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MICROPOLARITY-SURFACE ROUGHNESS INTERACTION IN HYDRODYNAMIC LUBRICATION OF LONG JOURNAL BEARINGS

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ABSTRACT

This paper describes a theoretical analysis of micropolarity-surface roughness interaction of infinitely long journal bearing. The modified averaged Reynolds type equation for the study of surface roughness has been derived on the basis of Eringen's micropolar fluid theory. The generalized stochastic random variable with non-zero mean, variance and skewness is used to mathematically model the surface roughness on the bearing surface. The more accurate Reynolds boundary conditions are utilized to solve the average Reynolds type equation. The numerical results are obtained for the fluid film pressure, load carrying capacity and the frictional force. It is found that the performance of the long journal bearing is dependent on the type of roughness pattern on the bearing surface and the lubricant additives. Further, it is found that the performance of the bearing improves due to the presence of negatively skewed surface roughness on the bearing surface and these effects are more pronounced for the micropolar lubricants.

Keywords: Surface roughness, journal bearing, micropolar fluid.

INTRODUCTION

The theoretical study of journal bearings becomes more realistic due to the consideration of many physical effects. The study of journal bearings with an assumption of smooth bearing surface will not predict the bearing performance accurately. Generally the size of the roughness asperity height is of the same order as the mean separation between the lubricated contacts. Hence, the finish surface that can improve the bearing performance has been sought. Consequently, several attempts have been made to study the surface roughness effects on the bearing performance by using both deterministic and stochastic methods. Out of these, the stochastic models are the best suited to characterize the surface roughness of the bearings for randomly distributed asperities. The stochastic study of Tzeng and Saibel (1967) attracted several investigators in the field of tribology. A new stochastic averaging approach for the study of roughness effects on the hydrodynamic lubrication was proposed by Christesen and Tonder (1969). Raj and Sinha (1983) have studied the effect of transverse surface roughness on the short journal bearings under dynamic loading. Lin *et al*. (2002) studied the Surface roughness effects on the oscillating squeeze film behavior of long partial journal bearings. Recently Naduvinamani and Gurubasavaraj (2004) studied the surface roughness effects on squeeze films in curved circular plates using Christensen's stochastic theory for rough surfaces. All these studies are based on the assumption that, the probability density function for the random variable characterizing the surface roughness is symmetric and has zero mean. However, in general it is valid as a first approximation. But in practice, due to non-uniform rubbing of surface the distribution of surface roughness may be asymmetrical. Andharia *et al*. (2001) proposed the study of effect of surface roughness on hydrodynamic lubrication of slider bearings by modeling the surface roughness by a stochastic random variable with non-zero mean, variance and skewness. Naduvinamani *et al*. (2003) extended the application of this theory to couple stress fluid lubrication of slider bearings with rough surfaces. Recently Naduvinamani and Biradar (2006) studied the effect of surface roughness on porous inclined slider bearings with micropolar fluid.

Several experimental studies show that the effectiveness of lubricating oil can be improved on blending small amounts of long chained polymer additives with Newtonian lubricants. Most of the modern lubricants are mainly the polymer thickened oils or lubricants with additives. These lubricants become heavily contaminated with suspended metal particles and they start to exhibit non-Newtonian behavior. To predict the accurate flow behavior of such lubricants with additives, several microcontinuum theories have been proposed. Eringen's (1966) micropolar fluid theory is the generalization of the classical theory of fluids, which accounts for polar effects. This theory accurately describes the rheological behaviors of lubricants with polymer additives Prakash

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assumption of perfectly smooth surfaces. In this paper, an attempt has been made to study the effect of random roughness on the performance of infinitely long journal bearings lubricated with micropolar fluids. A general type of surface roughness is mathematically modeled by a stochastic random variable with non-zero mean, variance and skewness.

Consider the flow of a micropolar lubricant in a wedge shaped film built up in a journal bearing operating with a surface velocity U under an imposed load W. Fig. 1(a) and 1(b) are schematic diagrams of journal bearing and its developed surfaces. It consists of two surfaces separated by a lubricant film. The inner surface of the journal bearing is moving with a constant velocity U in its own plane while the upper surface is at rest. It is assumed that, the bearing surfaces are rough. The shape of the lubricant film formed between the two surfaces is convergent.

To represent the surface roughness the mathematical expression for the film thickness is considered to be consisting of two parts

$$H(x) = h(x) + h_s \tag{1}$$

where $h(x) = C + e\cos\theta$ is the mean film thickness and h_s is a randomly varying quantity measured from the mean level and thus characterizes the surface roughness, e and C are as shown in figure 1(a).

Further, the stochastic part h_s is considered to have the probability density function $f(h_s)$ defined over the domain $-c \le h_s \le c$. Where c is a maximum deviation from the mean film thickness.

The mean α^*, the standard deviation σ^* and the parameter ε_1^*, which is the measure of symmetry of the random variable h_s are defined as

$$\alpha^* = E(h_s) \tag{2}$$

$$\sigma^* = E\left[(h_s - \alpha^*)^2\right] \tag{3}$$

$$\varepsilon_1^* = E\left[(h_s - \alpha^*)^3\right] \tag{4}$$

where E is an expectation operator defined by

$$E(\bullet) = \int_{-\infty}^{\infty} (\bullet)\, f(h_s)\, dh_s \tag{5}$$

The parameters α^*, σ^* and ε_1^* are all independent of x. The mean α^* and the parameter ε_1^* can assume both positive and negative values whereas σ^* can always assume positive values.

The lubricant in the film region formed between two surfaces of the bearings is assumed to be an incompressible, laminar fluid flow with negligible body forces and negligible inertia forces. The velocity and microrotation vectors for the micropolar lubricant are assumed to be $\vec{V} = (u_1, u_2, u_3)$ and $\vec{V} = (w_1, 0, w_3)$ respectively. Using the usual assumptions of lubrication theory Pinkus and Sternlitch (1961) the basic equations governing the flow of the micropolar lubricant under steady state conditions become

Conservation of mass:

$$\frac{\partial u_1}{\partial x} + \frac{\partial u_2}{\partial y} + \frac{\partial u_3}{\partial z} = 0 \tag{6}$$

Conservation of momentum:

$$(\mu + \mu_1)\frac{\partial^2 u_1}{\partial y^2} + \mu_1 \frac{\partial w_3}{\partial y} - \frac{\partial p}{\partial x} = 0 \tag{7}$$

$$\frac{\partial p}{\partial y} = 0 \tag{8}$$

$$(\mu + \mu_1)\frac{\partial^2 u_3}{\partial y^2} - \mu_1 \frac{\partial w_1}{\partial y} - \frac{\partial p}{\partial z} = 0 \tag{9}$$

Conservation of angular momentum:

$$\gamma \frac{\partial^2 w_3}{\partial y^2} - \mu_1 \frac{\partial u_1}{\partial y} - 2\mu_1 w_3 = 0 \tag{10}$$

$$\gamma \frac{\partial^2 w_1}{\partial y^2} - \mu_1 \frac{\partial u_3}{\partial y} - 2\mu_1 w_1 = 0 \tag{11}$$

where γ and μ_1 are the additional viscosity coefficients for the micropolar lubricants.

The modified Reynolds equation for smooth journal bearing lubricated with micropolar fluid was obtained by Zaheeruddin and Isa (1978) in the form

$$\frac{\partial}{\partial x}\left\{\frac{f(H,\overline{\mu_1})}{6(2\mu + \mu_1)}\frac{\partial p}{\partial x}\right\} + \frac{\partial}{\partial z}\left\{\frac{f(H,\overline{\mu_1})}{6(2\mu + \mu_1)}\frac{\partial p}{\partial z}\right\} = \frac{U}{2}\frac{\partial H}{\partial x} \tag{12}$$

where

$$f(H,\overline{\mu_1}) = H^3\left(1 + \frac{3\overline{\mu_1}}{2(1+\overline{\mu_1})\alpha_1^2 H^2} - \frac{3\overline{\mu_1}}{2(1+\overline{\mu_1})\alpha_1 H}\coth(\alpha_1 H)\right),$$

$$\alpha^2{}_1 = \frac{\lambda^2 C^2}{4} = \frac{M\overline{\mu_1}(2+\overline{\mu_1})}{(1+\overline{\mu_1})}, \quad M = \frac{\mu C^2}{4\gamma},$$

$$\lambda^2 = \frac{(2\mu + \mu_1)\mu_1}{(\mu + \mu_1)\gamma}, \quad \overline{\mu_1} = \frac{\mu_1}{\mu}.$$

For an infinitely long journal bearing equation (12) reduces to

$$\frac{\partial}{\partial x}\left\{\frac{f(H,\overline{\mu_1})}{6(2\mu+\mu_1)}\frac{\partial p}{\partial x}\right\}=\frac{U}{2}\frac{\partial H}{\partial x} \tag{13}$$

Multiplying equation (13) by $f(h_s)$ and integrating with respect to h_s from $-c \le h_s \le c$ and also the using equations (2) - (4), gives the averaged Reynolds type equation in the form

$$\frac{\partial}{\partial x}\left\{\frac{F(h,\alpha^*,\sigma^*,\varepsilon_1^*,\overline{\mu_1})}{6(2\mu+\mu_1)}\frac{\partial \overline{p}}{\partial x}\right\}=\frac{U}{2}\frac{\partial h}{\partial x} \tag{14}$$

where the $\overline{p}$ (=E (p)) is the expected value of the film pressure p.

Introducing the non-dimensional quantities

$$\alpha=\frac{\alpha^*}{C},\ \sigma=\frac{\sigma^*}{C^2},\ \varepsilon_1=\frac{\varepsilon_1^*}{C^3},\ h=\frac{\overline{h}}{C},\ P=\frac{\overline{p}C^2}{\mu\Omega R^2},$$

$$\overline{h_0}=\frac{h_0}{C}\quad x=R\theta,\qquad U=R\Omega$$

into equation (14), the non-dimensional Reynolds type equation is obtained in the form

$$\frac{\partial}{\partial\theta}\left\{\frac{F(\overline{h},\alpha,\sigma,\varepsilon_1,\overline{\mu_1})}{6\left(1+\frac{\overline{\mu_1}}{2}\right)}\frac{\partial P}{\partial x}\right\}=\frac{\partial\overline{h}}{\partial\theta} \tag{15}$$

where

$$F(\overline{h},\alpha,\sigma,\varepsilon_1,\overline{\mu_1})=B\left(1+\frac{3\overline{\mu_1}}{2(1+\overline{\mu_1})}\left[\frac{1-\alpha_1(\overline{h}+\alpha)(Q+S)}{\alpha_1^2(\overline{h}+\alpha^2+\sigma^2+2\overline{h}\alpha)}\right]\right),$$

$$\alpha_1=\sqrt{\frac{M\overline{\mu_1}(2+\overline{\mu_1})}{(1+\overline{\mu_1})}},$$

$$A=\frac{\alpha_1}{3}(\varepsilon_1+3\sigma^2\alpha+\alpha^3+\alpha),$$

$$B=\overline{h}^3+3\overline{h}^2\alpha+3\overline{h}(\alpha^2+\sigma^2)+\varepsilon_1+3\sigma^2\alpha+\alpha^3,$$

$$Q=\left(1-\frac{1}{\tanh^2(\alpha_1\overline{h})}\right)[\alpha_1\alpha-A]\ \text{and}$$

$$S=\frac{1}{\tanh(\alpha_1\overline{h})}\left(1-\alpha_1^2(\alpha^2+\sigma^2)\right).$$

Once integration of equation (15) and the use of boundary condition

$\frac{dP}{d\theta}=0$ at $h=h_o$ gives

$$\frac{dP}{d\theta}=\frac{6\left(1+\frac{\overline{\mu_1}}{2}\right)}{F(\overline{h},\alpha,\sigma,\varepsilon_1,\overline{\mu_1})}\left(\overline{h}-\overline{h_0}\right). \tag{16}$$

Assume the pressure distribution to be asymmetric and the boundary condition to be

$$P=0 \text{ at } \theta=0$$
$$P=0 \text{ at } \theta=\theta_c \tag{17}$$

Integrating equation (16) using boundary conditions (17) gives

$$\int_0^{\theta_c}\frac{1}{F(\overline{h},\alpha,\sigma,\varepsilon_1,\overline{\mu_1})}\left(\overline{h}-\overline{h_0}\right)d\theta=0 \tag{18}$$

where

$\overline{h_o}=1+\varepsilon\cos(\theta_c)$, θ_c is determined from equation (18) for different values of ε and are tabulated in Table 1. Using these values of θ_c equation (16) is solved numerically.

The load carrying capacity W of the bearing is defined as

$$W=\int_0^{2\pi}P\sin\theta\ LR\ d\theta\quad=-LR\int_0^{2\pi}\cos\theta\ \frac{dP}{d\theta}\ d\theta \tag{19}$$

which on using equation (19) gives the non-dimensional load carrying capacity is

$$\overline{W}=\frac{WC}{\mu LR^3\Omega} \tag{20}$$

$$=-6\left(1+\frac{\overline{\mu_1}}{2}\right)\int_0^{2\pi}\frac{\cos\theta\left(\overline{h}-\overline{h_0}\right)}{F(\overline{h},\alpha,\sigma,\varepsilon_1,\overline{\mu_1})}\ d\theta \tag{21}$$

After substituting $\frac{h_o}{C}=1+\varepsilon\cos(\theta_c)$ for the corresponding values of ε, the load carrying capacity is determined from equation (21).

Frictional force acting on the journal surface is given by

$$F=-\int_0^{2\pi}(\mu+\mu_1)\left(\frac{\partial u_1}{\partial y}\right)_{y=h}LR\ d\theta \tag{22}$$

where

$$\left(\frac{\partial u_1}{\partial y}\right)_{y=h}=\frac{(2\mu+\mu_1)}{(\mu+\mu_1)}\left[\frac{h}{2(2\mu+\mu_1)}\frac{1}{R}\frac{dP}{d\theta}-\frac{R\Omega\sinh\lambda h}{2\left(\beta(\cosh\lambda h-1)+\sinh\lambda h\right)}\right]$$

and

$$\beta=\frac{\lambda\gamma}{\mu_1}-\frac{2}{\lambda} \tag{23}$$

The non-dimensional frictional force is given by

$$\overline{F}=\frac{FC}{\mu LR^2\Omega}$$

$$\overline{F} = -\left(1+\frac{\overline{\mu_1}}{2}\right)\int_0^{2\pi}\left[\frac{3\left(\overline{h}-\overline{h_0}\right)}{F(\overline{h},\alpha,\sigma,\varepsilon_1,\overline{\mu_1})} - \frac{2\alpha_1}{2\alpha_1\left(\overline{h}+\alpha\right)-\frac{\overline{\mu_1}}{\left(1+\overline{\mu_1}\right)}[D+E*G]}\right]d\theta \quad (24)$$

where

$$D = \tanh(\alpha_1\overline{h})$$

$$E = 1-\tanh^2(\alpha_1\overline{h}) \quad and$$

$$G = \alpha_q\alpha - \frac{\alpha_1^{\ 3}}{3}(\varepsilon_1+\alpha^3+3\sigma^2\alpha) \quad (25)$$

The non-dimensional coefficient of friction is given by

$$\overline{C_f} = \frac{\overline{F}}{\overline{W}} \quad (26)$$

RESULTS AND DISCUSSION

The surface roughness effect on the performance characteristics of one-dimensional infinitely long journal bearings lubricated with micropolar fluid is analyzed through the dimensionless parameters $\alpha, \sigma, \varepsilon_1, M, \varepsilon, \overline{\mu_1}$. As the roughness parameters tend to zero, the results obtained in this paper reduce to smooth case studied by Zaheeruddin and Isa (1978). The following set of values are used for various non-dimensional parameters α = [-0.1 - 0.1]; ε_1 = [-0.1 - 0.1]; σ = [0.0 - 0.4]; ε = [0.1 – 0.4]; M = 5.0 and 10.0 and $\overline{\mu_1}$ = [0.0 - 0.4] for the numerical computations of journal bearing characteristics. The numerical values for the roughness parameters $\alpha, \sigma, \varepsilon_1$ are also so chosen that the corresponding film shapes are feasible.

The variation of non-dimensional pressure P with the angular coordinate θ for different values of ε, α, ε_1 and σ are depicted in the figures 2 - 5 respectively in all these figures it is assumed that $\overline{\mu_1}$ = 0.4. From figure 2 it is observed that P increases for increasing values of ε and decreases for increasing values of M. The effect of roughness parameter α and ε_1 on the variation of P with θ is shown in figures 3 and 4 respectively. It is observed that, P increases for negatively increasing values of α and ε_1 and decreases for positively increasing values of α and ε_1. From figure 5 it is observed that increasing values of σ causes P to decrease.

The variation of non-dimensional load carrying capacity $\overline{W}$ with $\overline{\mu_1}$ is shown in figures 6 – 9 for different values of ε, α, ε_1 and σ. It is observed that $\overline{W}$ increases for increasing values of $\overline{\mu_1}$ and ε. The effect of roughness parameter α and ε_1 on the variations of $\overline{W}$ and $\overline{\mu_1}$ is depicted in figures 7 and 8 respectively. It is observed that $\overline{W}$ increases for negatively skewed surface roughness whereas decreases for positively skewed surface roughness. From figure 9 it is observed that $\overline{W}$ decreases for increasing values of σ.

Figures 10,11,12 and 13 depicts the variation of non-dimensional frictional force $\overline{F}$ with $\overline{\mu_1}$ for different values of ε, α, ε_1 and σ respectively. $\overline{F}$ increases for increasing values of $\overline{\mu_1}$ and ε (Fig. 10). The negatively skewed surface roughness on the bearing surface causes an increasing $\overline{F}$ whereas, the negatively skewed surface roughness reduces the frictional force $\overline{F}$ (Figures 11 and 12). From figure 13 it is observed that increasing values of σ decreases $\overline{F}$.

Variation of non-dimensional coefficient of friction with $\overline{\mu_1}$ is shown in figures 14, 15 and16 for different values of ε, α and ε_1. It is observed that, $\overline{C}$ decreases for increasing values of $\overline{\mu_1}$ and ε. The negatively skewed surface roughness increases $\overline{C}$ where as the positively skewed surface roughness decreases $\overline{C}$.

CONCLUSIONS

On the basis of numerical computations and the results presented in figures 2 – 16, the following conclusions can be drawn.

1. The micropolar lubricants provide an increase in the load carrying capacity and decreases the coefficient of friction as compared to the corresponding Newtonian case ($\overline{\mu_1}$ =0).
2. The presence of negatively skewed surface roughness on the bearing surface increases the load carrying capacity, frictional force and the coefficient of friction, whereas the reverse trend is observed for the positively skewed surface roughness.
3. These results are more accentuated for micropolar fluids.

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Mathematical Formulation of the problem

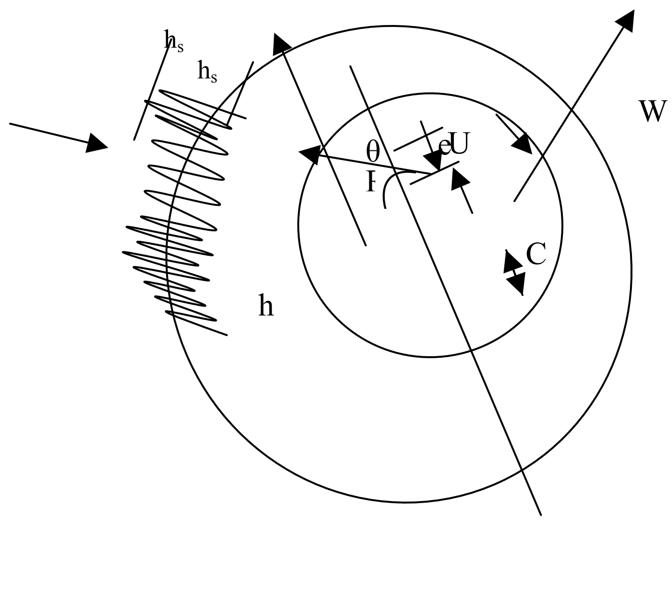

Fig. 1(a) Physical configuration of the problem

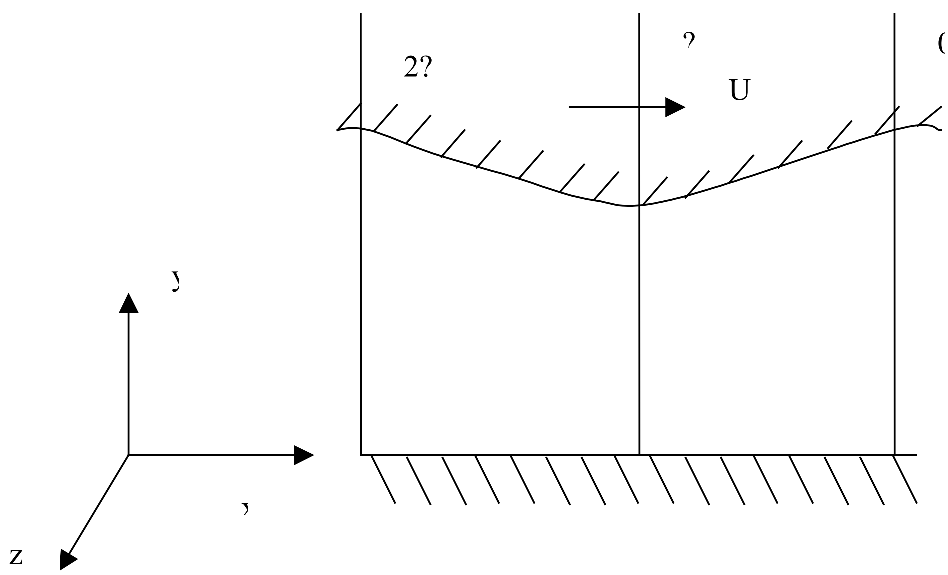

Fig.1. (b). Geometry of the developed surfaces of the journal and bush

Table. 1 Shows the $\frac{h_0}{C}$ for different values of ε (eccentricity ratio)

Eccentricity ratio ε	$\frac{h_0}{C}$
0.1	5.325699
0.2	5.388539
0.3	5.482800
0.4	5.608481

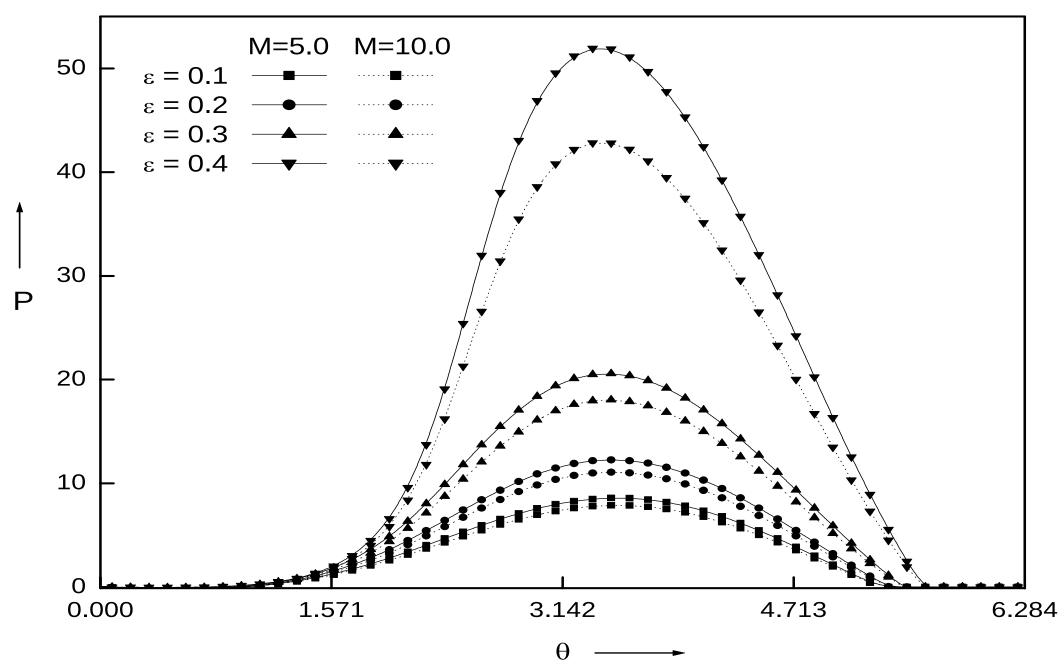

Fig.2 Variation of non-dimensional pressure P with θ for different values of ε with α = -0.1, σ = 0.1, ε_1 = -0.1.

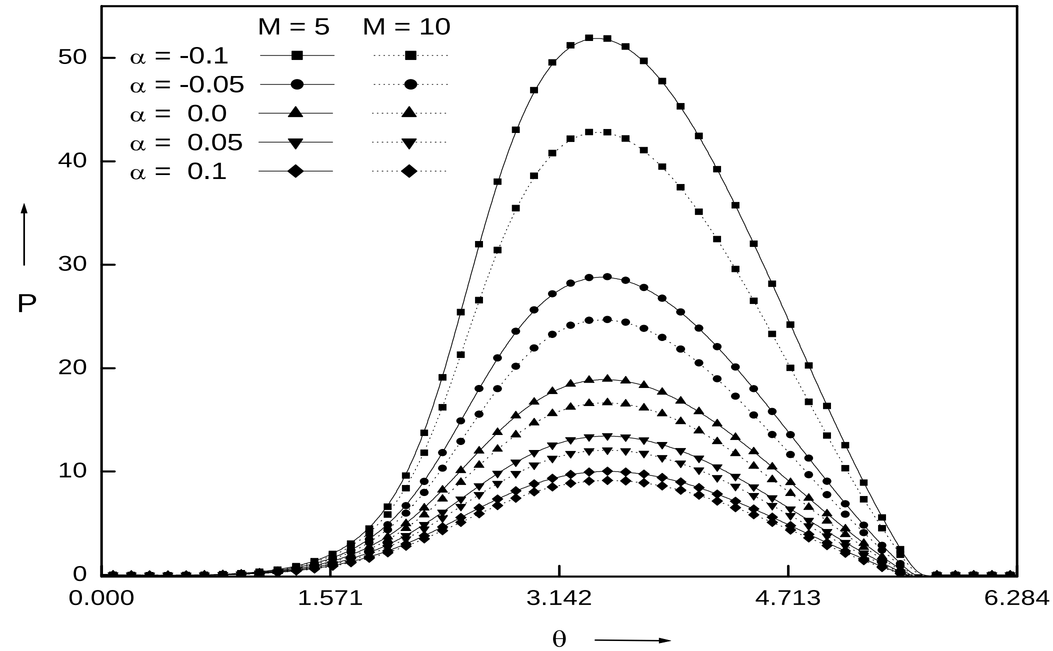

Fig.3 Variation of non-dimensional pressure P with θ for different values of α with $\sigma = 0.1$, $\varepsilon_1 = -0.1$.

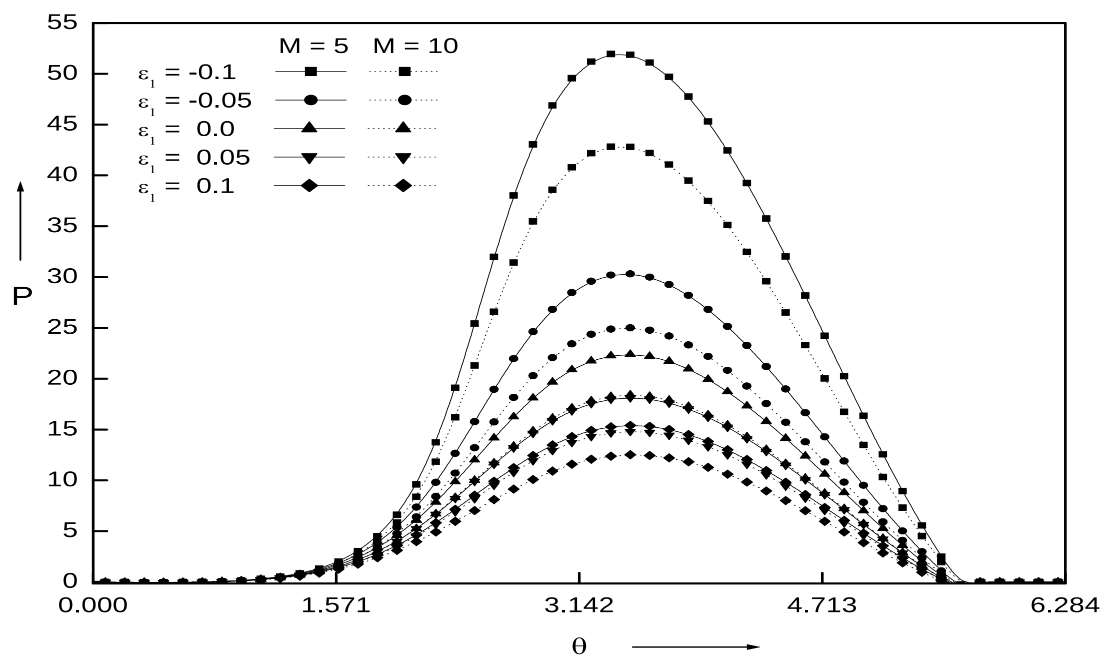

Fig. 4 Variation of non-dimensional pressure P with θ for different values of ε_1 with $\alpha = -0.1$, $\sigma = 0.1$.

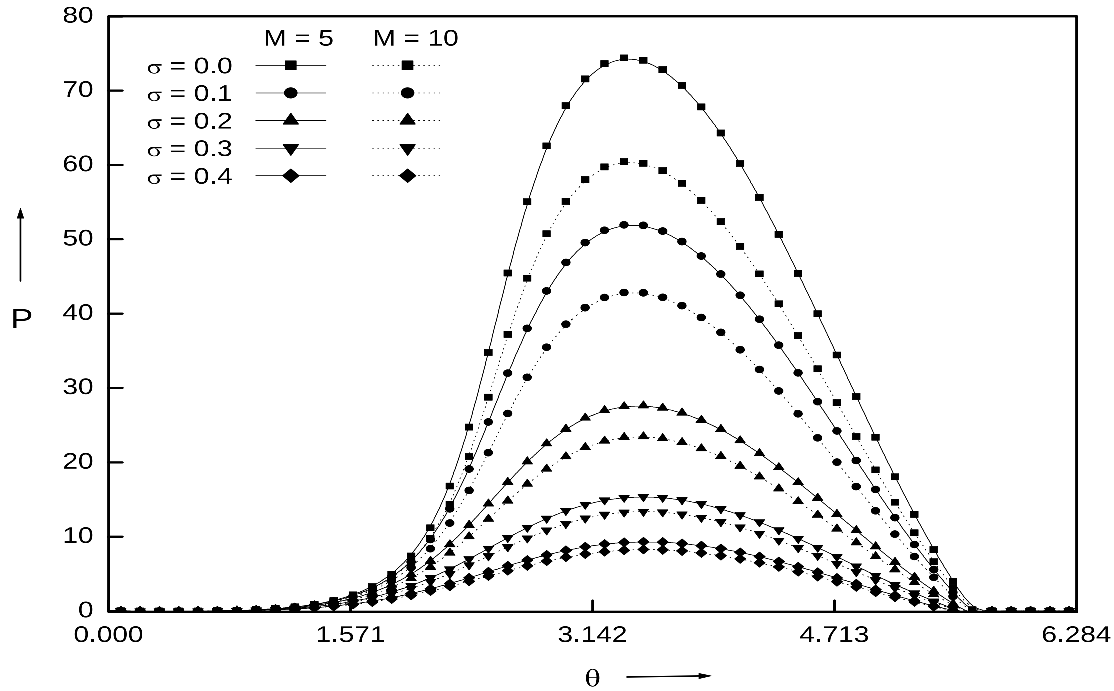

Fig. 5 Variation of non-dimensional pressure P with θ for different values of σ with α = -0.1, ε_1 = -0.1.

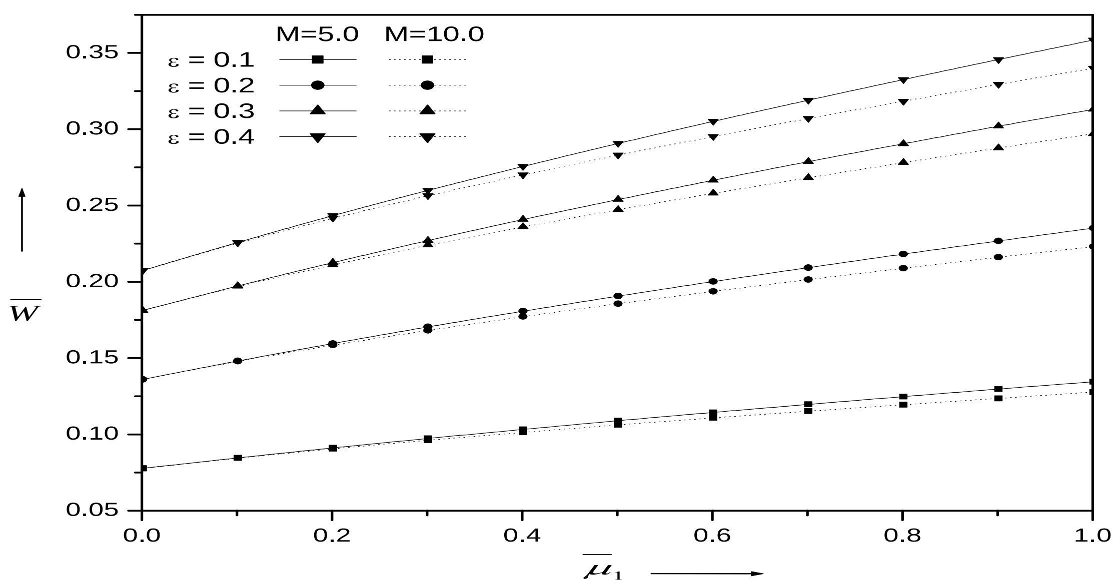

Fig. 6 Variation of non-dimensional Load $\overline{W}$ with $\overline{\mu}_1$ for different values of ε with α = -0.1, σ = 0.1, ε_1 = -0.1

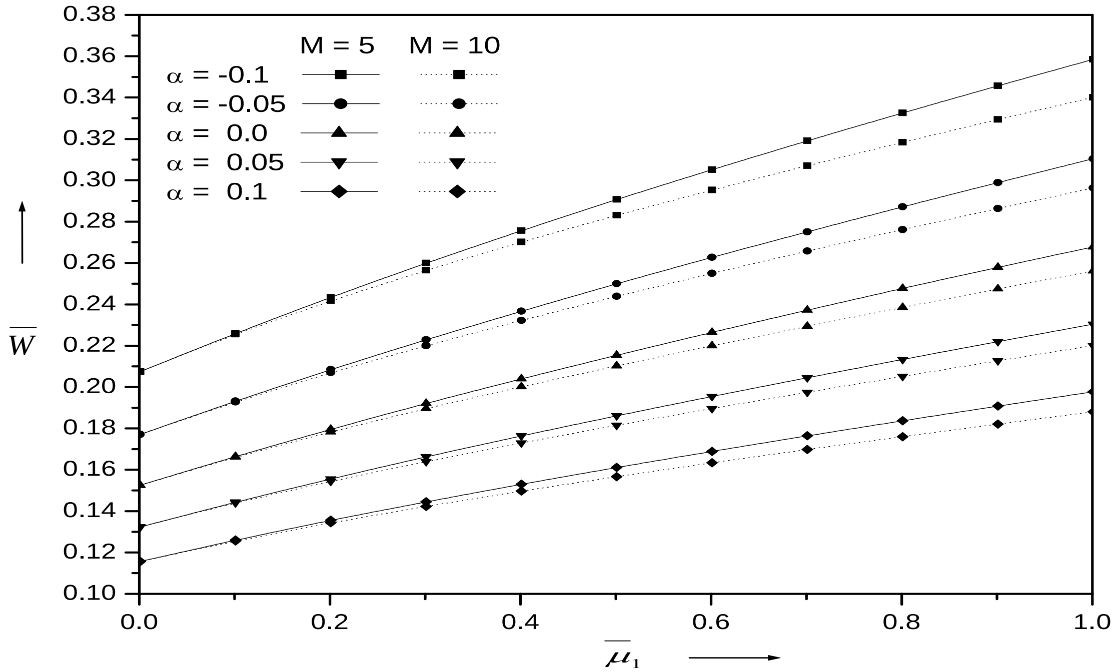

Fig. 7 Variation of non-dimensional Load $\overline{W}$ with $\overline{\mu}_1$ for different values of α with $\sigma = 0.1$, $\varepsilon_1 = -0.1$ and $\varepsilon = 0.4$

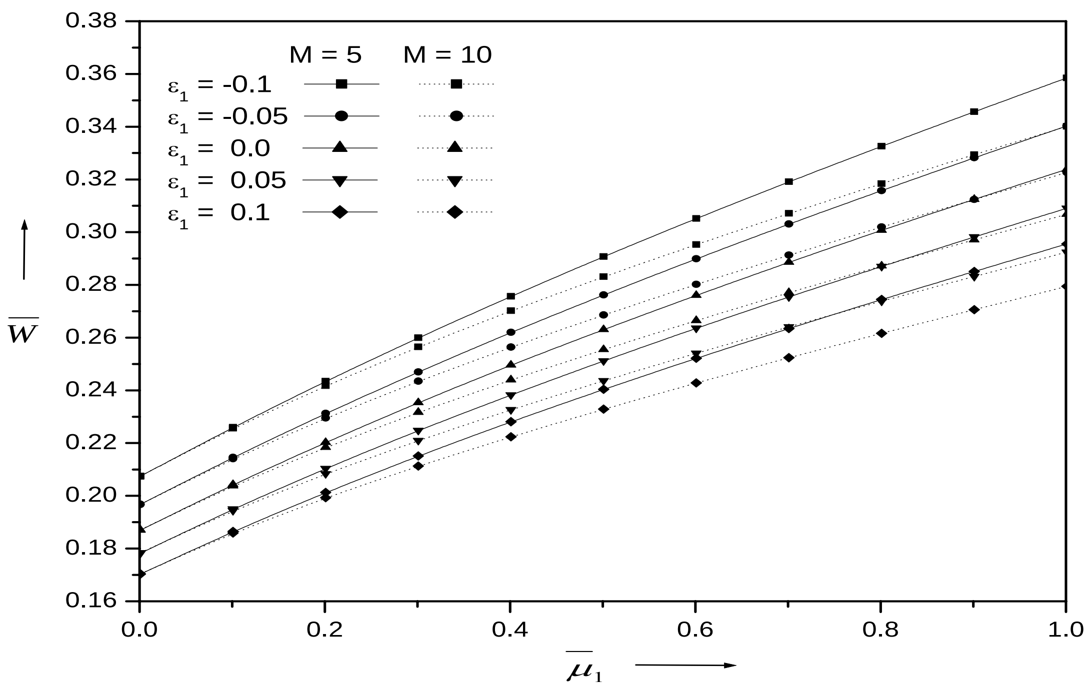

Fig. 8 Variation of non-dimensional Load $\overline{W}$ with $\overline{\mu}_1$ for different values of ε_1 with $\alpha = -0.1$, $\sigma = 0.1$ and $\varepsilon = 0.4$

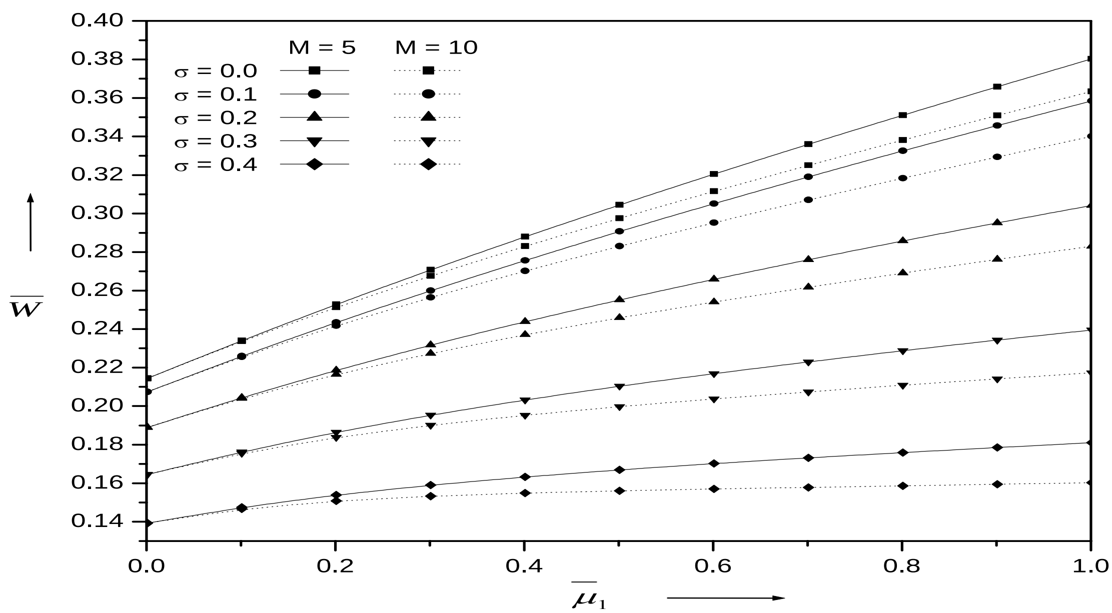

Fig. 9 Variation of non-dimensional Load $\overline{W}$ with $\overline{\mu_1}$ for different values of σ with α = -0.1, ε_1 = -0.1 and ε = 0.4

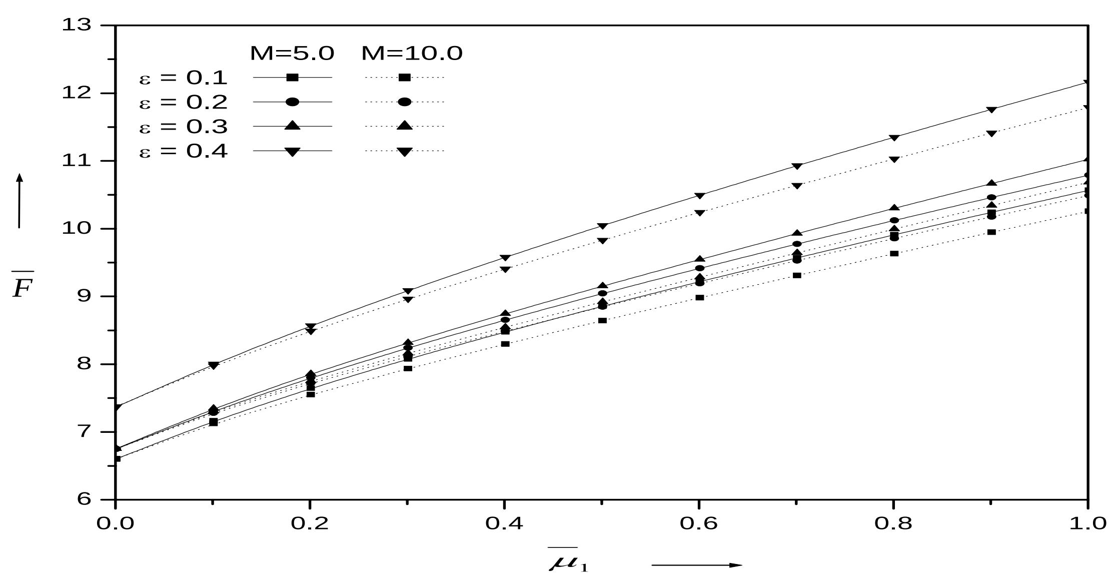

Fig. 10 Variation of non-dimensional Frictional force $\overline{F}$ with $\overline{\mu_1}$ for different values of ε with α = -0.1, σ = 0.1, ε_1 = -0.1

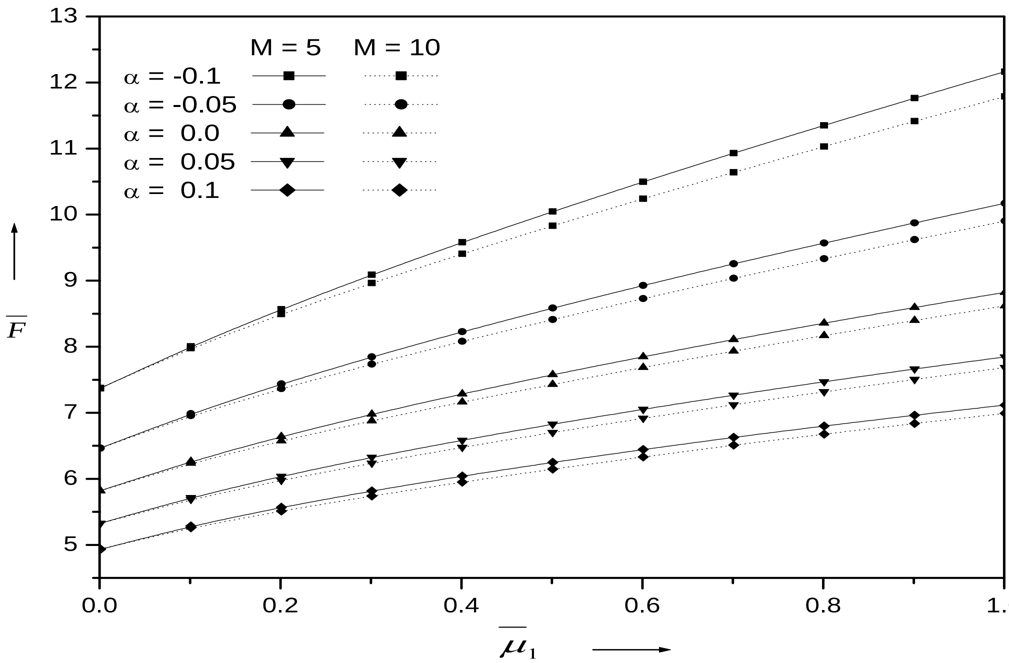

Fig. 11 Variation of non-dimensional Frictional force $\overline{F}$ with $\overline{\mu}_1$ for different values of α with $\sigma = 0.1$, $\varepsilon_1 = -0.1$ and $\varepsilon = 0.4$

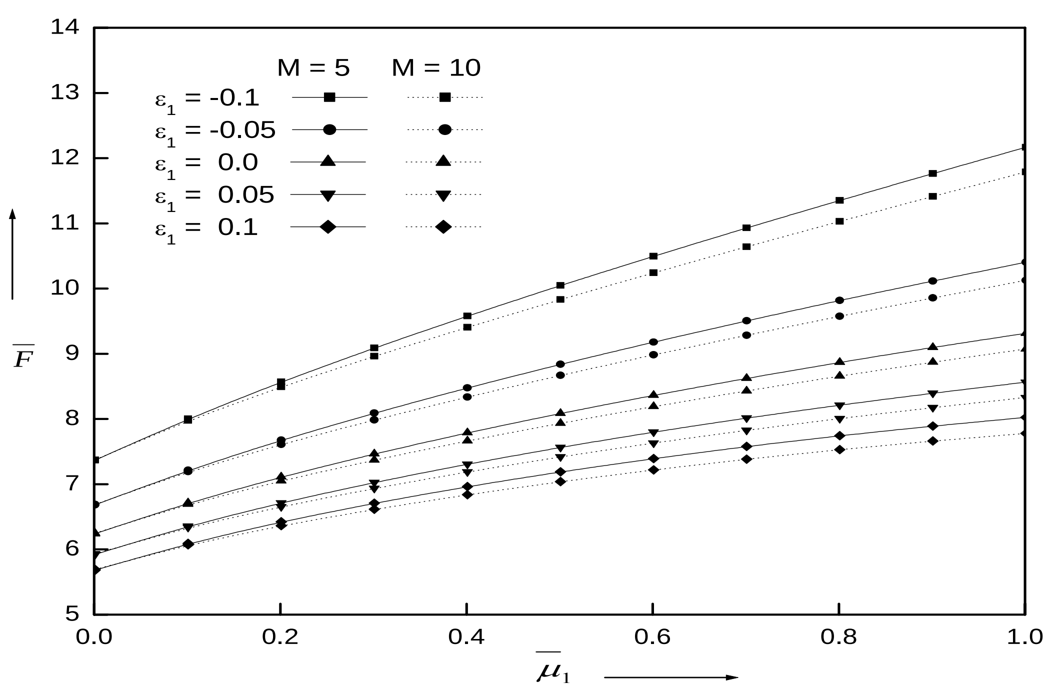

Fig. 12 Variation of non-dimensional Frictional force $\overline{F}$ with $\overline{\mu}_1$ for different values of ε_1 with $\alpha = -0.1$, $\sigma = 0.1$ and $\varepsilon = 0.4$

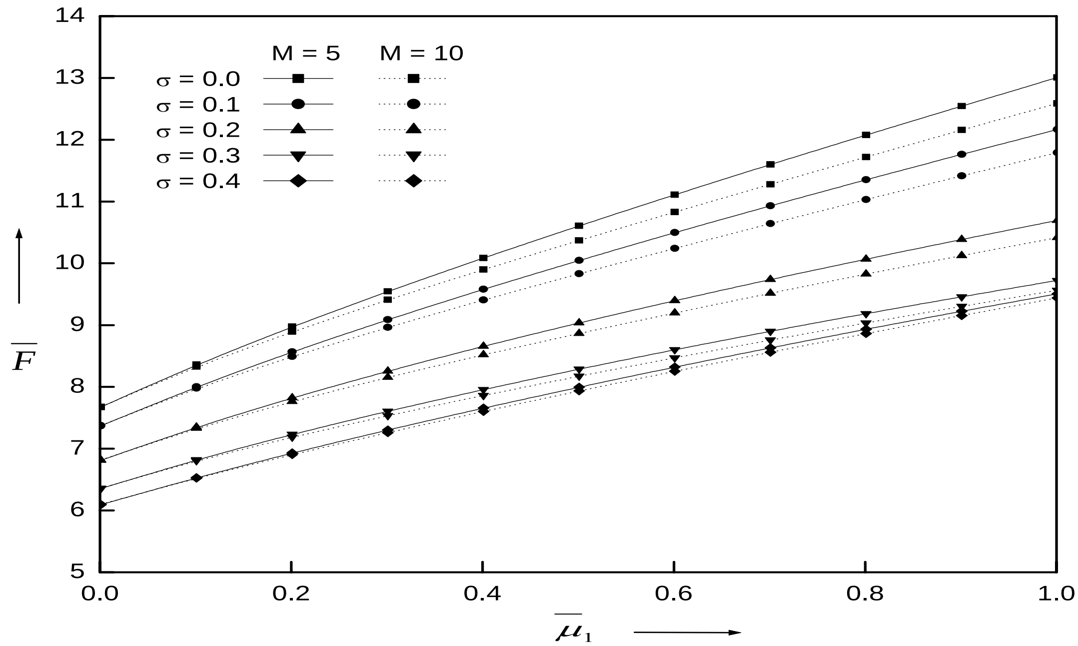

Fig. 13 Variation of non-dimensional Frictional force $\overline{F}$ with $\overline{\mu}_1$ for different values of σ with α = -0.1, ε_1 = -0.1 and ε = 0.4

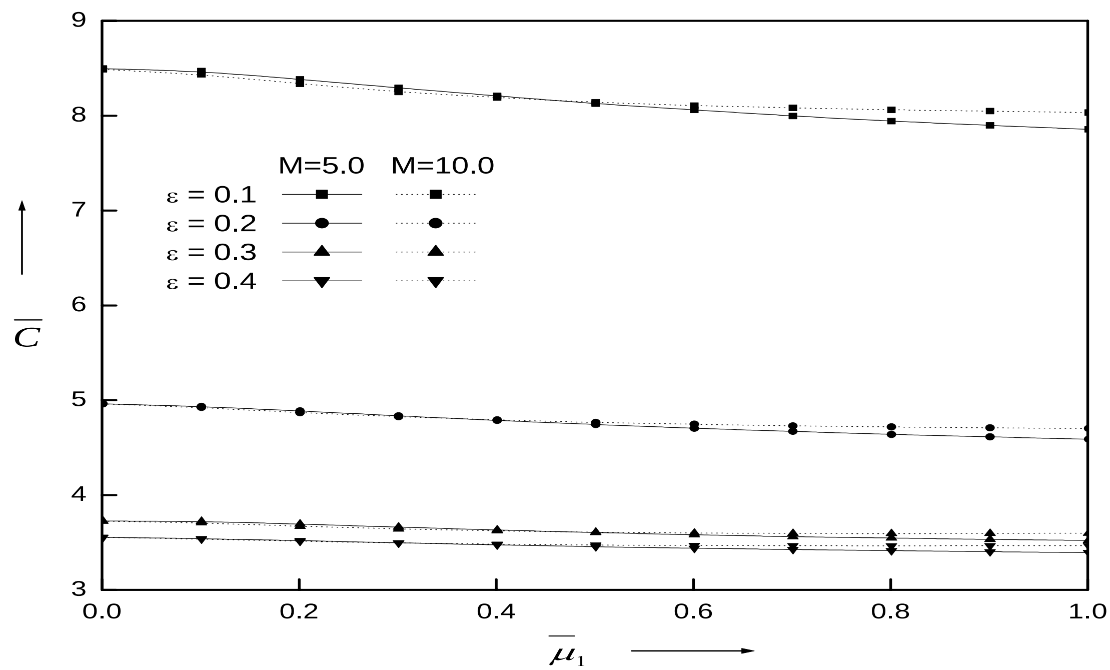

Fig. 14 Variation of non-dimensional Frictional coefficient $\overline{C}$ with $\overline{\mu}_1$ for different values of ε with α = -0.1, σ = 0.1 and ε_1 = -0.1

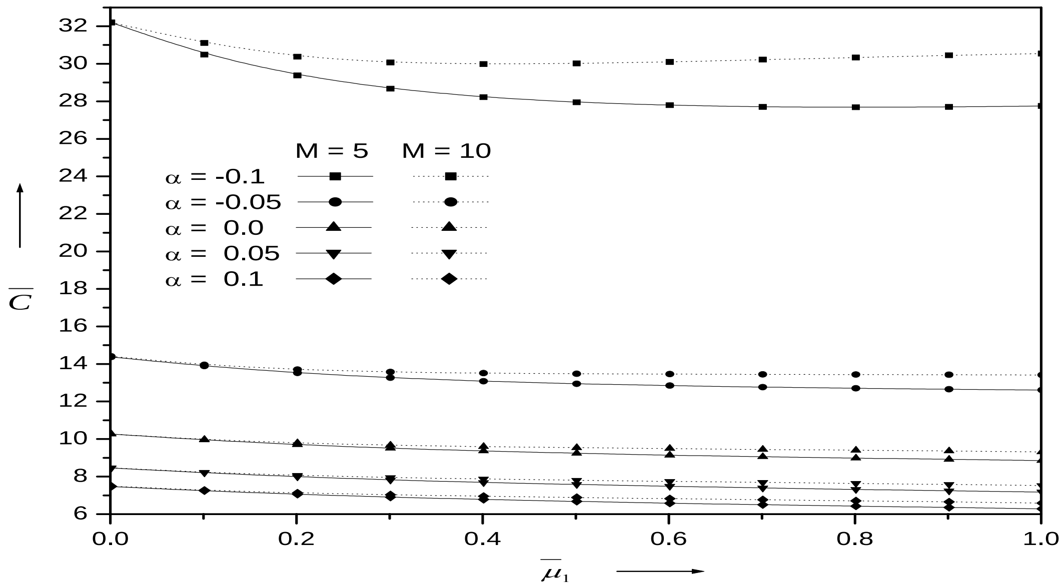

Fig. 15 Variation of non-dimensional Frictional Coefficient $\overline{C}$ with $\overline{\mu}_1$ for different values of α with $\sigma = 0.1$, $\varepsilon_1 = -0.1$ and $\varepsilon = 0.4$

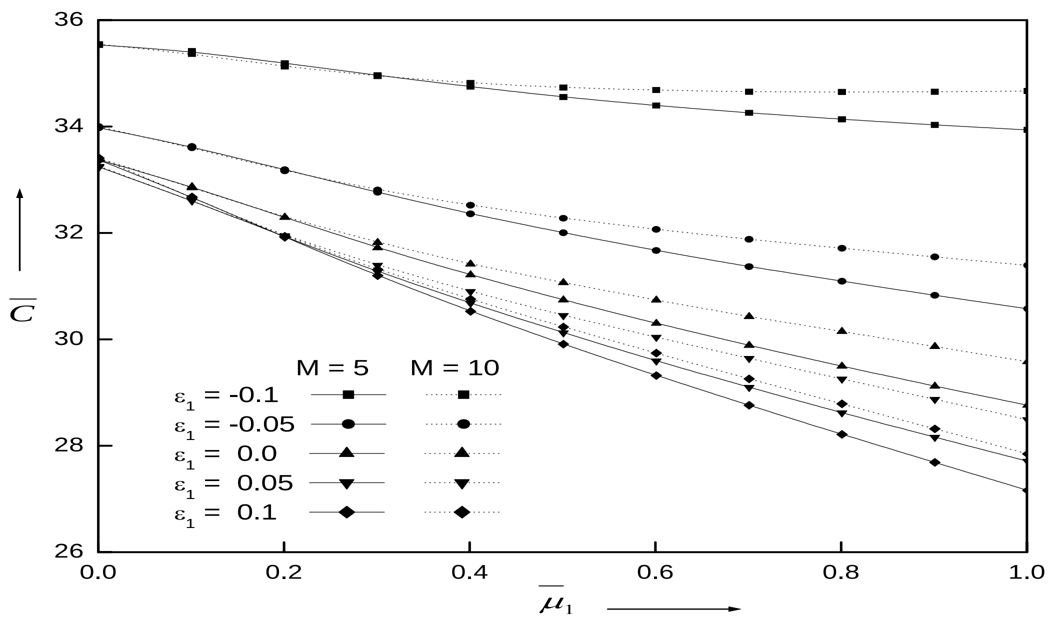

Fig. 16 Variation of non-dimensional Frictional coefficient $\overline{C}$ with $\overline{\mu}_1$ for different values of ε_1 with $\alpha = -0.1$, $\sigma = 0.1$ and $\varepsilon = 0.4$

NOMENCLATURE

C radial clearance
c maximum asperity height from the mean level
$\overline{C_f}$ non-dimensional frictional coefficient, $= \overline{F}/\overline{W}$
E expectancy operator defined by equation (5)
F frictional force
$\overline{F}$ non-dimensional frictional force, $= \frac{F_1 C}{\mu L R^2 \Omega}$
H fluid film thickness, $= h(x) + h_s$
h_s random variable
h mean film thickness, $= h(x) = C\ (1 + \varepsilon \cos\theta)$
$\frac{h_o}{C}$ non-dimensional film thickness at maximum pressure
L axial length of the bearing
M micropolar parameter, $= \frac{\mu C^2}{4\gamma}$
p pressure in the film region.
P non-dimensional pressure
R journal radius
u_1, u_2, u_3 components of fluid velocity in x, y, and z directions respectively
U surface velocity of the journal
w_1, w_2, w_3 microrotation velocity components
W load carrying capacity
$\overline{W}$ non-dimensional load carrying capacity $\overline{W} = \frac{W_1 C^2}{\mu L R^3 \Omega}$
r, θ, z cylindrical coordinates
x, y, z rectangular coordinates
α^* mean defined by equation (2)
α mean defined by equation $\left(= \frac{\alpha^*}{C}\right)$
ε_1^* roughness parameter defined by equation (4)
ε_1 non-dimensional form of ε_1^* $\left(= \frac{\varepsilon_1^*}{C^3}\right)$
ε eccentricity ratio, $\left(= \frac{e}{C}\right)$
σ^* standard deviation defined by equation (3)
σ non-dimensional standard deviation $\left(= \frac{\sigma^*}{C^2}\right)$
μ viscosity coefficient of the base lubricant
$\overline{\mu_1}$ non-dimensional viscosity coefficient, $\overline{\mu_1} = \mu_1/\mu$
μ_1, γ viscosity coefficient for micropolar fluid
Ω angular velocity of the journal

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