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Hydromagnetic squeeze film between conducting transversely rough curved circular plates lying along the surfaces determined by exponential and hyperbolic functions

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ABSTRACT:

The problem of squeeze films in the presence of transverse magnetic field between electrically conducting rough surfaces with electrically conducting lubricant is investigated for circular shape of the bearing surfaces. The surfaces of the bearings are known to be rough transverse. The bearing surface's roughness is based on a stochastic random variable with non-zero mean, variance, and skewness. The Reynolds' expression is stochastically about the parameter of random roughness. This is then solved with suitable boundary conditions to obtain the distribution of pressure, which is then used to obtain the capacity of load carrying. The findings are displayed in graphical format. It is found that, in general, the bearing suffers due to transverse surface roughness. Nevertheless, in the case of negatively skewed roughness, the condition can be recovered to some degree particularly when negative variance arises by selecting the conductivities of the plate suitably. Furthermore, it is observed that even when there is no flow, the bearing with magnetic field can support a load.

KEYWORDS
Squeeze film, Hydromagnetic lubrication, Reynolds' equation, Transverse roughness, Load profile.

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NOMENCLATURE

a	Radius of the plates (meter)
r	Radial coordinate
R	= r/a : Non dimensional radial coordinate
B	Curvature parameter of the upper plate (m)
C	Curvature parameter of the lower plate (m)
B^*	$= Ba^2$: Dimensionless curvature parameter of the upper plate
C^*	$= Ca^2$: Dimensionless curvature parameter of the lower plate
h	Lubricant film thickness (meter)
H	Magnetic field component
K	Permeability
m	Porosity of the porous matrix
M	$= B_0 h \left(\frac{s}{\mu}\right)^{1/2}$ = Hartmann number
p	Pressure distribution (N/m^2)
P	Non-dimensional pressure
s	Electrical conductivity of the lubricant
w	Load carrying capacity (kgm/s^2)

W	Dimensionless load carrying capacity
B_0	Uniform transverse magnetic field applied between the plates.
c^2	$= 1+\frac{KM^2}{h^2 m}$
$h_0^{'}$	Surface width of the lower plate (meter)
$h_1^{'}$	Surface width of the upper plate (meter)
s_0	Electrical conductivity of lower surface
s_1	Electrical conductivity of upper surface
Δt	Response time
ΔT	Non-dimensional response time
$\phi_0(h)$	$= \frac{s_0 h_0^{'}}{sh}$ = Electrical permeability of the lower surface
$\phi_1(h)$	$= \frac{s_1 h_1^{'}}{sh}$ = Electrical permeability of the upper surface
ψ	$= \frac{KH}{h^3}$ = Porosity
μ	Viscosity (kg/ms)
$\overline{\mu}$	Magnetic susceptibility (m^3/kg)
μ_0	Permeability of the free space (N/A^2)
σ^*	Non-dimensional standard deviation ($\sigma/\ \overline{h}$)
α^*	Non-dimensional variance ($\alpha/\ \overline{h}$)
ε^*	Non-dimensional skew ness ($\varepsilon/\ \overline{h}^3$)

I. INTRODUCTION

It is well known that if liquid metals like Mercury and Sodium could be pumped or held between the bearing's moving surfaces, the application of a strong magnetic field could accommodate greater loads. The possibilities of electromagnetic pressurization from the application of an external magnetic field have been explored and investigated because of the large electrical conductivity of liquid metals. This electromagnetic pressurization comes into effect when a large external electromagnetic field is applied to induce circulating currents through the electrically conductive lubricant, which in turn interacts with the magnetic field to create a body force that pumps the fluid between the bearing surfaces. Since the liquid metals are good electrical conductors, the load carrying capacity can be improved by using the electromagnetic force, thereby overcoming the defect associated with lubricants at high temperatures and thus alleviating the disadvantage of low viscosity. With the use of superconducting magnets, considerably high increases in load carrying capacity are possible, while the magnetic field requires very little power.

A variety of theoretical and experimental studies have been conducted on the hydromagnetic lubrications for both porous and plane metal bearings. In liquid metal lubrication Elco and Huges (1962) studied magnetohydrodynamic pressurization. The behavior of magnetohydrodynamic squeeze films was studied by Kuzma (1964) and Kuzma, Maki and Donelly (1964). Shukla (1965) studied the hydromagnetic theory of squeeze films in the presence of a transverse magnetic field to conduct lubricants between two non-conducting non-porous surfaces. Shukla and Prasad (1965) discussed the behavior of hydromagnetic squeeze films between two non-porous surfaces conducting and studied the effect of surface conductivity on bearing system performance. A number of theoretical and experimental studies were devoted to magnetohydrodynamic lubrication (Dodge F.T., Osterle J.F., and Rauleau W.T.(1965); Maki, Kuzma and Donelly (1966); Snyder (1962)). Sinha and Gupta (1973; 1974) addressed the study of hydromagnetic effect on porous squeeze films in which annular plates and rectangular plates were considered. This effect was observed by Patel and Hingu (1978) for squeeze films between circular disks. Patel and Gupta (1979) used the Morgan – Cameron approximation and simplified the study of a variety of geometric shapes for hydromagnetic squeeze films between parallel plates. Patel (1975) studied the behavior of hydromagnetic squeeze films with tangential velocity slips between porous annular disks.

It is a well-known fact that the bearing surfaces develop ruggedness after some run-in and wear. Several authors studied the influence of surface roughness (Davies (1963); Burton (1963); Michell (1950); Tonder (1967, 1977); Tzeng and Saibel (1967); Christensen and Tonder (1969.a; 1969.b; 1970); Berthe and Godet (1973), Mishra et. al. (2018)). Christensen and Tonder (1969.a; 1969.b; 1970) proposed a comprehensive general analysis for the roughness of both the transverse and longitudinal surfaces. The method of Christensen and Tonder formed the basis for the analysis of the effect of surface roughness in several investigations (Ting (1975); Prakash and Tiwari (1982); Prajapati (1991; 1992); Guha (1993); Gupta and Deheri (1996); Andharia, Gupta and Deheri (1997; 1999)). Patel and Deheri (2003; 2004) analyzed the performance of a magnetic fluid-based squeeze film between circular plates and annular plates, and studied the effect of surface roughness on bearing system performance.

While, in the case of circular plates, the increase in plate conductivity leads to improved performance (Prajapati (1995)), here it is shown that the transverse roughness of the bearing surfaces adversely affects system performance. Of course, in the case of negatively skewed roughness, the situation can be retrieved to some extent especially when there is negative variance. The objective of this article is to study hydromagnetic squeeze film between conducting transversely rough curved circular plates lying along the surfaces determined by exponential and hyperbolic functions. Here, it is found that the curvature parameters of both plates a crucial role in augmenting the performance of the bearing system.

II. ANALYSIS

The bearing configuration is presented below.

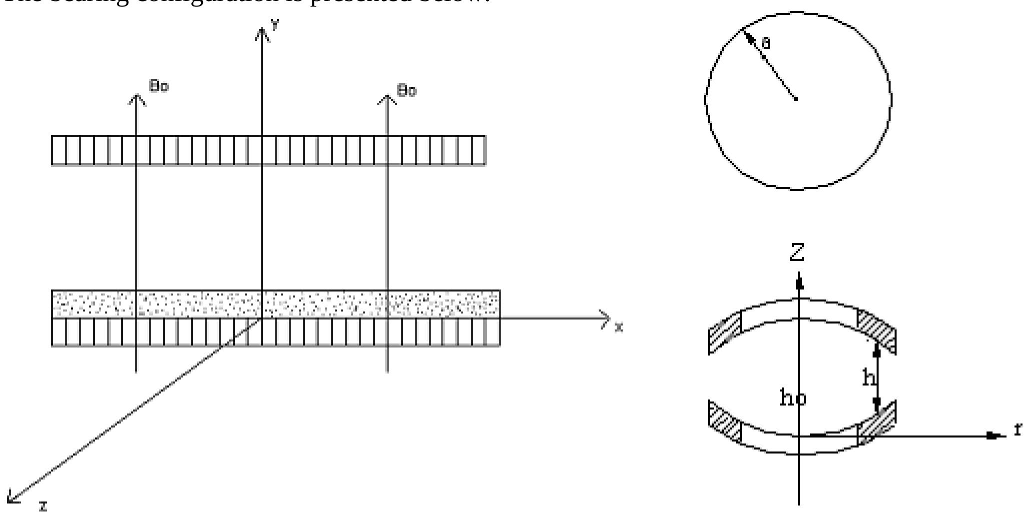


Fig. 1 Squeeze film bearing geometry and configuration

It is assumed that the lower plate with porous facing is fixed, while the upper plate moves towards the lower plate along its normal course. The plates are considered to be electrically conductive, and a lubricant that conducts electrically fills the clearance space between them. A uniform transverse magnetic field between the plates is applied. The flow into the porous medium obeys Darcy's modified form of rule. (Cf. Prajapati (1995)), whereas, the hydromagnetic lubrication theory equations hold in the film region. Tzeng and Saibel (1967) utilized a strategy for irregular examining and accepted the one-dimensional film thickness to be of the shape

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

where $\bar{h}(x)$ is the mean film thickness and $h_s(x)$ is the random deviation from the mean film thickness that exemplifies the random roughness of the bearing. Thickness h(x) is regarded as a random variable whose probability density function is either a Gaussian normal distribution function or Beta distribution function given by

$$f(h_s) = \frac{35}{32}\left(1-\frac{{h_s}^2}{c^2}\right)^3, \quad -c \le h_s \le c; \text{ and } f(h_s) = 0, \text{ elsewhere}$$

This distribution function is utilized for normal out the physical amounts in the Reynolds' equation as for film thickness. Other than this mathematical form, numerous other numerical methodologies have likewise come up in various examinations. The mean α, the standard deviation σ and the parameter ε which is the measure of symmetry , of random variable h_s, are determined by relationships

$\alpha = E\,(h_s)$

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

and

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

where E denotes the expected value defined by

$$E\,(R) = \int_{-c}^{c} R f(h_S)\,dh_S$$

The particulars regarding the characterization of roughness aspect can be borrowed from Christensen and Tonder (1969.a; 1969.b; 1970).

Using equations (A_{15} - A_{19}) in the equation of continuity and simplifying it one gets the modified Reynolds' equation as

$$\nabla^2 p = \frac{dh/dt}{\left[\frac{2h^3}{\mu M^3}\left\{\left(\tanh\frac{M}{2} - \frac{M}{2}\right) - \frac{\psi h^3}{\mu C^2}\right\}\right]\left[\frac{\varphi_0 + \varphi_1 + 1}{\varphi_0 + \varphi_1 + \frac{\tanh(M/2)}{M/2}}\right]} \qquad \dots (1)$$

where $\psi = \dfrac{KH}{h^3}$

Under standard hydromagnetic lubrication assumptions and using expressions (A_1 to A_{19}) the modified equation of Reynolds' type for the lubricant film pressure is (Prajapati (1995), Vadher et. al. (2008))

$$\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial p}{\partial r}\right) = \frac{\mu \dot{h}}{g(h)AB} \qquad \dots (2)$$

where $g(h) = h^3 + 3\sigma^2 h + 3h^2\alpha + 3h\alpha^2 + 3\sigma^2\alpha + \alpha^3 + \varepsilon$,

$$h(r) = h_0\left[e^{-Br^2} - \frac{1}{Cr+1} + 1\right],$$

$$A = \left[\frac{\psi}{c^2} - \frac{2}{M^3}\{\tanh(M/2) - (M/2)\}\right], \quad B = \left[\frac{\phi_0 + \phi_1 + 1}{\phi_0 + \phi_1 + \frac{\tanh(M/2)}{(M/2)}}\right]$$

and

$$M = B_0 h\left(\frac{s}{\mu}\right)^{1/2}$$

Solving this equation with the use of boundary conditions

$p(a) = 0;$

$$\frac{\partial p}{\partial r} = 0 \text{ at } r = 0 \qquad \dots (3)$$

and applying following non dimensional terms

$B^* = Ba^{2,}$ $\quad C^* = Ca^2$, $\quad R = \dfrac{r}{a}$,

$\sigma^* = \dfrac{\sigma}{h_0}$, $\quad \alpha^* = \dfrac{\alpha}{h_0}$, $\quad \varepsilon^* = \dfrac{\varepsilon}{h_0^3}$

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

$$\bar{h} = \frac{h(r)}{h_0} = e^{-B^*R^2} - \frac{1}{C^*R+1} + 1$$

gives the pressure distribution in dimensionless form as

$$P = \frac{-ph^3}{\mu \dot{h} \pi a^2}$$

$$P = -\frac{1}{\pi AB}\int_R^1 \frac{\int_R^1 \frac{R}{g(\bar{h})}dR}{R}dR \qquad \dots (4)$$

Then the load carrying capacity given by

$$w = 2\pi\int_0^a p(r)\cdot rdr \qquad \dots (5)$$

is obtained in non dimensional form as

$$W = -\frac{wh^3}{\mu \dot{h} \pi^2 a^4}$$

$$W = -\frac{1}{\pi AB}\int_0^1 RPdR \qquad \dots (6)$$

Lastly, if the time taken for the plate to move from the film thickness $h = h_0$ at $t = t_0$ to the film thickness $h = h_1$ at $t = t_1$ then the non-dimensional squeeze time ΔT is derived from equation (4) as

$$\Delta T = \int_0^{t_1/t_0} \frac{Wh_0^2}{\mu\pi^2 a^4}dt$$

which means

$$\Delta T = \frac{1}{8\pi} I \qquad \dots \dots \dots (7)$$

where I is given by

$$I = -h_0^2 \int_1^{h_1/h_0} \frac{dh}{\left[\frac{2A}{M^3}\left(\tanh\frac{M}{2} - \frac{M}{2}\right) - \frac{KH}{c^2}\right]} \bullet \frac{1}{\left[\frac{\phi_0 + \phi_1 + 1}{\phi_0 + \phi_1 + \frac{\tanh(M/2)}{(M/2)}}\right]}$$

III. RESULTSAND SISCUSSIONS

It is observed from equations (4) and (6) that the pressure and the load carrying capacity depend on various parameters such as M, ψ , $\phi_0 + \phi_1$, σ^* , ε^*, α^* and curvature parameters. The effect of conductivity on the pressure distribution, load and the response time ΔT come through the factor

$$\left(\frac{\phi_0 + \phi_1 + \frac{\tanh(M/2)}{(M/2)}}{\phi_0 + \phi_1 + 1}\right)$$

For large values of M this tends to

$$\frac{\phi_0 + \phi_1}{\phi_0 + \phi_1 + 1}$$

as tanhM ~ 1, 2 / M ~ 0.

Figures (1) – (7) present the variation of load carrying capacity with respect to the magnetization parameter M, for various values of $\phi_0 + \phi_1$, ψ, σ^* , α^*, ε^*, B* and C* respectively .

Figure: 1 Variation of load carrying capacity with respect to M and $\phi_0 + \phi_1$

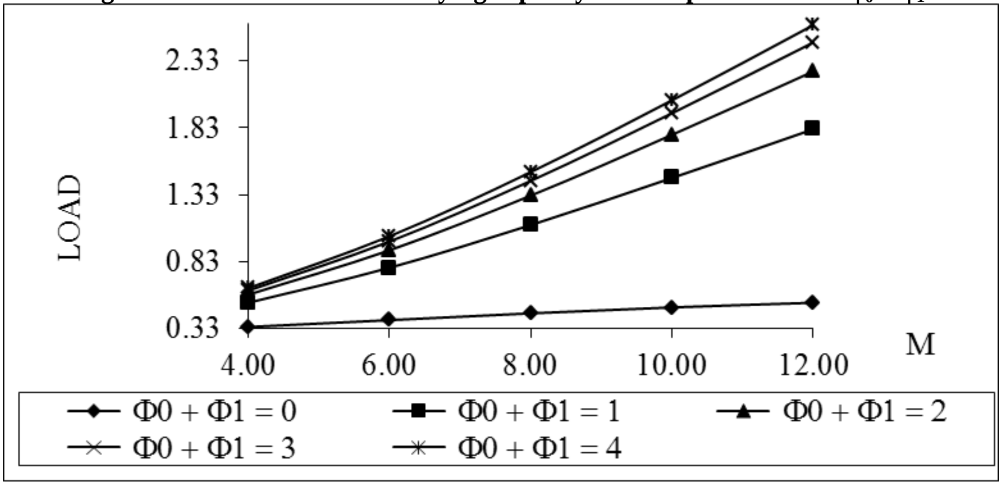


Figure: 2 Distribution of load bearing capacity with respect to M and ψ

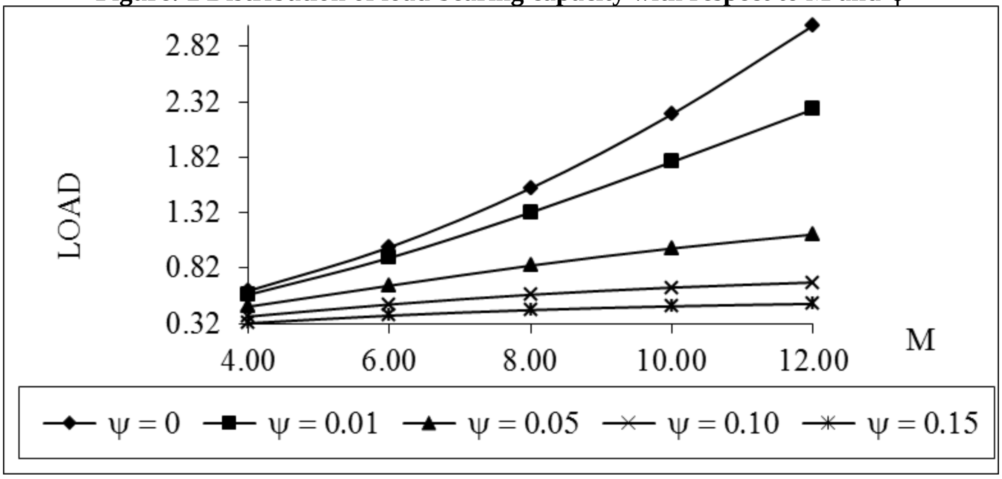


Figure: 3 Profile of load with respect to M and σ*

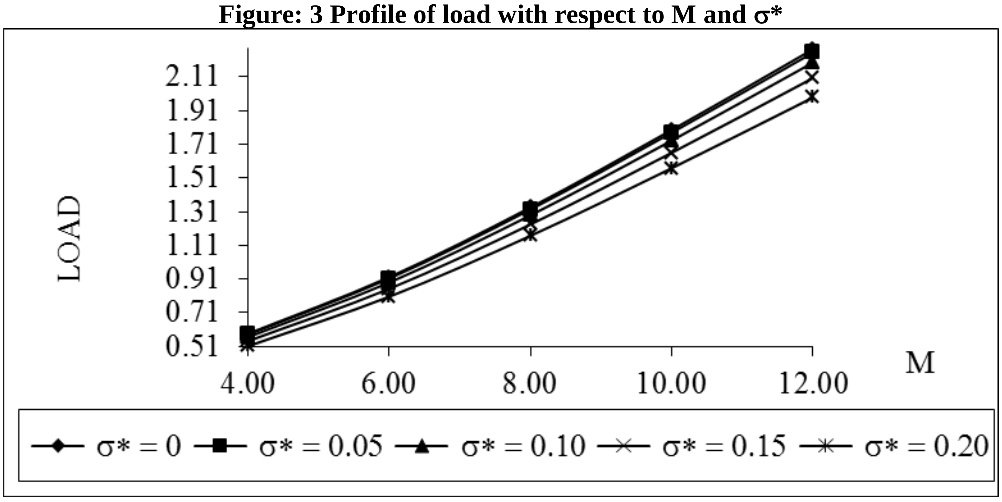


Figure: 4 Variation of load carrying capacity with respect to M and α*

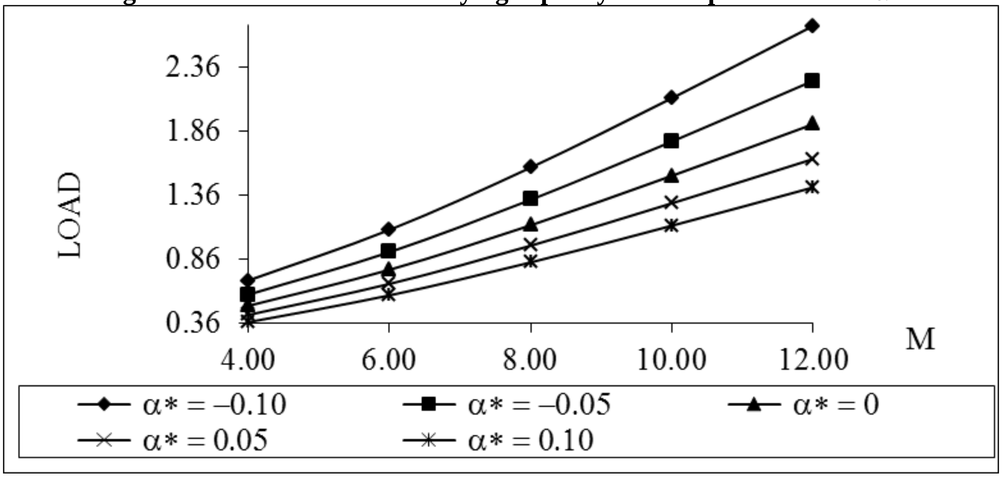


Figure: 5 Distribution of load bearing capacity with respect to M and ε*

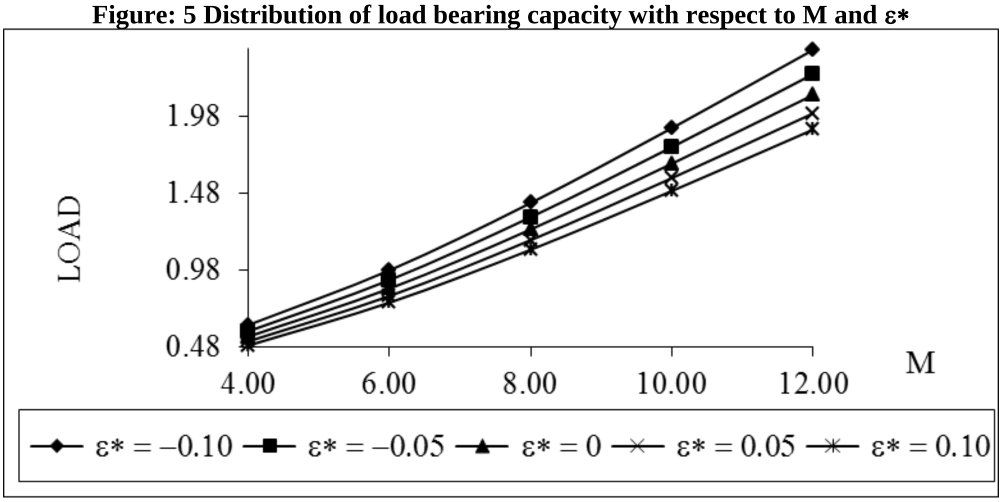


Figure: 6 Profile of load bearing capacity with respect to M and B*

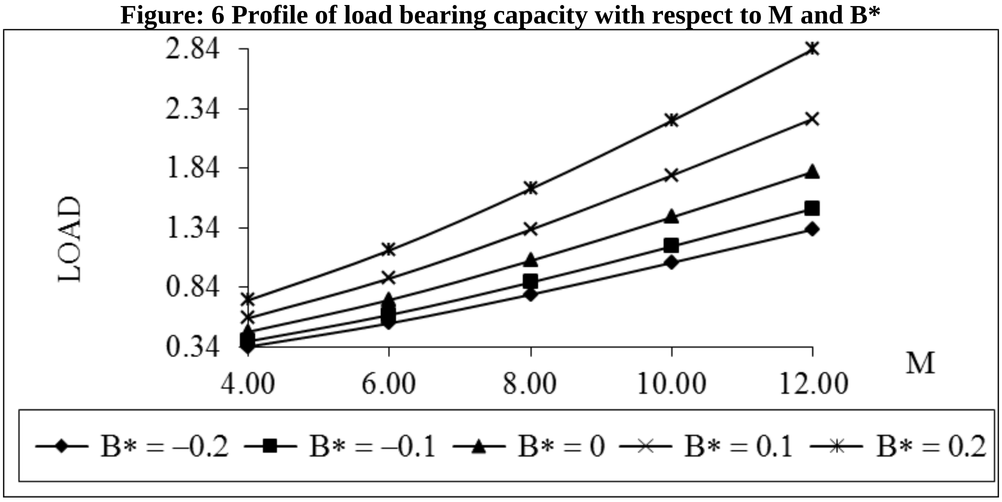
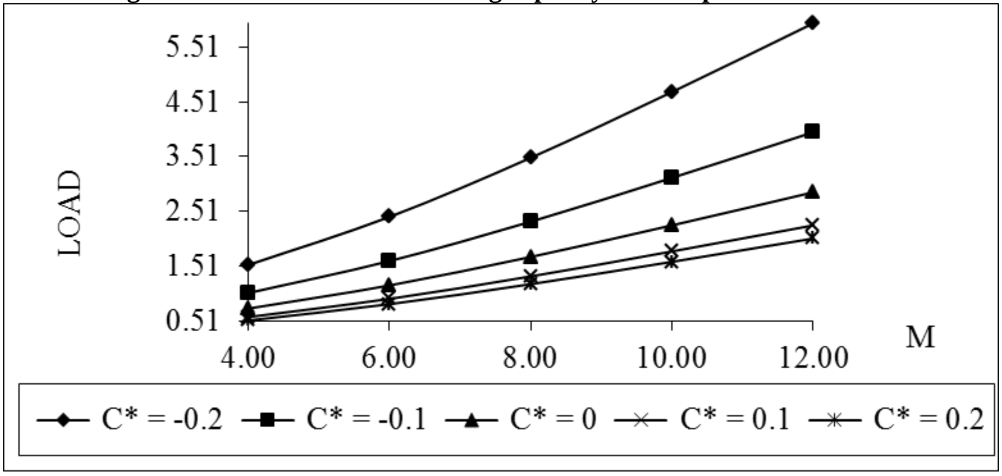


Figure: 7 Variation of load bearing capacity with respect to M and C*



A closed glance at these figures reveals that the load increases considerably. In addition, negatively skewed roughness tends to enhance the performance of the bearing system. Same is the case with negative variance. It is interesting to note that the increase in load bearing capacity induced by variance (-ve) is better and the variance in general. It is observed that the surface roughness affects the bearing system adversely.

The distribution of load bearing capacity with respect to $\phi_0 + \phi_1$ for various values of ψ, σ^*, α^*, ε^*, B* and C* presented in Figures (8) to (13), these figures tend to suggest that the load carrying capacity increases significantly with respect to $\phi_0 + \phi_1$.

Figure: 8 Distribution of load with respect to $\phi_0 + \phi_1$ and ψ*

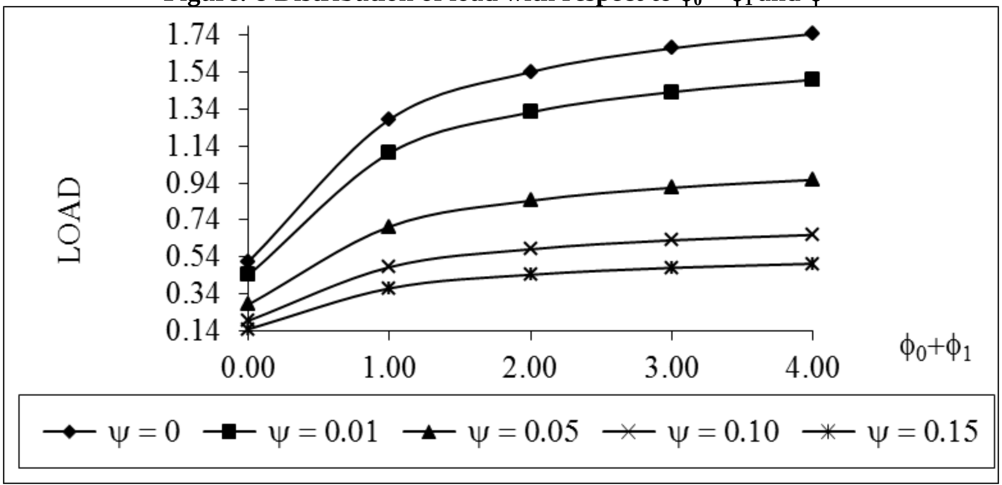


Figure: 9 Profile of load carrying capacity with respect to $\phi_0 + \phi_1$ and σ*

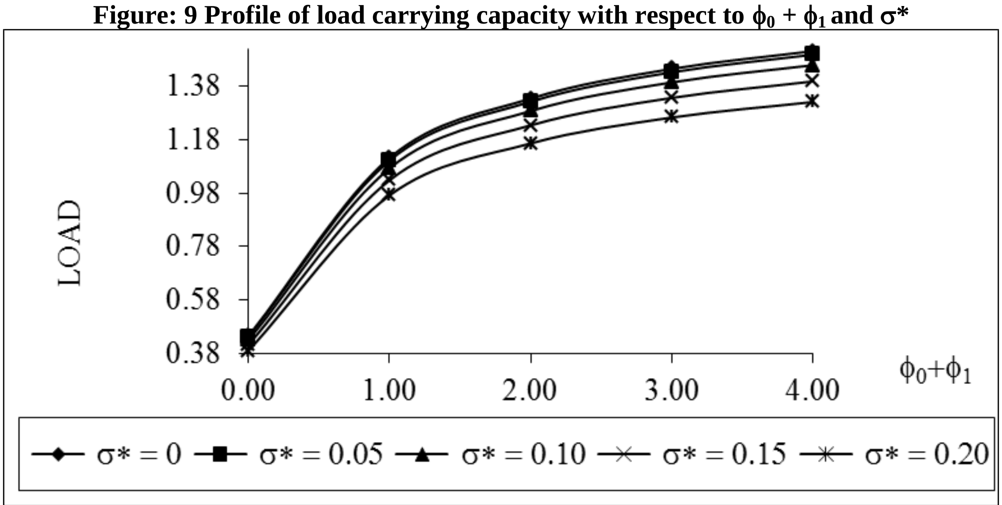


Figure: 10 Variation of load bearing capacity with respect to $\phi_0 + \phi_1$ and α^*

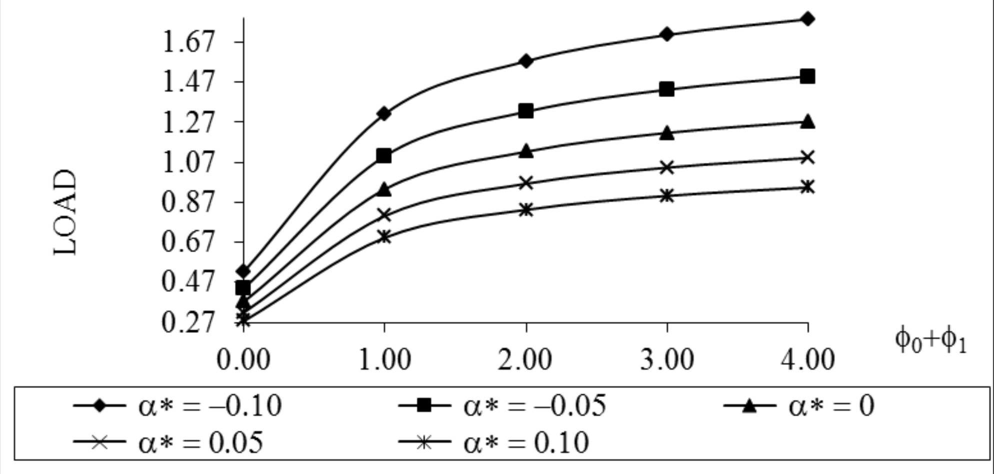


Figure: 11 Distribution of load with respect to $\phi_0 + \phi_1$ and ε^*

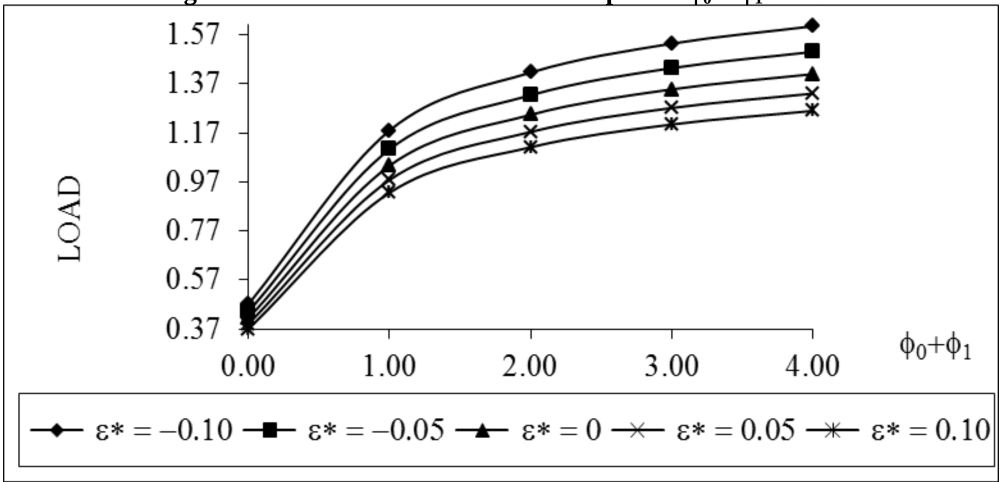


Figure: 12 Profile of load with respect to $\phi_0 + \phi_1$ and B*

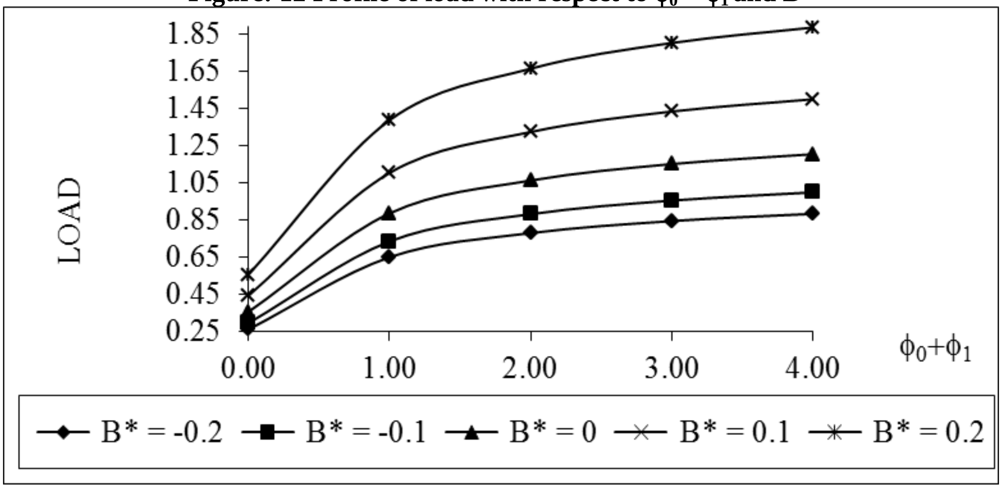
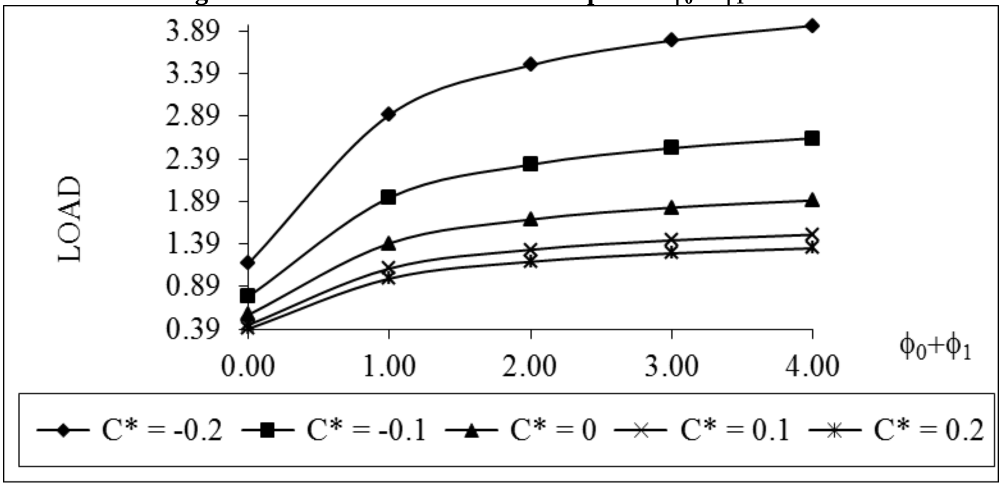


Figure: 13 Variation of load with respect to $\phi_0 + \phi_1$ and C*



Here also, the variance has a very sharp impact in this increase in load carrying capacity, while the standard deviation has a nominal role. Further, it is depicted from above all the figures that the curvature parameter associated the upper plate enhances the performance marginally.
Figures (14) to (18) presented below gives the profile of load with different values of porosity parameter.

Figure:14 Distribution of load carrying capacity with respect to ψ and σ*

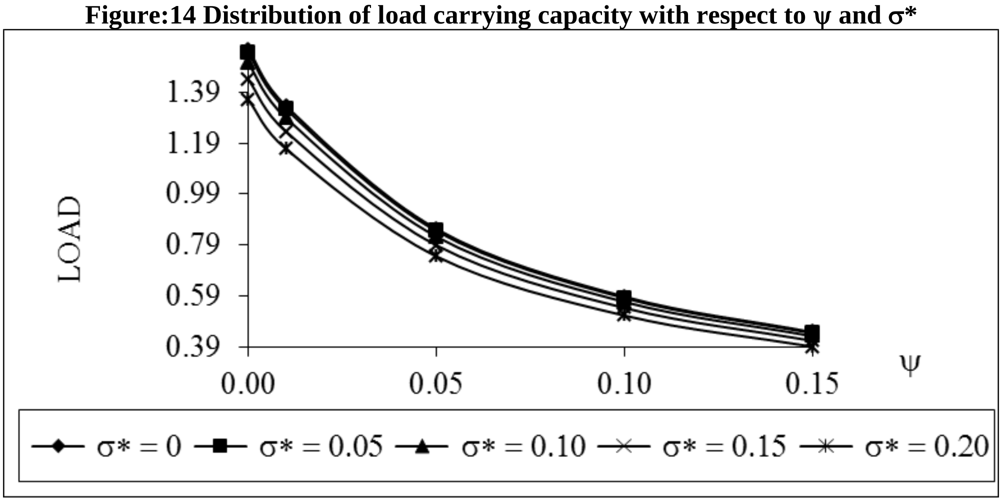


Figure: 15 Profile of load bearing capacity with respect to ψ and α*

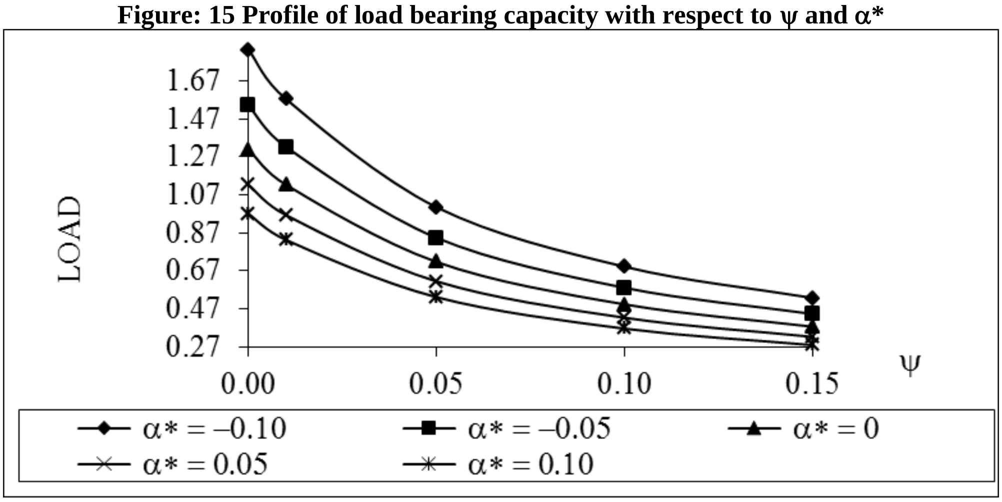


Figure: 16 Variation of load with respect to ψ and ε*

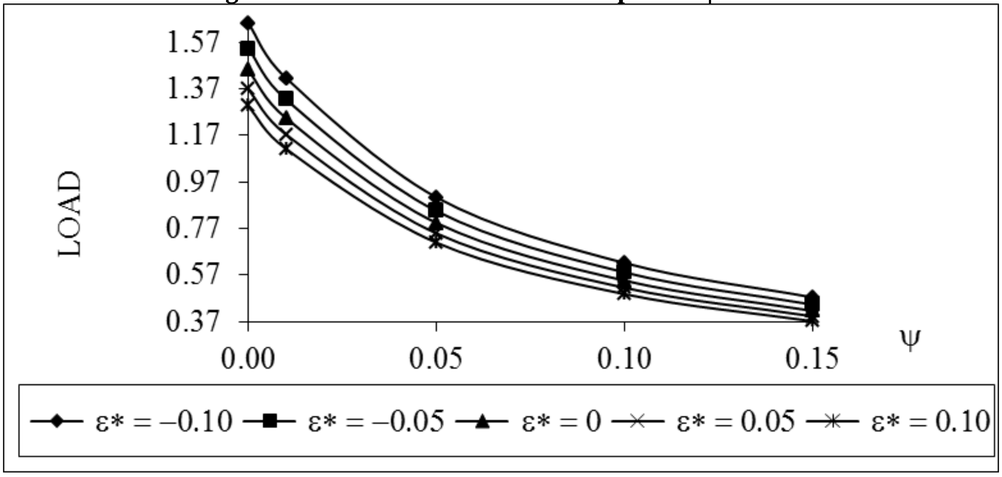


Figure: 17 Distribution of load with respect to ψ and B*

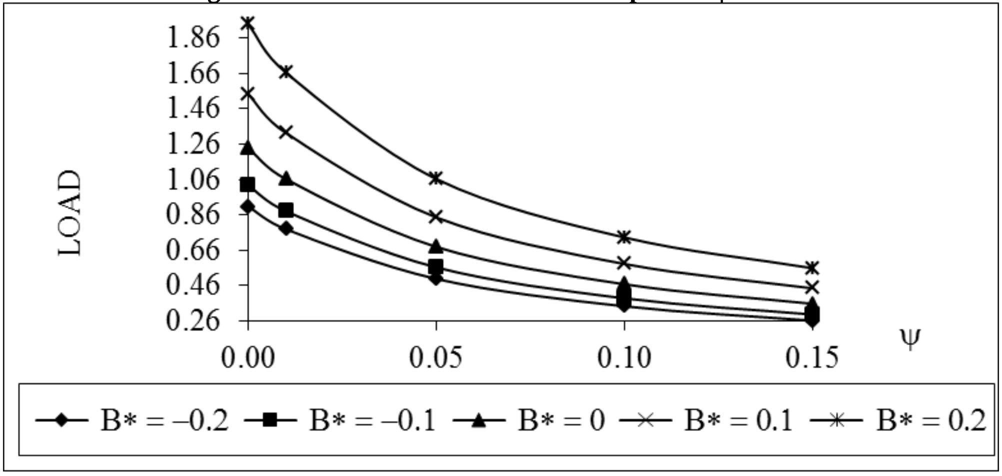
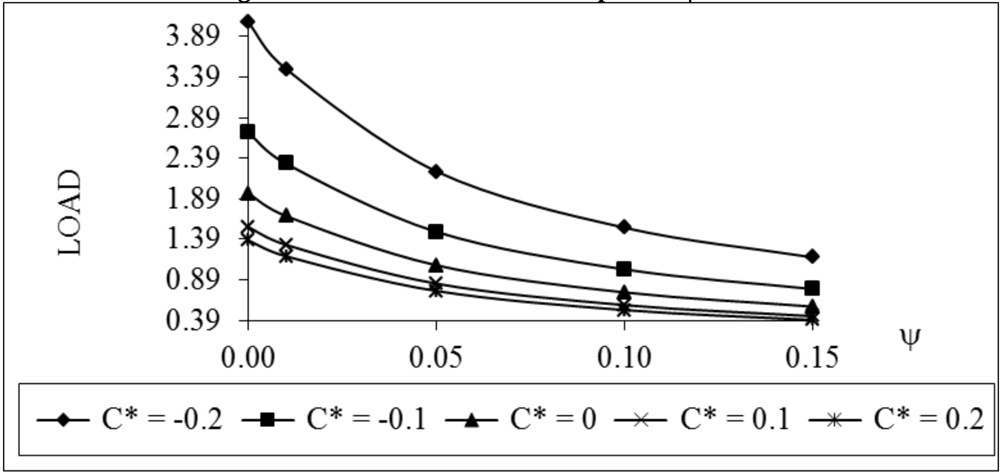


Figure: 18 Profile of load with respect to ψ and C*



One concludes from these figures that the combined effect of the porosity and the roughness parameters is significantly negative but this situation can be retrieved up to certain extent in the case of negatively skewed roughness. Also the combined effect of curvature parameters plays an important role.

The load bearing capacity for the combinations of roughness parameters such as standard deviation, variance and skewness are presented in Figures (19) to (27) respectively.

Figure: 19 Variation of load bearing capacity with respect to σ* and α*

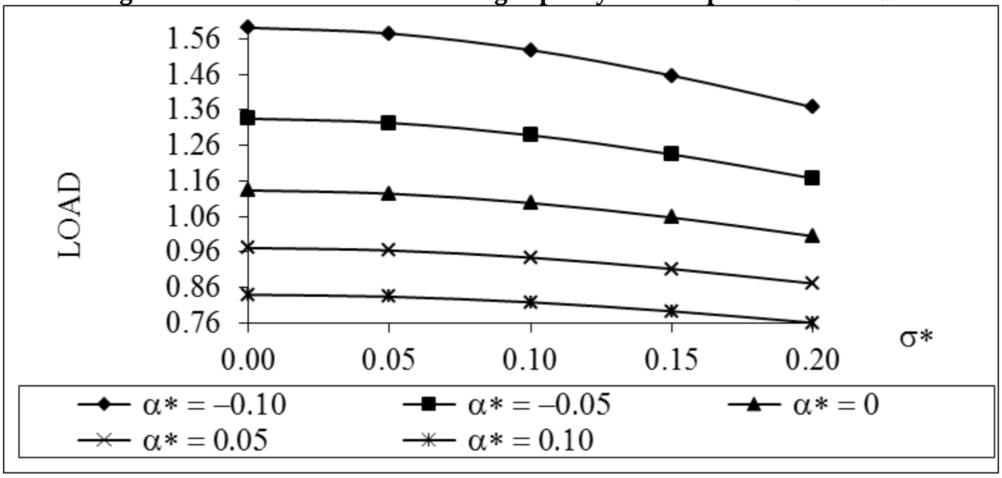


Figure: 20 Distribution of load carrying capacity with respect to σ and ε*

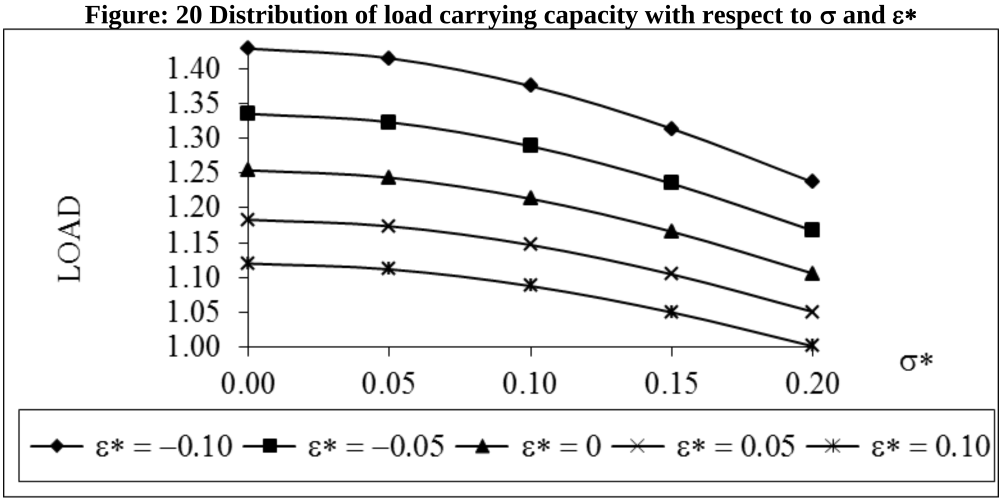


Figure: 21 Profile of load with respect to σ* and B*

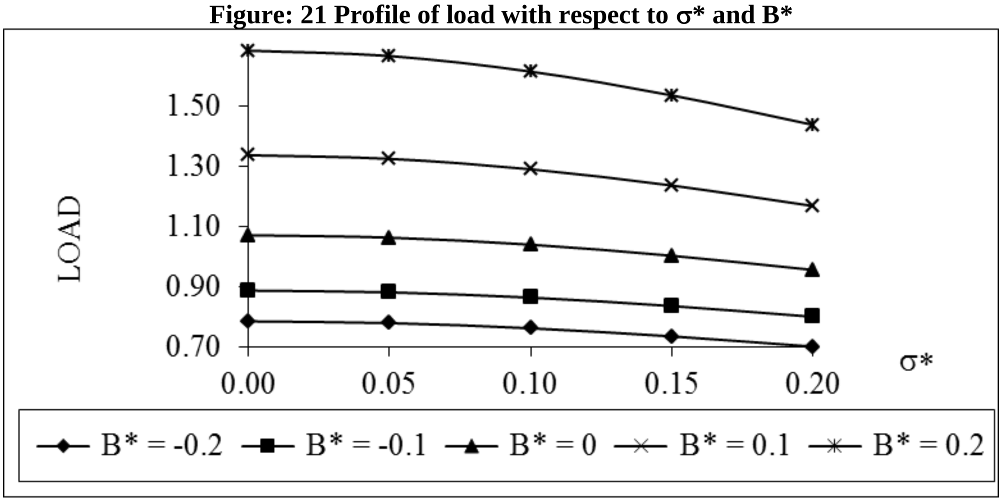


Figure: 22 Variation of load bearing capacity with respect to σ* and C*

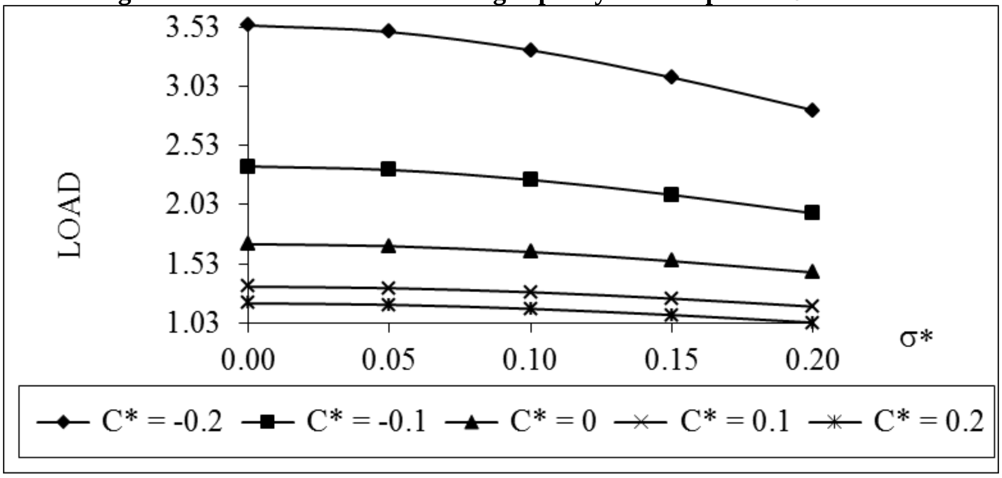


Figure: 23 Distribution of load carrying capacity with respect to α* and ε*

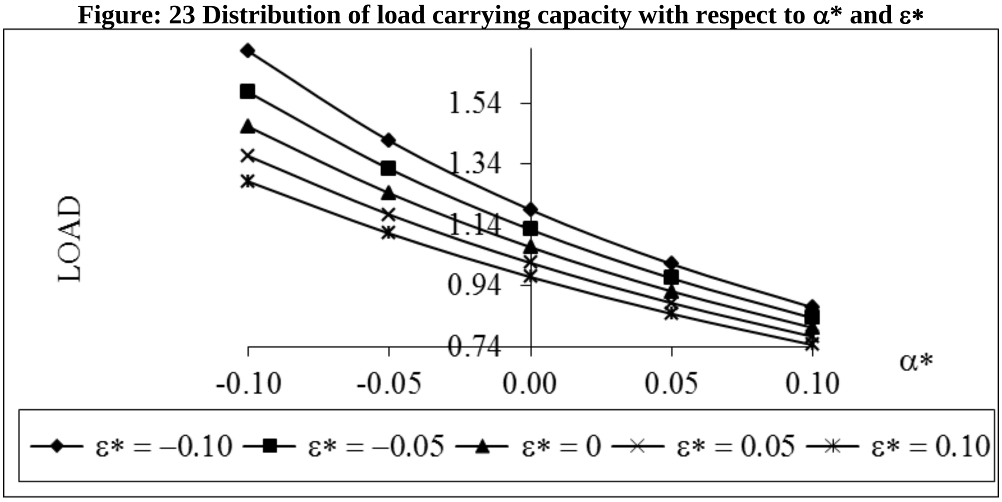


Figure: 24 Profile of load with respect to α* and B*

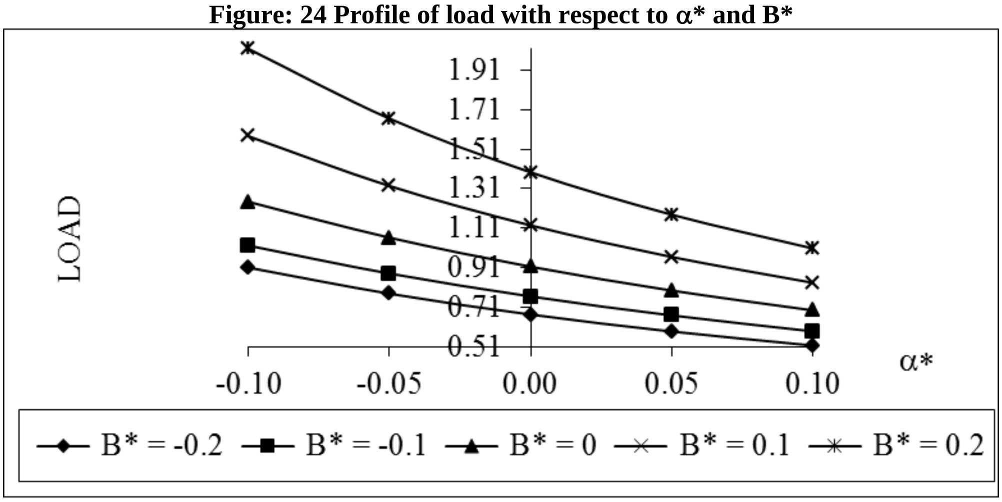


Figure: 25 Variation of load bearing capacity with respect to α* and C*

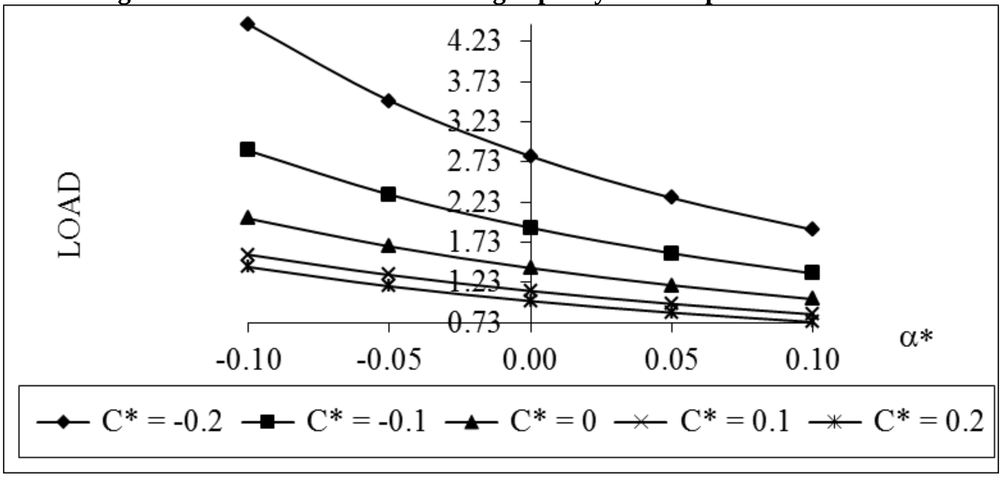


Figure: 26 Distribution of load carrying capacity with respect to ε* and B*

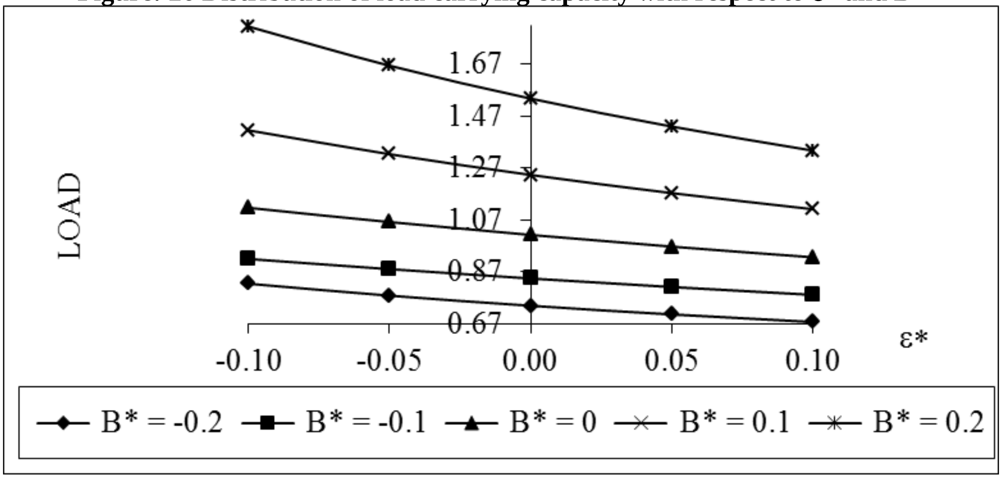
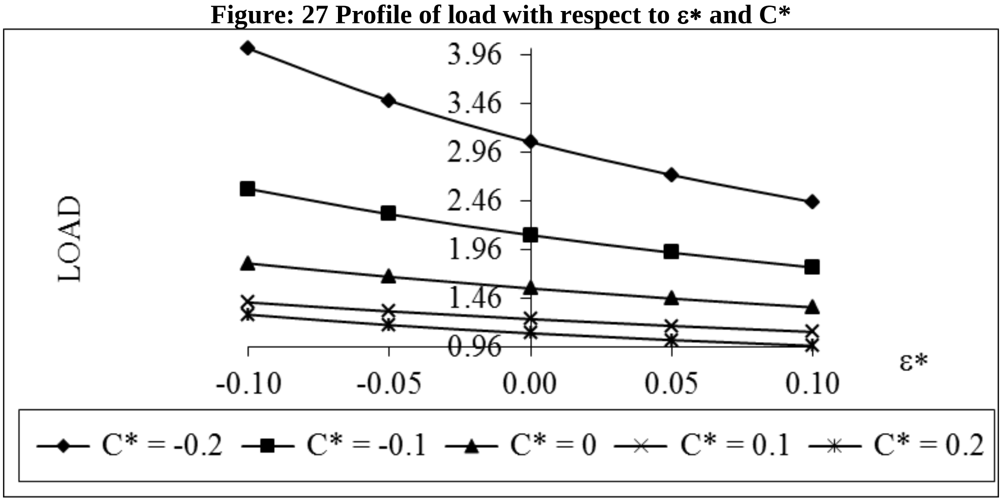


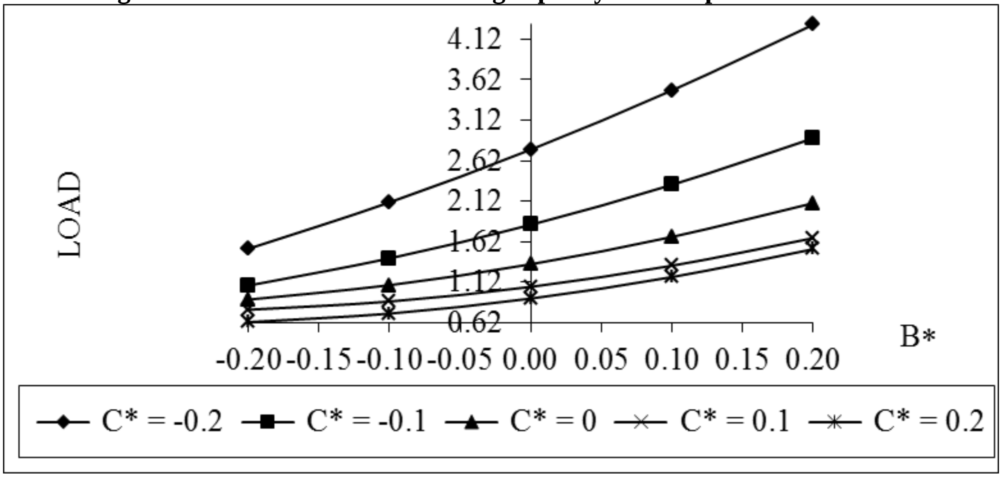
Figure: 27 Profile of load with respect to ε* and C*



So far as the combined effect of roughness parameters is concerned, it is clear that the effect of the roughness parameters causes reduced load. In this reduction the prominent roles are played by the positive variance and the standard deviation.

Figure (28) presents the effect of curvature parameters for the fixed values of the other parameters such as: M, ψ , $\phi_0 + \phi_1$, σ^* , ε^*, α^*.

Figure: 28 Variation of load bearing capacity with respect to B* and C*



This article offers that there ample scopes for augmenting the performance of the bearing system in the case of negatively skewed roughness as specially when negative variance is involved. The above consideration alone suggests that the life period of the machine can be extended in the case of rough bearings.

Justifications:

- For non- porous smooth conducting plate the results for squeeze film analyzed by Shukla and Prasad (1965) are recovered when $\psi = 0$ and the roughness parameters σ^*, α^* and ε^* are equal to zero.
- For non–magnetic porous squeeze films the results of Prakash and Vij (1973) are obtained in the limiting case when we take $M \rightarrow 0$ and the roughness parameters $\sigma^* = \alpha^* = \varepsilon^* = 0$.The results of Patel and Gupta (1979) are obtained when ϕ_0 and ϕ_1 are taken as zero.
- This study reduces to the analysis of Prajapati (1995) when the roughness parameters are assumed to be zero. For non – porous conducting plates the results for hydromagnetic squeeze films between two conducting rough surfaces are obtained. In the limiting case of $M \rightarrow 0$, the study tends to non – magnetic porous squeeze films between rough porous plates.
- When ϕ_0 and ϕ_1 are taken to be zero this analysis essentially, describes the study of the behavior of a hydromagnetic squeeze film between rough porous plates.

Conclusions:

This paper tend to suggest that for both small as well as large values of M the bearing suffers when the plates are considered electrically conducting in comparison with the hydromagnetic case when the plates are non-conducting. Probably, this is due to the occurrence of fringing phenomena when the plates are electrically conducting.

Further, it is clearly noticed that as the plate conductivities, upper plate's curvature parameter and negative variance increase, the lubricant pressure, load carrying capacity and the response time increase. Also, this article clearly tells that roughness parameters, curvature parameter must be given due consideration while designing the bearing system, even if a proper selection of M and $\phi_0 + \phi_1$ has been taken into account.

Conflict of Interest:

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