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Parametric Characterization of Load Capacity of Infinitely Wide Parabolic Slider Bearing with Couple Stress Fluids

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***Abstract*—**A mathematical model for the hydrodynamic lubrication of parabolic slider bearings with couple stress lubricants is presented. A numerical solution for the mathematical model using finite element scheme is obtained using three nodes isoparametric quadratic elements. Stiffness integrals obtained from the weak form of the governing equations were solved using Gauss Quadrature to obtain a finite number of stiffness matrices. The global system of equations was obtained for the bearing and solved using Gauss Seidel iterative scheme. The converged pressure solution was used to obtain the load capacity of the bearing. Parametric studies were carried out and it was shown that the effect of couple stresses and profile parameter are to increase the load carrying capacity of the parabolic slider bearing. Numerical experiments reveal that the magnitude of the profile parameter at which maximum load is obtained increases with decrease in couple stress parameter. The results are presented in graphical form.

***Keywords*—**Finite element, numerical, parabolic slider.

I. 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 [2]. He employed the iterative transfer matrix approach to suggest optimum film profile parameters for reduced friction coefficient. [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. [4] has studied a slider bearing with exponential film thickness profile and obtained analytically 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. [5] obtained a perturbation solution for pressure distribution in a slider bearing with a Powel-Eyring fluid as lubricant. [6] used a second grade fluid in a taper flat slider bearing similar to that used by [2] and constructed a Von – korman momentum integral solution. [7] computed values for the bearing characteristics for a secant shaped slider bearing using a magnetic fluid lubricant.

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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 micros – continuum theories have been proposed [8]. [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 [6]. They reported that poroelastic bearings with couple stress fluid as lubricant provide augmented pressure distribution and ensured significant load carrying capacity. [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. [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 [12] and [6]. [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. [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 [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.

The open literature is replete with slider bearing design with couple stress fluids as lubricants using finite difference method as the numerical tool for analysis as can be deduced from the literature cited above. Previous researchers seem not to have exploited the applicability of finite element methods in slider bearing design. The finite element methods are probably the most accurate and versatile, but tend to be very time consuming and require high knowledge, not assessable to the common designer [17], hence its obvious absence in the perused literature. It is this gap that the present paper seeks to fill. In particular, this work centers on the use of continuous Galerkin finite element method for parametric characterization of the load capacity of parabolic slider bearings lubricated with couple stress fluids.

II. Modified Reynolds' Equation

The parabolic slider bearing under consideration is shown in Fig. 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 Fig. 1.

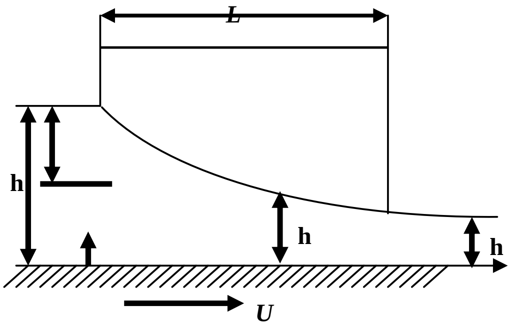

Fig. 1 Bearing geometry of a parabolic shaped slider

The film profile is described by the expression below:

$$h(x) = h_m + h_p(x) = h_m + \frac{d}{L}\left(L - 2x + \frac{x^2}{L}\right) \tag{1}$$

Where h_m is the minimum film thickness at the exit of the slider and d represents the shoulder height of the bearing. The oil film profile can be non dimensionalised by introducing some non dimensional quantities and we obtain:

$$h^* = h_m^* + h_p^* = h_m^* + \partial(1 - 2x^* + x^{*2}) \tag{2}$$

Where $h^* = \frac{h}{h_0}$, $h_m^* = \frac{h_m}{h_0}$, $h_p^* = \frac{h_p}{h_0}$ and ∂ stands for the bearing profile parameter.

$$\partial = \frac{d}{h_o}$$

The continuity and momentum equation can be written in non dimensional form as in equations 3 and 4 respectively.

$$\frac{\partial u^*}{\partial x^*} + \frac{\partial w^*}{\partial z^*} = 0 \tag{3}$$

$$\frac{\partial p^*}{\partial x^*} = \frac{\partial^2 u^*}{\partial z^{*2}} - l^* \frac{\partial^4 u^*}{\partial z^{*4}} \tag{4}$$

$$\frac{\partial p}{\partial z} = 0 \tag{5}$$

Calculations on slider bearing lubrication are frequently performed in non dimensional form [18, 19, 20]. We define the following non dimensional parameters.

$$x^* = \frac{x}{L},\ z^* = \frac{z}{h_0},\quad u^* = \frac{u}{U},\quad w^* = \frac{l}{Uh_0} w \tag{6}$$

$$P^* = \frac{p h_o^{\,2}}{\mu UL},\quad l^* = \frac{l}{h_o} = \frac{(\eta/\mu)^{\frac{1}{2}}}{h_o} \tag{7}$$

In these equations u^* and w^* represents the non dimensional velocity components in x and z directions respectively. p^* is the non dimensional pressure. 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. Its value can be determined by some experiments as discussed by Stokes. The dimension of $l = \left(\frac{\eta}{\mu}\right)^{\frac{1}{2}}$ is of length. The length could be identified as the characteristic material length or the molecular length of the polar suspensions in a non polar fluid. The effects of couple stress are therefore dominated through the dimensionless couple stress parameter $l^* = \frac{l}{h_o}$. If $\eta = 0$, therefore $l^* = 0$,and the classical form of the Newtonian lubricant is obtained. The boundary conditions are the no slip conditions and the non couple stress conditions.

The non dimensional modified Reynolds equation governing the hydrodynamic film pressure is given by:

$$\frac{d}{dx^*}\left[f\left(h^*, l^*\right) \frac{dp^*}{dx^*} \right] = 6\frac{dh^*}{dx^*} \tag{8}$$

Where the function is $f\left(h^*, l^*\right)$ defined by:

$$f\left(h^*, l^*\right) = h^{*3} - 12 l^{*2}\left[h^* - 2l^* \tanh\left(\frac{h^*}{2l^*}\right)\right] \tag{9}$$

As the value of l^* approaches zero, equation 9 is reduced to the classical form for a Newtonian lubricant case.

III. Weak Formulation

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 [21].

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

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:

$$\int_{x_1}^{x_3} w_i\left[\frac{d}{dx}\left(f(h,l^*)\frac{dp}{dx}\right)-6\frac{dh}{dx}\right]dx=0 \tag{11}$$

Integrating the first term of equation 11, we obtain equation below.

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

$$i=1,2,......n \tag{12}$$

Equation 11 now becomes:

$$\int_{x_1}^{x_3}\frac{dw_i}{dx}f\left(h,l^*\right)\frac{dp}{dx}+\left[wf\left(h,l^*\right)\frac{dp}{dx}\right]_{x_1}^{x_3}-\int_{x_1}^{x_3}6\frac{dh}{dx}dx \tag{13}$$

Now we assume a trial solution for the nodal degree of freedom of the form of equation 8 below.

$$p=\sum_{j=1}^{n}p_j\varphi_j \tag{14}$$

Obtaining the first derivative of equation 14 and substituting into equation13 with the weight functions set identical to the trial functions, we obtain the Galerkin finite element model for the parabolic slider problem equation 15. The integration is over a typical element as shown in the equation.

$$\sum_{j=1}^{n}\left[\int_e\frac{d\varphi_j^e}{dx}f\left(h,l^*\right)\frac{d\varphi_i^e}{dx}\right]p_j+\left[f\left(h,l^*\right)\frac{dp}{dx}\varphi_i^e\right]^e$$
$$-\int_e 6\frac{dh}{dx}\varphi_i^e dx=0 \tag{15}$$

IV. Boundary Conditions

The boundary conditions needed to solve equation 15 is the specification of the pressure at the end of the bearing. The pressure at the ends of the bearing is to 0.

V. Numerical Results and Discussion

Microsoft Visual Basic .Net software developed by the authors was used to carry out the finite element simulation of the slider bearing under consideration. It is necessary to set base parameters that will be used for the simulation. The most commonly used couple stress parameter in literature is in the range 0 – 0.5. The effect of selected couple stress parameters in this range on the pressure distribution profile is presented in what follows. Base parameters used for this analysis include a uniform mesh of 50 uniform C^0 quadratic isoparametric elements, minimum film thickness parameter (h_m^*) equal 1.0, three gauss points for computation of element stiffness integrals and profile parameter (δ) equal 1.4 representing the middle of the range of the most frequently used values in literature. Graphical representation of the results obtained for increasing non Newtonian effects from $l^*=0-0.5$ is shown below.

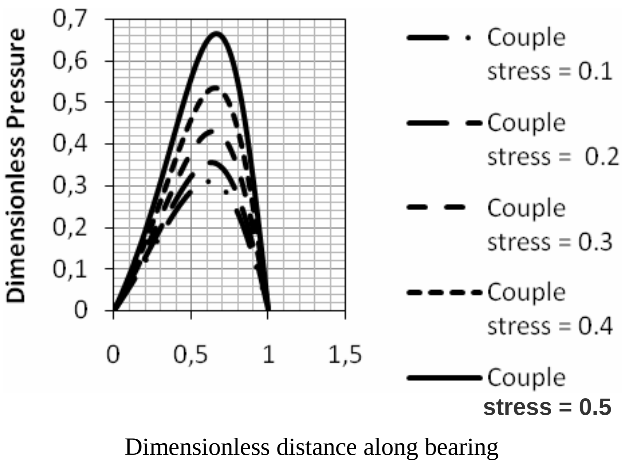

Fig. 2 Dimensionless pressure against dimensionless distance

A plot of dimensionless pressure for different values of couple stress parameters is shown in Fig. 2. The plot shows that increase in couple stress indicating increasing non Newtonian effect augments the pressure distribution in the lubricant film contained in the clearance zone. In particular, it can be observed that the peak pressure increases with increase in couple stress parameter. It is concluded that couple stresses improves the bearing performance as more pressure generation increases the load carrying capacity due to increased wedge effect.

The profile parameter determines the shape and thickness of the film between the journal and the slider. The profile parameter influences the pressure distribution along the bearing. The effect of this parameter will be investigated numerically using finite element simulation via the instrumentality of the developed software. Using the parameters $l^*=0.25, h_m^*=1$, three gauss points and a uniform mesh of 50 C^0 quadratic isoparametric elements, we vary the profile parameter within the range $\delta=0.2-2.8$ in steps of 0.2 units.

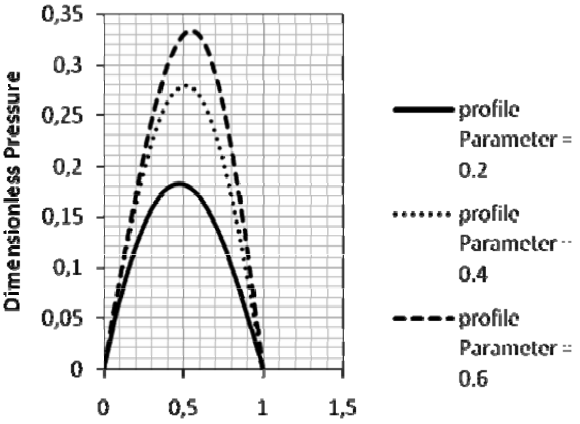

Fig. 3 Dimensionless Pressure vs dimenionles distance

Fig. 3 shows the results for the parametric analysis of the variation of pressure distribution for different values of profile parameter for given couple stress parameters. The graph shows that the pressure generated in the film of the lubricant increases with increase in profile parameter. This implies improvement of bearing performance from the stand point of load.

VI. Effect of Profile Parameter on Load Capacity

The effect of increase in profile parameter on the load carrying capacity of the bearing is shown in the Fig. 4.

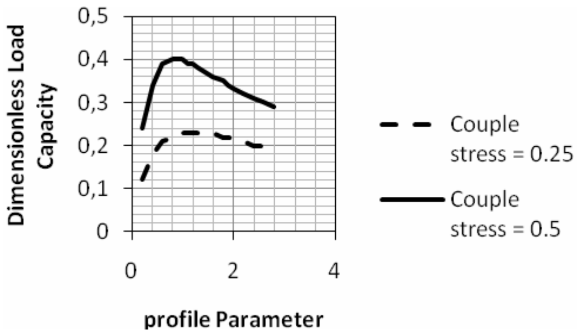

Fig. 4 Dimensionless load vs profile parameter

Fig. 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 capacity of the bearing is increased. However, for given couples stress parameter the load capacity first increases with profile parameter, attains a maximum and then decreases with further increase in profile parameter. The profile parameter at which maximum load is obtained increases with decrease in couple stress parameter. A couple stress parameter of 0.5 is observed to produce dimensionless bearing load of 0.4 at a profile parameter of 0.8 whereas a couple stress parameter of 0.25 produces a maximum bearing load of 0.23 at a profile parameter of 1.1. The optimum profile parameter for a given couple stress parameter is important from the stand point of design of parabolic slider bearing. Knowledge of this can aid design engineers in specifying optimum structural dimensions for bearings during design.

VII. Effect of Couple Stress Parameter on Load

The influence of couple stress parameter on load carrying capacity is investigated by simulating the bearing design software for different values of couple stress parameter for a fixed profile parameter. The results for the simulation for a range of couple stress parameter for profile parameter $\delta = 1.0$ and $\delta = 1.5$ using a mesh of 40 uniform C^0 isoparametric quadratic elements is shown in Fig. 5. The minimum film thickness ratio is set at unity.

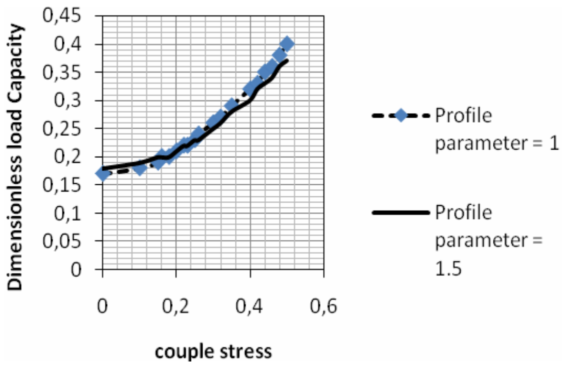

Fig. 5 Dimensionless load capacity vs couple stress

Fig. 5 above 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 a parabolic bearing lubricated with a Newtonian fluid brings about a higher load carrying capacity of the bearing. With increase in the profile parameter, the load capacity of the bearing increases slightly with increasing non Newtonian behavior irrespective of the profile parameter within the range $l^* = 0.22 - 0.25$, there is no noticeable change in the bearing performance (Load consideration). 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.

VIII. Conclusion

The effect of couple stresses and profile parameter upon the bearing load and pressure profile of an infinitely wide parabolic slider bearing has been considered using the finite element method. The modified Reynolds equation which is derived from Navier Stokes equations that accounts for the effect of couple stresses arising from blending a Newtonian lubricant with various additives. The effect of structural and lubricant rheological properties on load capacity has been presented. An optimum parameter value for maximum load capacity has also been revealed. The present study provides Engineers with useful information in bearing design.

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