{L}, P^* = \frac{h_0^{\,3}}{U\eta L^2}P, \mu^* = \frac{h_0^{\,3}\;K\mu_0\bar{\mu}}{U\eta}, \varepsilon^* = \frac{\varepsilon}{h_0^3}, \sigma^* = \frac{\sigma}{h_0},$$
$$\alpha^* = \frac{\alpha}{h_0}, \quad A_1 = \varepsilon^* + 3\sigma^{*2}\alpha^* + \alpha^{*3}, A_2 = 3\left(\sigma^{*2} + \alpha^{*2}\right), A_3 = 3\alpha^* \qquad (2)$$

and fixing the following symbols;

$$m_1 = A_2 - A_3 + 1, E = A_2 - A_1, F = A_1 - A_2 + A_3, \quad E = A_2 - A_1, F = A_1 - A_2 + A_3, G = A_1 - E$$

$$K_1 = ln\left(\frac{2}{2 - cL}\right) + \frac{E}{2A_1}\; ln\left(\frac{A_1 + E + F}{A_1(1 - cL)^2 + E(1 - cL) + F}\right)\frac{(E^2 - 2A_1F)}{\sqrt{A_1}\sqrt{4A_1F - E^2}}$$
$$\times tan^{-1}\left(\frac{cL\sqrt{4A_1F - E^2}}{cLE + 2A_1cL - 2A_1 - 2E - 2F}\right)$$

$$K_2 = GCL + m_1 \ln\left(\frac{2-cL}{2}\right) \times \frac{(GA_3)}{2A_1} \ln\left(\frac{A_1(1-cL)^2 + E(1-cL) + F}{A_1 + E + F}\right)$$
$$-\left((G-m_1) - E\frac{(GA_3 - m_1 E)}{2A_1}\right) \times \frac{2\sqrt{A_1}}{\sqrt{4A_1F - E^2}}$$
$$\times \tan^{-1}\left(\frac{cL\sqrt{4A_1F - E^2}}{CLE + 2A_1cL - 2A_1 - 2E - 2F}\right)$$

$$Q_1 = K_2 + K_1 m_1, \qquad Q_2 = \frac{k_1(GA_3 - m_1E)}{2A_1} - \frac{EK_2}{2A_1} \qquad Q_4 = -K_1 G$$

$$Q_3 = \left[\frac{K_2(E^2 - 2A_1F)}{\sqrt{A_1}\sqrt{4A_1F - E^2}} + K_1\left((G-m_1)\frac{E(GA_3 - m_1E)}{2A_1}\right) \times \frac{2\sqrt{A_1}}{\sqrt{4A_1F - E^2}} \tan^{-1}\left(\frac{2A_1(1+cL) + E}{\sqrt{4A_1F - E^2}}\right)\right]$$

The pressure distribution in non-dimensional form comes out be

$$P^* = \frac{\mu^*}{2} + 6\left(Q_1 \ln\left(\frac{2+x^*}{2}\right) + Q_2 \ln\left(\frac{A_1(1+x^*)^2 + E(1+x^*) + F}{A_1 + E + F}\right)\right.$$

$$\left. + Q_3 \tan^{-1}\left(\frac{cx^*\sqrt{4A_1F - E^2}}{Ex^* + 2A_1x^* - 2A_1 - 2E - 2F}\right) + Q_4 x^*\right) \tag{3}$$

The dimensionless load carrying capacity of the bearing is given by,

$$W^* = \frac{h_0^3}{U\eta L^4} W$$
$$= \int_{-1}^{0} P^* \, dx$$

which results in

$$= \frac{\mu^*}{12} + W_1 + W_2 + W_3 + W_4 \tag{4}$$

where

$$W_1 = Q_1\left[(2-x^*)\ln\left(\frac{2-x^*}{2}\right)\right] + x^* \qquad W_4 = Q_4\left[\frac{(x^*)^2}{2}\right]$$

$$W_2 = Q_2\left[(1-x^*) + \frac{A_3 - 1}{2A_1} \times \ln\left(\frac{A_2 + 1}{A_1(1-x^*)^2 + (A_3 - 1)(1-x^*) + 1}\right) - 2x^*\right.$$
$$\left. + \frac{\sqrt{4A_1 - (A_3-1)^2}}{\sqrt{A_1}} \tan^{-1}\left(\frac{x^*\sqrt{4A_1 - (A_3-1)^2}}{(A_3-1)(x^*-2) + 2A_1(x^*-1) - 2}\right)\right]$$

$$W_3 = Q_3\left[\left(\frac{2A_1 + A_3 - 1 - 4A_1x^*}{4A_1}\right) \times \tan^{-1}\left(\frac{x^*\sqrt{4A_1 - (A_3-1)^2}}{(A_3-1)(x^*-2) + 2A_1(x^*-1) - 2}\right)\right.$$

$$\left. - \frac{\sqrt{4A_1F - (A_3-1)^2}}{4\sqrt{A_1}} \times \ln\left(\frac{A_2 + 1}{A_1(1-x^*)^2 + (A_3-1)(1-x^*) + 1}\right)\right]$$

The total friction F is,

$$F = \int_{-L}^{0} \tau \, dx$$

Now friction

$$= L\int_{-L}^{0}\eta\left(\frac{\partial u}{\partial z}\right)\,dx$$

Where,

$$\left(\frac{\partial u}{\partial z}\right) = \frac{\partial p}{\eta\partial x}\left(z-\frac{h}{2}\right)+\frac{U_1-U_2}{h}$$

The friction force is needed on the two surfaces $z = h$ and $z = 0$.Writing F_h for the first term and F_0 for the second, one observer the form to be

$$\frac{F_{h,0}}{L} = \int_{-L}^{0}\left[\pm\left(\frac{\partial}{\partial x}\left(p-\frac{\mu_0\bar{\mu}H^2}{2}\right)\frac{h}{2}\right)+\frac{(U_1-U_2)\eta}{h}\right]dx$$

It is shown that the upper pad is titled there will be a horizontal component of force, which exactly equals the difference in frictional drag between the top and bottom surfaces, if this term is added to the friction of the upper surface F_h it cancels the minus sign. The details can be seen from [3]. It gives directly non-dimensional friction

$$F^* = \frac{ch_0F}{U\eta L}$$

$$= \frac{\mu^*}{8}+F_1+F_2+F_3+F_4 \quad (5)$$

where

$$F_1 = \frac{Q_1}{2}\left[2\,ln\left(\frac{2-x^*}{2(1-x^*)}\right)-\frac{x^* ln2}{1-x^*}\right], \quad F_4 = \frac{Q_4}{2}\left[-ln(1-x^*)-\frac{x^*}{1-x^*}\right]+x^*(1-x^*)+\frac{(x^*)^3}{3}$$

$$F_2 = \frac{Q_2}{2}\left[-\frac{E}{F}ln(1-x^*)-\frac{E}{2F}\times ln\left(\frac{A_1+E+F}{A_1(1-x^*)^2+E(1-x^*)+F}\right)\right.$$

$$\left.-\sqrt{A_1}\sqrt{4A_1F-E^2}\times tan^{-1}\left(\frac{x^*\sqrt{4A_1F-E^2}}{Ex^*+2A_1x^*-2A_1-2E-2F}\right)+\left(\frac{x^*}{1-x^*}\right)ln(A_1+E+F)\right]$$

$$F_3 = \frac{Q_3}{2}\left[\frac{1}{F}ln(1-x^*)-\frac{1}{2F}\times ln\left(\frac{A_1+E+F}{A_1(1-x^*)^2+E(1-x^*)+F}\right)+\frac{E}{F}\frac{\sqrt{A_1}}{\sqrt{4A_1F-E^2}}\right.$$

$$\left.\times\ tan^{-1}\left(\frac{x^*\sqrt{4A_1F-E^2}}{Ex^*+2A_1x^*-2A_1-2E-2F}\right)\quad +tan^{-1}\left(\frac{2A_1+E}{\sqrt{4A_1F-E^2}}\right)\frac{x^*}{1-x^*}\right]$$

3. Results and discussion

It is clearly observed that the dimensionless pressure distribution is presented by equation (3), while the profile of load carrying capacity in non-dimensional form is determined by equation (4). In addition, equation (5) gives the friction in non-dimensional form. It is seen that the dimensional less pressure increases by (0.5) μ^* while increase in load carrying capacity comes out to be (0.08) μ^* as compared to the conventional lubricant. The performance of the corresponding magnetic fluid based squeeze film for the smooth bearing system can be obtained by assuming the roughness parameters to be zero.

Figures (2) – (5) dealing with the variation of load carrying capacity with respect to μ^* for various values of $\sigma^*, \varepsilon^*, \alpha^*$ and x^* suggest as the load carrying capacity increases significantly due to the magnetic fluid lubricant. This increase is relatively less in case of σ^*. Figures (6)-(8) present the variation of non-dimensional load profile with respect to σ^* for different values of ε^*, α^* and x^* respectively. It can be easily seen that the effect of standard deviation is considerably adverse, in the sense that the load carrying capacity decreases considerably due to σ^*.

The combined effect of ε^* *and* α^* on the distribution of load carrying capacity is presented in Figure (9). Here, it is observed that the positively skewed roughness decreases the load carrying capacity, while load carrying capacity increases sharply due to the negatively skewed roughness. The trends of α^* is identically with that of trends of ε^*. Thus, the combined effect of negatively skewed roughness and (-ve) variance is significantly positive. Further, decrease the load carrying capacity due to variation (+ve) gets further decrease owing to positively skewed roughness.

The variation of non-dimensional friction can be seen from the profile presented in Figures (12)-(21). It is observed that the standard deviation and magnetization decrease the friction while negatively skewed roughness increases the friction. Further, positively skewed roughness and variance (+ve) decrease the friction. However, the friction decreases more rapidly in the case of x^* which can be seen from figure (15). Interesting, it is noticed that friction decreases considerably due to the magnetic fluid lubricant unlike the case of exponential slider bearing. A close glance at some of the figures reveals that the negative effect of transverse surface roughness can be minimized by the positive effect of magnetization in the case of negatively skewed roughness especially, when negative variance occurs.

4. Conclusion

This investigation establishes that the bearing can support a load even when there is no flow. Besides, this study makes it clear that the roughness must be given due considerations while designing the bearing system even, if suitable values of magnetization is considered. This is all the more necessary from bearing's life period point of view

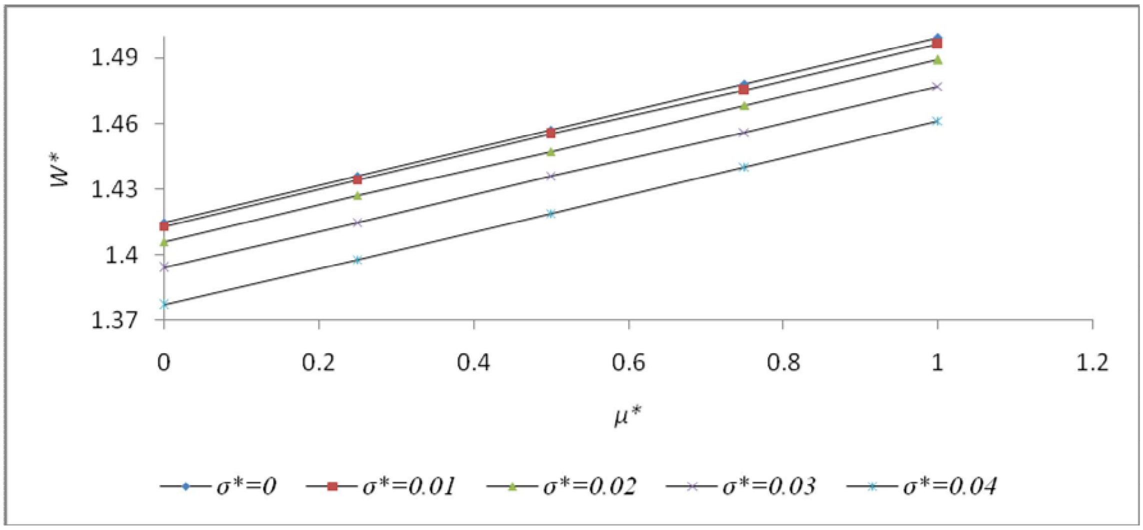

Fig.2 The variation of load carrying capacity with respect to μ^* and σ^*

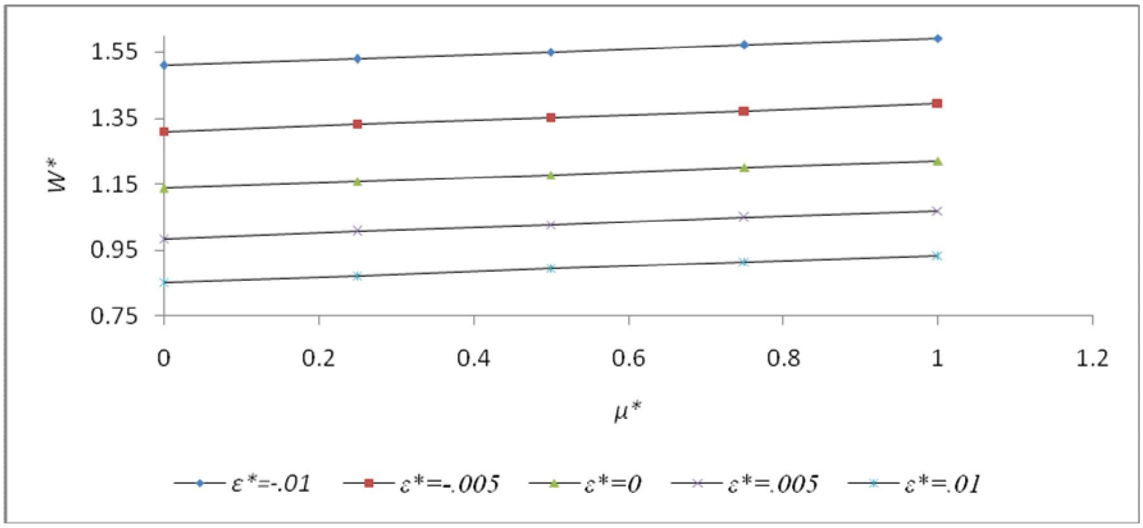

Fig.3 The variation of load carrying capacity with respect to μ^* and ε^*

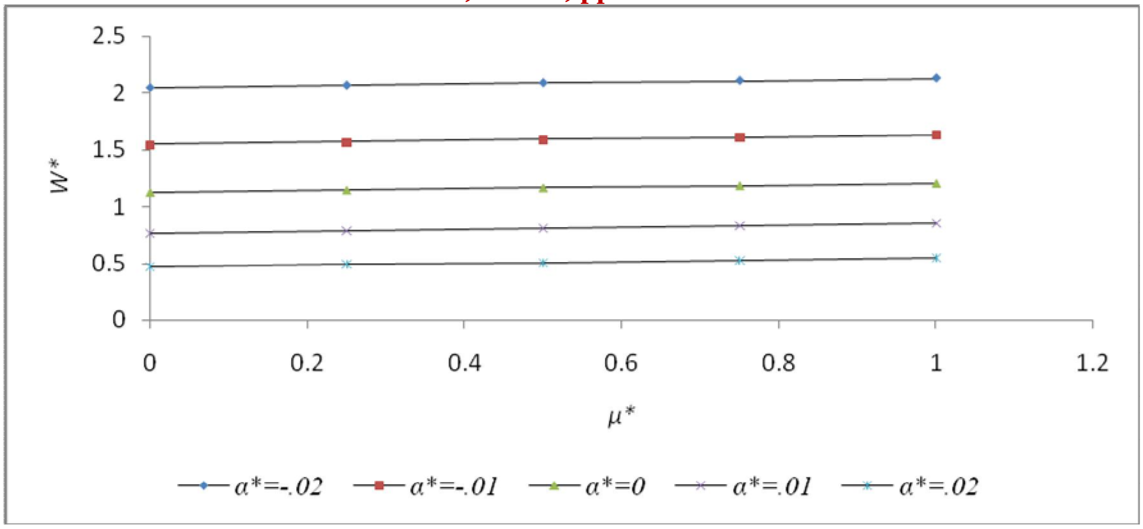

Fig.4 The variation of load carrying capacity with respect to μ^* and α^*

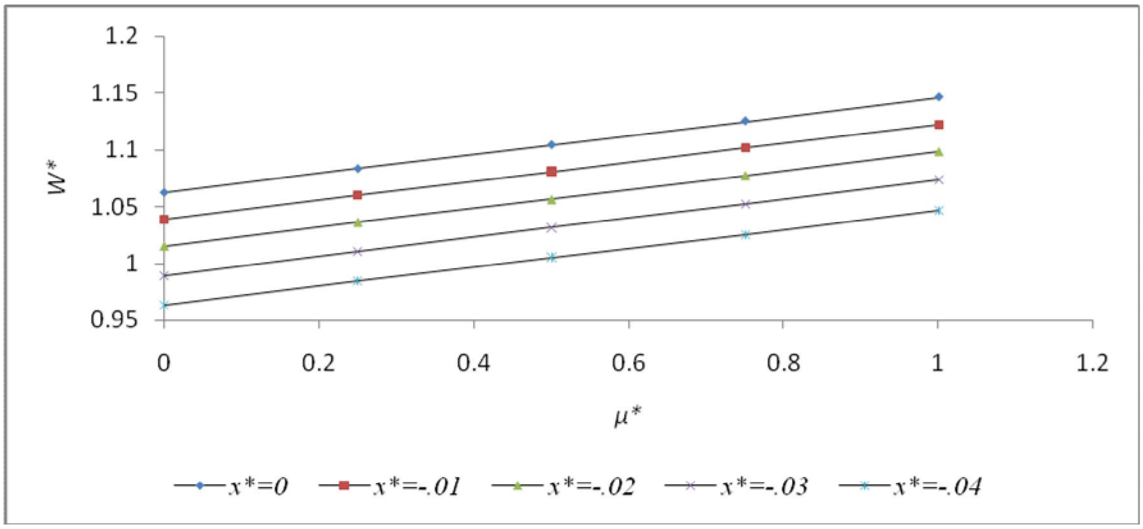

Fig.5 The variation of load carrying capacity with respect to μ^* and x^*

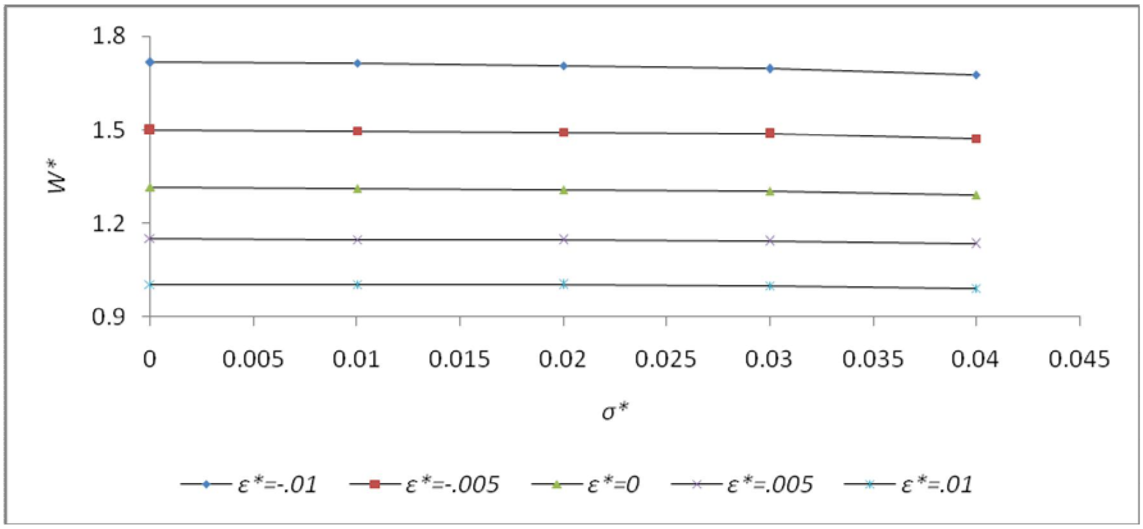

Fig.6 The variation of load carrying capacity with respect to σ * and ε *

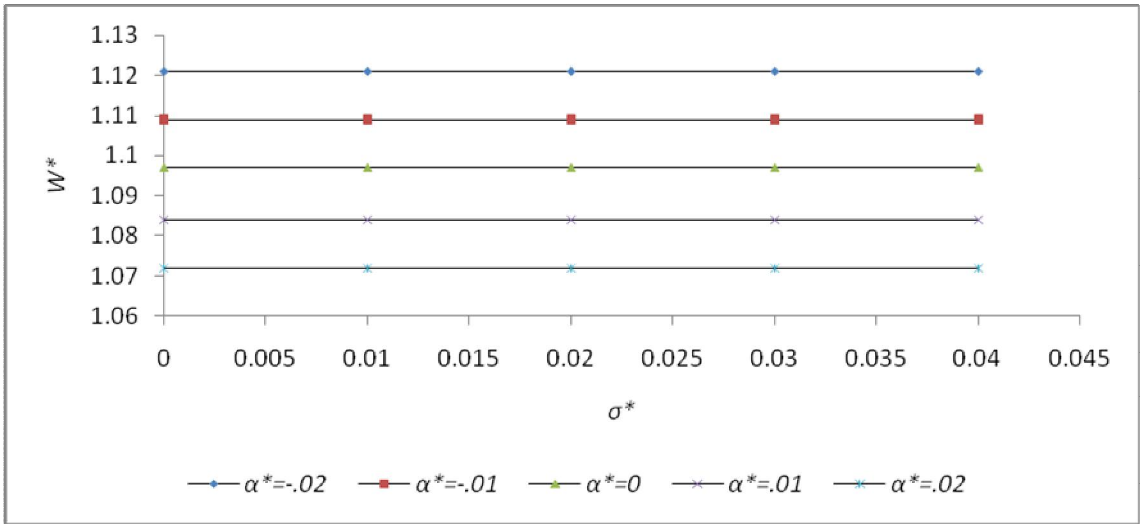

Fig.7 The variation of load carrying capacity with respect to σ * and α *

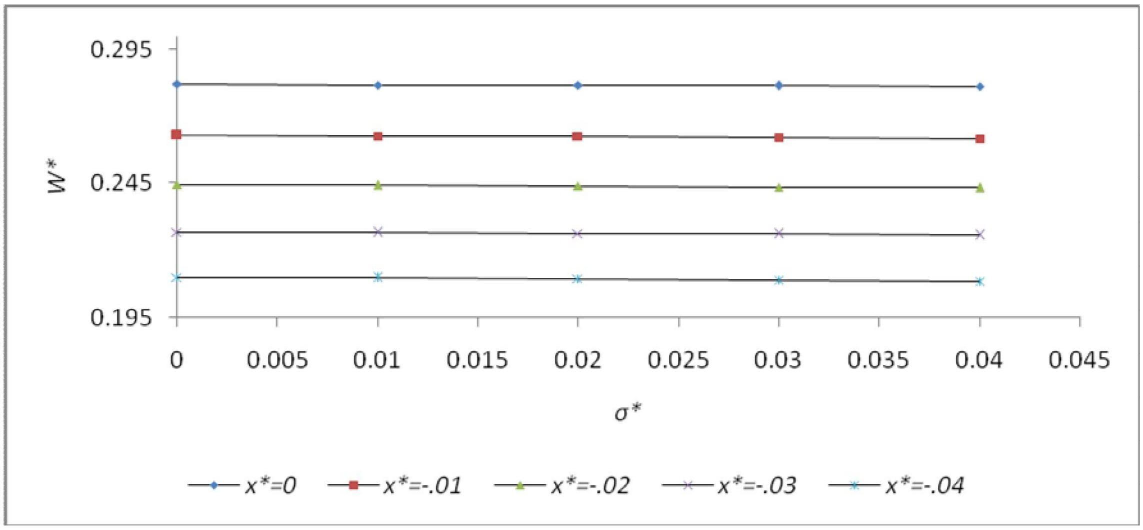

Fig.8 The variation of load carrying capacity with respect to σ * and x *

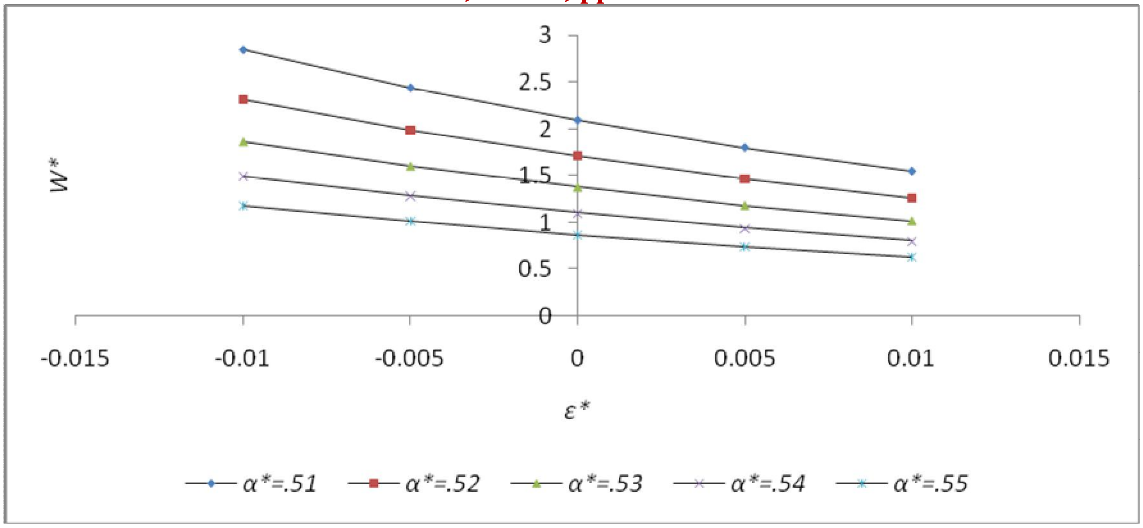

Fig.9 The variation of load carrying capacity with respect to ε * and α*

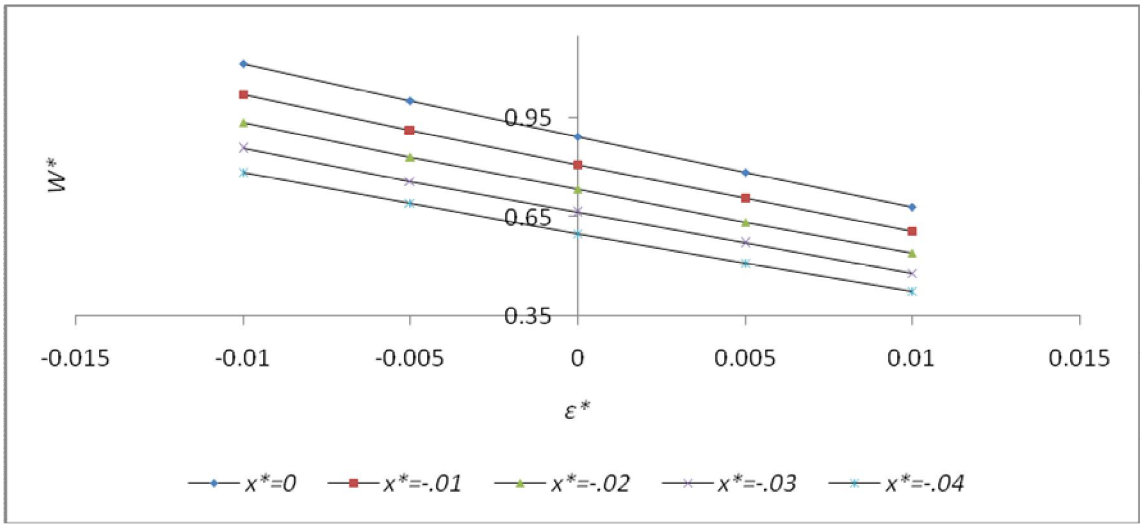

Fig.10 The variation of load carrying capacity with respect to ε * and x*

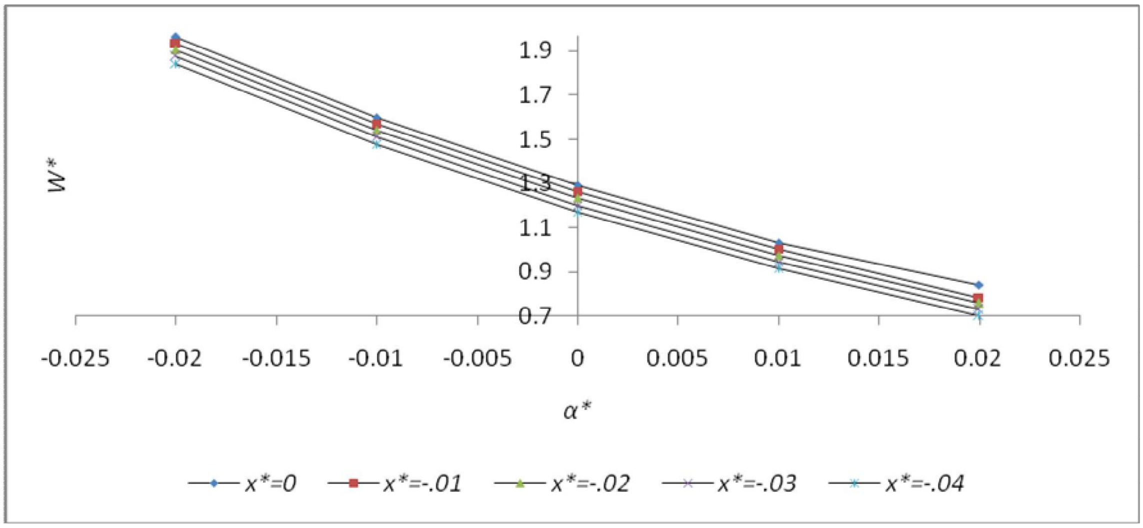

Fig.11 The variation of load carrying capacity with respect to α * and x*

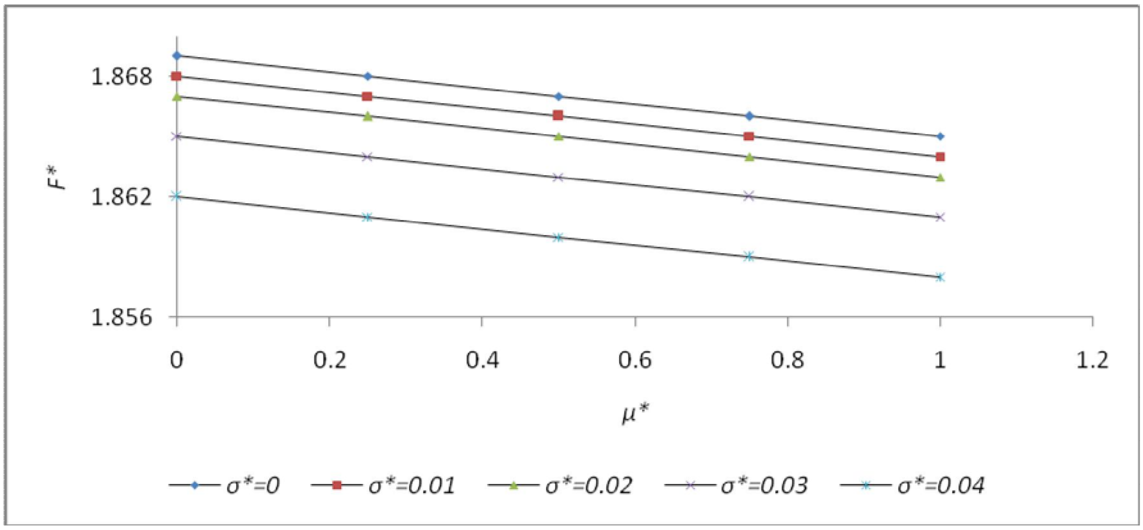

Fig.12 The variation of friction with respect to μ* and σ*

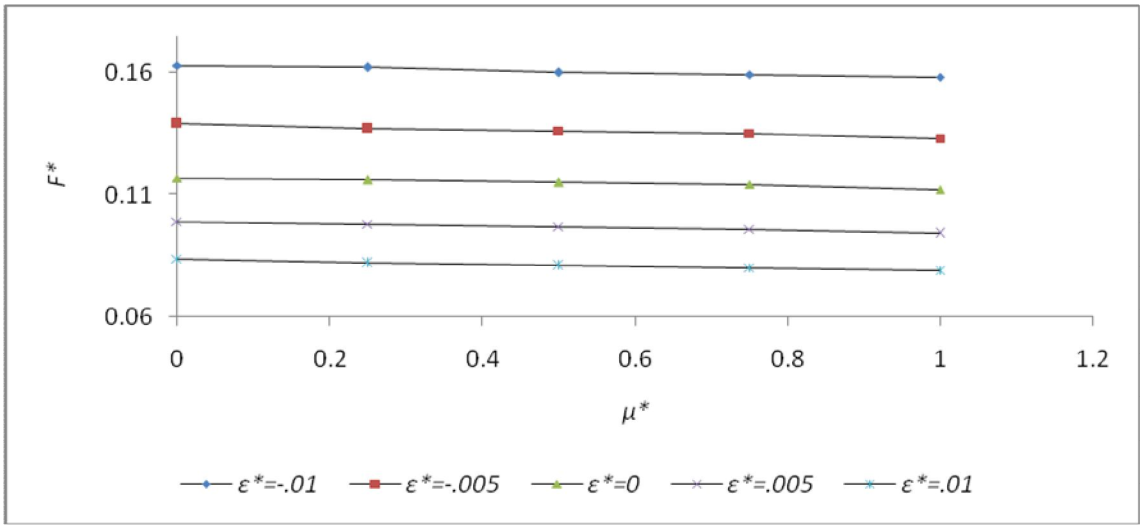

Fig.13 The variation of friction with respect to μ* and ε*

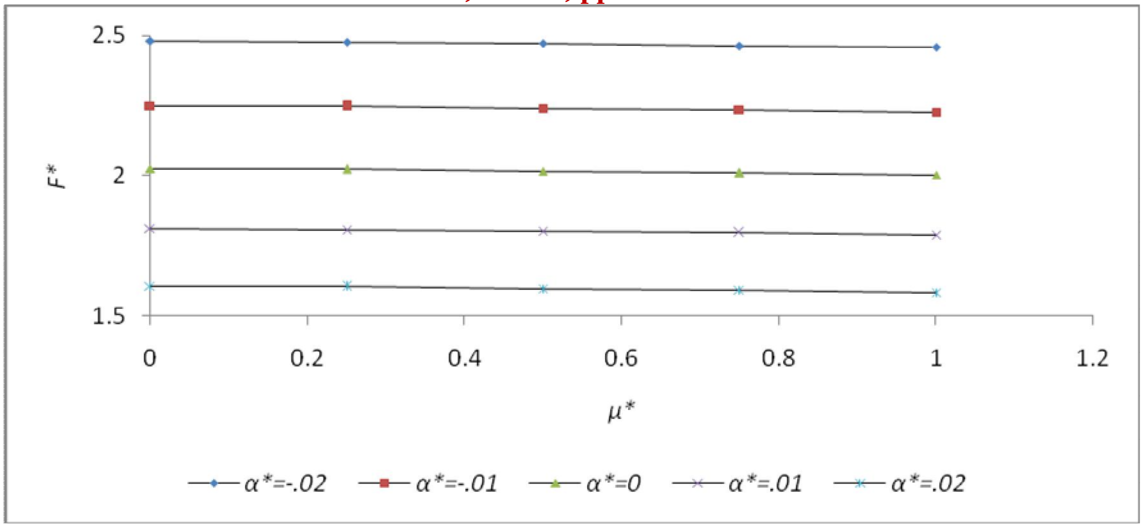

Fig.14 The variation of friction with respect to μ^* and α^*

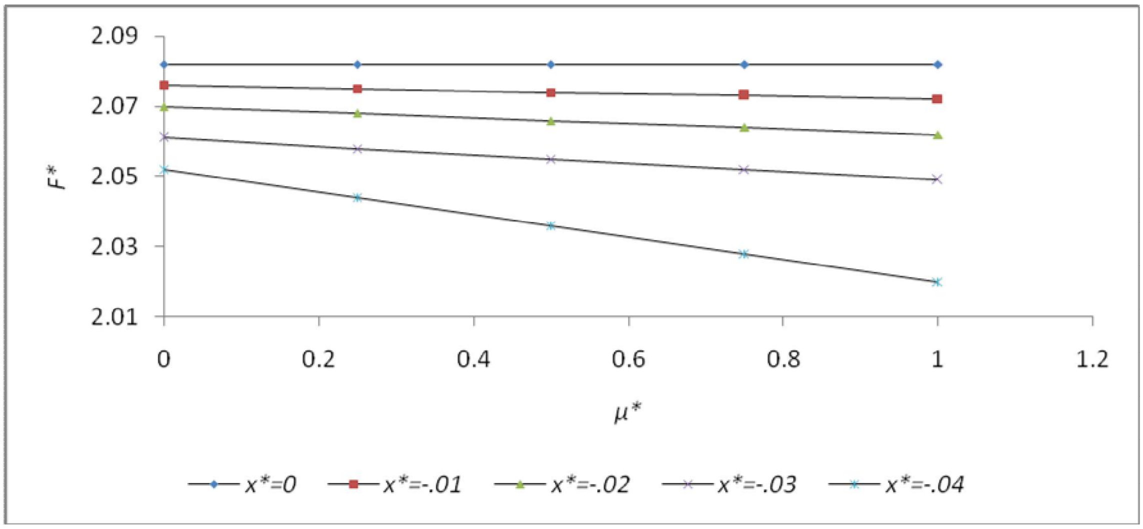

Fig.15 The variation of friction with respect to μ^* and x^*

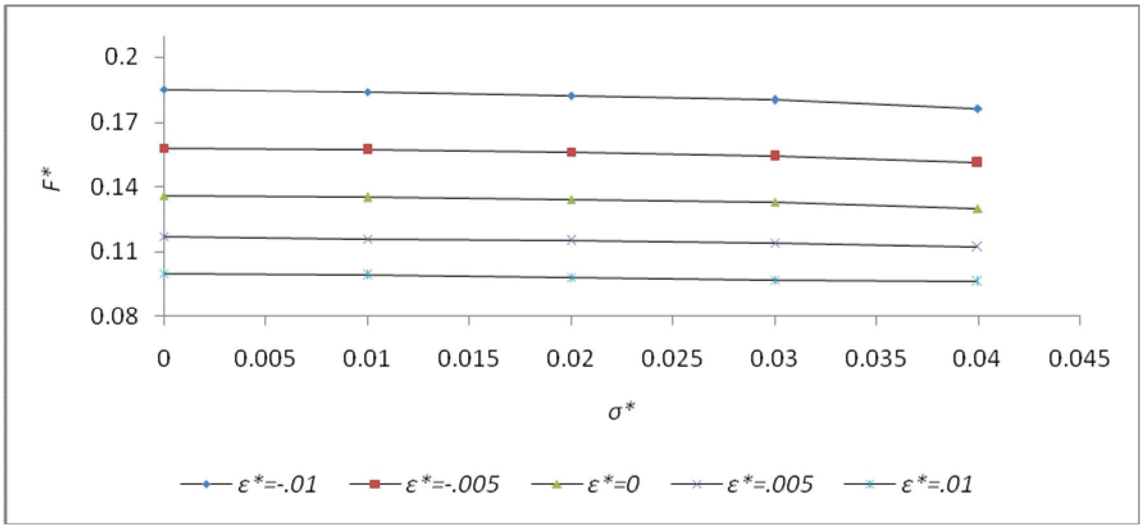

Fig.16 The variation of friction with respect to σ * and ε *

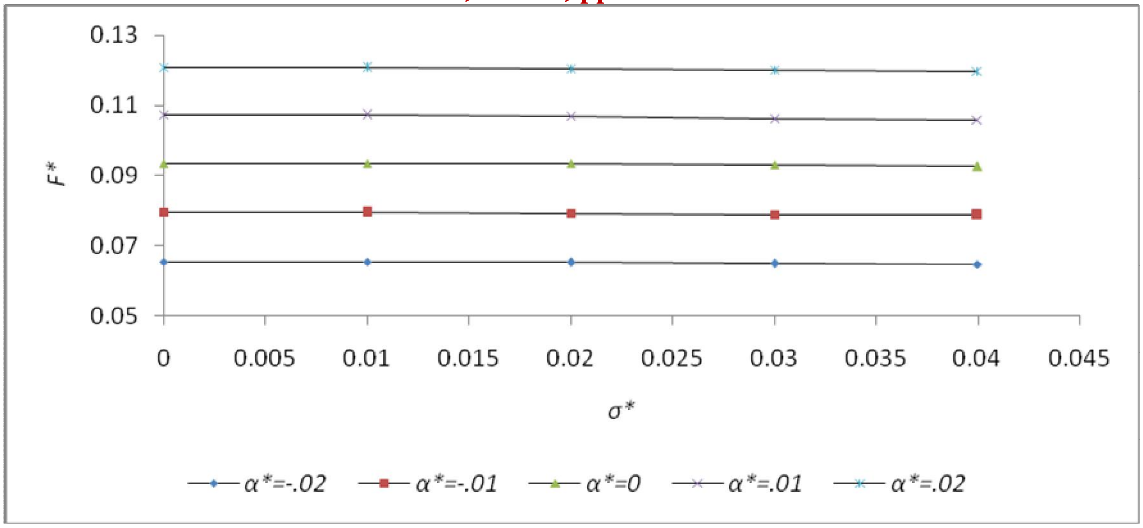

Fig.17 The variation of friction with respect to σ * and α *

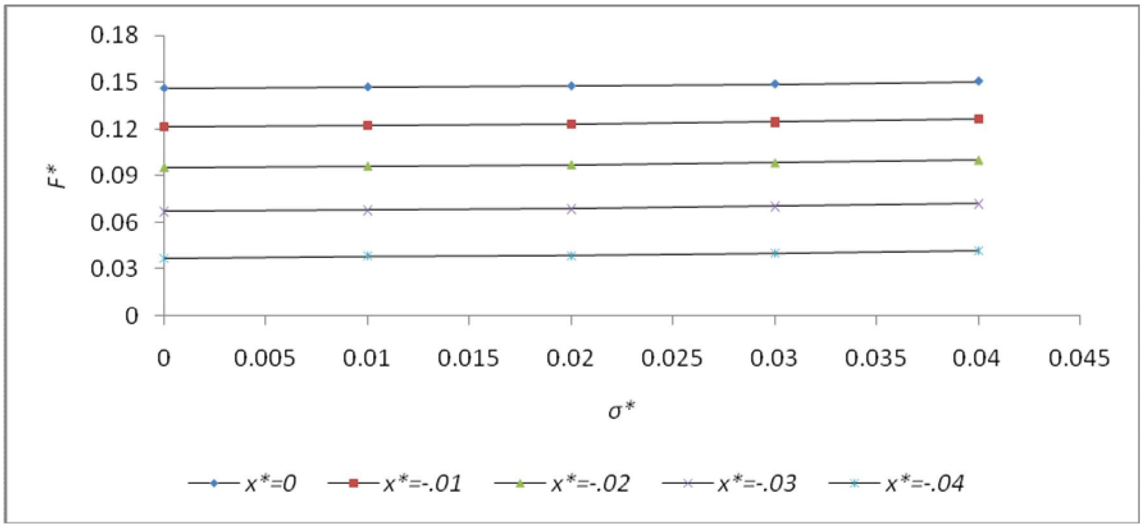

Fig.18 The variation of friction with respect to σ * and x *

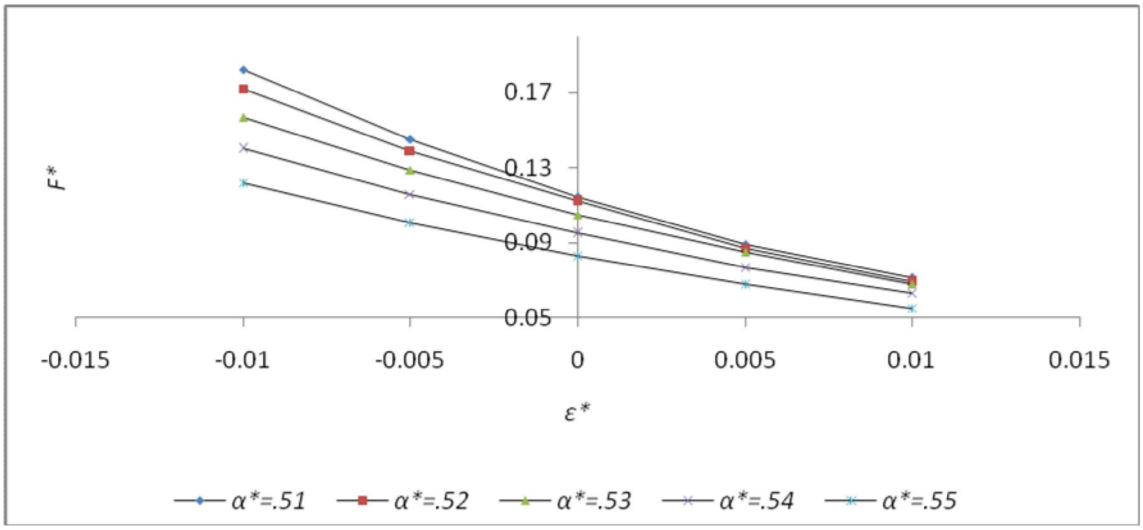

Fig.19 The variation of friction with respect to ε * and α*

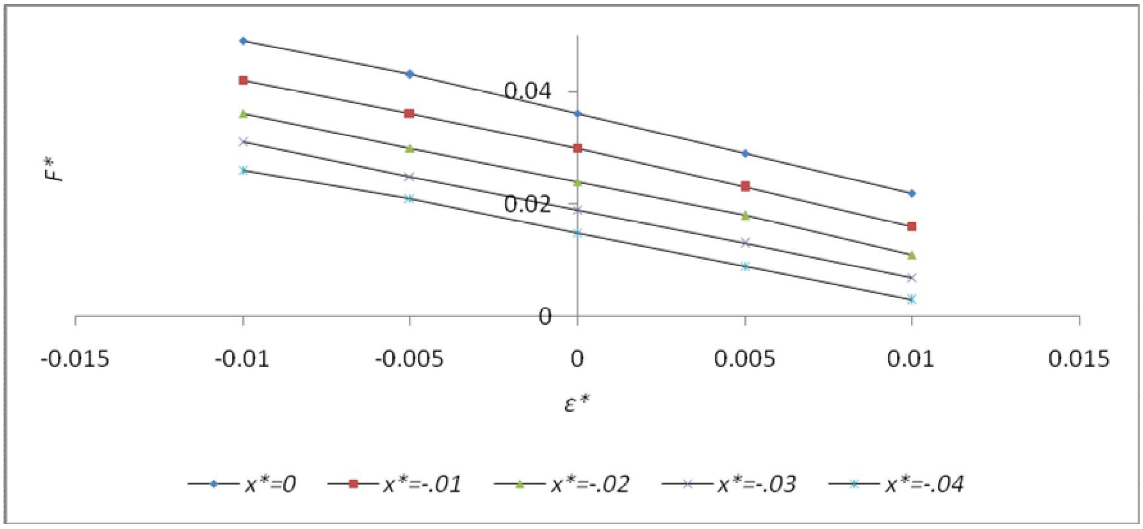

Fig.20 The variation of friction with respect to ε * and x*

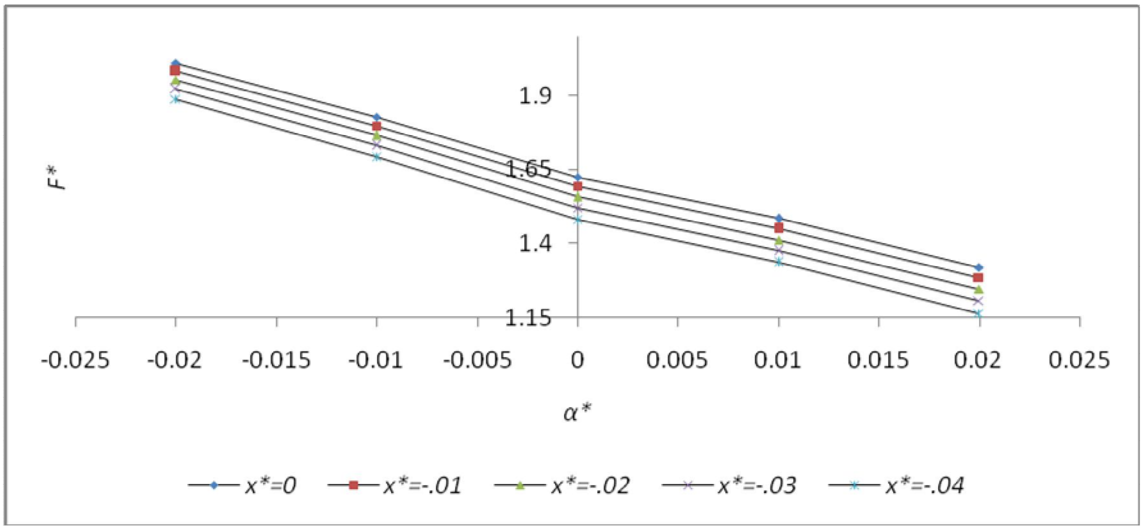

Fig.21 The variation of friction with respect to α * and x*

References

[1] Purday, H.P.F. (1949). *Steam Line Flow, constable,* London.
[2] Pinkus, O. and Stemlitch,B. (1961), *Theory of Hydrodynamic Lubrication,* McGraw Hill Book Company , New-York.
[3] Cameron, A., (1972). *Basic Theory of Lubrication,* Ellis Harwood Limited, Halsted Press, John Willy & Sons, New-York.
[4] Prakash,J. and Vij,S.K. (1973). "Hydrodynamic lubrication of a porous slider", *Journal of .Mechanical Engineering and science,* Vol.15, pp.222-234.
[5] Bhat,M.V. and Patel,C.M. (1981)."The squeeze film in an inclined porous slider bearing", *WEAR,* Vol.66, pp.189-193.
[6] Bagci,C. and Singh,A.P. (1983), "Hydrodynamic lubrication of a finite slider bearing, effect of one dimensional film shape and their computer aided optimum designs", *ASLE Transaction journal of Lubrication and Technology* , Vol.105, pp.48-66 .
[7] Ajwaliya, M.B.,.(1984),"Hydrodynamic lubrication of a secant shaped porous slider, Thesis, submitted to S.P. University, India.

[8] Hamrock,B.J. (1994). *Fundamentals of fluid film lubrication*, McGraw Hill Book Company, New-York.

[9] Mitchell,A.G.M (1950). *Lubrication its principle and practice*, Blackie, London.

[10] Davis,M.G. (1963). "The generation of pressure between rough fluid lubricated moving deformable surfaces", *Lubrication of Engineering*, Vol.19, p.-246.

[11] Burton,R.A. (1963). "Effect of two dimensional sinusoidal roughness on the load support characteristics of a lubricant film", *Journal of Basic Engineering Transaction of .ASME*, Vol.85, pp.258-264.

[12] Tzeng,S.T. and Saibel,E. (1967). "Surface roughness effect on slider bearing lubrication", *Transaction of ASME, Journal of Lubrication and Technology*, Vol. 10, pp.334-338.

[13] Christensen,H. and Tonder,K.C. , 1969(a) Tribology of rough surfaces: Stochastic models of hydrodynamic lubrication. Technical Report No. SINTEF 10/69-18

[14] Christensen,H. and Tonder,K.C. , 1969(b). Tribology of rough surfaces: parametric study and comparison of lubrication models, Technical Report No. SINTEF 22/69-18

[15] Christensen,H. and Tonder,K.C. , 1970. The hydrodynamic lubrication of rough bearing surfaces of finite width. ASME-ASLE Lubrication conference, Cincinnati, Ohio. Paper No.70, Lub-7.

[16] Tonder,K.C. (1972). "Surface distributed waviness and roughness", *Proc. First world conference in industrial Tribology* , New-Delhi (India) Vol.3A, pp.01-08.

[17] Berthe,D. and Godet,M. (1973). "A More general form of Reynolds equation application to rough surfaces", *WEAR*, Vol.27, pp.345-357.

[18] Ting,L.L. (1975). "Engagement behavior of lubricated porous annular disks Part-1 squeeze film phase surface roughness and elastic deformation effects", *WEAR*, Vol.34, pp.159-182.

[19] Prakash, J. and Tiwari,K. (1982). "Lubrication of porous bearing with surface corrugations", *Journal of Lubrication and Technology Transaction ASME*, Vol.104, pp.127-134.

[20] Prakash, J. and Tiwari,K. (1983). "Roughness effect in porous circular squeeze plate with arbitrary wall thickness", *Journal of Lubriction and Technology Transaction ASME*, Vol. 105, pp.90-95.

[21] Prajapati, B.L. (1991). "Behavior of squeeze film between rotating porous circular plates; surface roughness and elastic deformation effects", *Pure and Applied Mathematical Science*, Vol.33, No.1-2, p-27-36.

[22] Prajapati , B.L. (1992). "Squeeze behavior of between rotating porous circular plates with a concentric circular pocket: surface roughness and elastic deformation effects", *WEAR*, Vol.152, p-301-307.

[23] Guha, S.K. (1993). "Analysis of dynamic characteristics of hydrodynamic journal bearing with isotropic roughness effect.", *WEAR*, Vol.167, pp.173-179.

[24] Gupta,J.L. and Deheri,G.M. (1996). "Effects of roughness on the behavior of squeeze film in a spherical bearing", *Tribology Transactions*, Vol.39, pp.99-102.

[25] Andharia,P.I. , Gupta,J.L. and Deheri,G.M. (1997). "Effects of longitudinal surface roughness on hydrodynamic lubrication of slider bearing", *Proc. Tenth International conference on surface modification Technologies*, The Institute of materials, London, pp.872-880.

[26] Andharia,P.I. , Gupta,J.L. and Deheri,G.M. (1999). "Effects of transverse surface roughness on the behavior of squeeze film in a spherical bearing", *Journal of Applied Mechanics and Engineering*, Vol.4, pp.19-24.

[27] Verma,P.D.S. (1986). "Magnetic fluid based squeeze film", *International Journal of Engineering Science*, Vol.24, No.3, pp.395-401.

[28] Agrwal,V.K. (1986). "Magnetic fluid based porous inclined slider bearing", *WEAR*, Vol. 107, pp.133-139.

[29] Bhat,M.V. and Deheri,G.M. (1991). "Squeeze film behavior in porous annular discs lubricated with magnetic fluid", *WEAR* , Vol.151, pp.123-128.

[30] Bhat,M.V. and Deheri,G.M. (1991). "Porous composite slider bearing lubricated with magnetic fluid", *Japanese Journal of Applied Physics*, Vol.30, pp.2513-2514.

[31] Bhat,M.V. and Deheri,G.M. (1995). "Porous slider bearing with squeeze film formed by a magnetic fluid", *Pure and Applied Mathematical Sciences*, Vol.39, No. 1-2 , pp.39-43.

[32] Bhat,M.V. (2003). *Lubrication with a magnetic fluid*, Team spirit(India) Pvt.Ltd.

[33] Shah,R.C., and M.V.Bhat. (2003). "Effect of slip velocity in a porous secant shaped slider bearing with a Ferro fluid lubricant" FIZIKA A. – 12 pp.1-8.

[34] Deheri,G.M., Andharia,P.I. ,Patel,R.M. (2005). "Transversely rough slider bearing with squeeze

film formed by a magnetic fluid", *International journal of Applied Mechanics and Engineering*, Vol.10, No.1, pp.53-76.

Nomenclature:

c	The hyperbolic factor
h	fluid film thickness at any point
F	frictional force
H	magnitude of the magnetic field
L	length of the bearing
P	lubricant pressure
U	velocity
W	load carrying capacity
F^*	dimensionless frictional force
P^*	dimensionless pressure
W^*	dimensionless load carrying capacity
h_0	fluid film thickness at $x = 0$
η	dynamic viscosity of fluid
τ	shear stress
σ	standard deviation
ε	skewness
α	variance
σ^*	non-dimensional standard deviation
ε^*	non-dimensional skewness
α^*	non-dimensional variance
μ^*	magnetization parameter
$\bar{\mu}$	magnetic susceptibility
μ_0	permeability of the free space