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PERFORMANCE MODELING OF INFINITELY WIDE EXPONENTIALLY SHAPED SLIDER BEARING LUBRICATED WITH COUPLE STRESS FLUIDS

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ABSTRACT

In this paper, the pressure distribution and load capacity of an infinitely wide exponentially shaped slider bearing lubricated with a non Newtonian couple stress fluid is presented. Based on Stokes micro continuum theory, the effect of couple stresses on the pressure distribution and load are computed using continuous Galerkin finite element method The domain is discretized into uniform mesh of quadratic isoparametric elements and Gauss quadrature used to numerically integrate the stiffness integrals to obtain stiffness matrices for all elements which are subsequently assembled to form the global system of equations. The resulting global matrix is solved using Gauss Seidel iterative scheme. The numerical method adopted for the solution is shown to produce convergent results when implemented on a sequence of successively finer meshes. Having established the reliability of the numerical method, parametric studies are carried out to show the effects of couple stresses and aspect ratio on the pressure distribution and load. Computations put forward show that the effect of couple stresses is to enhance the load carrying capacity of the bearing. It has also been shown that compared to the Newtonian case, the pressure developed in the clearance zone of the bearing is augmented.

Keywords: slider bearing, performance modeling, couple stresses, fluid, pressure distribution, load capacity.

INTRODUCTION

In most mechanical systems where relative motion occurs between two parts, lubricants are introduced to reduce friction and wear. The geometry of the contacting elements determines the shape of the lubricant film [1]. Various researchers have considered different configurations of the lubricating film in the clearance zone in their analysis. The contacting surfaces can be narrowing geometrically in linear style as considered by Ozalp [2]. He employed the iterative transfer matrix approach to suggest optimum film profile parameters for reduced friction coefficient. Bayrakpeken *et al*., [3] carried out a comparative study of inclined and parabolic slider bearings using a non-Newtonian fluid in the clearance zone. and developed closed form expressions for the performance metrics. Shah *et al*., [4] has studied a slider bearing with exponential film thickness profile and obtained analytical expressions for variation of dimensionless pressure, friction, coefficient of friction and centre of pressure. A ferrofluid was used between the contacting surfaces of the bearing. Yurusoy [5] obtained a perturbation solution for pressure distribution in a slider bearing with a Powel-Eyring fluid as lubricant. Bujurke *et al*., [6] used a second grade fluid in a taper flat slider bearing similar to that used by Ozalp [2] and constructed a Von - korman momentum integral solution. Shah [7] computed values for the bearing characteristics for a secant shaped slider bearing using a magnetic fluid lubricant.

Different types of fluids have been used in the clearance zone of slider bearings and their performance investigated as shown in the previous works cited above. However, in order to enhance lubricating performance, the increasing use of a Newtonian lubricant which has been blended with long chain polymers has been observed. Since the conventional micro - continuum theory cannot accurately describe the flow of these kinds of fluids, various micro - continuum theories have been proposed [8]. Stokes [9] proposed the simplest micro - continuum theory which permits the presence of couple stresses, body couples and non symmetric tenors [10].

A number of researchers have investigated the effect of the couple stress fluid model on the steady state performance of different slider bearing configurations using different numerical schemes. In recent times, most numerical work in hydrodynamic lubrication has involved the use of the Reynolds equation and the finite difference method [11]. A finite difference multigrid approach was used to investigate the squeeze film behavior of poroelastic bearing with couple stress fluid as lubricant by Bujurke *et al*., [6]. They reported that poroelastic bearings with couple stress fluid as lubricant provide augmented pressure distribution and ensured significant load carrying capacity. Serangi *et al*., [12] solved the modified Reynolds equation extended to include couple stress effects in lubricants blended with polar additives using the Finite difference method with a successive over relaxation scheme. They reported increase in load carrying capacity and reduction in friction coefficient as compared to Newtonian lubricants. Lin [13] used the conjugate Method of iteration to build up the pressure generated in a finite journal bearing lubricated with a couple stress fluids. The results obtained including increase in the load carrying capacity agrees with those obtained by Serangi *et al*., [12] and Bujurke *et al*., [6]. Elsharkawy [14] provided a numerical solution for a mathematical model for hydrodynamic lubrication of misaligned journal bearings with couple stress fluids as lubricants using the Finite Difference Method. Lin [15] calculated the steady and perturbed pressures of a two dimensional plane inclined slider bearing incorporating a couple stress fluid using the conjugate gradient method and reported improved steady and dynamic performance compared to the Newtonian case especially for higher aspect ratios. Nada and Osman [16] investigated the problem of finite hydrodynamic journal bearing lubricated by magnetic fluids with couple stresses using the finite difference method. For different couple stress parameter and magnetic coefficient, they obtained the pressure distribution. They concluded that fluids with couple stresses are better compared with the Newtonian case after comparison of the bearing static characteristics. Recently, Oladeinde and Akpobi [17] studied an infinitely wide parabolic slider bearing using finite element method and showed the effect of couple stress lubricants on the bearing load.

From the cited literature it can be seen that the exponential slider bearing lubricated with couple stress fluids has not been given attention. It is this gap that this paper tries to fill. In particular, this work centers on the of continuous Galerkin finite element method to study the effect of couple stress on load capacity and pressure distribution on an infinitely wide exponentially shaped slider bearing.

MODIFIED REYNOLDS' EQUATION

The exponential slider bearing under consideration is shown in Figure-1. The lubricant in the clearance zone is taken to be a couple stress fluids. The slider bearing has a length L and moves with a velocity U as shown in Figure-1.

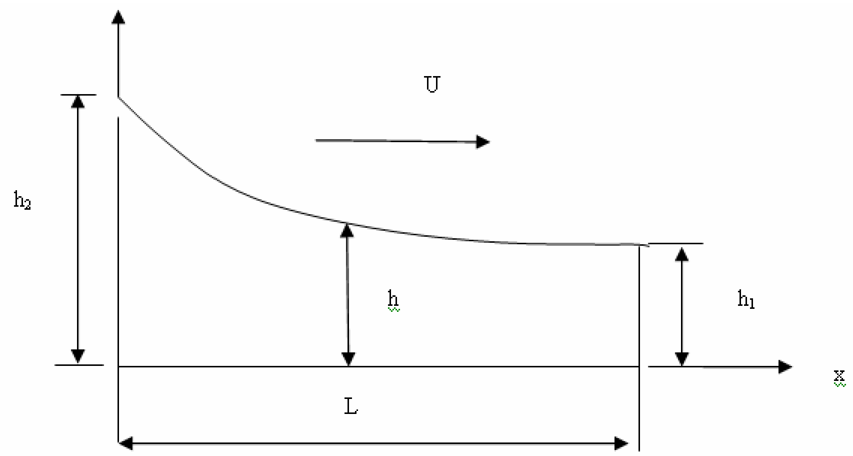

Figure-1. Physical geometry of a wide slider bearing with an exponential film profile.

The film profile is described by equation 1[18]

$$h(x) = h_2 e^{-x\ln(\frac{a}{L})} \tag{1}$$

Where L is the length of the bearing; h_2 is the film thickness at the entry of the slider bearing and a is the film thickness ratio

$$a = \frac{h_2}{h_1} \tag{2}$$

The film thickness ratio is approximated by δ+1, where δ is the profile parameter defined by equation 3. In equation 3, d is the shoulder height defined as the difference between the maximum and minimum film thickness

$$\partial = \frac{d}{h_1} \tag{3}$$

Calculations on slider bearing lubrication are frequently performed in non dimensional form (19, 20, 21). Using the non dimensional parameters in equation 4, the film thickness profile can be cast in dimensionless form as shown in equation 5

$$x^* = \frac{x}{L},\ h^* = \frac{h}{h_2} \tag{4}$$

$$h^* = e^{-x^* \ln a} \tag{5}$$

The dimensionless modified Reynolds equation governing the hydrodynamic film pressure for a slider bearing lubricated by couple stress fluid is given by equation 6 [21]

$$\frac{d}{dx^*}\left[f\left(h^*, l^*\right) \frac{dp^*}{dx^*} \right] = 6 \frac{dh^*}{dx^*} \quad 0 \le x^* \le 1 \tag{6}$$

In equation 6, l^* is the dimensionless couple stress parameter. The couple stress parameter can be obtained by some experiments as described by stokes. It can be computed by using equation 7

$$l = \left(\frac{\eta}{\mu}\right)^{\frac{1}{2}} \tag{7}$$

In equation 7, μ is the shear viscosity and η is a new material constant with the dimension of momentum and is responsible for the couple stress property. The effect of couple stress is determined through the couple

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stress parameter defined as $l^* = \frac{l}{h_o}$. If $\eta = 0$, therefore $l^* = 0$, and the classical form of the Newtonian lubricant is obtained.

The function $f\left(h^*, l^*\right)$ in equation 6 is defined by equation 8

$$f\left(h^*, l^*\right) = h^{*3} - 12l^{*2}\left[h^* - 2l^* \tanh\left(\frac{h^*}{2l^*}\right)\right] \tag{8}$$

As the value of l^* approaches zero, equation 8 is reduced to the classical form for a Newtonian lubricant case.
The boundary conditions are a specification of the pressure at the ends of the bearings. The boundary conditions are given in equation 9

$$p^*(x^* = 0) = 0 \quad p^*(x^* = 1) = 0 \tag{9}$$

WEAK FORMULATION

In order to obtain the pressure distribution on the bearing using the finite element method, we first obtain the residual of the governing equation by taking all terms on the right hand side to the left hand side to obtain equation 10 below. A Galerkin formulation was utilized in order to apply the finite element method

$$R(x, p) = \frac{d}{dx}\left[f(h, l)\frac{dp}{dx}\right] - 6\frac{dh}{dx} \tag{10}$$

Multiplying equation 10 by a weight function w_i and integrating over a typical element with end nodes x_1 and x_3, we obtain

$$\int_{x_1}^{x_3} w_i\left[\frac{d}{dx}\left(f(h, l^*)\frac{dp}{dx}\right) - 6\frac{dh}{dx}\right]dx = 0 \tag{10}$$

Integrating the first term of equation 10, we obtain equation 11

$$\int_{x_1}^{x_3} w_i\left[\frac{d}{dx}\left(f\left(h, l^*\right)\frac{dp}{dx}\right)\right]dx = \int_{x_1}^{x_3}\frac{dw_i}{dx} f\left(h, l^*\right)\frac{dp}{dx} + \left[w_i f\left(h, l^*\right)\frac{dp}{dx}\right]_{x_1}^{x_3} \tag{11}$$

$$i = 1, 2, \ldots\ldots n$$

Equation 10 now becomes

$$\int_{x_1}^{x_3}\frac{dw_i}{dx} f\left(h, l^*\right)\frac{dp}{dx} + \left[wf\left(h, l^*\right)\frac{dp}{dx}\right]_{x_1}^{x_3} - \int_{x_1}^{x_3} 6\frac{dh}{dx}dx \tag{12}$$

Now we assume a trial solution for the nodal degree of freedom of the form of equation 13

$$p = \sum_{j=1}^{n} p_j \varphi_j \tag{13}$$

Obtaining the first derivative of equation 13 and substituting into equation12 with the weight functions set identical to the trial functions, we obtain the Galerkin finite element model for the parabolic slider problem shown in equation 14. The integration is over a typical element.

$$\sum_{j=1}^{n}\left[\int_e \frac{d\varphi_j^e}{dx} f\left(h, l^*\right)\frac{d\varphi_i^e}{dx}\right]p_j + \left[f\left(h, l^*\right)\frac{dp}{dx}\varphi_i^e\right]^e - \int_e 6\frac{dh}{dx}\varphi_i^e dx = 0 \tag{14}$$

NUMERICAL RESULTS AND DISCUSSIONS

Validation of results

The finite element results are only approximate in nature and in using the results to predict the load capacity, it is essential that the reliability of the results is first examined. In numerical analysis, in particular grid point methods, the finite element solution eventually converges to the exact solution as the mesh is refined progressively. Consequently, the numerical model is first examined for its convergence characteristic using meshes of 4, 8 and 10 quadratic isoparametric elements for $l^* = 0.1$ and $a = 0.5$ Numerical experimentation shows that the dimensionless pressure at the middle of the bearing is 0.0816 for a mesh of 4 quadratic elements, 0.0822 for a mesh of 8 isoparametric quadratic elements and 0.0822 for a mesh of 10 isoparametric quadratic elements. The finite element model applied clearly exhibits convergence behavior. Increasing the mesh density more than 10 quadratic elements only increases the computational time and no appreciable effect on the accuracy of the solution. Hence a mesh density of 10 isoparametric elements was used for the parametric studies.

Pressure

The variation of dimensionless pressure distribution with dimensionless distance along the bearing is shown in Figure-2 for a Newtonian case where the couple stress parameter is set equal to zero. It can be deduced from the Figure that the effect of increase in profile parameter for an exponentially shaped slider lubricated with a Newtonian lubricant is to augment the pressure distribution in the bearing. This is attributable to the increase in wedge effect on the bearing as the profile parameter increases. Compared to a parabolic slider bearing of a similar profile parameter, the exponential slider bearing produces a smaller pressure build up in the lubricating film. However, the exponential slider bearing exhibits a similar trend in pressure build up with increase in profile parameter as reported by Oladeinde and Akpobi

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(2009). Figure-3 shows the variation of dimensionless pressure with distance for an exponential slider bearing using a non Newtonian couple stress fluid with couple stress parameter equal 0.1, 0.2 and 0.3 respectively.

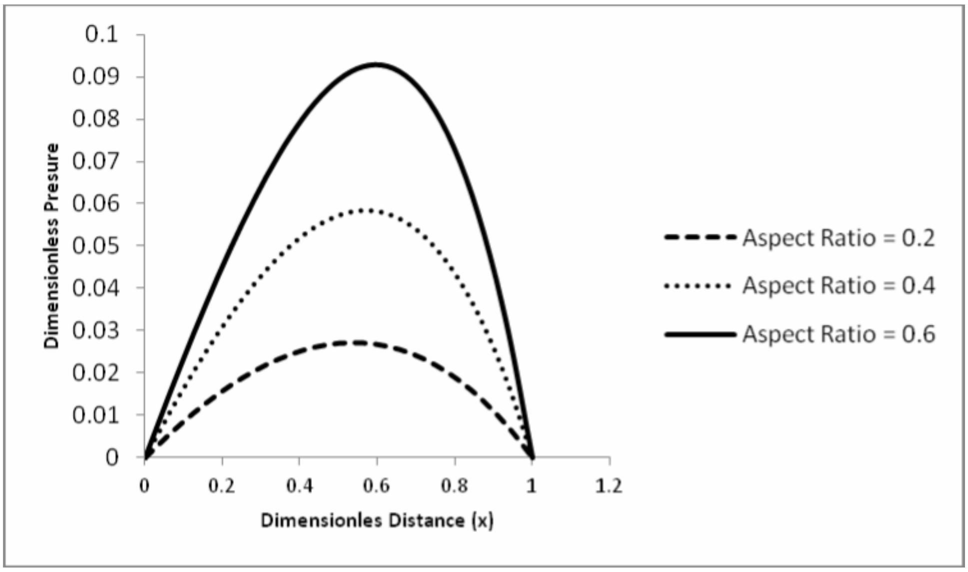

Figure-2. Variation of dimensionless pressure with distance along the bearing for different aspect ratios.

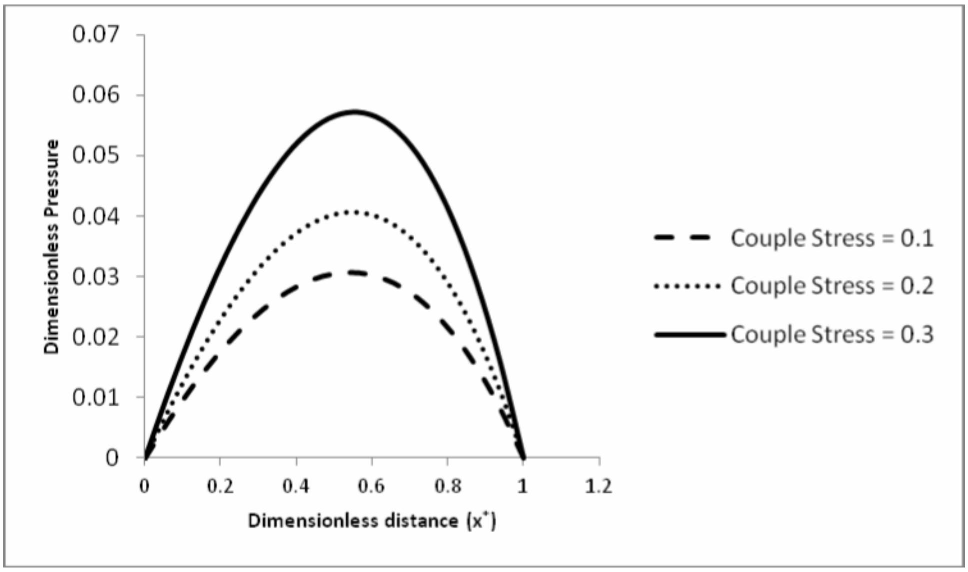

Figure-3. Variation of dimensionless pressure with distance along exponential slider bearing with different couple stress parameters.

The plot shows that the pressure in the clearance zone of the bearing increases with increase in couple stress parameter. This finding is consistent with that obtained by Oladeinde and Akpobi (2009) for a parabolic slider bearing. With increase in couple stress parameter from 0.1 to 0.3, the maximum pressure in the bearing increases by a factor approximately equal to 2. However, in contrast with the results obtained by Oladeinde and Akpobi (2009) for a parabolic slider bearing, the position of the maximum pressure is not influenced by introduction of polar additives in the Newtonian lubricant accounted for by the couple stress parameter for a given profile parameter. Computations show that with Increase in the profile parameter, the position of the maximum pressure moves towards the exit of the bearing for different couple stress parameter.

Load capacity

The effect of profile parameter on load capacity for different couple stress parameters is shown in Figure-4.

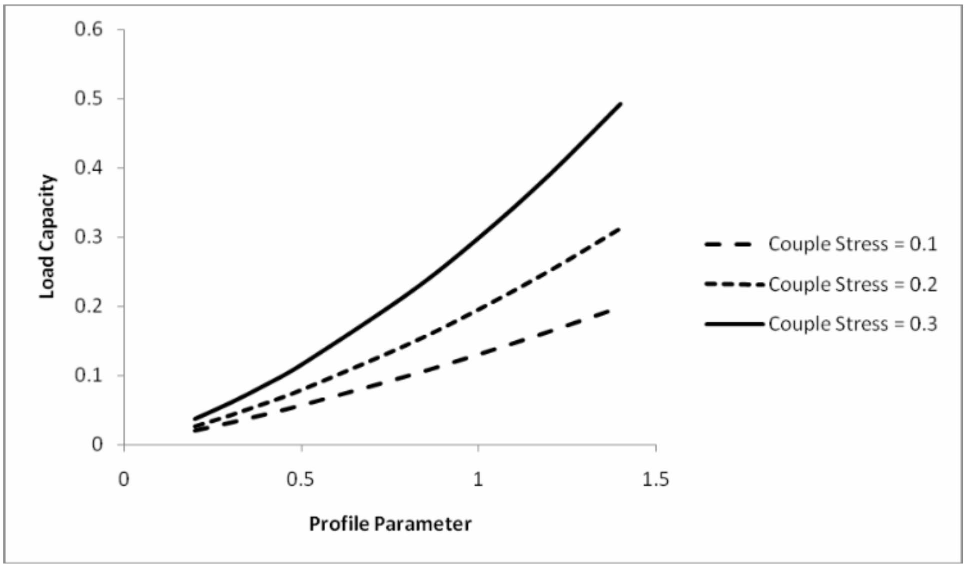

Figure-4. Variation of dimensionless load capacity with profile parameter for different couple stress parameters.

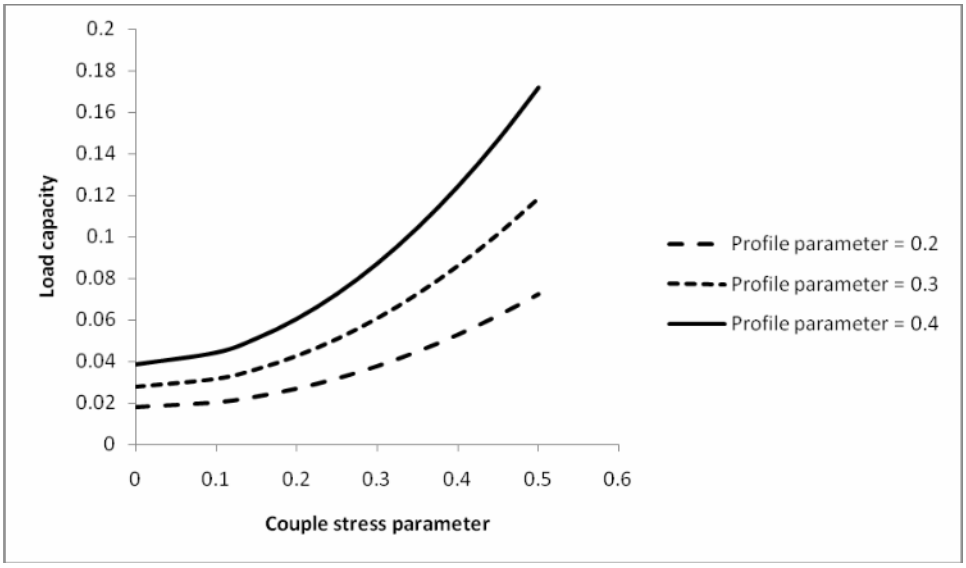

Figure-5. Variation of dimensionless load capacity with couple stress parameter for different profile parameters.

Figure-4 shows the dimensionless load capacity as a function of profile parameter for different couple stress parameter. Since the effect of couple stress provide an increase in the oil film pressure, the load carrying capacity is similarly influenced. In general, with increase in non Newtonian behavior, the load carrying of the bearing is increased. The plot shows that the increase in load capacity with couple stresses is greater at higher profile parameters. Compared to the parabolic slider case which exhibits optimum profile parameter after which the benefit derived from the introduction of polar additives decreases, the benefit derived from the addition of polar additives increases with increase in profile parameter for an exponentially shaped slider bearing. The plot also shows that for coupe stress greater than 0.3, the variation of dimensionless load capacity with profile parameter becomes linear in nature. Figure-5 shows the results of the simulation of load capacity against couple stress parameter for different values of profile parameter. As illustrated in the graph, a higher profile parameter for an exponential slider bearing lubricated with a Newtonian fluid brings about a higher load carrying capacity. The plot shows that the bearing load increases with couple stress parameter for a given profile parameter. The improvement in load capacity is due to the increase in the pressure generated in the lubricant film with increase in profile parameter due to wedge effect.

CONCLUSIONS

The effect of couple stresses and profile parameter on the bearing load and pressure profile of an infinitely wide exponential slider bearing has been considered using the finite element method. The effect of structural and lubricant rheological property (couple stress) on load capacity and pressure distribution has been presented. The present study provides Engineers with useful information in bearing design.

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