International Journal of Mathematical Archive-5(2), 2014, 10-19
Available online through www.ijma.info ISSN 2229 – 5046

SURFACE ROUGHNESS EFFECT IN FINITE ELLIPTICAL JOURNAL BEARING OPERATING WITH COUPLE STRESS FLUID

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(Received on: 19-01-14; Revised & Accepted on: 10-02-14)

ABSTRACT

This paper is to theoretically investigate the behavior of Finite Elliptical journal bearing lubricated with non-Newtonian couple stress fluid lubricant and surface roughness. The modified Stochastic Reynold's Equation is derived on the basis of Stokes micro-continuum theory for couple stress fluid and Christensen Stochastic theory for the rough surfaces. Closed form solution of the stochastic Reynold's equation is derived. The importance of roughness and couple stress on bearing characteristics are presented in terms of pressure and load carrying capacity. It is observed that, effect of couple stress fluid, and surface roughness is more pronounced in Elliptical bearing than to circular bearing.

Keywords: *Couple stress fluid, Rough Surface, Circular bearing, Elliptical bearing. Finite Difference Method.*

1. INTRODUCTION

Hydrodynamic journal bearings have been widely used to support high speed rotating machinery such as Turbines, Compressors, Pumps, Engine Cranks, Gear Boxes, Large Milling Systems, etc. In such machinery, non-circular bearings like Elliptical in shape, etc. are used to support the Oil Whirl. Lubricants in classical hydrodynamic lubrication analyses are assumed to behave as Newtonian fluid constitutive approximation is not a satisfactory engineering approach to lubrication problems. Experimental results showed that the addition of small amounts of long-chained additives to a Newtonian fluid gives the most desirable lubricants. The behavior of additives added oils is no longer Newtonian rather it is non-Newtonian. A number of micro continuum theories have been derived to explain the peculiar behaviour of the fluids containing a structure such as polymeric fluids [1, 2]. The micro continuum theory derived by Stokes [22, 23] is the simplest generalization of the classical theory of fluids which allows for the polar effects such as the presence of couple stresses and body couples. Jayachandra Reddy *et.al*, [12] analyzed the effect of viscosity variation on the static performance of a narrow journal bearing operating with couple stress fluid. Narasimha Reddy *et. al*., [17] have studied effects of viscosity variation due to additives on squeeze film characteristics of long partial journal bearing operating with couple stress fluid.

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The couple stress effects are considered as a result of the action of one part of a deforming body on its neighborhood. It is well known in the tribology literature that the bearing surfaces develop roughness after having some run-in and wear. The chemical degradation of lubricants leading to the contamination of lubricants is also one of possible reasons for developing the roughness on bearing surfaces in some cases. Since the surface roughness distribution is random in nature, a stochastic approach to model the surface roughness mathematically has to be adopted. Several approaches have been proposed to study the effect of roughness of the bearing surfaces on the performance of hydrodynamic bearings. Davies [7] employed a saw tooth curve to mathematically model the surface roughness. Burton [4] modeled the roughness by a Fourier series type approximation. Mitchell [16] described surface roughness by a high frequency Sine curve. Christensen [6] developed a stochastic model for the study of rough surfaces in hydrodynamic lubrication of solid bearings. Prakash and Tiwari [19] used the Christensen stochastic theory to study the effect of surface roughness on the characteristics of porous bearing. Bujurke and Naduvinamani [5] have studied the effect of surface roughness on the squeeze film lubrication between anisotropic porous rectangular plates and found sensitivity of the maximum load to the roughness parameter. Gururajan and Prakash [8] have studied the effect of surface roughness on the performance characteristics of porous journal bearing operating under steady conditions.

The performance characteristics of Elliptical Bearings have been presented by various authors: considering the static performance, Pinkus, 1956 [18]; Kumar *et al.*, 1980 [14]; Li *et al.*, 1980 [15]. The stability limits to support the flexible rotors, Ajeet Singh and Gupta, 1982 [3]; two different two-lobe bearings of elliptical form under the turbulent flow conditions, Hashinoto *et al*., 1984 [9], 1985 [10]; the effect of Ellipticity Ratio on the static and dynamic characteristics in a laminar flow conditions, Vaidynanathan and Keith, 1989[24]; whereas the dynamic behavior of Short-Elliptical Bearings was analyzed theoretically, Hiromu Hashimoto, 1992 [11] and on the performance of Short–Elliptical Bearing operating with the Couple Stress Fluid, Jaya Chandra Reddy *et.al.*, 2007 [13]. Micropolar lubricant effects on the performance of noncircular lobed bearings using finite element method, Rahmatabadi *et.al* [20].

All the investigations mentioned above are confined to study the surfaces are rough and they interact with the lubricant when the film thickness is very small. Thus keeping this in view in this paper a generalized Reynolds equation for rough surfaces by taking roughness interaction zones near the surfaces and purely hydrodynamic zone in the middle layer is derived and is applied to Finite Elliptical bearings. The lubricant is taken as a couple stress fluids and also solved by using Finite Difference Method.

2. MATHEMATICAL FORMULATION AND SOLUTION OF THE PROBLEM

Consider the flow of an incompressible lubricant between two moving rough surfaces as shown in Fig. 1

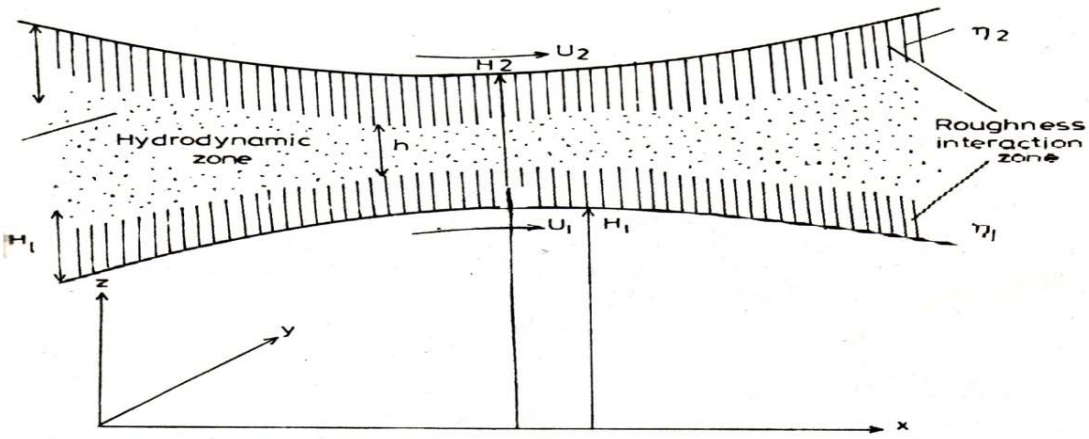

Fig. 1: Couple stress fluid lubrication between two rough surfaces

$$\rho\frac{DV}{Dt} = -\nabla p + \rho F + \frac{1}{2}\nabla\times(\rho C) + \mu\nabla^2 V - \eta\nabla^4 V \tag{1}$$

$$\nabla\cdot V = 0 \tag{2}$$

where the vectors V, F and C represent the velocity, body force per unit mass, and body couple per unit mass; ρ, the density ; μ is the shear viscosity, and η is a new material constant responsible for the couple stress fluid property.

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0 \tag{3}$$

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$$\mu \frac{\partial^2 u_2}{\partial y^2} - \eta \frac{\partial^2 u_2}{\partial y^4} = \frac{\partial p}{\partial x} \tag{4}$$

$$\mu \frac{\partial^2 u_2}{\partial y^2} - \eta \frac{\partial^2 u_2}{\partial y^4} = \frac{\partial p}{\partial z} \tag{5}$$

$$\frac{\partial p}{\partial y} = 0 \tag{6}$$

u, v, and w are the velocity components in the x, y, and z – directions respectively. The boundary conditions at the bearing surfaces are

$$\frac{\partial^2 u_2}{\partial y^2} = 0 \;\; at\; z = H_1 + H_l \;\; and \;\; z = H_2 - H_f \tag{7}$$

$$\frac{\partial^2 v_2}{\partial y^2} = 0 \;\; at\; z = H_1 + H_l \;\; and \;\; z = H_2 - H_f \tag{8}$$

Integrating equation (3) across the film thickness i.e., from H_1 to H_2, And using the boundary conditions (7) & (8), the Reynolds equation becomes

$$\frac{\partial}{\partial x}\left(F \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial y}\left(F \frac{\partial p}{\partial y} \right) = \frac{U}{2} \frac{d}{dx}(h + 2a) \tag{9}$$

Equation (9) can be applied to elliptical bearing to analyze the roughness and couple stress effects in the consequent section.

3. BASIC EQUATION

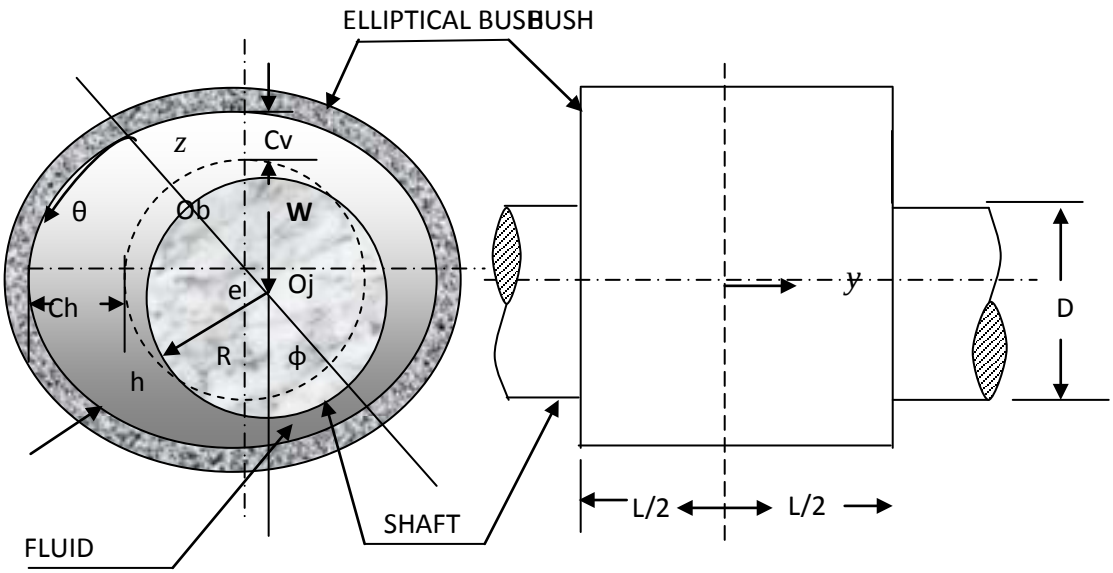
The equations governing the fluid flow in the bearing is given by equation (9) as

$$\frac{\partial}{\partial x}\left[\frac{F}{12\mu} \frac{\partial p}{\partial x} \right] + \frac{\partial}{\partial y}\left[\frac{F}{12\mu} \frac{\partial p}{\partial y} \right] = \frac{U}{2} \frac{dh_n}{dx} \tag{10}$$

where
$$F = 6 \frac{M - T\,\mathrm{anh}\, M}{M^3} \frac{(2h_s)^3}{k} + \left[(h_n - 2h_s) + 2h_s \frac{Tanh(M)}{M} \right]^2 12h_s \frac{Tanh(2M)}{k(2M)}$$
$$+ \left(h_n - 2h_s \right)^3 - 12l^2 \left(h_n - 2h_s \right) + 24l^3 Tanh\left(\frac{h_n - 2h_s}{2l} \right)$$
$$M = \alpha h_s, \quad \alpha = \sqrt{\frac{1}{k\varphi}} \;\; , \;\; l = \left(\frac{\eta}{\mu} \right)^{1/2} \tag{11}$$

Here h_n is the nominal film thickness, h_s is the mean height of surface asperities, φ is the roughness parameter of the bearing surface, k is the ratio of lubricant viscosities at the rough surface to purely hydrodynamic zone and l characteristics the couple stress property of the lubricant.

The physical configuration of an elliptical hydrodynamic bearing is shown in the following Fig.2. The film thickness is expressed by the following continuous function



R = Shaft Radius
C_H = Horizontal Clearance
O_B = Bearing Center
e = Eccentricity
L = Length of the Bearing
C_V = Vertical Clearance
O_J = Journal Center
h = Fluid Film Thickness.

Fig. 2: Geometry of an Elliptical Bearing.

$$h_n = c_H\left(1 - m + \varepsilon\cos\theta + m\sin^2(\theta+\phi)\right) \tag{12}$$

where $m = \dfrac{C_H - C_V}{C_H}$ = Pre - load factor or Ellipticity ratio

C_H = Horizontal clearance $\qquad C_V$ = Vertical clearance

θ = Circumferential angle of the bearing and ϕ Attitude angle

4. METHOD OF SOLUTION

The dimensionless modified Reynolds equation is solved numerically using finite difference method [21]. The lubricating oil film is divided by the grid spacing and the equation (11) can be written as

$$4\lambda^2\frac{1}{\Delta\theta}\left[F_{i+\frac{1}{2},j}\frac{P_{i+1,j}-P_{i,j}}{\Delta\theta} - F_{i-\frac{1}{2},j}\frac{P_{i,j}-P_{i-1,j}}{\Delta\theta}\right] + \frac{1}{\Delta\bar{z}}\left[F_{i,j+\frac{1}{2}}\frac{P_{i,j+1}-P_{i,j}}{\Delta\bar{z}} - F_{i,j-\frac{1}{2}}\frac{P_{i,j}-P_{i,j-1}}{\Delta\bar{z}}\right]$$
$$= 48\lambda^2\left(-\varepsilon\sin\theta + m\sin 2(\theta+\varphi)\right) \tag{13}$$

$$P_{i+1,j}\left[4\lambda^2 r^2 F_{i+\frac{1}{2},j}\right] - P_{i,j}\left[4\lambda^2 r^2\left[F_{i+\frac{1}{2},j} + F_{i-\frac{1}{2},j}\right] + \left[F_{i,j+\frac{1}{2}} + F_{i,j-\frac{1}{2}}\right]\right] + P_{i-1,j}\left[4\lambda^2 r^2 F_{i+\frac{1}{2},j}\right]$$
$$+P_{i-1,j}\left[4\lambda^2 r^2 F_{i+\frac{1}{2},j}\right] + P_{i,j+1}\left[F_{i,j+\frac{1}{2}}\right] + P_{i,j-1}\left[F_{i,j-\frac{1}{2}}\right] = 48\lambda^2 r^2\left(-\varepsilon\sin\theta + m\sin 2(\theta+\varphi)\right) \tag{14}$$

$$C_0 P_{i,j} = C_1 P_{i+1,j} + C_2 P_{i-1,j} + C_3 P_{i,j+1} + C_4 P_{i,j-1} + C_5 \tag{15}$$

$$C_0 = 4\lambda^2 r^2\left[F_{i+\frac{1}{2},j} + F_{i-\frac{1}{2},j}\right] + \left[F_{i,j+\frac{1}{2}} + F_{i,j-\frac{1}{2}}\right]$$

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$$C_1 = 4\lambda^2 r^2 F_{i+\frac{1}{2},j}\ ;\ C_2 = 4\lambda^2 r^2 F_{i-\frac{1}{2},j}\ ; C_3 = F_{i,j+\frac{1}{2}}\ ;\ C_3 = F_{i,j-\frac{1}{2}}$$
$$C_5 = 48\lambda^2 \Delta\bar{z}^2(-\varepsilon\sin\theta + m\sin 2(\theta+\phi); r = \frac{\Delta\bar{z}}{\Delta\theta}; \Delta\bar{z} = 0.05; \Delta\theta = 9^0 \tag{16}$$

$$F = 6\frac{M - TanhM}{M^3}\frac{(2h_s)^3}{k} + \left[\left(h_n - 2h_s\right) + 2h_s\frac{Tanh(M)}{M}\right]^2 12h_s\frac{Tanh(2M)}{k(2M)} + \left(h_n - 2h_s\right)^3$$
$$-12l^2\left(h_n - 2h_s\right) + 24l^3 Tanh\frac{h_n - 2h_s}{2l}$$

5. BEARING STATIC CHARACTERISTIC

Once the fluid film pressure is known from the above equation (16), the bearing load carrying capacity can be obtained

Load Carrying Capacity: The load carrying capacity is obtained by integrating the fluid film pressure acting on the journal. The load component along the line of centers in the non dimensional form is

$$\overline{W}_0 = \overline{W}\ \cos\phi = \int_{-\frac{1}{2}}^{\frac{1}{2}}\int_0^{\pi}\overline{P}\cos\theta\, d\theta\, d\bar{z} = \sum_{i=0}^{m}\sum_{j=0}^{m}\overline{P}_{i,j}\cos\theta_i\Delta\theta\Delta\bar{z}$$
$$\overline{W}_{\frac{\pi}{2}} = \overline{W}\ \sin\phi = \int_{-\frac{1}{2}}^{\frac{1}{2}}\int_0^{\pi}\overline{P}\sin\theta\, d\theta\, d\bar{z} = \sum_{i=0}^{m}\sum_{j=0}^{m}\overline{P}_{i,j}\sin\theta_i\Delta\theta\Delta\bar{z} \tag{17}$$
$$\overline{W} = \sqrt{\overline{W}_0^{\,2} + \overline{W}_{\frac{\pi}{2}}^{\,2}} \quad where\ \overline{W} = \frac{WC_h^{\,2}}{\mu ULR^2}$$

6. RESULT AND DISCUSSIONS

The variation of non-dimensional pressure $\bar{P}$ for different eccentricity ratios ε and ellipticity ratios i.e., m=0 represents circular bearing and m=0.15 represents elliptical bearing at $\bar{l} = 0.12, \bar{h}_s = 0.02, \phi = \frac{\pi}{6}, k = 2, \bar{\varphi} = 0.001$ is shown in figure-3. It is observed that as the eccentricity ratio increases the non dimensional pressure also increases for both the bearings, due to the decrease in the oil film thickness. The value of $\bar{P}$ is more for elliptical bearing than circular bearing, due to more convergent portion. Increase in $\overline{p}$ is more pronounced for larger values of eccentricity ratio and preload factor.

The figure -4 represents the variation in non dimensional pressure $\bar{P}$ for different $\bar{h}_s$, for both circular and elliptical bearing at $\bar{l} = 0.12, \varepsilon = 0.4, \varphi = \frac{\pi}{6}, k = 2, \bar{\phi} = 0.001.$ It is observed that as the $\bar{h}_{s.}$ It is observed that as the $\bar{h}_s$ increases, $\overline{p}$ also increases, for both the bearings, due to the less side leakage of the lubricant oil. For the same condition the pressure rise in elliptical bearing is two times the pressure in circular bearing.

If $\bar{\varphi}$ value is less than 1 represents the transverse roughness where as greater than 1 represents the longitudinal roughness.

Figure-5, 6 & 7 indicate, the dimension less load carrying capacity $\bar{w}$ verses surface roughness parameter, $\bar{h}_s$, it is seen that from Figure-5 that the load carrying capacity increases for k > 1 as $\bar{h}_s$ increases when there is no roughness and decreases for k < 1, i.e., the load carrying capacity increases as the thickness of peripheral layer increases, when the peripheral layer viscosity more than the middle layer viscosity. In both the cases i.e,, for circular bearing and elliptical bearing. Viscous layer parameter (k) increases with respect to surface roughness $\bar{h}_s$ the load carrying capacity is increases.

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From Figure-6, it is observed that dimensionless load carrying capacity $\bar{w}$ is depicted with surface roughness parameter ($\bar{h}_s$), the mean height of roughness asperities for different couple stress parameter ($\bar{l}$). As the mean height of roughness parameter $\bar{h}_s$ increases, the load carrying capacity $\bar{w}$ increases for both the bearings.

From Figure -7, it is indicated that the dimension less load carrying capacity $\bar{w}$ is plotted with surface roughness parameter $\bar{h}_s$, for different values of directional roughness parameter $\bar{\varphi}$, for low values of $\bar{\varphi}$, i.e., for transverse roughness $\bar{w}$ increases and for higher values of $\bar{\varphi}$, i.e., for longitudinal roughness, $\bar{w}$ is decreases. However, as the chain length of the additive molecules increases the load carrying capacity increases for both the bearings.

From Figure-8, the dimension less load carrying capacity $\bar{w}$ is depicted against the couple stress parameter $\bar{l}$ for varies eccentricities (ε). As $\bar{l}$ increases, the load carrying capacity $\bar{w}$ increases with increase in ε . As the higher values of 'ε' the bearing is lubricated with couple stress fluids shows a larger load carrying capacity in both the bearings.

CONCLUSIONS

A modified Reynolds equation is derived to study the combined effect of couple stress and surface roughness.As the roughness parameter $\bar{h}_s$ increases, the dimensionless pressure $\bar{p}$ also increases, for both the bearings, due to the less side leakage of the lubricant oil. For the same condition the pressure rise in elliptical bearing is two times the pressure in circular bearing.

It is concluded that as the surface roughness parameter increases load carrying capacity also increases for both the Circular and Elliptical Bearings, due to the reduction of side leakage of the lubricant as surface roughness asperities presence.

The load carrying capacity is more for elliptical bearing compared to the Circular bearing because of more convergent film formation in the upper lobe of the elliptical bearing.

In both the cases i.e., for circular bearing and elliptical bearing, Viscous layer parameter (k) increases with respect to surface roughness,$\bar{h}_s$, the load carrying capacity is also increases. As the mean height of roughness parameter $\bar{h}_s$ increases, the load carrying capacity $\bar{w}$ increases for both the bearings. This means that enhanced viscosity near the surfaces and the large chain length of the molecule in the lubricant increase the bearing life considerably for both the bearings.

NOMENCLATURE

a	The mean height of surface asperities in the symmetric roughness case
C_H	Horizontal clearance
C_V	Vertical clearance
h_n	Nominal film thickness
h_s	Mean height of roughness asperities
H_l, H_f	Average height of surface asperities at lower and upper rough surfaces
k	Ratio of the viscosities near the surface to the purely hydrodynamic zone
$k_{1x}, k_{1y}, k_{2x}, k_{2y}$	The resistances offered to the flow near the roughness interaction zones in x and y directions and at lower and upper surfaces
l	Couple stress parameter
m	Pre – load factor or Ellipticity ratio
p	Hydrodynamic pressure
Q_x, Q_y	The lubricant flux in *x* and *y* directions
R	Radius of the shaft
η_1, η_2	The effective viscosities adjacent the bearing surfaces
$\varphi_{1x}, \varphi_{1y}, \varphi_{2x}, \varphi_{2y}$,	Permeability of roughness interaction zones at the lower and upper surfaces in *x* and *y* directions
u_1, u_2, u_3 v_1, v_2, v_3	The lubricant velocities at different zones in *x* and *y* directions

U_1, U_2	Surface velocities of lower and upper surfaces
U_i, U_j, V_i, V_j	The interface velocities in *x* and *y* directions
U	Velocity of the surfaces in the case of one-dimensional form
V_1, V_2	The surface velocities of lower and upper surfaces in y-direction
u, v, w	The velocity of the fluid in *x, y* and *z* directions
x, y, z	Cartesian coordinates
μ	Viscosity of the base lubricant
φ	Roughness parameter
θ	Circumferential angle of the bearing
ϕ	Attitude angle
η	Material coefficient of the couple stress fluid
φ_x, φ_y	The permeability of roughness interaction zones in the symmetric roughness case

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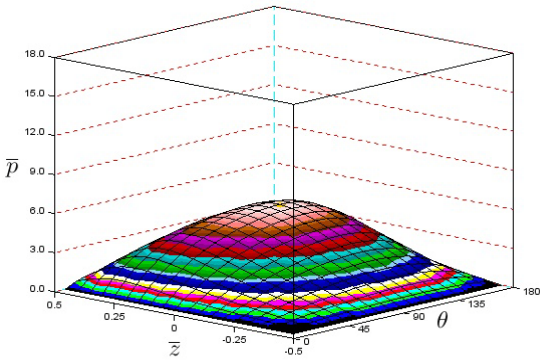
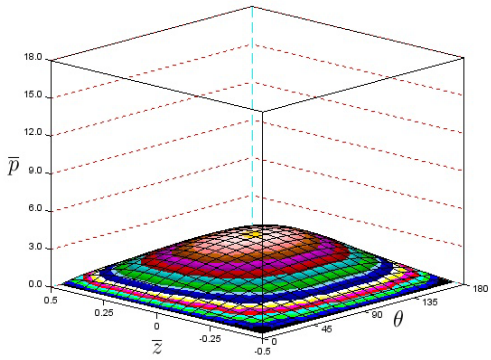


ε=0.4, m=0.15



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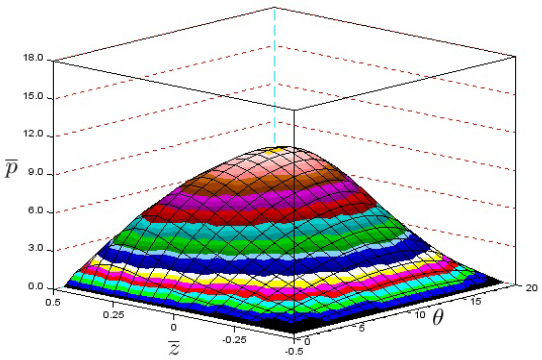

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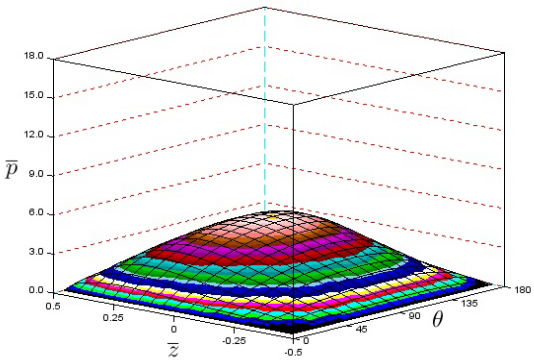
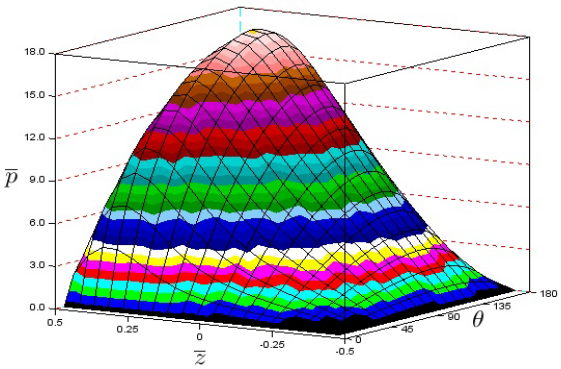
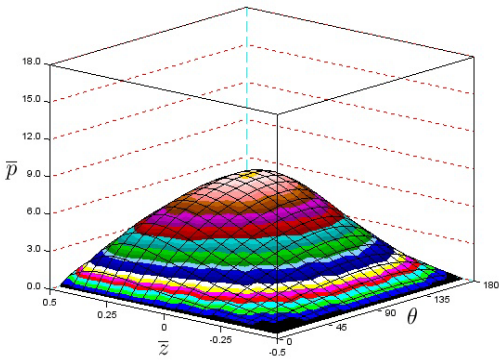


ε=0.5,m=0



ε=0.6,m=0.15



ε=0.6,m=0.0

Figure-3: $\bar{l} = 0.12, \bar{h}_s = 0.02, \phi = \frac{\pi}{6}, k = 2, \bar{\varphi} = 0.001$

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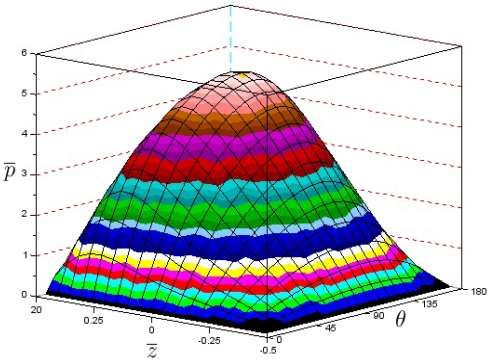
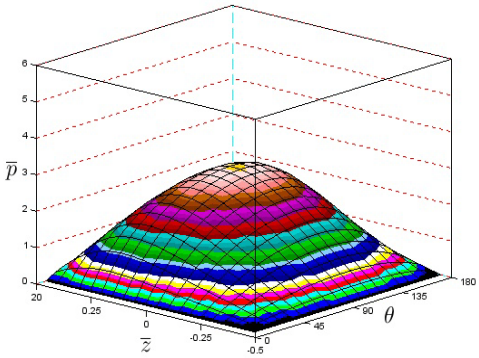
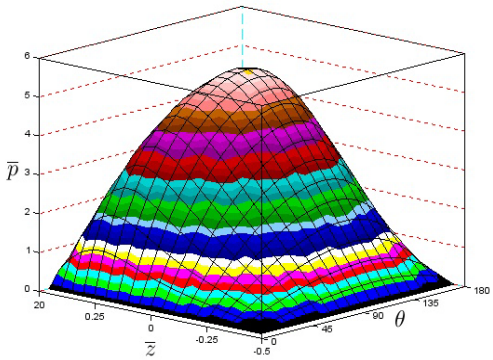


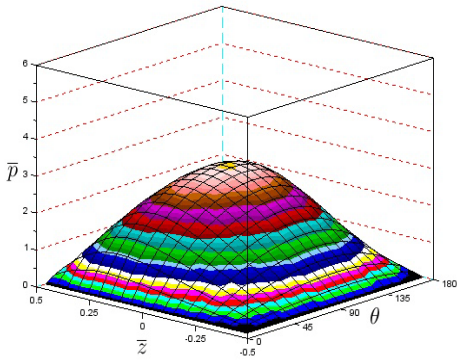
$\overline{h_s} = 0.005, m = 0.15$



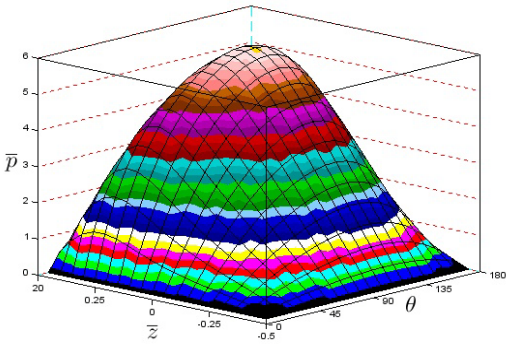
$\overline{h_s} = 0.005, m = 0.0$



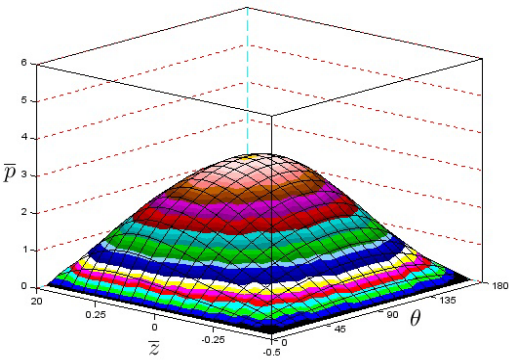
$\overline{h_s} = 0.025, m = 0.15$



$\overline{h_s} = 0.025, m = 0.0$



$\overline{h_s} = 0.04, m = 0.15$



$\overline{h_s} = 0.04, m = 0.0$

Figure-4: $\bar{l} = 0.12, \varepsilon = 0.4, \phi = \dfrac{\pi}{6}, k = 2, \overline{\varphi} = 0.001.$

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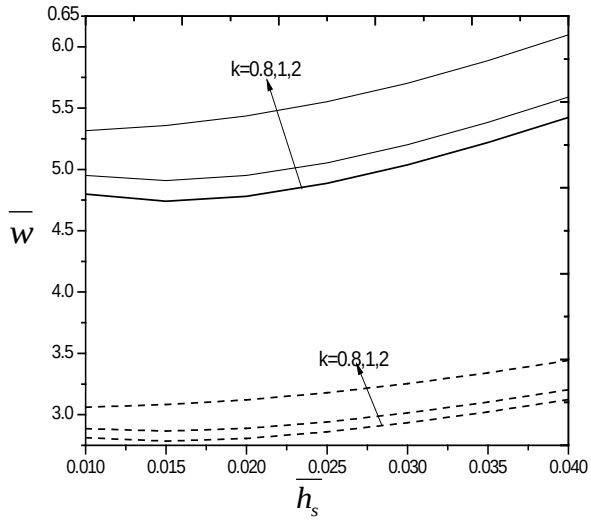

Figure-5: Dimension load Vs $\bar{h}_s$ For Different k with $\bar{\varphi} = 0.001, \varepsilon = 0.4, \bar{l} = 0.06$

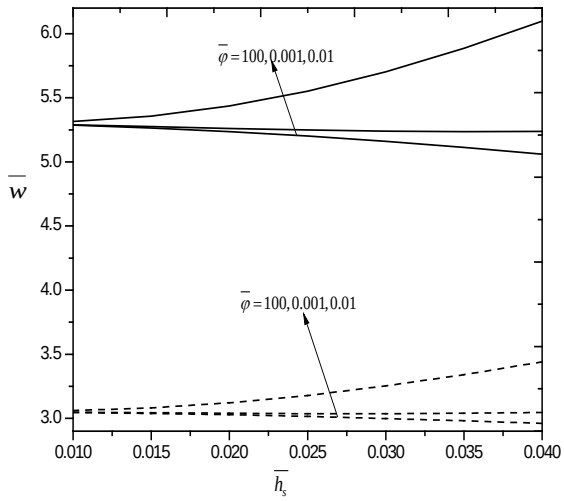

Figure-7: Dimension load Vs $\bar{h}_s$ for different $\bar{\varphi}$ with $\bar{l} = 0.04, \varepsilon = 0.4, k = 2$

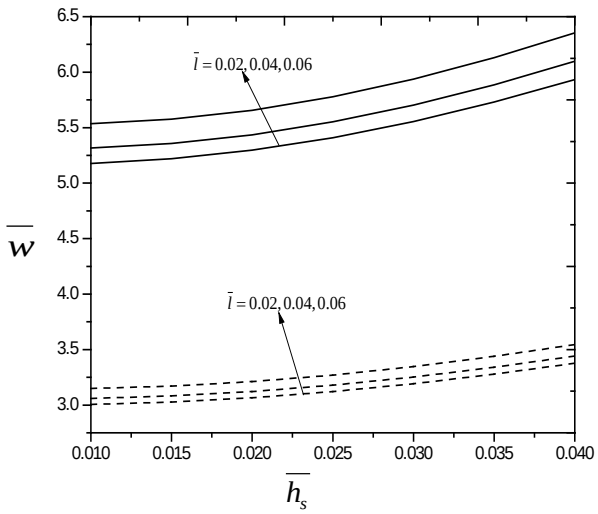

Figure-6: Dimension load Vs $\bar{h}_s$ for different $\bar{l}$ with $\bar{\varphi} = 0.001, \varepsilon = 0.4, k = 2$

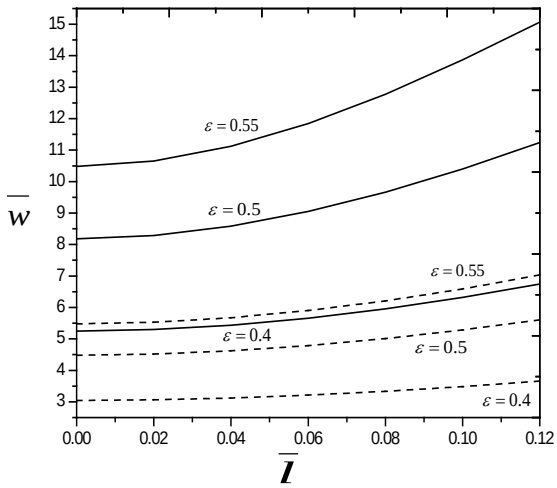

Figure-8: Dimension load Vs $\bar{l}$ for different values of ε with $\bar{h}_s = 0.02, \bar{\varphi} = 0.001, k = 2$

Source of support: Nil, Conflict of interest: None Declared

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