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Analysis of Journal Bearing Considering the Effects of Surface Layer and Couple Stress Fluids

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ABSTRACT

This study presents an analysis of journal bearing lubricated with couple stress fluids considering the effects of a layer adhered to bearing surface. The modified classical Reynold's equation is derived considering the effects of surface layer and couple stress fluids. In the present study, the effects of couple stresses on the steady state journal bearing performance characteristics are analyzed based on Stokes micro-continuum theory. The Reynold's boundary conditions are used in the analysis. Results of non-dimensional load capacity and coefficient of friction are presented.

Key words: Journal bearing, surface layer, couple stress fluid, load capacity, coefficient of friction

INTRODUCTION

Structure and properties of fluid film are important aspects in the analysis of journal bearing lubrication. Szeri (2010) advocated the modification of structure of fluid film to a composite film that combines both high and low viscosity fluid. Composite film bearings provide significant savings in power loss due to viscous friction in hydrodynamic bearings. Tichy (1995) developed a rheological model for thin film lubrication in which surface layers appears to have higher viscosity compared to conventional viscosity.

The increasingly severe technological requirements led to considerable thermal and mechanical deformations and consequently compromise minimum film thickness in bearings (Mansouri *et al*., 2007). The couple stresses fluid takes into account the properties of lubricants with additives. Couple stress fluid model based on micro-continuum theory is the simplest generalization of classical theory of fluids. The experimental study of Oliver (1988) predicted load enhancement and friction reduction effects in a short journal bearing due to dissolved polymer in lubricant. Bujurke and Naduvinamani (1991) presented characteristics of narrow porous journal bearing lubricated with couple stress fluids. Couple stress fluid theory predicts improvement in load capacity and reduction in coefficient of friction in journal bearings compared to those predicted using Newtonian fluids. Lin (1997) investigated the effects of couple stresses on lubrication of finite journal bearing. Mokhiamer *et al*. (1999) presented a study of couple stress fluid taking into account the elasticity of the liner in journal bearing. Li and Chu (2004) and Elsharkawy (2005) derived both porous media and couple stress model to study the effects of lubricant additives on the bearing performance characteristics. Siddiqui *et al*. (2006) presented an analytical study of an infinite slider bearing lubricated with Powell-Eyring fluid.

The purpose of this study is to investigate the non-dimensional load capacity and coefficient of friction in long journal bearing using both surface layer and couple stress fluid effects. A modified Reynold's equation is derived considering Newtonian surface layer and Stokes' couple stress fluid model. The non-dimensional pressure and shear stress expressions are derived. Reynold's boundary conditions are used to solve the pressure distribution. The influence of surface layer and couple stress effects on the bearing characteristics are presented.

ANALYSIS

In this theoretical study, the influence of Newtonian surface layer and Stokes' couple stress fluid film on the journal bearing load capacity and coefficient of friction are investigated. The schematic of journal bearing with surface layer and couple stress fluid film is shown in Fig. 1.

Using the assumptions of thin film lubrication theory, the momentum equations for Newtonian surface layer ($0 \le y \le \delta$) and the equations of motion for Stokes' couple stress fluid film ($\delta \le y \le h$) are:

$$\frac{1}{\mu_1}\frac{dp}{dx} = \frac{d^2u_1}{dy^2} \tag{1}$$

$$\frac{1}{\eta}\frac{dp}{dx} = \frac{\mu_2}{\eta}\frac{d^2u_2}{dy^2} - \frac{d^4u_2}{dy^4} \tag{2}$$

The boundary conditions are that velocities and viscous shear stresses are continuous at the interface, the couple stresses vanish at the interface, and no-slip at the bearing surfaces:

$$y = 0 : u_1 = 0 \tag{3}$$

$$y = \delta : u_1 = u_2 = u_{12},\ \mu_1\frac{du_1}{dy} = \mu_2\frac{du_2}{dy} \text{ and } \frac{d^2u_2}{dy^2} = 0 \tag{4}$$

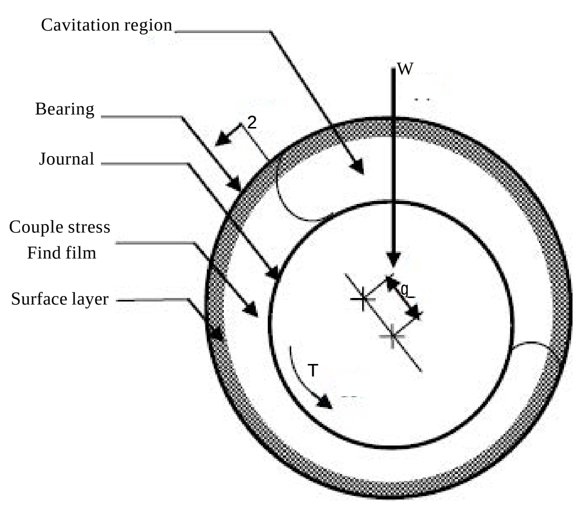

Fig. 1: Geometry of journal bearing with surface layer and couple stress fluid film

$$y = h : u_2 = u_j \text{ and } \frac{d^2u_2}{dy^2} = 0 \tag{5}$$

Integrating the Eq. 1 and 2, using the boundary conditions in Eq. 3 and 5, the non-dimensional velocity distribution in the surface layer ($0 \le Y \le \Delta$) and couple stress fluid film ($\Delta \le Y \le H$) is expressed as:

$$0 \le Y \le \Delta : U_1 = U_{12}\frac{Y}{\Delta} + \frac{1}{2\alpha}\frac{dP}{d\theta}\left(Y^2 - \Delta^2\right) \tag{6}$$

$$\Delta \le Y \le H : U_2 = U_{12} + \frac{1}{2}\frac{dP}{d\theta}(Y-\Delta)(Y-H) + (1-U_{12})\left(\frac{Y-\Delta}{H-\Delta}\right) + \lambda^2\frac{dP}{d\theta}C(Y) \tag{7}$$

Where:

$$C(Y) = 1 + \frac{\sinh\left(\frac{Y-H}{\lambda}\right)}{\sinh\left(\frac{H-\Delta}{\lambda}\right)} - \frac{\sinh\left(\frac{Y-\Delta}{\lambda}\right)}{\sinh\left(\frac{H-\Delta}{\lambda}\right)} \tag{8}$$

$$U_{12} = F_1 - \frac{dP}{d\theta}F_2 \tag{9}$$

$$F_1 = \frac{\Delta}{\left[\alpha(H-\Delta)+\Delta\right]},\ F_2 = F_1(H-\Delta)\left(\frac{1}{2}H - \lambda H_1^*\right) \tag{10}$$

$$H_1^* = \left[\coth\left(\frac{H-\Delta}{\lambda}\right) - \operatorname{csc h}\left(\frac{H-\Delta}{\lambda}\right)\right] \tag{11}$$

$$H = (1 + \varepsilon\cos\theta) \tag{12}$$

The equation of continuity across the film is:

$$Q = \int_0^{\Delta} U_1 dY + \int_{\Delta}^{H} U_2 dY \tag{13}$$

Simplifying the equation of continuity across the film, yields:

$$\frac{dP}{d\theta} = \frac{G_1 - Q}{G_2} \tag{14}$$

Where:

$$G_1 = \frac{1}{2}\left[H(F_1+1) - \Delta\right] \tag{15}$$

$$G_2 = \frac{1}{12\alpha}\Delta^3 + \frac{1}{12}(H-\Delta)^3 - \lambda^2(H-\Delta) + 2\lambda^3 H_1^* + \frac{1}{2}F_2 H \tag{16}$$

For $\Delta = 0$, G_1 and G_2 in Eq. 15 and 16 reduce to:

$$G_1 = \frac{1}{2}H \text{ and } G_2 = \frac{1}{12}H^3 - \lambda^2 H + 2\lambda^3 H_1^* \tag{17}$$

For $\Delta = 0$ and $\lambda = 0$, G_1 and G_2 in Eq. 17 reduce to:

$$G_1 = \frac{1}{2}H \text{ and } G_2 = \frac{1}{12}H^3 \tag{18}$$

The Reynold's boundary conditions are:

$$P\big|_{\theta=0} = 0,\ P\big|_{\theta=\theta_r} = 0 \text{ and } \frac{dP}{d\theta}\bigg|_{\theta=\theta_r} = 0 \tag{19}$$

Integrating the Eq. 14 and substituting the first boundary condition given in Eq. 19, yields the non-dimensional pressure profile as:

$$P = \int_0^\theta \frac{G_1}{G_2}d\theta - Q\int_0^\theta \frac{1}{G_2}d\theta \tag{20}$$

Substitution of the Reynold's boundary conditions for non-dimensional pressure at film rupture in Eq. 20 and simplifying results in Q as:

$$Q = \frac{\int_0^{\theta_r} \frac{G_1}{G_2}d\theta}{\int_0^{\theta_r} \frac{1}{G_2}d\theta} \tag{21}$$

Substituting the pressure gradient boundary condition given in Eq. 19 in the expression for non-dimensional pressure gradient in Eq. 14, results in:

$$Q = G_1\big|_{\theta=\theta_r} \tag{22}$$

The Newton-Raphson iterative procedure is used to solve simultaneously both θ_r and Q using Eq. 21 and 22.

The radial and tangential non-dimensional load capacity obtained by integration of non-dimensional pressure along and perpendicular to line of centers are expressed as:

$$W_\varepsilon = -\int_0^{\theta_r} P\cos\theta d\theta,\ W_\phi = \int_0^{\theta_r} P\sin\theta d\theta \tag{23}$$

The non-dimensional load capacity is expressed as:

$$W = \sqrt{W_\varepsilon^2 + W_\phi^2} \tag{24}$$

The non-dimensional shear stress in the journal bearing at Y = H is obtained as:

$$\Pi\big|_{Y=H} = \frac{dU_2}{dY}\bigg|_{Y=H} \tag{25}$$

The non-dimensional friction force on the journal surface is obtained by integrating the shear stress along the journal surface as:

$$F = \int_0^{\theta_c} \Pi d\theta \tag{26}$$

The non-dimensional friction coefficient is calculated as:

$$C_f = \left(\frac{R}{C}\right)\frac{f}{w} = \frac{F}{W}$$

RESULTS AND DISCUSSION

A journal bearing considering the effects of surface layer and couple stress fluids is considered in the analysis. The parameters used in the analysis of journal bearing are: eccentricity ratio (ε) = 0.5; couple stress parameter (λ) = 0.1, 0.2, 0.3, 0.4; dynamic viscosity ratio of surface layer to couple stress fluid film (α) = 10, 100 and non-dimensional surface layer thickness (Δ) = 0.025, 0.05, 0.075, 0.1, 0.2, 0.3, 0.4. The influence of couple stress parameter and high viscosity surface layer on the load capacity enhancement and coefficient of friction reduction for journal bearing are analyzed.

Figure 2a and b show the non-dimensional load capacity (W) with variation in non-dimensional surface layer thickness (Δ = 0.025-0.1 and Δ = 0.1-0.4) for various values of (1) couple stress parameter (λ) and (2) dynamic viscosity ratio of surface layer to couple stress fluid film (α).

The non-dimensional load capacity (W) increases (1) with increase in couple stress parameter (λ) and (2) increase in non-dimensional surface layer thickness (Δ) in the range 0.025-0.1. The non-dimensional load capacity (W) for α = 100 increases significantly with increase in non-dimensional surface layer thickness (Δ = 0.1-0.4). The non-dimensional load capacity (W) increases with increase in non-dimensional surface layer thickness (Δ = 0.1-0.4) for α = 10 and λ = 0.1-0.2. The non-dimensional load capacity (W) increases with increase in non-dimensional surface layer thickness (Δ = 0.1-0.3) for α = 10 and λ = 0.3-0.4. It is observed that non-dimensional load capacity (W) increases with increase in couple stress parameter (λ) especially for higher values of dynamic viscosity ratio of surface layer to couple stress fluid film (α = 100).

Figure 3a and b show the coefficient of friction (C_f) with variation in non-dimensional surface layer thickness (Δ = 0.025-0.1 and Δ = 0.1-0.4). The coefficient of friction (C_f) decreases (i) with increase in couple stress parameter (λ) and (ii) increase in non-dimensional surface layer thickness (Δ) in the range 0.025-0.1. For a given value of non-dimensional surface layer thickness (Δ),

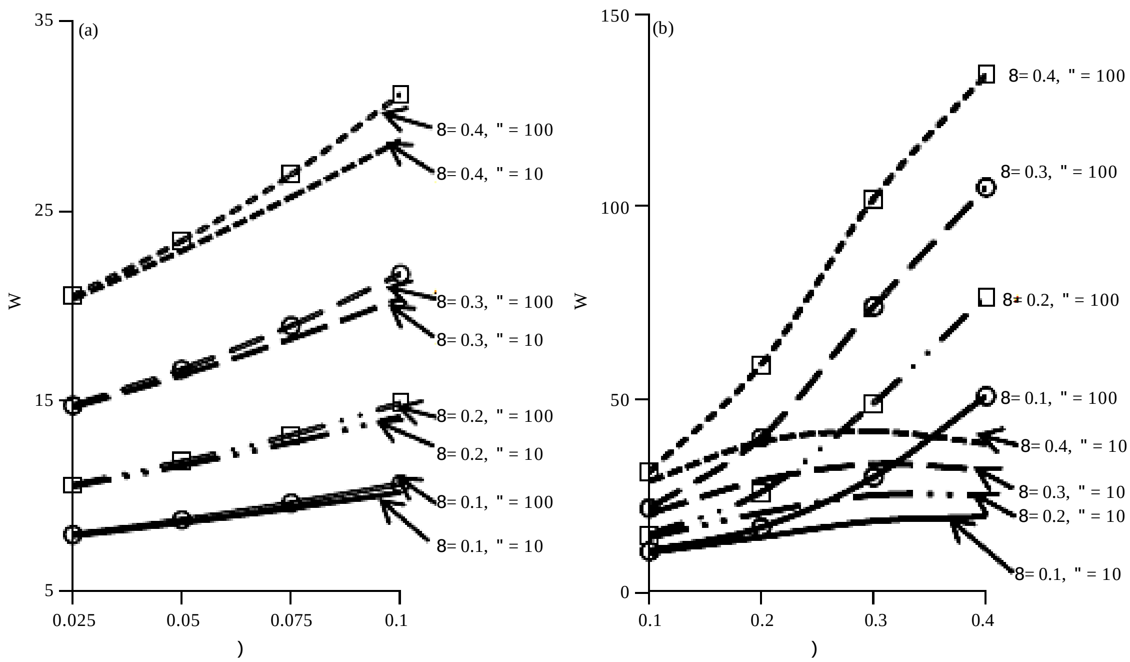

Fig. 2(a-b): Non-dimensional load capacity (ε = 0.5)

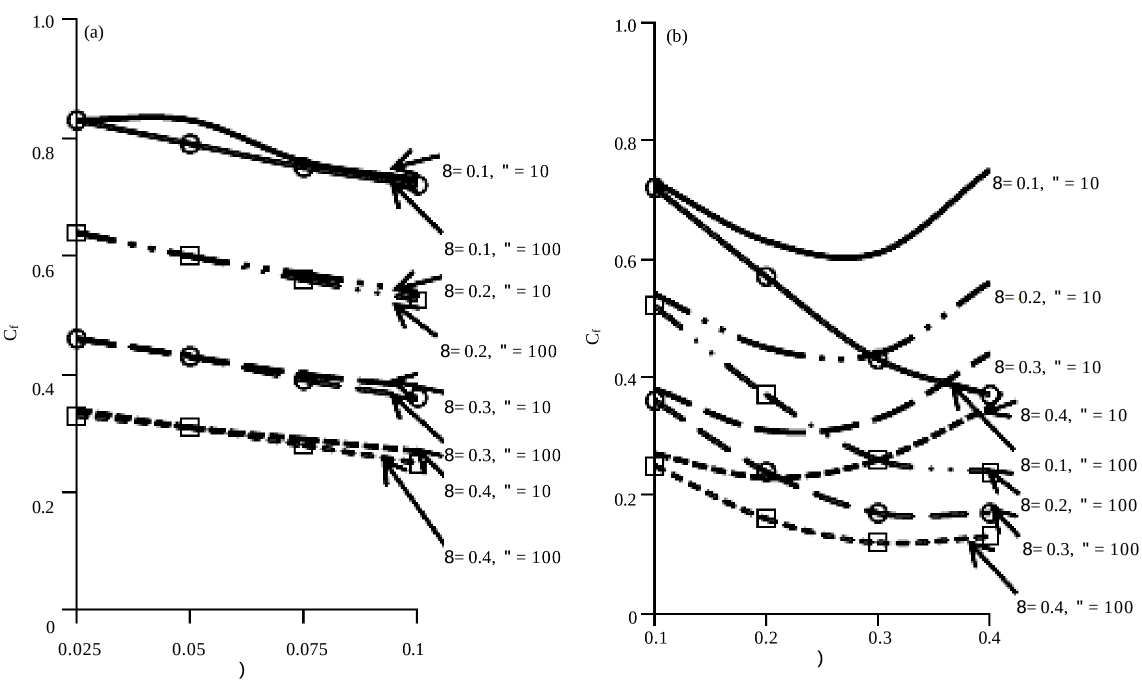

Fig. 3(a-b): Coefficient of friction (ε = 0.5)

coefficient of friction decreases with increase in (1) couple stress parameter (λ) and (2) dynamic viscosity ratio of surface layer to couple stress fluid film (α). Also for a given value of couple stress parameter (λ), the reduction in coefficient of friction with non-dimensional surface layer thickness (Δ = 0.3-0.4) is higher for α = 100 compared to α =10.

CONCLUSION

The present study evaluates on improvement in load capacity and reduction in friction coefficient for a journal bearing considering couple stress parameter and high viscosity surface layer. Modified Reynold's equation is derived for long journal bearing taking into consideration of surface layer and couple stress effects. Both surface layer and couple stress fluid film improve the characteristics of journal bearing. The conclusions based on the analysis presented in this study are:

- The non-dimensional load capacity (W) increases significantly with increase in non-dimensional surface layer thickness (Δ) for higher values of ratio of dynamic viscosity of surface layer and couple stress fluid film (α). The non-dimensional load capacity (W) also increases significantly with increase in couple stress parameter (λ) for higher values of dynamic viscosity ratio of surface layer to couple stress fluid film (α) and non-dimensional surface layer thickness (Δ)
- The coefficient of friction (C_f) decreases with increase in non-dimensional surface layer thickness (Δ) and couple stress parameter (λ), for higher values of dynamic viscosity ratio of surface layer to couple stress fluid film (α)

High viscosity bearing surface layer and couple stress effects increase the load carrying capacity and reduce the coefficient of friction in a journal bearing.

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NOMENCLATURE

C	=	Radial clearance (m)
f	=	Friction force, N; $F = fC/\mu uRL$
h, H	=	Film thickness (m), $H = h/C$
L	=	Length of the journal bearing (m)
p	=	Pressure distribution, (N m^{-2}), $P = pC^2/\mu u_j R$
q	=	Volume flow rate per unit length along film thickness, (m^2 sec^{-1}), $Q = q/u_j C$
R	=	Journal radius (m)
u	=	Velocity component along θ direction (m sec^{-1}), $U = u/u_j$
u_i, i = 1, 2	=	Velocity component along θ direction in thin fluid film layer and fluid film layer with couple stress fluid respectively (m sec^{-1})
u_{12}	=	Velocity component along θ direction at the interface of thin fluid film layer and fluid film layer with couple stress fluid respectively (m sec^{-1})
u_j	=	Journal velocity along θ direction (m sec^{-1})
w	=	Static load, N; $W = wC^2/\mu u_j R^2 L$
W_ε, W_ϕ	=	Non-dimensional radial and tangential static load for journal bearing
x	=	Coordinate along circumferential (x) direction, m; $\theta = x/R$

y	=	Coordinate along radial (y) direction (m), Y = y/c
α	=	Dynamic viscosity ratio of surface layer to couple stress fluid film; $\alpha = \mu_1/\mu_2$
δ	=	Thickness of surface layer (m), $\Delta = \delta/C$
ε	=	Journal bearing eccentricity ratio
μ_i, i = 1, 2	=	Dynamic viscosity of surface layer and fluid film layer with couple stress fluid respectively (Ns m^2)
η	=	Material constant for couple stress (kg sec^{-1})
λ	=	Couple stress parameter; $\lambda = \sqrt{\eta/\mu}/C$
θ	=	Angular coordinate measured from the position of maximum film thickness in journal bearing
θ_r	=	Angular extent of film rupture for journal bearing
τ	=	Shear stress component (N m^{-2}), $\Pi = \tau C/\mu u_j$
ω	=	Angular velocity of journal bearing (rad sec^{-1})
r	=	Extent of outlet film in journal bearing measured
ϵ	=	Along the radial direction
ϕ	=	Along the tangential direction

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