On the Performance of Finite Journal Bearings Lubricated With Micropolar Fluids

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ON THE PERFORMANCE OF FINITE JOURNAL BEARINGS LUBRICATED

WITH MICROPOLAR FLUIDS

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ABSTRACT

A Study of the performance parameters for a journal bearing of finite length lubricated with micropolar fluids is undertaken. Results indicate that a significantly higher load carrying capacity than Newtonian fluids may result depending on the size of material characteristic length and the coupling number. It is also shown that although the frictional force associated with micropolar fluid is in general higher than that of a Newtonian fluid, the friction coefficient of micropolar fluids tends to be lower than that of the Newtonian.

NOMENCLATURE

c   nominal radial clearance, m

d   journal diameter, m

e   eccentricity, m

f   nondimensional friction coefficient

F   friction force, N

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*Member, STLE.

$\bar{F}$ nondimensional friction force $\dfrac{c}{\mu U R} F$

h   film thickness, m

h_{cav} film thickness at the point of cavitation, m

$\bar{h}$ nondimensional film thickness $= \dfrac{h}{c}$

L   bearing length, m

ℓ_m nondimensional characteristic length of micropolar fluid $= \dfrac{c}{\Lambda}$

N   coupling number

P   hydrodynamic pressure, N/m^2

$\bar{P}$ nondimensional pressure $= \dfrac{p}{\mu} \dfrac{R}{U} \left(\dfrac{c}{R}\right)^2$

Q   leakage flow rate, m^3/s

$\bar{Q}$ nondimensional leakage $= \dfrac{6L}{cUR^2} Q$

R   journal radius, m

W_R component of load along the radial, N

$\bar{W}_R$ nondimensional load component $= \dfrac{W_x c^2}{\mu U L R^2}$

W_φ component of load along the tangential direction, N

$\bar{W}_\varphi$ nondimensional load component $= \dfrac{W_y c^2}{\mu U L R^2}$

W   resultant of load, N

$\bar{W}$ nondimensional load $= \dfrac{W c^2}{\mu U L R^2}$

x,y,z   coordinate system, m

$\bar{y}$ nondimensional coordinates $= \dfrac{y}{L}$

γ micropolar viscosity coefficient, N-s

ε eccentricity ratio, $\varepsilon = e/c$

θ nondimensional coordinate, x/R

θ_{cav} angular coordinate where the film cavitates

κ micropolar viscosity coefficient, $N \cdot s/m^2$

Λ characteristic length of micropolar fluids, m

μ viscosity of the Newtonian fluid, $N \cdot s/m^2$

Φ function defined in Eq. (4)

$\bar{\Phi}$ normalized form of Φ

φ attitude angle, rad

INTRODUCTION

In recent years there has been a considerable amount of interest in the
lubricating effectiveness of non-Newtonian fluids. Generally speaking, in
non-Newtonian lubricants the shear stresses do not always remain proportional
to the shear rates, particularly for high shear rates. To model such fluids,
various mathematical formulations such as the power law and Ree-Ering models
have been proposed. This paper presents a study of the non-Newtonian behavior
of the lubricating oil (in a finite length journal bearing) from the modern
continuum point of view. In continuum mechanics, the continuous media is
regarded as a set of structured particles possessing individual mass and
velocity. These continuum particles are assigned a sub-structure which can
translate, rotate, or even deform independently. In other words, the
microvolume is considered to have independent microvolume elements that possess
individuality.

Theoretical work dealing with the mechanics of fluids with microstructure
began with the work of Jeffery (Ref. 1) in 1922 who studied the problem of a
Newtonian fluid motion in the presence of suspended ellipsoids. His results
indicated that the viscosity of the fluid with such particles was higher than

that of its base Newtonian fluid. Later in 1957, Prager (Ref. 2) derived constitutive equations for stress in a noninteracting (dilute) dumbbell particle suspension. Since unlike the classical fluid mechanics the microcontinua is regarded as sets of structured particles, the theory required a complete re-examination of the classical concepts that has to be augmented with additional balance laws and constitutive relations.

The general theory of microcontinua is attributed to Eringen who developed the fluid mechanics of deformable microelements (Ref. 3) in an article entitled "Simple Microfluids." The Eringen definition of a simple microfluid is a viscous medium whose behavior and properties are affected by the local motion of particles in its microvolume. These fluids are characterized by 22 viscosity and material constants and when applied to flow problems the result is a system of 19 partial differential equations with 19 unknowns that may not be amenable to solutions. Eringen (Ref. 4) subsequently introduced a subclass of fluids which he named micropolar fluids that ignores the deformation of the microelements but still allows for the particle micromotion to take place. Since then many other papers have appeared in the literature that present different views of the subject based on the continuum mechanics. A comprehensive review of these theories is provided by Ariman et al. (Ref. 5).

As for the application of the microcontinuum theory to real flows, the blood rheology appears to be a very promising area (Refs. 6 and 7). Continuum approach to blood flow in the arterioles takes the rotation and deformation of red blood cells into account. Such consideration may be particularly important because of the relative size of the cells as compared with the diameter of the arterioles. Another area where microcontinuum theory may be applied is in the study of rheological behavior of complex microstructures such as liquid crystals which are known to possess complex molecular structure and orientation patterns (Ref. 8).

4

Microcontinuum theory in general may also be applied to the study of the non-Newtonian behavior of certain polymer solutions such as polyisobutylene which are known to have long chains of molecular structure. To study such fluids, Hand (Ref. 9) put forward a theory of so called anisotropic fluids and presented theoretical results with comparison to those measured experimentally. Hand showed that theories due to Jeffery (Ref. 1) and Prager (Ref. 2) are contained in his anisotropic fluids as special cases and that the theory can be applied to fluids with microstructure such as polyisobutylene that are spherical at rest but elongate under shear field as do immiscible drops suspended in fluids. Comparing results with experimental measurements, Hand reports good agreement for the dependence of apparent viscosity and normal stress particularly at low shear rates. His success is indicative of the potential of applicability of modern continuum theory for non-Newtonian fluids.

Theoretical works on the non-Newtonian bearings from microcontinuum viewpoint began with the work of Allen and Kline (Ref. 10). They stated that in addition to polymeric fluids with long molecular chains, the continuum model can be used, as a first approximation, for modeling bearings that are contaminated with dirt and particles. Under such conditions, the lubricant can be considered as a fluid suspension. They studied the one-dimensional slider bearing problem and concluded that the theory predicts a higher load capacity and a lower friction coefficient compared to those of a Newtonian fluid, thereby signifying an improvement in the lubrication of a one-dimensional bearing lubricant with micropolar fluids.

Since that article was published, many other papers appeared in the literature that dealt with the lubrication of one-dimensional bearings. To put things in perspective, the following references can be mentioned: for slider or step bearings (Refs. 11 to 17), thrust bearings (Refs. 18 and 19),

journal bearings (Refs. 20 to 25), squeeze film bearings (Refs. 26 to 30),
rolling element and elastohydrodynamic lubrication (Refs. 31 and 32).

Governing Equations

The general form of the governing equations for the steady state motion
of micropolar fluids can be written in tensor notation as follows (Ref. 31).
conservation of mass:

$$\nabla \cdot (\rho V) = 0 \tag{1}$$

Conservation of linear momentum:

$$(\lambda + 2\mu) \nabla\nabla \cdot V - \frac{1}{2}(2\mu + \kappa) \nabla \times V + \kappa\nabla \times \upsilon - \nabla\pi + \rho F_B = 0 \tag{2}$$

Conservation of angular momentum:

$$(\alpha + \beta + \gamma)\nabla\nabla \cdot \upsilon - \gamma\nabla \times \nabla \times \upsilon + \kappa\nabla \times V - 2\kappa\upsilon + \rho \mathcal{L}_B = 0 \tag{3}$$

where V is the velocity vector and υ is the microrotational
velocity vector. ρ is the mass density, π is the thermodynamic pressure, B
F_B is the body force per unit mass, $\mathcal{L}_B$ is the body couple (moment) per unit
mass, μ and λ are the familiar viscosity coefficients of the classical
fluid mechanics, while α, β, γ, and κ are additional viscosity coefficients
for micropolar fluids.

Equation (2) has the familiar form of the Navier-Stokes equation but it
is coupled with Eq. (3) which essentially describes the motion of the particles
inside the microvolume as they undergo microrotational effects represented by
the microrotational vector υ. For fluids with no microstructure, this
parameter vanishes. For Newtonian fluids, Eqs. (2) and (3) decouple since
$\kappa = 0$. These equations can be simplified considerably for incompressible
fluids (ρ = constant) in the absence of body forces and inertial forces since

$\nabla \cdot V = 0$ and F_B and $\mathcal{L}_B$ are assumed to be negligible. Furthermore, the thermodynamic pressure π can be replaced by P since:

$$\pi = - \left[\frac{\partial E}{\partial \rho^{-1}} \right] = P$$

where E is the internal energy.

Having made the aformentioned assumptions, a so-called generalized Reynolds equation for micropolar fluids can be derived, see Ref. 14. The resulting equation is given below (Ref. 25):

$$\frac{\partial}{\partial x} \left[\frac{h^3}{\mu} \Phi(N,\Lambda,h) \frac{\partial P}{\partial x} \right] + \frac{\partial}{\partial y} \left[\frac{h^3}{\mu} \Phi(N,\Lambda,h) \frac{\partial P}{\partial y} \right] = \frac{U}{2} \frac{\partial h}{\partial x} \qquad (4)$$

where

$$\Phi(N,\Lambda,h) = \frac{1}{12} + \frac{\Lambda^2}{h^2} - \frac{N\Lambda}{2h} \coth \left(\frac{Nh}{2\Lambda} \right)$$

$$N = \left(\frac{\kappa}{2\mu + \kappa} \right)^{\frac{1}{2}}$$

$$\Lambda = \left(\frac{\gamma}{4\mu} \right)^{\frac{1}{2}}$$

The parameter μ is the viscosity of the base fluid as in the case of Newtonian fluids and κ and γ are two additional viscosity coefficients for micropolar fluids. These viscosity parameters are grouped in the form of two parameters N and Λ. N is a dimensionless parameter called the coupling number, for it characterizes the coupling of the linear and angular momentum equations. When N is identically zero, the equations of linear and angular momentum are decoupled and the equation of the linear momentum reduces to the classical Navier-Stokes equation. The parameter Λ is called the characteristic length for it characterizes the interaction between the micropolar fluid and the film gap. This parameter, carrying the dimension of

length, is a function of the size of the lubricant molecule. As Λ approaches zero, the effect of microstructure becomes less important and in the limiting case, when Λ vanishes, the function Φ becomes identical to 1/12 and the Reynolds equation reduces to its classical form.

Nondimensionalization

Let:

$$\ell_m = \frac{c}{\Lambda} \qquad \theta = \frac{x}{R} \qquad \bar{y} = \frac{y}{L} \qquad \bar{h} = \frac{h}{c} \qquad \bar{P} = \frac{p}{\mu U} \frac{R}{R} \left(\frac{c}{R}\right)^2$$

Then the generalized Reynolds equation for micropolar fluids becomes:

$$\frac{\partial}{\partial\theta}\left[\bar{h}^3 \, \bar{\Phi} \, \frac{\partial \bar{P}}{\partial\theta}\right] + \left(\frac{R}{L}\right)^2 \frac{\partial}{\partial\bar{y}}\left[\bar{h}^3 \, \bar{\Phi} \frac{\partial \bar{P}}{\partial\bar{y}}\right] = 6 \frac{\partial\bar{h}}{\partial\theta} \tag{5}$$

where:

$$\bar{\Phi} = 1 + 12 \frac{1}{\ell_m^2 \bar{h}^2} - 6N \frac{1}{\ell_m \bar{h}} \coth\left(\frac{N\bar{h}\ell_m}{2}\right)$$

Boundary conditions are:

$$\bar{P} = 0 \qquad \text{at } \theta = 0$$
$$\bar{P} = \frac{d\bar{P}}{d\theta} = 0 \qquad \text{at } \theta = \theta_{cav} \tag{6}$$
$$\bar{P} = 0 \qquad \text{at } \bar{y} = 0 \text{ and } \bar{y} = 1$$

where $\theta = \theta_{cav}$ is the angular coordinate at which the film cavitates.

Equation (5) is a fully elliptic partial differential equation for which no analytical solution exists. Analytical solutions for a number of bearing configurations are available as cited earlier for either infinitely long bearings (L/D >> 1) or short bearings (L/D << 1). To evaluate the performance of a bearing for a finite length, one must resort to numerical schemes for solution. Equation (5) was solved using the finite difference method: the equation was discretized and the difference equations were solved on a computer using the Successive-Over-Relaxation method (S.O.R.). Convergence was assumed

8

when the relative maximum error between two successive iterations fell below a tolerance value of 10^{-3}. The Reynolds cavitation boundary conditions were satisfied by setting all negative pressures equal to zero during the computations. Once the pressure distribution was evaluated, the following performance parameters were computed numerically.

Load capacity and attitude angle:

$$\bar{W}_R = \frac{W_R c^2}{\mu U L R^2} = \int_0^1 \int_0^{\theta_{cav}} p \cos\theta \, d\theta \, dy \qquad (7)$$

$$\bar{W}_\varphi = \frac{W_\varphi c^2}{\mu U L R^2} = \int_0^1 \int_0^{\theta_{cav}} p \sin\theta \, d\theta \, dy \qquad (8)$$

$$\bar{W} = \frac{W c^2}{\mu U L R^2} = \sqrt{\bar{W}_R^2 + \bar{W}_\varphi^2} \qquad (9)$$

$$\varphi = \tan^{-1}\left(-\frac{\bar{W}_\varphi}{\bar{W}_R}\right) \qquad (10)$$

Leakage flow rate:

$$\bar{Q} = \frac{6L}{cUR^2} Q = \int_0^{\theta_{cav}} \bar{\Phi} \, \bar{h}^3 \, \frac{\partial \bar{P}}{\partial y} \, d\theta \qquad (11)$$

Friction force:

$$\bar{F} = \frac{c}{\mu U R} F = \int_0^{\theta_{cav}} A \, d\theta + \int_{\theta_{cav}}^{2\pi} A \, \frac{\bar{h}(\theta)}{\bar{h}_{cav}} \, d\theta \qquad (12)$$

where:

$$A = \frac{\bar{h}}{2} \frac{d\bar{P}}{d\theta} + \frac{1}{\bar{h} - \frac{2N}{L}\left[\frac{\cosh(NL\bar{h}) - 1}{\sinh(NL\bar{h})}\right]}$$

Friction coefficient:

$$f\left(\frac{R}{c}\right) = \frac{\bar{F}}{\bar{W}} \qquad (13)$$

Equations (7) to (13) were numerically evaluated using a Simpson's integration scheme.

RESULTS

To study the lubricating effectiveness of fluids that exhibit micropolarity, a parametric study was performed. Results are obtained as a function of two-nondimensional parameters, N and ℓ_m which represent the characteristics of micropolar fluid. Unless otherwise stated, the computations pertain to a finite journal bearing of L/D = 1.

Pressure Distribution and Load Capacity

Figure 1 presents a number of three-dimensional pressure profiles for a steadily loaded journal bearing at an eccentricity ratio of ε = 0.5. Figure 1(a) depicts the Newtonian case while Fig. 1(b) and (c) show the effect of micropolar lubricant on the pressure distribution. In Fig. 1(b), the nondimensional characteristic length, ℓ_m, was held constant at ℓ_m = 9.0 while the coupling number parameter, N^2, was varied from 0.1 to 0.9. Computations indicate that although the shape of the pressure distribution remains the same, the magnitude of the pressure and thus the load capacity varies significantly as a function of the coupling number. From thermodynamic considerations parameter N is limited to $0 \leq N \leq 1$, see Eringen (Ref. 3). At high values of N the effect of microstructures becomes significant whereas at low values of N the individuality of the substructure is much less pronounced. In the limiting case, as N approaches zero, the micropolarity is lost and the lubricant is considered to behave as Newtonian. Under such a limiting case, the equation of linear momentum reduces to that of the Navier-Stokes.

The effect of the nondimensional characteristic length on the pressure distribution is depicted in Fig. 1(c). As shown, at low ℓ_m, the effect of microstructure is very pronounced. Physically, lower values of ℓ_m correspond

10

to larger values of the characteristic length ($\ell_m = c/\Lambda$, where c - the bearing clearance - is assumed to remain constant). Small ℓ_m, therefore, indicates that characteristic length of the substructure is large compared to that of the clearance dimension. At high values of ℓ_m, the effect of substructure becomes less and less significant and in the limiting case when ℓ_m approaches infinity, the effect of individuality of the microstructure is lost so that the pressure distribution approaches that of the Newtonian.

Variation of the load carrying capacity as a function of the eccentricity ratio, ε, is depicted in Fig. 2 for various values of N, keeping the nondimensional characteristic length constant, $\ell_m = 9$, for all cases. The enhancement in the load capacity for higher values of the coupling number is pronounced. A possible explanation for such phenomena can be attributed to the increase in the viscosity due to the additives, characterized by micropolar fluids. There are some experimental works that support this view. Henniker's extensive review (Ref. 33) entitled, "The Depth of the Surface Zone of a Liquid," (Ref. 33) attest to the evidence of abnormally high viscosity in the vicinity of a solid surface. He refers to some experimental work by Deryagin et al. (Ref. 34) who found the viscosity of a turbine oil within 5000 A of solid wall to increase tenfold when aluminum naphthionate of up to 2 percent were added in.

Figure 3 depicts how the load carrying capacity varies as a function of ℓ_m for a number of N^2, keeping the eccentricity constant. It clearly shows an appreciable rise in the load capacity at small values off ℓ_m and that the load approaches to that of the Newtonian fluids in the limiting case as $\ell_m \rightarrow \infty$. These results are consistent with the analytical solutions due to Cowin (Ref. 35) who showed that for polar fluids in couette flow and Poiseuille flow the effective viscosity μ_E approaches the Newtonian fluid value μ as $\ell_m \rightarrow \infty$.

Friction Coefficient

Variation of the coefficient of friction as a function of eccentricity is shown in Fig. 4 for a number of N^2 while $\ell_m = 9.0$. Also shown is the corresponding variation in friction coefficient for the Newtonian fluid case. This figure indicates that, under the conditions simulated, micropolar fluids exhibit a beneficial effect in that the coefficient of friction is lower than the Newtonian fluids. Furthermore, as the coupling number increases, a lower friction coefficient may result. To further explore this effect, the coefficient of friction was plotted as a function of the ℓ_m for a number of N^2 as presented in Fig. 5. This figure clearly shows that micropolar friction coefficient approaches that of Newtonian results as $\ell_m \to 0$ and as $\ell_m \to \infty$ or as $N^2 \to 0$. The most interesting result is that there exists an optimum material length at which the friction coefficient is minimum.

It should be noted that although the friction coefficient of lubricant micropolarity tends to be lower than that of the Newtonian fluids, the friction force tends to be higher as depicted in Fig. 6. The reason for this is of course due to the load capacity of the micropolar fluids increasing more than the corresponding increase in (Fig. 3) which more than compensates for the frictional force. Similar behavior were predicted by analytical work of Allen and Kline (Ref. 10) for a one-dimensional slider bearing.

CONCLUDING REMARKS

A study of the lubricating effectiveness of micropolar fluids in a finite journal bearing is presented. Micropolar fluids are a subclass of microfluids in which the fluids exhibit microrotational effects i.e., the fluid is considered to possess microstructure that can rotate independently of the

12

microvolume but the deformation (microstretch) is not allowed. These fluids have potential in describing the effect of polymeric additives as the classical Navier-Stokes theory has no provision for the effects of microstructure in fluids.

The results presented in this paper indicate that for a steadily loaded finite journal bearing, the micropolar fluids do indeed exhibit a beneficial effect in that the load carrying capacity is significantly increased and the friction coefficient is less than that of the Newtonian lubricant.

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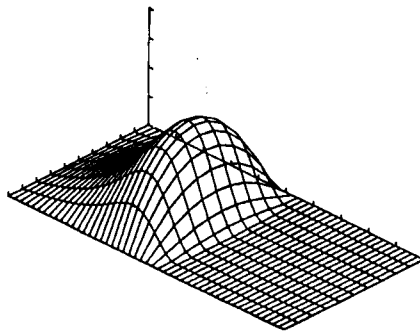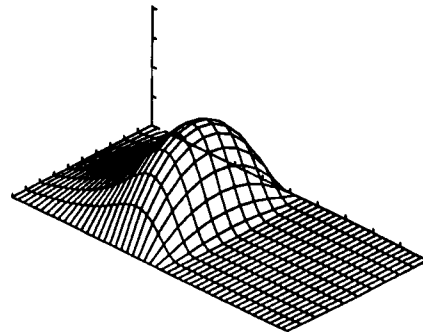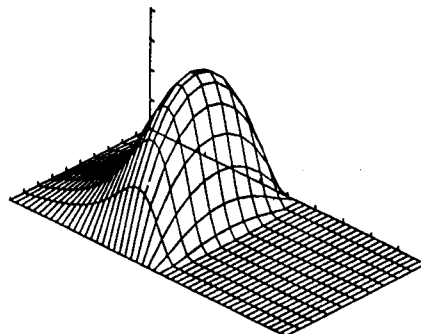
(A) NEWTONIAN.

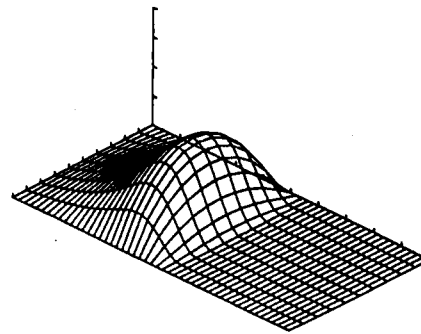$N^2 = 0.1$, $\ell_m = 9.0$

$\ell_m = 1.0$, $N^2 = 0.5$

$N^2 = 0.5$, $\ell_m = 9.0$

$\ell_m = 10.0$, $N^2 = 0.5$

$N^2 = 0.9$, $\ell_m = 9.0$
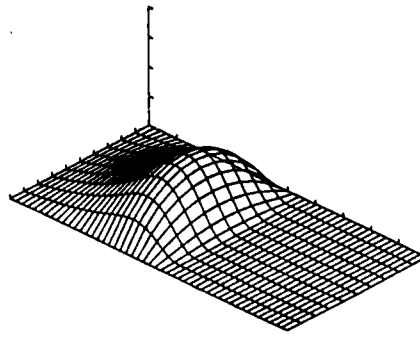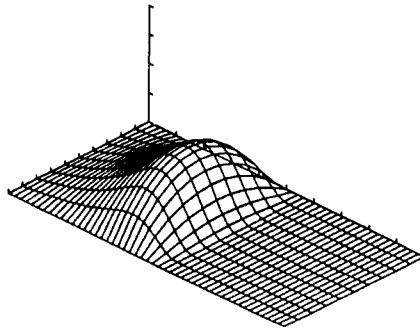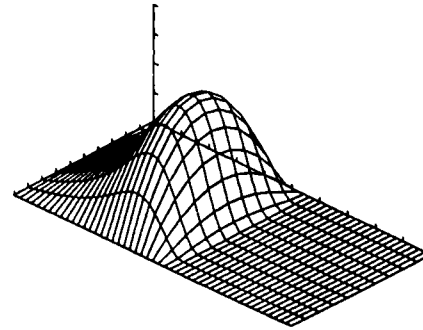
$\ell_m = 20.0$, $N^2 = 0.5$

(B) EFFECT OF COUPLING NUMBER, N^2.

(C) EFFECT OF ℓ_m.

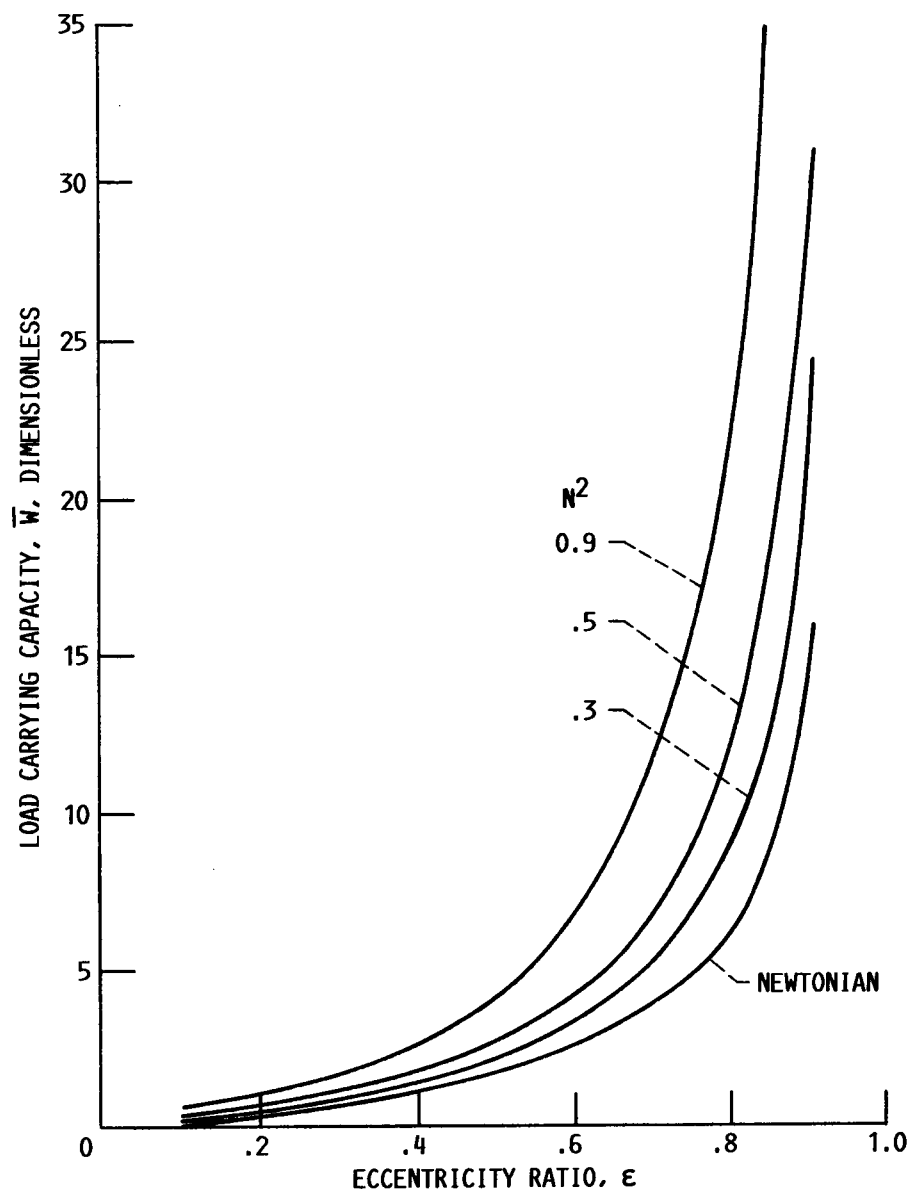FIGURE 1. - PRESSURE DISTRIBUTION FOR NEWTONIAN FLUID AND MICRO-POLAR FLUID. L/D = 1, ε = 0.5.

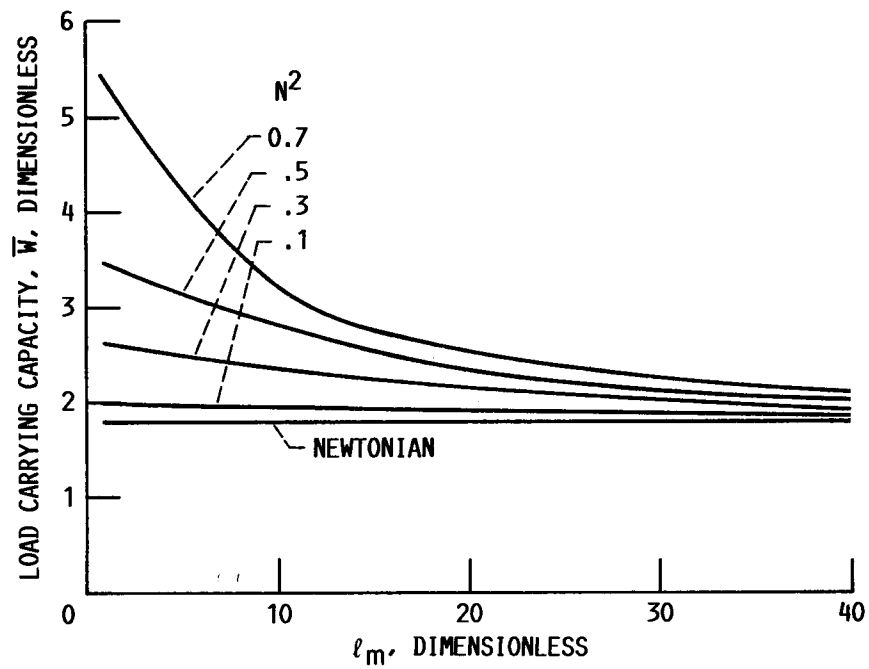FIGURE 2. - VARIATION OF LOAD CAPACITY AS A FUNCTION OF ECCENTRICITY. $L/D = 1$, $\ell_m = 9$.

FIGURE 3. – VARIATION OF LOAD CARRYING CAPACITY AS A
FUNCTION OF ℓ_m AND N^2. L/D = 1, ε = 0.5.
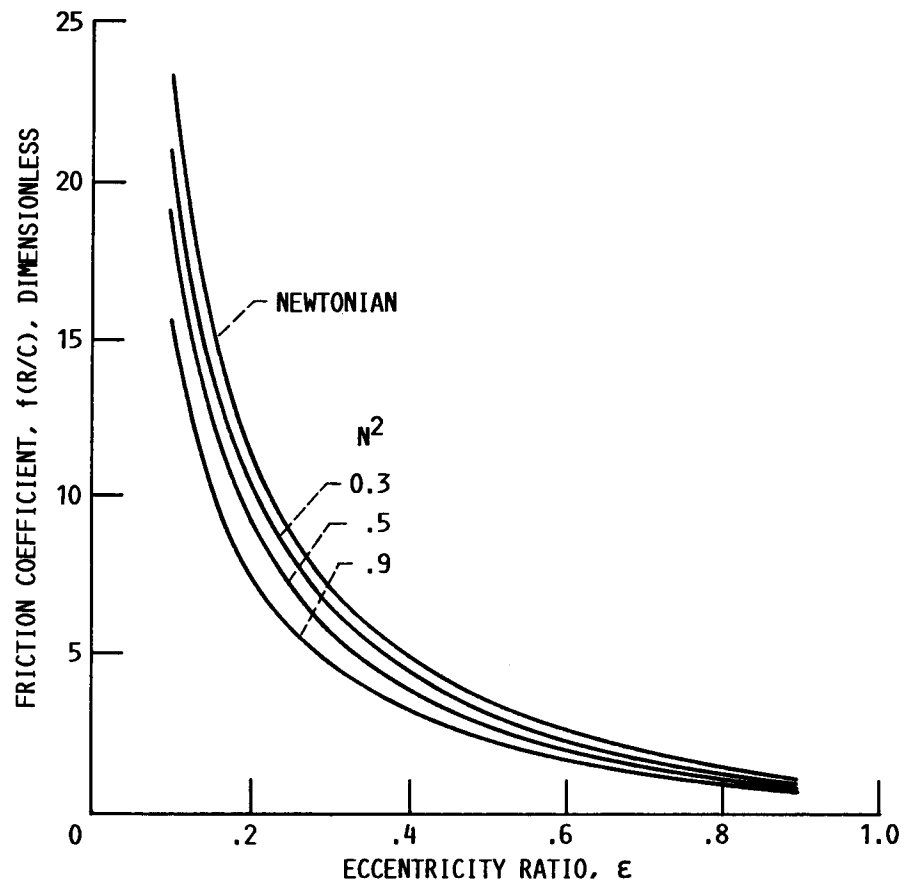

FIGURE 4. – FRICTION COEFFICIENT AS A FUNCTION OF ECCEN-
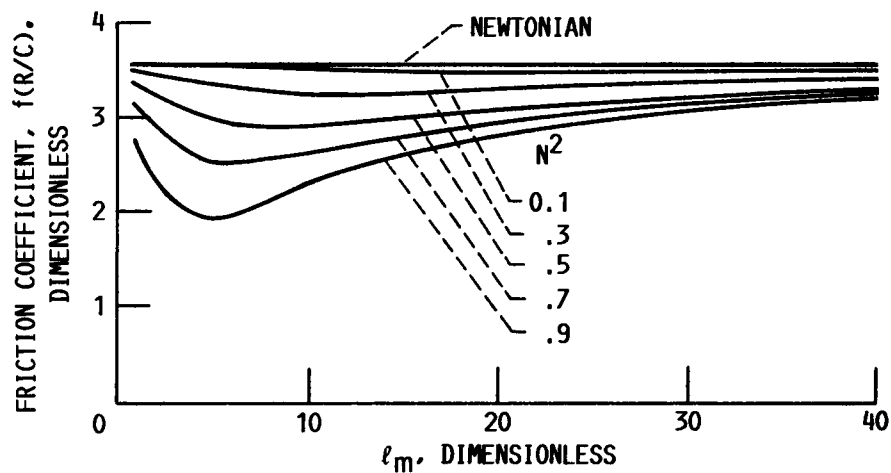TRICITY. L/D = 1, ℓ_m = 9.

FIGURE 5. - VARIATION OF FRICTION COEFFICIENT AS A FUNC-
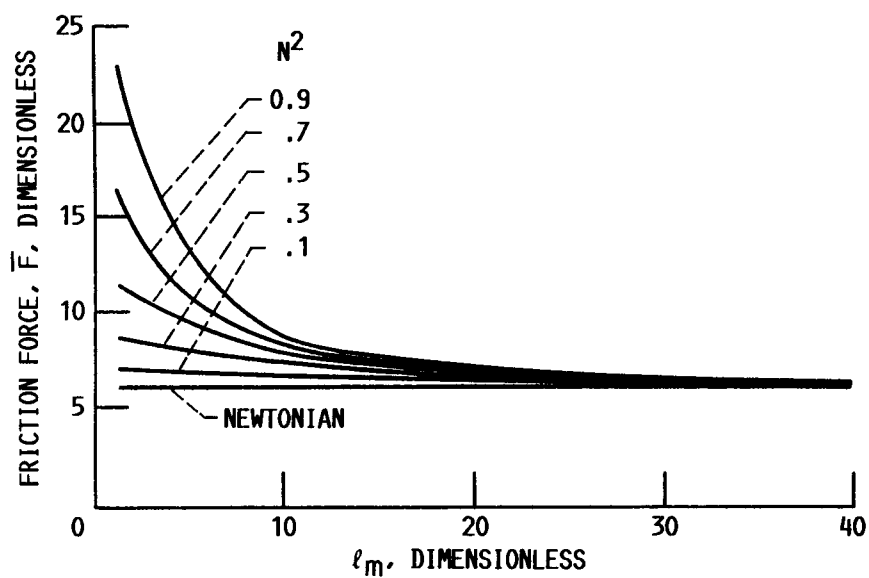TION OF ℓ_m AND N^2. L/D = 1, ε = 0.5.

FIGURE 6. - VARIATION OF FRICTION FORCE AS A FUNCTION OF
ℓ_m AND N^2. L/D = 1, ε = 0.5.

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15. Supplementary Notes

16. Abstract

A study of the performance parameters for a journal bearing of finite length lubricated with micropolar fluids is undertaken. Results indicate that a significantly higher load carrying capacity than Newtonian fluids may result depending on the size of material characteristic length and the coupling number. It is also shown that although the frictional force associated with micropolar fluid is in general higher than that of a Newtonian fluid, the friction coefficient of micropolar fluids tends to be lower than that of the Newtonian.

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