World Academy of Science, Engineering and Technology
International Journal of Civil and Environmental Engineering
Vol:10, No:9, 2016

EHD Effect on the Dynamic Characteristics of a Journal Bearing Lubricated with Couple Stress Fluids

B. Chetti, W. A. Crosby

Open Science Index, Civil and Environmental Engineering Vol:10, No:9, 2016 publications.waset.org/10005222/pdf

***Abstract*—**This paper presents a numerical analysis for the dynamic performance of a finite journal bearing lubricated with couple stress fluid taking into account the effect of the deformation of the bearing liner. The modified Reynolds equation has been solved by using finite difference technique. The dynamic characteristics in terms of stiffness coefficients, damping coefficients, critical mass and whirl ratio are evaluated for different values of eccentricity ratio and elastic coefficient for a journal bearing lubricated with a couple stress fluids and a Newtonian fluid. The results show that the dynamic characteristics of journal bearings lubricated with couple stress fluids are improved compared to journal bearings lubricated with Newtonian fluids.

***Keywords*—**Circular bearing, elastohydrodynamic, stability, couple stress.

I. Introduction

LUBRICANTS in classical hydrodynamic and elastohydrodynamic lubrication analyses are assumed to behave as Newtonian fluids. Advances in technology and in many practical lubrication applications necessitate the development of improved lubricants where the Newtonian fluids constitutive approximation is not a satisfactory engineering approach to the lubrication problems. Experimental results showed that the addition of a small amount of a long-chain polymer to a Newtonian fluid gives the most desirable lubricant. A number of theories of the microcontinuum have been developed to explain the behavior of these fluids as polymeric fluids [1], [2].

Among the microcontinuum theories, the simplest theory derived by Stokes [3] generalizes the classical theory to allow for polar effects such as the presence of couple stresses and body couples.

The couple stresses might be expected to appear at noticeable magnitudes in liquids containing additives with large molecules. These couple stresses may be significant particularly under lubrication conditions where thin films usually exist. The performance characteristics of hydrodynamic journal bearings using lubricants with couple stress have been studied by many researchers. Sinha et al. [4] studied the effects of couple stresses on various bearing characteristics for an infinitely long journal bearing taking into consideration the effect of cavitation. It was found that the presence of a couple stress produces an increase in the load capacity and a decrease in the coefficient of friction. Lin [5] investigated theoretically the theological effects of a couple stress fluid on the lubrication performance of a finite journal bearing. An inverse solution for finite journal bearings lubricated with couple stress fluids to estimate the eccentricity ratio and the couple stress parameter for a given experimentally measured pressure distribution was presented by El-Sharkawy et al. [6]. The effects of couple stresses on dynamic characteristics of a journal bearing have been examined in many studies. Swamy et al. [7] calculated stiffness and damping characteristics of finite width journal bearings with a non-Newtonian film. Lin [8] presented theoretical analysis of a linear stability threshold of a rotor-bearing system lubricated with couple stress fluids. Recently, Guha et al. [9] dealt with the rheological effects of couple stress fluids on the static and dynamic characteristics of finite journal bearings. It was found that the couple stress effects improve the dynamic characteristics of the journal bearing system. In all these studies, bearings were considered to be absolutely rigid. Another recent attempt model by Crosby et al. [10] studied static and dynamic characteristics of two-lobe journal bearings lubricated with couple stress fluids. Nada et al. [11] analyzed the thermal effect on the static characteristics of circular journal bearing lubricated by magnetic fluids with couple stresses.

In heavily loaded journal bearings, the bearing deformation may quite often affect the clearance space geometry of the bearing to an extent such that the actual performance characteristics may become significantly different from those computed with rigid bearings. Many studies are available on the effects of bearing deformation [12]-[16]. Recently, Mokhiamer et al. [17] investigated the effects of the couple stress parameter on the static characteristics of finite journal bearings with flexible bearing linear material.

In the present work, dynamic characteristics in terms of stiffness coefficient, damping coefficient, journal critical mass and whirl frequency ratio are presented for a circular journal rigid and a deformable bearing lubricated with a couple stress fluids.

II. Analysis

Based on the Stokes' [3] microcontinuum theory, when the body forces and body moments are absent, the momentum equation and the continuity equation of an incompressible couple stress fluid are given by:

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

B. Chetti is Faculty of Sciences and Technology, University of Djilali Bounaama-Khemis Miliana, Ain Defla, Algeria (e-mail: b.chetti@gmail.com).

W. A. Crosby is with the Mechanical Department, Faculty of Engineering, University of Alexandria, Alexandria, Egypt.

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

where the vectors V, F, and C represent the velocity, body force per unit mass and body couple per unit mass, respectively. p and ρ are pressure and density; μ is the classical viscosity and η is a new parameter responsible for couple stress property.

Fig. 1 depicts the physical configuration of a finite journal bearing with a journal of radius R rotating with a uniform tangential velocity U .

Making the usual assumptions of hydrodynamic lubrication applicable to thin films, the equations of motion are reduced to:

$$\frac{\partial p}{\partial x} = \mu \frac{\partial^2 u}{\partial y^2} - \eta \frac{\partial^4 u}{\partial y^4} \tag{3}$$

$$\frac{\partial p}{\partial y} = 0 \tag{4}$$

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

The boundary conditions at the bearing surface and the journal surface are

$$u\left(x,0,z\right) = \mathrm{w}\left(x,0,z\right) = 0 \tag{6a}$$

$$\left.\frac{\partial^2 u}{\partial y^2}\right|_{y=0} = \left.\frac{\partial^2 w}{\partial y^2}\right|_{y=0} = 0 \tag{6b}$$

$$u\left(x,h,z\right) = U,\ \mathrm{w}\left(x,h,z\right) = 0 \tag{7a}$$

$$\left.\frac{\partial^2 u}{\partial y^2}\right|_{y=h} = \left.\frac{\partial^2 w}{\partial y^2}\right|_{y=h} = 0 \tag{7b}$$

By applying the above boundary conditions, the velocity components u and w are solved from (3) and (5), respectively:

$$u = U\frac{y}{h} + \frac{1}{2\mu}\frac{\partial p}{\partial x}\left\{ y\left(y-h\right) + 2\ell^2\left[1 - \frac{\cosh\left[\left(2y-h\right)/\left(2\ell\right)\right]}{\cosh\left[h/\left(2\ell\right)\right]}\right]\right\} \tag{8}$$

and

$$w = \frac{1}{2\mu}\frac{\partial p}{\partial z}\left\{ y\left(y-h\right) + 2\ell^2\left[1 - \frac{\cosh\left[\left(2y-h\right)/\left(2\ell\right)\right]}{\cosh\left[h/\left(2\ell\right)\right]}\right]\right\} \tag{9}$$

where

$$\ell = \left(\frac{\eta}{\mu}\right)^{1/2}$$

A. Modified Reynolds Equation

Integrating the continuity equation (2) with respect to y with the boundary conditions for v as given by [8];

$$v\left(x,0,z\right) = 0\ ,\ v\left(x,h,z\right) = -\frac{\partial h}{\partial t} \tag{10}$$

The modified Reynolds equation is derived as,

$$\frac{\partial}{\partial x}\left[G\left(h,\ell\right)\frac{\partial p}{\partial x}\right] + \frac{\partial}{\partial z}\left[G\left(h,\ell\right)\frac{\partial p}{\partial z}\right] = 6\mu U\frac{\partial h}{\partial x} + 12\mu\frac{\partial h}{\partial t} \tag{11}$$

where

$$G\left(h,\ell\right) = h^3 - 12\ell^2 h + 24\ell^3\tanh\left(\frac{h}{2\ell}\right) \tag{12}$$

To obtain the modified Reynolds equation in dimensionless form, the following dimensionless variables are introduced:

$$\bar{x} = \frac{x}{R},\ \bar{z} = \frac{z}{L},\ \varepsilon = \frac{e}{c},\ \bar{p} = \frac{pc^2}{\mu UR},\ \bar{h} = \frac{h}{c},\ \bar{\ell} = \frac{\ell}{c},\quad \tau = \omega t$$

Thus, the modified Reynolds equation becomes:

$$\frac{\partial}{\partial\theta}\left[\bar{G}\left(\bar{h},\bar{\ell}\right)\frac{\partial\bar{p}}{\partial\theta}\right] + \left(\frac{R}{L}\right)^2\frac{\partial}{\partial\bar{z}}\left[\bar{G}\left(\bar{h},\bar{\ell}\right)\frac{\partial\bar{p}}{\partial\bar{z}}\right] = 6\frac{\partial\bar{h}}{\partial\theta} + 12\frac{\partial\bar{h}}{\partial\tau} \tag{13}$$

where

$$\bar{G}\left(\bar{h},\bar{\ell}\right) = \bar{h}^3 - 12\bar{\ell}^2\bar{h} + 24\bar{\ell}^3\tanh\left(\frac{\bar{h}}{2\bar{\ell}}\right) \tag{14}$$

B. Pressure Boundary Conditions

$$\bar{p} = 0 \quad \text{at} \quad \bar{z} = 0 \tag{15a}$$

$$\bar{p} = 0 \quad \text{at} \quad \bar{z} = 1 \tag{15b}$$

$$\bar{p} = 0 \text{ at } \theta = 0 \tag{15c}$$

$$\bar{p} = \frac{\partial p}{\partial\theta} = 0 \text{ at } \quad \theta = \theta^* \tag{15d}$$

Equations (15a) and (15b) result from the fact that the ends of the bearing are exposed to the ambient pressure, while (15c) and (15d) are the Reynolds (Swift-Stieber) conditions (θ^* is the position of the trailing edge).

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C. The Film Thickness

The film thickness at any angle θ for a rigid bearing is given by

$$h = c + e\cos\theta \tag{16}$$

and in dimensionless form

$$\bar{h} = 1 + \varepsilon\cos\theta \tag{17}$$

The film thickness at any angle θ taking into account the elastic deformation of the bearing liner is [13]:

$$h = c\left(1 + \varepsilon\cos\theta\right) + \delta \tag{18}$$

where

$$\delta = \frac{pt}{E}\left(1 - \nu^2\right) \tag{19}$$

in dimensionless form:

$$\bar{h} = 1 + \varepsilon\cos\theta + Co\,\bar{p} \tag{20}$$

where Co is the elastic coefficient defined as:

$$Co = \frac{\mu URt\left(1 - \nu^2\right)}{c^3 E} \tag{21}$$

III. Solution Procedure

The governing equations are solved numerically by using the finite difference method. 72 intervals are used in the circumferential direction and 20 intervals across the bearing length. The global iterative scheme is as follows: an initial value for the pressure field is given to calculate the film geometry at each point along and across the film. Modified Reynolds equation is substituted at each mesh point thus producing a new pressure field. The iterative procedure is stopped when at each point the error between two successive iterations fell below a tolerance ratio of 0.001.

The dimensionless radial and tangential load components are found from,

$$\begin{aligned} W_r &= \int_0^1\int_0^\theta p\cos d\theta dz \\ W_t &= \int_0^1\int_0^\theta p\sin d\theta dz \end{aligned} \tag{22}$$

The load and attitude angle are given by,

$$W = (W_r + W_t)^{1/2}\ , \quad \phi = \tan^{-1}(W_t / W_r) \tag{23}$$

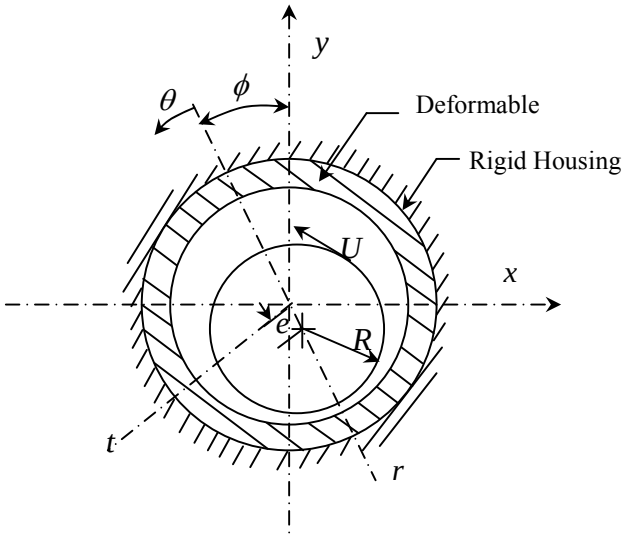


Fig. 1 Journal bearing geometry

IV. Stability Analysis

The stiffness coefficients are found from the following expressions,

$$\begin{aligned} \bar{K}_{xx} &= \left[\frac{\partial W_r}{\partial\varepsilon}\sin^2\phi + \frac{W_r}{\varepsilon} - \frac{\partial W_t}{\partial\varepsilon}\sin\phi\cos\phi\right]/W \\ \bar{K}_{xy} &= \left[\frac{\partial W_r}{\partial\varepsilon}\sin\phi\cos\phi - \frac{\partial W_t}{\partial\varepsilon}\cos^2\phi - \frac{W_t}{\varepsilon}\right]/W \\ \bar{K}_{yy} &= \left[\frac{\partial W_r}{\partial\varepsilon}\cos^2\phi + \frac{\partial W_t}{\partial\varepsilon}\sin\phi\cos\phi\right]/W \\ \bar{K}_{yx} &= \left[\frac{\partial W_r}{\partial\varepsilon}\sin\phi\cos\phi + \frac{\partial W_t}{\partial\varepsilon}\sin^2\phi\right]/W \end{aligned} \tag{24}$$

In these equations, the partial derivatives of ϕ are found by changing the attitude angle by a small value, keeping the eccentricity ratio constant and noting the corresponding change in the values of radial and tangential loads. Similarly, the partial derivatives of ε are found by changing the eccentricity by a small amount, keeping the attitude angle corresponding to the static equilibrium of the bearing.

The expressions of the damping coefficients are,

$$\bar{C}_{xx} = \left[\frac{\partial W_r}{\partial \varepsilon} \sin^2\phi + \frac{\partial W_r}{\partial \phi} \frac{\sin\phi\cos\phi}{\varepsilon} - \frac{\partial W_t}{\partial \varepsilon} \sin\phi\cos\phi - \frac{\partial W_t}{\partial \phi} \frac{\cos^2\phi}{\varepsilon} \right] / W$$

$$\bar{C}_{xy} = \left[\frac{\partial W_r}{\partial \varepsilon} \sin\phi\cos\phi - \frac{\partial W_r}{\partial \phi} \frac{\sin^2\phi}{\varepsilon} - \frac{\partial W_t}{\partial \varepsilon} \cos^2\phi + \frac{\partial W_t}{\partial \phi} \frac{\sin\phi\cos\phi}{\varepsilon} \right] / W$$

$$\bar{C}_{yy} = \left[\frac{\partial W_r}{\partial \varepsilon} \cos^2\phi - \frac{\partial W_r}{\partial \phi} \frac{\sin\phi\cos\phi}{\varepsilon} + \frac{\partial W_t}{\partial \varepsilon} \sin\phi\cos\phi - \frac{\partial W_t}{\partial \phi} \frac{\sin^2\phi}{\varepsilon} \right] / W \quad (25)$$

$$\bar{C}_{yx} = \left[\frac{\partial W_r}{\partial \varepsilon} \sin\phi\cos\phi + \frac{\partial W_r}{\partial \phi} \frac{\cos^2\phi}{\varepsilon} + \frac{\partial W_t}{\partial \varepsilon} \sin^2\phi + \frac{\partial W_t}{\partial \phi} \frac{\sin\phi\cos\phi}{\varepsilon} \right] / W$$

The partial derivatives of $\dot{\varepsilon}$ and $\dot{\phi}$ are obtained by giving a small value to $\dot{\varepsilon}$ and $\dot{\phi}$, respectively to the rotor corresponding to the equilibrium position of the bearing.

The critical mass and the whirl ratio γ are calculated by;

$$\overline{M}\gamma^2 = \frac{\bar{C}_{xx}\bar{K}_{yy} + \bar{C}_{yy}\bar{K}_{xx} - \bar{C}_{xy}\bar{K}_{yx} - \bar{C}_{yx}\bar{K}_{xy}}{\bar{C}_{xx} + \bar{C}_{yy}} \quad (26)$$

$$\gamma^2 = \frac{\left(\bar{K}_{xx} - \overline{M}\gamma^2\right)\left(\bar{K}_{yy} - \overline{M}\gamma^2\right) - \bar{K}_{yx}\bar{K}_{xy}}{\bar{C}_{xx}\bar{C}_{yy} - \bar{C}_{xy}\bar{C}_{yx}} \quad (27)$$

V. Results and Discussion

Computer solutions for a finite journal bearing have been computed for a journal radius to bearing length R/L of 0.5, couple stress parameter ranging from 0.0 to 0.4 and the elastic coefficient ranging from 0.0 to 0.1. Compared the results obtained in this work with the results published, the comparison is good for the case of rigid journal bearing with couple stress fluids [8].

The stability charts for different values of couple stress parameter of rigid and deformable circular journal bearing are given in Fig. 2. From these curves, it can be seen that the stable region increases with the increase of both the elastic coefficient and the couple stress parameter. The increase is very pronounced at higher values of eccentricity ratio. This increase of the stable region is due to the increase of the load carrying capacity using the couple stress fluids.

Fig. 3 shows the influence of couple stress parameter and the elastic coefficient on the whirl ratio. These results indicate that the whirl ratio decreases with an increase of both couple stress parameter and elastic coefficient, and the effect is very important at high values of eccentricity ratio.

The stability margins and whirl ratio in relation with the elastic coefficient for eccentricity ratios of 0.3 and 05 and for journal bearings lubricated with Newtonian and couple stress fluid are shown in Figs. 4 and 5. From these curves, it can be observed that the critical mass increases and the whirl ratio decreases with an increase of both eccentricity ratio and couple stress parameter. The increase of the critical mass with the increasing of the eccentricity ratio is produced by the increase of the load carrying capacity. At low eccentricity ratio, the variation of the critical mass and whirl ratio with elastic coefficient is not important, while at high eccentricity ratio the influence is more significant.

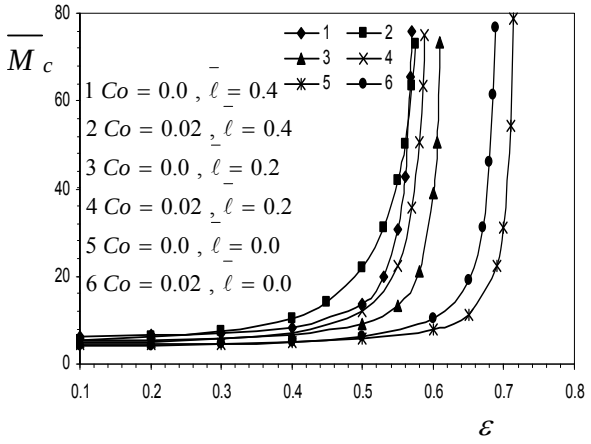


Fig. 2 $\overline{M}_c$ versus ε for various values of couple stress parameter $\bar{\ell}$

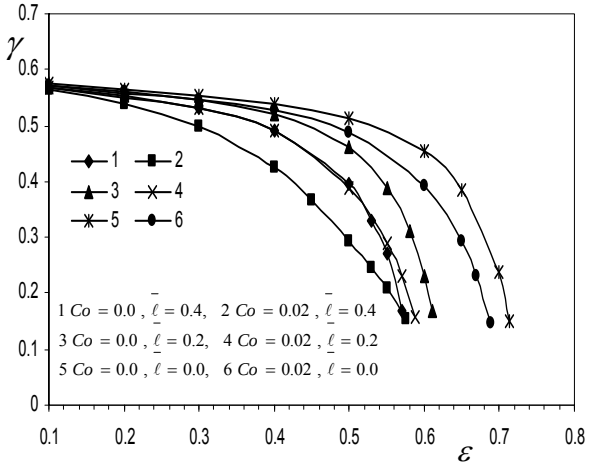


Fig. 3 γ versus ε for various values of couple stress parameter $\bar{\ell}$

VI. Conclusions

From the results and discussions of the effect of the couple stress parameter and the elastic coefficient on the dynamic characteristics of a finite journal bearing, the following conclusions are drawn:

- The stability of the journal bearing increases with an increase of both of elastic coefficient and couple stress parameter.
- The whirl ratio decreases with an increase of both of elastic coefficient and couple stress parameter.
- At high eccentricity ratios, the elastic coefficient has more influence on the stability and the whirl of the journal bearing lubricated with couple stress fluid or with a Newtonian fluid.

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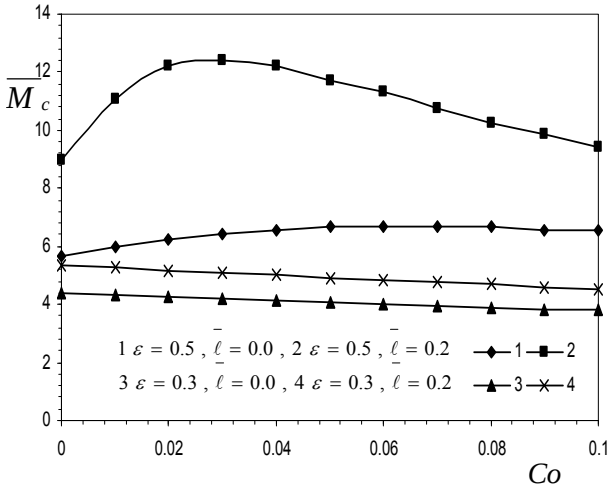


Fig. 4 $\overline{M}_c$ versus Co for various value of couple stress parameter $\bar{\ell}$

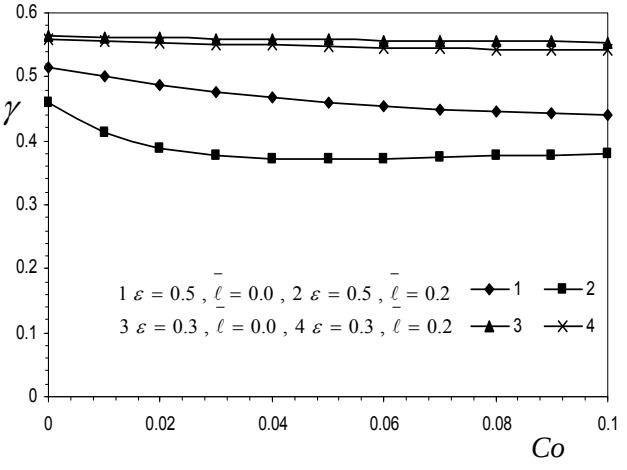


Fig. 5 γ versus Co for various values of couple stress parameter $\bar{\ell}$

NOMENCLATURE

c	The radial clearance
Co	Elastic Coefficient
C_f	Friction Coefficient
C_{ij}	Damping coefficients
$\overline{C}_{ij}$	Dimensionless damping coefficients, $C_{ij}c\omega / W$
e	Eccentricity
E	Modulus of elasticity of the bearing
F_h	Friction force
$\overline{F}_h$	Dimensionless friction force, $\overline{F}_h = F_h c / RLU\mu$
h	Oil film thickness
$\overline{h}$	Dimensionless oil film thickness, h / c
K_{ij}	Stiffness coefficients
$\overline{K}_{ij}$	Dimensionless stiffness coefficients, $K_{ij}c / W$
L	Bearing length
M	Mass of journal
M_c	Critical mass of journal
$\overline{M}_c$	Dimensionless critical mass of journal, $M_c c\omega^2 / W$
p	Pressure
$\bar{p}$	Dimensionless pressure, $p(c / R^2) / \mu\omega$
Q_s	Side leakage flow
$\overline{Q}_s$	Dimensionless side leakage flow, $Q_s L / UR^2 c$
R	Journal radius
S	Sommerfeld number, $1 / (2\pi\overline{W})$
t	Time
u, v, w	Velocity components
U	Velocity of the journal
W	Bearing load
W_r, W_t	Load components in radial and tangential direction respectively
$\overline{W}$	Dimensionless bearing load, $W\left(c / R\right)^2 / \mu\omega RL$
x, y, z	Circumferential, radial and axial coordinates respectively
$\bar{x}, \bar{y}, \bar{z}$	Dimensionless circumferential, radial and axial coordinates respectively, $x / R, y / h, z / L$
ℓ	Characteristics length of additives, $\ell = (\eta / \mu)^{1/2}$
$\bar{\ell}$	Couple stress parameter, ℓ / c
δ	The bearing liner deformation
ε	Eccentricity ratio, e / c
η	Material constant responsible for the couple stress property
θ	Angular coordinate
μ	Lubricant Viscosity
ρ	Lubricant density
ν	Whirl frequency
γ	Whirl ratio
ϕ	Attitude angle
ω	Angular velocity of the journal

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