Ferro fluid squeeze film in infinitely long porous rough rectangular plates

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Abstract:

This article aims to analyse the behaviour of a magnetic fluid based squeeze film between infinitely long, porous and transversely rough rectangular plates by considering an unusual form of the magnitude of the magnetic field. The stochastic method of Christensen and Tonder has been employed to evaluate the effect of surface roughness. The concerned stochastically averaged Reynolds' type equation is solved with appropriate boundary conditions. The results presented in graphical form establish that magnetization offers help to a limited extent in order to minimize the adverse effect of roughness. But, this investigation establishes that by considering a suitable form of magnitude of the magnetic field the situation can be made better.

Introduction:

By now it is well known that the roughness of the bearing surface has a significant effect on the performance characteristics of a squeeze film bearing system.

The roughness appears to be random in character which was recognized by several investigators who adopted a stochastic approach to mathematically model the roughness of the bearing surfaces. Christensen and Tonder's work [1, 2, 3] which was based upon the concept of viewing the film thickness as a stochastic process, resulted in a averaged Reynolds' type equation. Christensen and Tonder [1, 2, 3] presented a comprehensive general analysis for both versions of surface roughness namely, transverse and longitudinal. Later on, this approach of Christensen and Tonder [1, 2, 3] formed the basis of the analysis to analyze the effect of surface roughness on the performance of the bearing system in a number of investigations [4, 5]. Andharia, Gupta and Deheri [6] dealt with the analysis of the effect of surface roughness on the performance of a squeeze film bearing using the general stochastic

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analysis (without making use of a specific probability distribution) for describing the random roughness. Verma [7] and Agrawal [8] investigated the application of magnetic fluid as a lubricant. While Verma [7] considered tangential slip velocity at the porous matrix lubricant interface, Agrawal [8] discussed no slip condition. Patel et. al. [9] considered the performance characteristics of a magnetic fluid based squeeze film between infinitely long rough porous rectangle plates. This study offered the enhanced performance of the bearing system by picking up a suitable combination of magnetization parameter and aspect ratio in the case of negatively skewed roughness.

Shukla and Deheri [10] investigated the effect of slip velocity on the performance of a magnetic fluid based squeeze film in porous rough infinitely long parallel plates. This article suggested that for a better performance of the bearing system the slip velocity deserved to be kept at minimum, even if the magnetic strength was chosen suitably.

Shah and Bhatt [11] analyzed the porous exponentially slider bearing lubricated with a ferrofluid considering slip velocity. Increase in the slip parameter did not significantly affect the position of centre of pressure although it increased the load carrying capacity.

Patel et. al. [12] studied the effect of non-uniform thickness on the magnetic fluid based squeeze film in rough porous long plates. It was observed that the variable film thickness offered assistance to the magnetization in minimizing the adverse effect of porosity and roughness.

Here, it has been deemed proper to analyse the behaviour of a ferrofluid squeeze film in infinitely long porous rough rectangle plates resorting to a different form of magnitude of the magnetic field.

Analysis:

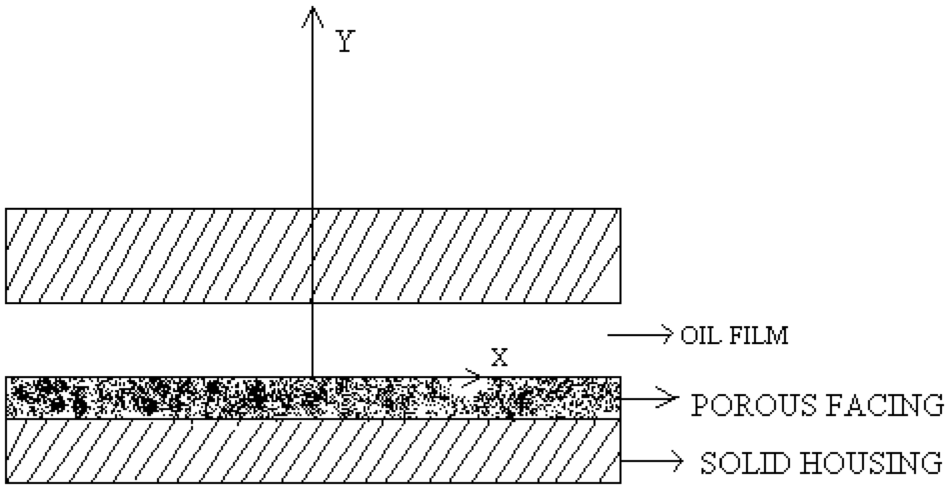

Figure – I Configuration of the bearing

Figure – I shows the configuration of the bearing system consisting of geometrical structure of the rectangular plates. The upper plate moves normally towards the lower plate with uniform velocity $\dot{h} = \frac{dh}{dt}$ both the plates are considered to have transversely rough surfaces.

Following the discussions of Christensen and Tonder [1, 2, 3] the geometry of local film thickness is considered as

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

where $\bar{h}(x)$ is the mean film thickness and $h_s(x)$ is the deviation from the mean film thickness characterizing the random roughness of the bearing surfaces. $h_s(x)$ is stochastic in nature and described by a probability density function $f(h_s)$ $-c \le h_s \le c$ where c is the maximum deviation from the mean film thickness.

The details regarding the probability density function, the mean α, the standard deviation σ and the measure of symmetry ε can be had from Christensen and Tonder [1, 2, 3].

Assuming axially symmetry flow of the magnetic fluid between the rectangular plates under an oblique magnetic field $\bar{H}$, whose magnitude H is a function of z vanishing at $\pm\frac{b}{2}$, the modified Reynolds' equation governing the film pressure p turns out to be

$$\frac{d^2}{dz^2}\left(P - 0.5\mu_0\bar{\mu}\,H^2\right) = \frac{12\mu\dot{h}}{g(h) + 12\phi H_0} \qquad \dots (2)$$

where

$$g(h) = h^3 + 3\sigma^2 h + 3h^2\alpha + 3h\alpha^2 + 3\sigma^2\alpha + \alpha^3 + \qquad \dots (3)$$

and

$$H^2 = kb^2\left(\frac{3\pi\bar{z}}{2} + \frac{3\pi}{4}\right)\cos\left(\frac{3\pi\bar{z}}{2} - \frac{\pi}{4}\right) \qquad \dots (4)$$

while k_1 is a suitable chosen constant to produce a magnetic field of required strength. μ is the viscosity, μ_0 is the permeability of the free space, $\bar{\mu}$ is the magnetic susceptibility, σ is the standard deviation, α is the variance and ε is the measure of the skewness.

Integrating the above equation with respect to the boundary conditions $p\left(\pm\frac{b}{2}\right) = 0$ leads to the expression for the distribution.

$$p = 0.5\mu_0\bar{\mu}\,H^2 + \frac{6\mu\dot{h}b^2}{g(h) + 12\phi H_0}\left(\frac{z^2}{b^2} - \frac{1}{4}\right) \qquad \dots (5)$$

Introducing the non dimensional quantities

$$\bar{z} = \frac{z}{b} \qquad \mu^* = -\frac{h^3\mu_0\bar{\mu}}{\mu\dot{h}} \qquad \psi = \frac{\phi H_0}{h^3}$$

$$\sigma^* = \frac{\sigma}{h} \qquad \alpha^* = \frac{\alpha}{h} \qquad \varepsilon^* = \frac{\varepsilon}{h^3}$$

$$\bar{h} = \frac{h}{h_0} \qquad \bar{h}_1 = \frac{h_1}{h_0} \qquad \bar{h}_2 = \frac{h_2}{h_0}$$

$$\bar{p} = -\frac{h^3 p}{\mu\,\dot{h}\,a\,b} \qquad \bar{w} = -\frac{h^3\,w}{\mu\,\dot{h}\,a^2\,b^2}$$

the nondimensional pressure distribution turns outs to be

$$\bar{p} = \frac{\mu^*}{2} k_1 \left(\frac{3\pi \bar{z}}{2} + \frac{3\pi}{4} \right) \cos \left(\frac{3\pi \bar{z}}{2} - \frac{\pi}{4} \right) + \frac{6k_1}{g(\bar{h}) + 12\psi} \left(\bar{z}^2 - \frac{1}{4} \right) \qquad \dots (6)$$

where

$$g\left(\bar{h}\right) = 1 + 3\sigma^{*^2} + 3\alpha^{*^2} + 3\varepsilon\alpha^* + 3\sigma^{*^2}\alpha^* + \alpha^{*^3} + {}^* \qquad \dots (7)$$

The load carrying capacity of the bearing in non dimensional form can be expressed as

$$\bar{w} = \frac{1.2122\, \mu^*}{2k_1} + \frac{1}{k_1\left(g\left(\bar{h}\right) + 12\psi\right)} \qquad \dots (8)$$

Lastly the response time Δt taken by the upper plate to reach a film thickness h_2 from an initial film thickness h_1 can be determined in non-dimensional form as

$$\Delta\bar{t} = -\bar{w} \int_{-\bar{h}_1}^{\bar{h}_2} \frac{d\bar{h}}{\bar{h}^3 + 3\bar{\sigma}^2\bar{h} + 3\bar{h}\psi\bar{\alpha} + 3\bar{h}\bar{\alpha}^2 + 3\bar{\sigma}^2\bar{\alpha} + \bar{\alpha}^3 + \bar{\varepsilon} + 12}$$

Results and discussion:

It is clear from equation (8) that the non-dimensional load carrying capacity enhances by

$$\frac{(1.2122)\mu^*}{k_1},$$

as compared to the case of a conventional lubricant based bearing system. In the absence of magnetization this study reduces to the investigation of Prakash and Vij [13] for smooth bearing systems.

The results presented in graphical forms (figures (1)-(5)) make it clear that the load carrying capacity increases sharply owing to the magnetization. The variation of load carrying capacity with respect to σ^* presented in figures (6)-(9) establish that the standard deviation brings down the load carrying capacity. Further, the combined effect of porosity and standard deviation adversely affect the performance characteristics. Besides, the effect of porosity on the variation of load carrying capacity with respect to the standard deviation is almost marginal up to the porosity value $\psi = 0.001$.

The fact that positive variance decreases the load carrying capacity while, the load carrying capacity increases owing to variance (-ve), is manifest in figures (10) – (12). Here also, the effect of porosity is almost negligible up to the porosity value $\psi = 0.001$.

Figures (13)-(14) testify that skewness follows the path of variance so far as the distribution of load carrying capacity is concerned. In other words the negatively skewed roughness further increases the already increased load carrying capacity due to variance(-ve)

It is revealed that in this type of bearing system the combined effect of porosity and aspect ratio may play a crucial role in bearing design. (Figure (15))

However, the negative effect induced by porosity and standard deviation can be neutralized to a significant extent by the positive effect of magnetization at least in the case of negatively skewed roughness. This effect further enhances when variance (-ve) is in place.

The results presented in graphical forms make it clear that the ferrofluid lubrication goes a long way in reducing the adverse effect of surface roughness (Figure-(3)) by suitably choosing the aspect ratio (Figure (4)) in the case of negatively skewed roughness.

Conclusion:

This article makes it clear that roughness must be given priority while designing this type of bearing systems, even if a suitable form of the magnitude of the magnetic field has been considered. Besides, this type of bearing system supports a good amount of load even when there is no flow, which is very much unlikely in the case of conventional lubricant based bearing system. Besides, this study provides an additional degree of freedom through the form of the magnitude of the magnetic field.

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Figure: 1 Variation of load carrying capacity with respect to μ* and σ*

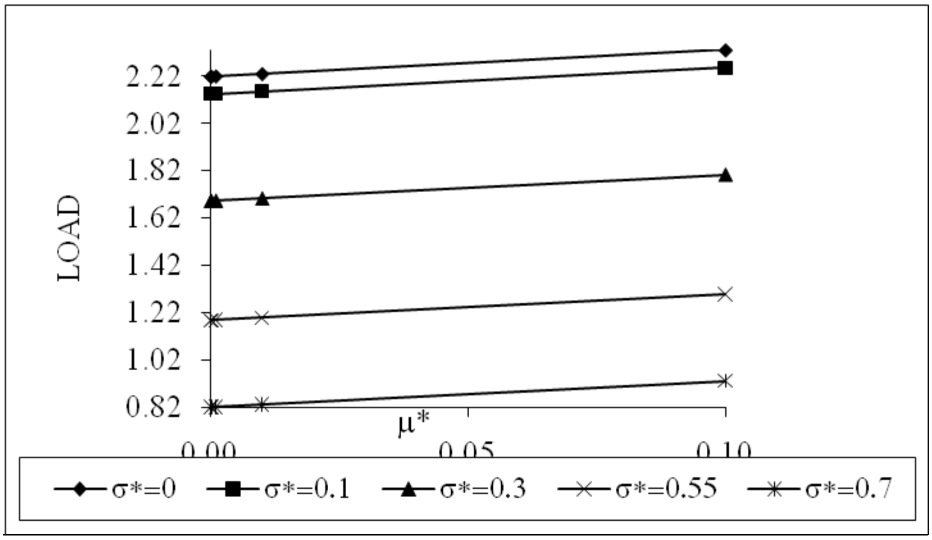

Figure: 2 Variation of load carrying capacity with respect to μ* and α*

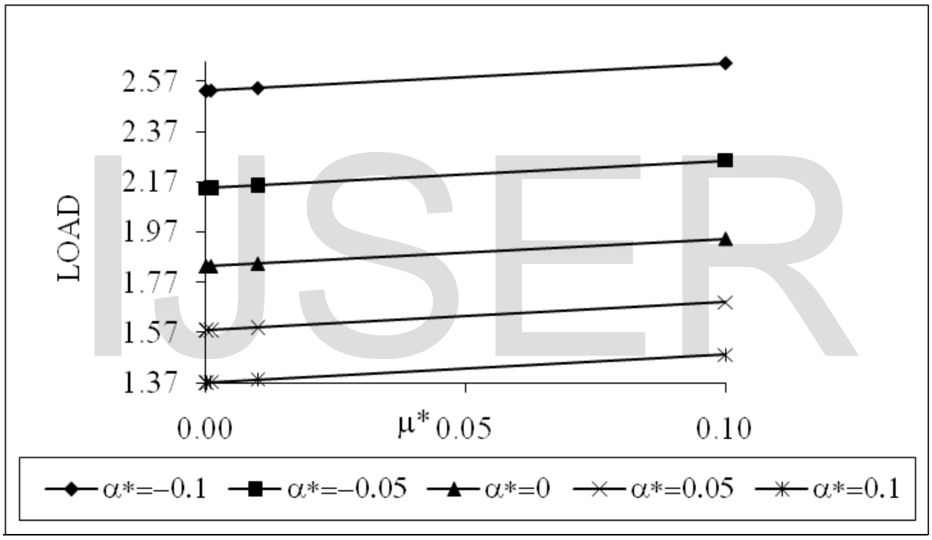

Figure: 3 Variation of load carrying capacity with respect to μ* and ε*

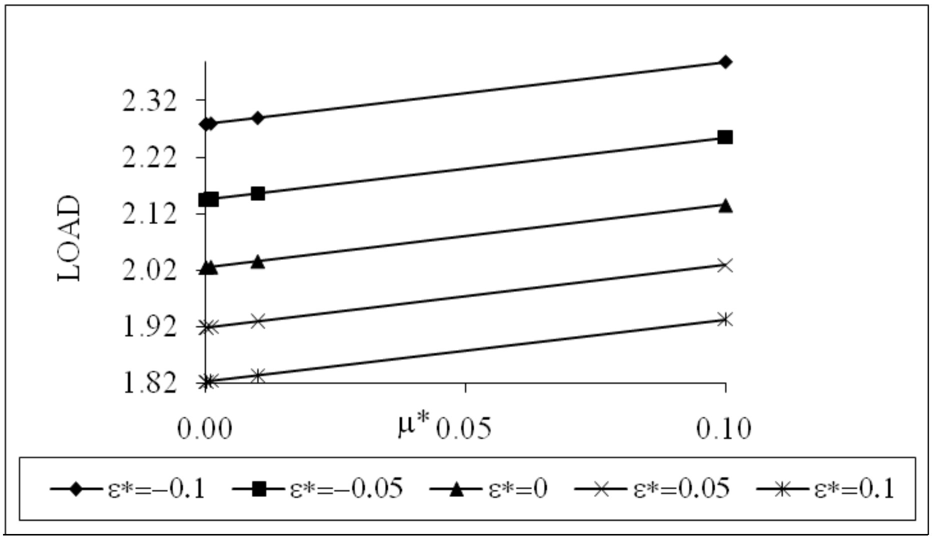

Figure: 4 Variation of load carrying capacity with respect to μ* and ψ

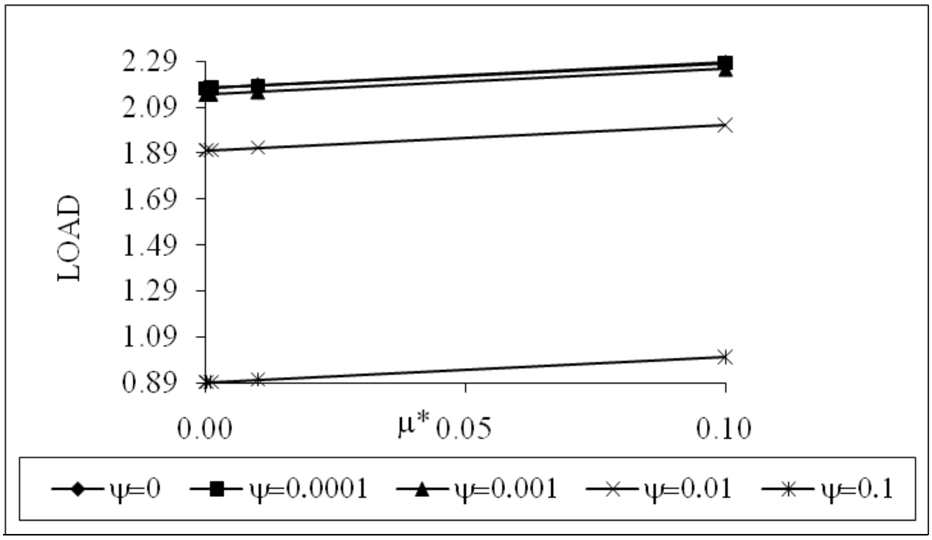

Figure: 5 Variation of load carrying capacity with respect to μ* and k_1

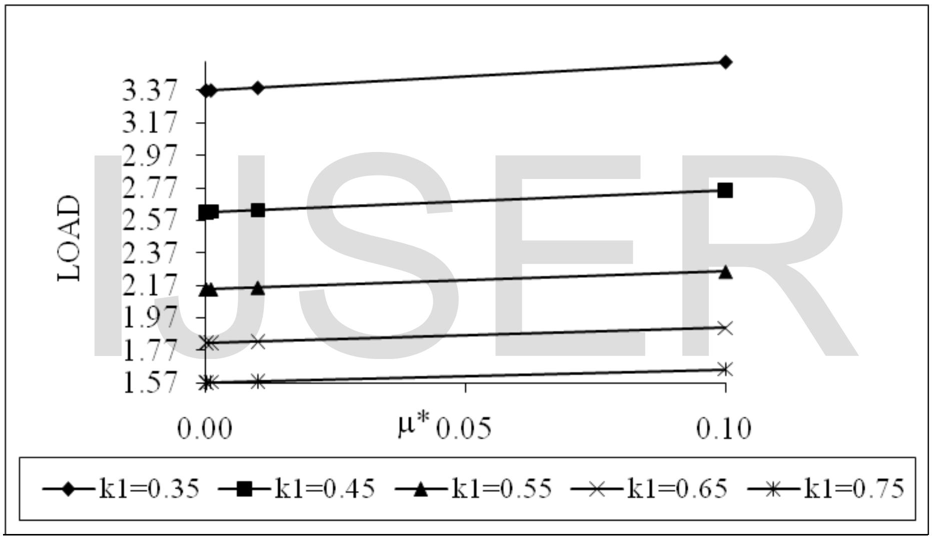

Figure: 6 Variation of load carrying capacity with respect to σ^* and α^*

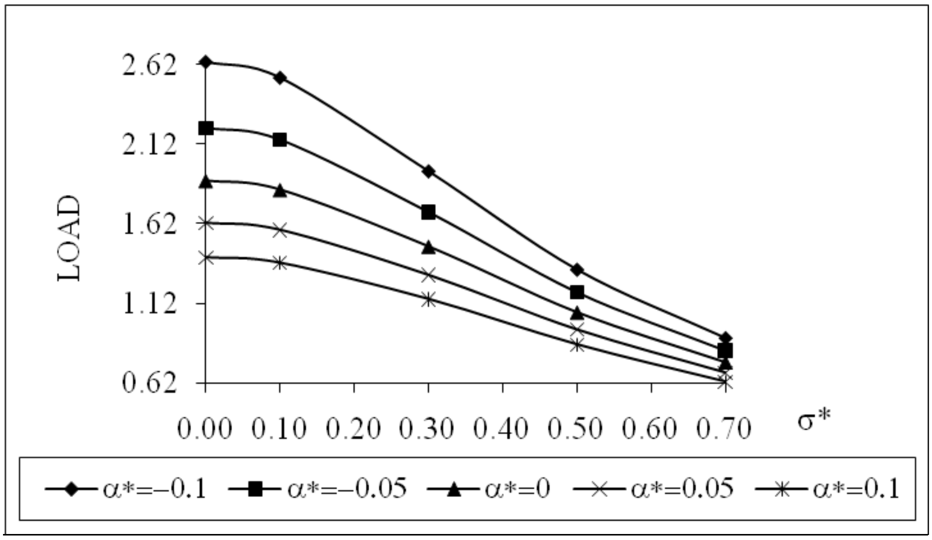

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Figure: 7 Variation of load carrying capacity with respect to σ^* and ε^*

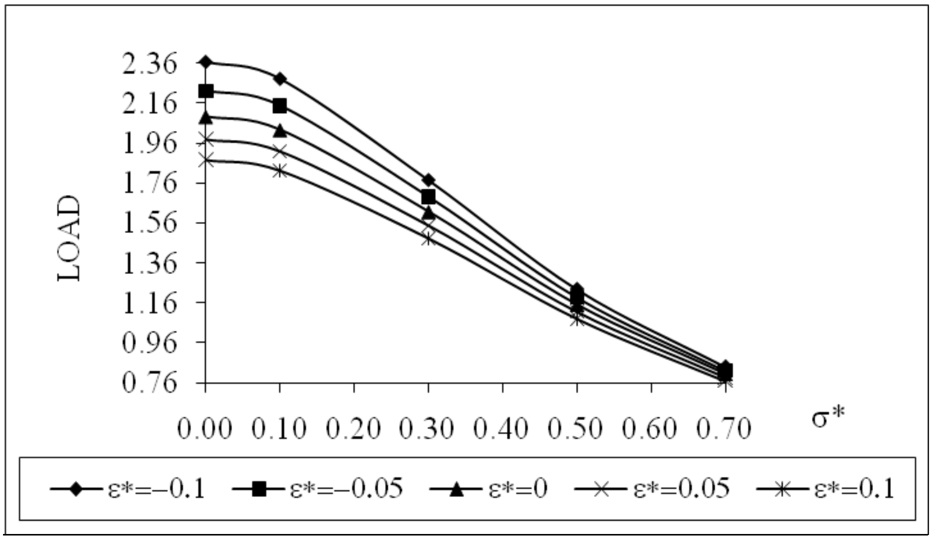

Figure: 8 Variation of load carrying capacity with respect to σ^* and ψ

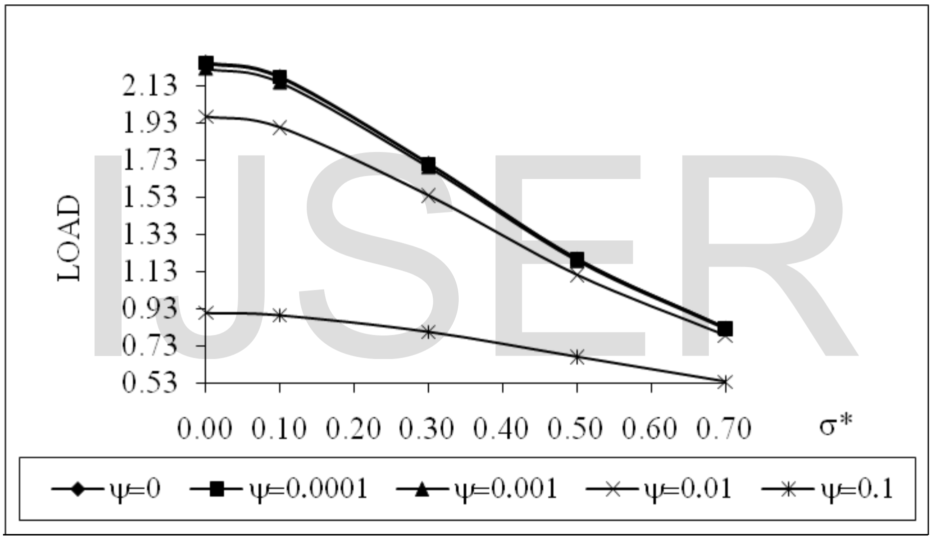

Figure: 9 Variation of load carrying capacity with respect to σ^* and k_1

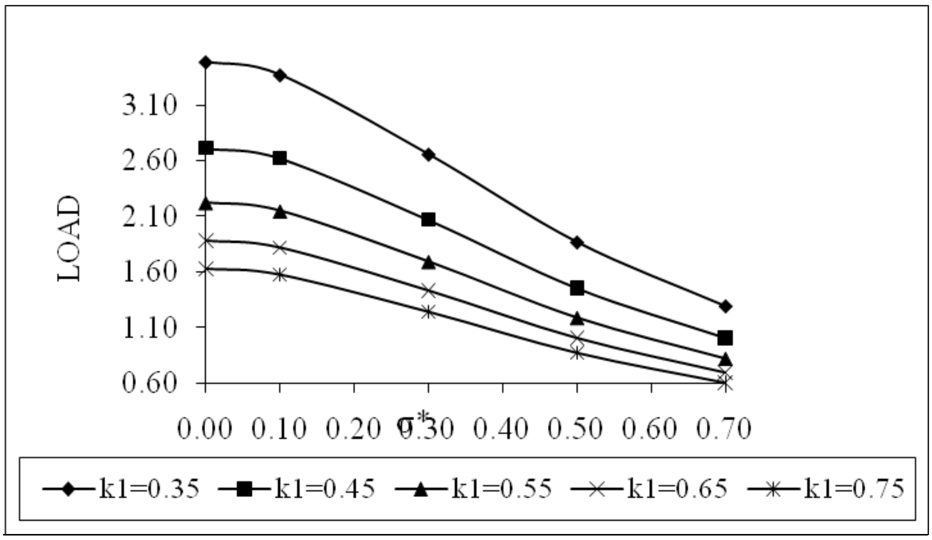

Figure: 10 Variation of load carrying capacity with respect to α^* and ε^*

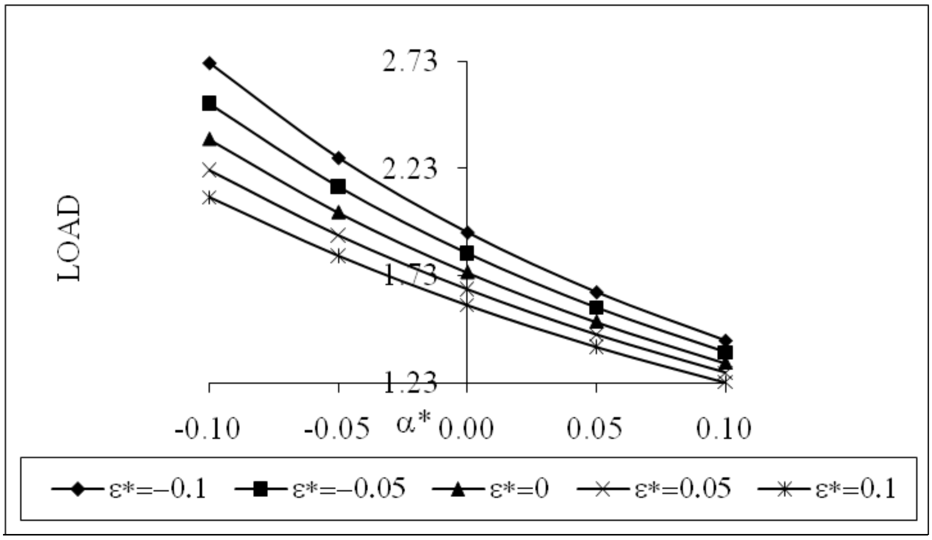

Figure: 11 Variation of load carrying capacity with respect to α^* and ψ

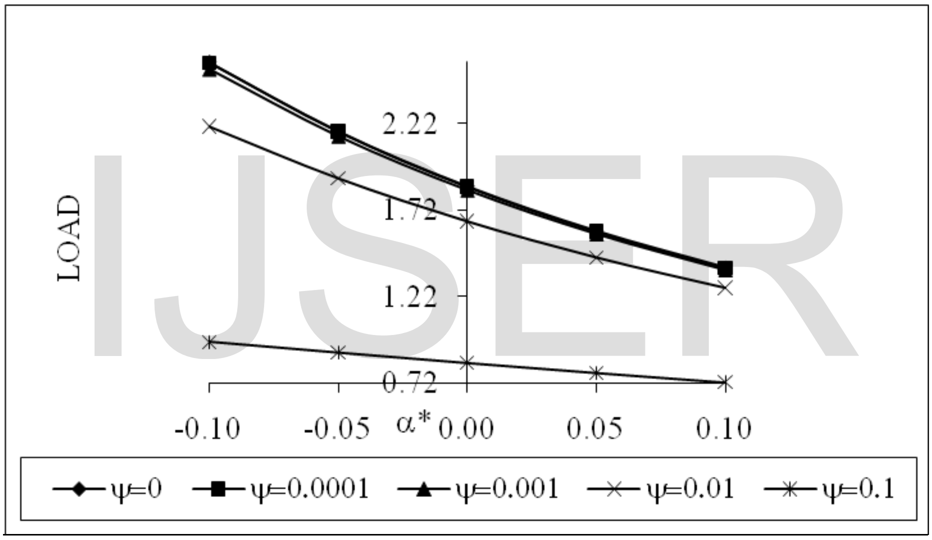

Figure: 12 Variation of load carrying capacity with respect to α^* and k_1

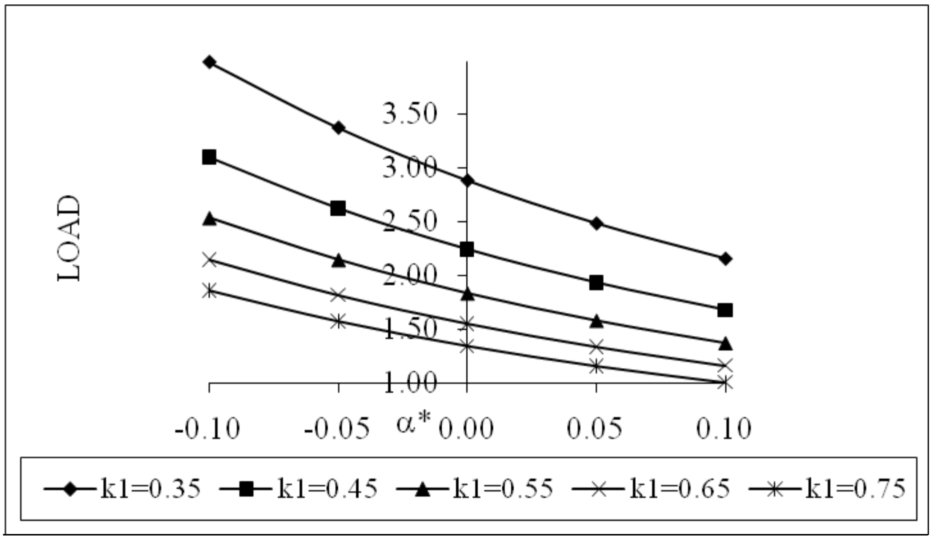

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Figure: 13 Variation of load carrying capacity with respect to ε^* and ψ

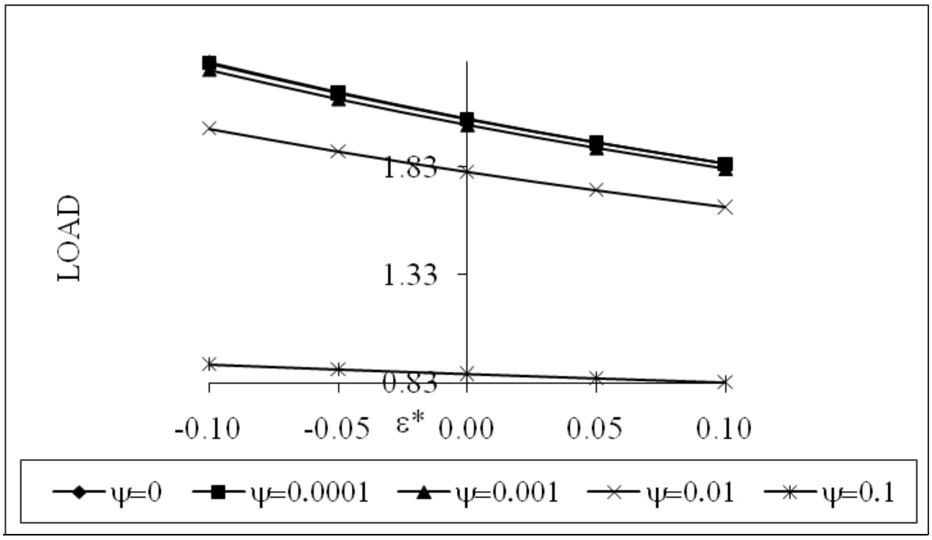

Figure: 14 Variation of load carrying capacity with respect to ε^* and k_1

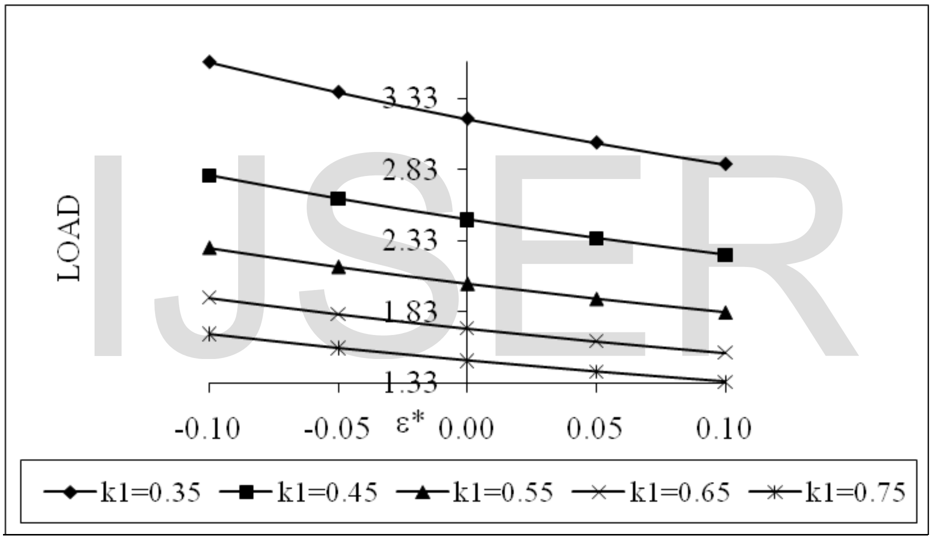

Figure: 15 Variation of load carrying capacity with respect to ψ and k_1