A Comparative study of Load Capacity and Pressure Distribution of Infinitely wide Parabolic and Inclined Slider Bearings

Mobolaji H. Oladeinde and John A. Akpobi

***Abstract*—** A mathematical model for the hydrodynamic lubrication of infinitely wide inclined and 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 for both configurations of bearings. 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 bearings and solved using gauss seidel iterative scheme with a convergence criterion on 10^{-10}. Numerical experiments indicate that when the slider bearing is modeled to include couple stress parameter, the finite element method produces stable and convergent result while the finite difference technique does not. Computations reveal that when compared using similar profile and couple stress parameter, greater pressure builds up in a parabolic slider compared to an inclined slider indicating greater wedge effect in the parabolic slider. It has also been shown that when load carrying capacity is the yardstick for comparison, the parabolic slider bearings is superior to the inclined case irrespective of the profile and couple stress parameter.

Index Terms — Hydrodynamic lubrication, parabolic slider, inclined slider, finite element

1. 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 of the slider bearings. He developed close form expressions for the performance characteristics of the bearings. Shah et al [4] 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 of 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. However, in order to enhance lubricating performance, the increasing use of 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. .Sarangi 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

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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 agree with those obtained by Sarangi et al [12] and Bujurke et al [6]. Elsharkawy [14] provided a numerical solution for a mathematical model describing the hydrodynamic lubrication of misaligned journal bearings with couple stress fluids as lubricants using the finite difference method. Lin [15] calculated the steady and perturbed pressure 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 parameters and magnetic coefficients, 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. Previous researchers seem not to have exploited the applicability of finite element methods in slider bearing design. The finite element method is probably the most accurate and versatile, but tends to be very time consuming and requires high knowledge, not assessable to the common designer [17], hence it's 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 carrying out a comparative study of pressure distribution and bearing load of infinitely wide parabolic and inclined slider bearings lubricated with couple stress fluids.

II. MODIFIED REYNOLDS' EQUATION

The geometry of parabolic and inclined slider bearings under consideration are shown in figs. 1 and 2 respectively. 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 the figs. 1 and 2

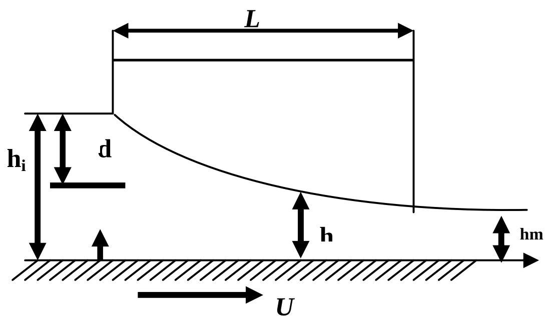

Fig. 1: Bearing geometry of a parabolic shaped slider

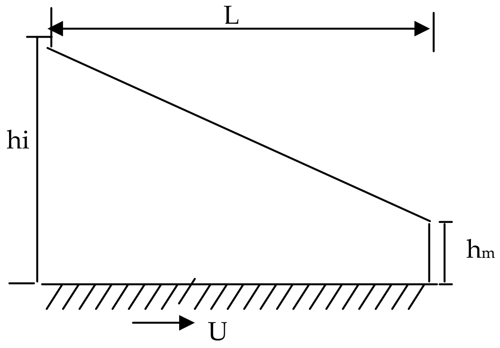

Fig. 2: Bearing Geometry of inclined shaped slider

The oil film profile for the parabolic slider in non dimensionalised form is as shown in (1). The corresponding film profile equation for the inclined slider bearing is shown in (2)

$$h = h_m + h_p = h_m + \partial(1 - 2x + x^2) \qquad (1)$$

$$h = h_m + \delta\left(1 - x\right) \qquad (2)$$

Where h_m is the minimum film thickness at the exit of the slider and δ represents the profile parameter of the bearing.
The continuity and momentum equation for a slider bearing can be written in non dimensional form as in (3) and (4) respectively.

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

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

$$\frac{\partial p}{\partial z} = 0 \qquad (5)$$

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

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

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

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

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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

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

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

111. 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 eq. 10. A Galerkin formulation was utilized in order to apply the finite element method [21].

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

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

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

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

$$\int_{x_1}^{x_3} w_i\left[\frac{d}{dx}\left(f\left(h^*,l^*\right)\frac{dp^*}{dx}\right)\right]dx \tag{12}$$

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

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

Eq. 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 (8).

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

Obtaining the first derivative of (14) and substituting into (13) with the weight functions set identical to the trial functions, we obtain the Galerkin finite element model for the parabolic slider problem shown in (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 (15) are the specification of the pressure at the end of the bearing. The pressures at the ends of the bearing are set to 0.

V. SOLUTION METHODOLOGY

The dimensionless physical domain $x = [0,1]$ is divided into uniform quadratic elements of length Δx. This results to a constant transformation to a local element co ordinate system in which the differential dx and d/dx are given by

$$dx = \Delta x d\xi \tag{16}$$

$$\frac{d}{dx} = \frac{1}{\Delta x}\frac{d}{d\xi} \tag{17}$$

Where ξ is the natural co-ordinate system in an element. The pressure in an element is approximated by basis or trial functions ϕ_I over each element and is given by (18)

$$p(\xi) = \sum_{j=1}^{3} p_j\varphi_j(\xi) \tag{18}$$

The transformed element integrals obtained from the weak formulation of the governing equation shown in (15) are evaluated numerically by Gauss Quadrature by making use of the transformed integral shown in (19)

$$\left[K^e\right] = \int_{-1}^{1}\frac{1}{J^e(\xi)}\frac{d\varphi(\xi)}{d\xi}\left(f\left(x^e,l(\xi)\right)\right)^3\frac{1}{J^e(\xi)}\frac{d\varphi(\xi)}{d\xi}J^e(\xi)d(\xi) \tag{19}$$

$J^e(\xi)$ is the Jacobian and is defined by the expression below

$$J^e\left(\xi\right)=\frac{dx}{d\xi}=\sum_{i=1}^{3}x_i^e\frac{d\varphi_i\left(\xi\right)}{d\xi}$$

The element stiffness integrals are evaluated iteratively using three gauss points and weights according to the expression in (20)

$$\int_{-1}^{+1}I\left(\xi\right)d\xi\simeq\sum_{l=1}^{n}w_{nl}I\left(\xi_{nl}\right) \qquad (20)$$

After writing the global integral as a sum of the individual element stiffness integrals, a system of algebraic equations is obtained. Boundary conditions are imposed on the global system of equations resulting in a condensed matrix which is solved by gauss seidel iterative scheme to obtain the pressure solution. Parametric studies are carried out to determine the effect of bearing parameters on the pressure distribution and bearing load

VI. NUMERICAL RESULTS AND DISCUSSION

In the following section, the results of the finite element based simulation of the slider bearing configurations under consideration are presented. The validity of the results of the finite element simulation is examined. In particular, we investigate the convergence characteristics of the results obtained using the finite element method and compare same with those obtained using the finite difference method.

VII. VALIDATION OF RESULTS

TABLE I

NODAL PRESSURES OBTAINED USING FINITE ELEMENT METHOD FOR INFINITELY WIDE INCLINED SLIDER BEARING WITH COUPLE STRESS PARAMETER

Node position	10 elements	20 elements	40 elements	80 elements
0	0.000000	0.0000000	0.00000	0.000000
0.05	.0239242	.0239243	.0239243	.0239243
0.1	.0462179	.0462179	.0462179	.0462179
0.15	.0667703	.0667705	.0667705	.0667705
0.20	.0854649	.0854649	.0854649	.0854649
0.25	.1021762	.1021764	.1021764	.1021764
0.30	.1167726	.1167726	.1167726	.1167726
0.35	.1291120	.1291122	.1291122	.1291122
0.40	.1390451	.1390451	.1390451	.1390451
0.45	.1464109	.1464112	.1464112	.1464112
0.50	.1510400	.1510400	.1510400	.1510400
0.55	.1527494	.1527497	.1527497	.1527497
0.60	.1513464	.1513465	.1513465	.1513465
0.65	.1466229	.1466233	.1466233	.1466233
0.70	.1383594	.1383594	.1383594	.1383594
0.75	.1263183	.1263188	.1263188	.1263188
0.80	.1102493	.1102493	.1102493	.1102493
0.85	.0898809	.0898815	.0898815	.0898815
0.90	.0649268	.0649268	.0649268	.0649268
0.95	.0350759	.0350765	.0350765	.0350765
1.0	0.000000	0.000000	0.000000	0.000000

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TABLE II

NODAL PRESSURES OBTAINED USING FINITE DIFFERENCE METHOD FOR INFINITELY WIDE INCLINED SLIDER BEARING WITH COUPLE STRESS PARAMETER

Node position	10 elements	20 elements	40 elements	80 elements
0	0.00000	0.00000	0.00000	0.00000
0.1	.0480833	.0477022	.0509494	.0526036
0.2	.0893103	.0949955	.0979870	.0995095
0.3	.1284146	.1335880	.1363043	.1376854
0.4	.1574209	.1620400	.1644595	.1656884
0.5	.1746474	.1786640	.1807626	.1818271
0.6	.1781126	.1814722	.1832225	.1841091
0.7	.1654700	.1681102	.1694812	.1701747
0.8	.1339235	.1357722	.1367288	.1372119
0.9	.0801187	.0810921	.0815938	.0818467
1	0.00000	0.00000	0.00000	0.00000

The results presented in table I show that the convergence rate of the solution varies spatially along the bearing length. There are points which exhibit fast convergence rate whereas other converges less quickly. For a mesh of ten (10) quadratic elements, it can be seen that the finite element solution converges at points 0.20. 0.30, 0.40 and 0.70. Since these points converge quickly, we can apply adaptive mesh refinement to the other areas to accelerate the convergence rate. For a mesh of 80 elements, the solution converges for all the points considered. From the standpoint of cost of computation, it is uneconomical to simulate the bearing with greater number of elements. The global convergence of the finite element method demonstrates that the method can be used to correctly simulate the hydrodynamic scheme of inclined slider bearing when the lubricant is a couple stress fluid.

Table II shows the results obtained using the finite difference method for an infinitely wide inclined slider bearing with couple stress fluid. It can be seen that there is no evidence of convergence of the solution at all points even with a mesh density of 80 elements. The solution obtained point wise appears to be increasing without bound. The reason for this is the non linearity introduced by the couple stress parameter into the governing equation. The finite difference method is therefore unsuitable for simulating this class of bearings with couple stress fluids.

The behaviour of the finite element and finite difference results for parabolic slider bearing configurations have also been examined and a conclusion similar to the inclined case has been obtained. In order to reveal the cause of the instability of the finite difference scheme, the simulation was carried out without the use of a couple stress parameter. This is equivalent to modeling the bearing with Newtonian lubricants. The results shown in table III were obtained for an inclined slider bearing.

TABLE III

NODAL PRESSURES OBTAINED USING FINITE DIFFERENCE METHOD FOR INFINITELY WIDE INCLINED SLIDER BEARING WITHOUT COUPLE STRESS PARAMETER

Node position	10 elements	20 elements	40 elements	80 elements
0	0.000000	0.000000	0.000000	0.000000
0.1	.0352482	.0352543	.0352558	.0352562
0.2	.0648426	.0648540	.0648569	.0648576
0.3	.0881179	.0881337	.0881377	.0881387
0.4	.1043344	.1043536	.1043583	.1043595
0.5	.1126691	.1126902	.1126955	.1126968
0.6	.1122049	.1122263	.1122317	.1122330
0.7	.1019186	.1019384	.1019434	.1019446
0.8	0806671	.0806832	.0806872	.0806882
0.9	.0471718	.0471814	.0471838	.0471844
1	0.000000	0.000000	0.000000	0.000000

Table III shows that the finite difference method converges when the inclined slider bearing is modeled without the use of couple stress parameter. It is concluded that in the present case where the bearings are modeled to include couple stress parameter, the finite difference method is unsuitable.

VIII. PARAMETRIC STUDIES

To provide information for engineers involved with slider bearing design, comparison of the performance of the two bearing profiles is made. First, comparison of the pressure distribution under the same bearing parameters is investigated through numerical experiments. Table IV shows the dimensionless pressure obtained for the parabolic and inclined slider bearings respectively for $\delta = 1$,

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$l^* = 0.2$ minimum film thickness =1, and a uniform mesh of 10 elements

Fig 3 shows the dimensionless pressure generated under the same conditions for inclined and parabolic slider bearings with couple stress. The graph shows that a lower maximum pressure is generated using the same bearing parameters for an inclined slider bearing with couple stress compared to a parabolic bearing. The maximum dimensionless pressure generated for parabolic slider bearing is 0.34 compared to 0.31 for the inclined slider bearing. Under the magnitude of the parameters considered, the parabolic slider bearing generates a load capacity of 0.21 compared to 0.19 for the inclined slider bearing. For dimensionless distance up to 0.8 the pressure generated in the parabolic slider bearing is greater than that in the inclined slider bearing, but downstream from 0.8 to the end of the bearing, the pressure in the latter is greater. To observe the performance of the two bearings without couple stress, fig. 4 is presented. Fig. 4 shows the dimensionless pressure distribution of the two bearings without couple stress for $\delta = 1$. The graph shows that the parabolic slider bearing retains its higher maximum pressure as in the non Newtonian lubricant case. However, the effect of simulating the two bearings without couple stress is to decrease the pressure generated from 0.34 to 0.28 and from 0.31 to 0.25 for the parabolic and inclined slider bearings respectively. Increasing the profile parameter from 1(fig. 4) to 1.4(fig. 5) results in a decrease in the maximum pressure generated in the parabolic slider in contrast to the inclined case where it has a positive effect. Increasing the profile parameter is the same as increasing the wedge effect which ultimately results in greater pressure in the lubricating film of the inclined slider bearing. The net result of simulating the bearings without couple stress is to decrease the load carrying capacity as shown in fig. 4

Fig. 4 shows that the load carrying capacity for a parabolic slider bearing is higher than that for the inclined case for the same couple stress parameter. Parabolic slider bearings are therefore superior at higher couple stress parameters than at lower couple stress regimes

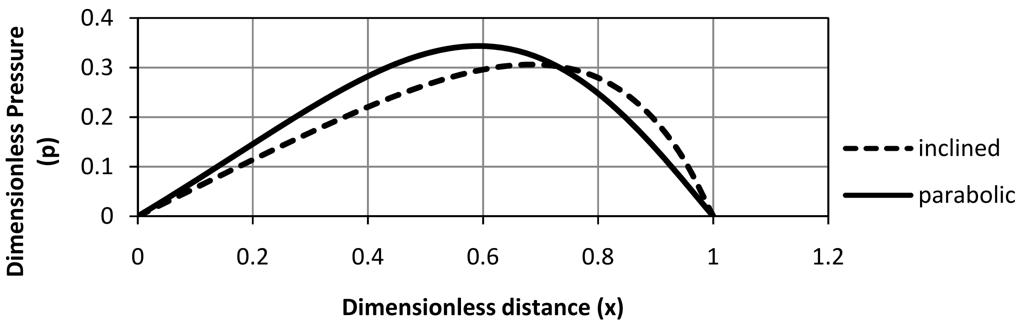

Fig. 3: Dimensionless pressures for infinitely wide parabolic and inclined slider bearing with couple stress

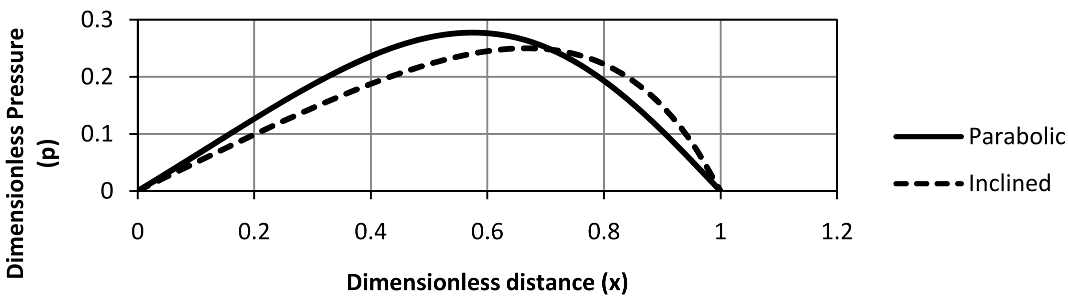

Fig. 4: Dimensionless pressure against dimensionless distance for inclined and parabolic slider without couple stress. $\delta = 1$

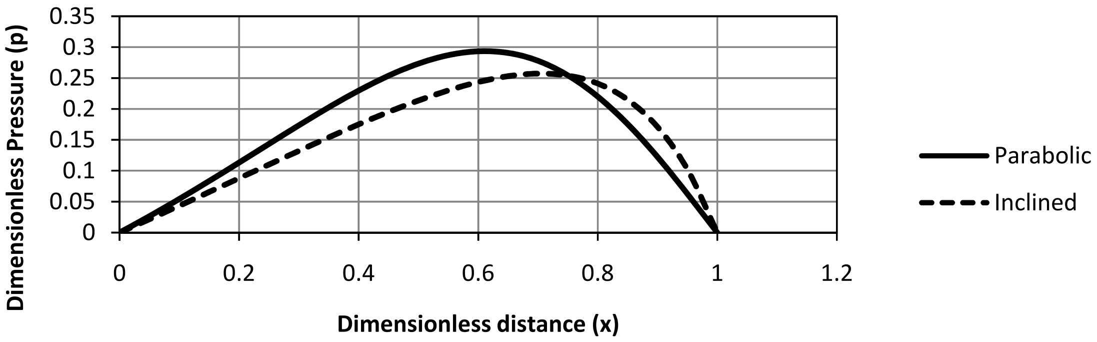

Fig. 5: Dimensionless pressure against dimensionless distance for infinitely wide inclined and parabolic slider bearing. Without couple stress ($\delta = 1.4$)

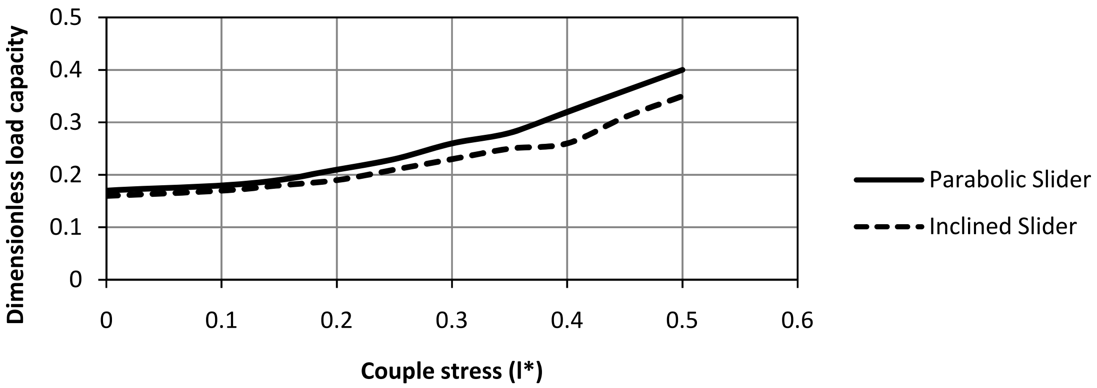

Fig. 6: Dimensionless Load capacity against dimensionless couple stress for parabolic and inclined slider bearings

TABLE IV

DIMENSIONLESS PRESSURE AT SELECTED POINTS ALONG THE BEARING FOR PARABOLIC AND INCLINED SLIDER BEARINGS

Distance(x^*)	Pressure(Inclined)	Pressure(Parabolic)
0	0	0
0.1	0.0285	0.0352
0.15	0.057	0.0715
0.2	0.0856	0.1086
0.25	0.1139	0.1459
0.3	0.1691	0.1828
0.35	0.1955	0.2186
0.4	0.2206	0.2522
0.45	0.2438	0.2826
0.5	0.2647	0.3083
0.55	0.2824	0.3281
0.6	0.2961	0.3403
0.65	0.3044	0.3436
0.7	0.3058	0.3367
0.75	0.2983	0.3187
0.8	0.2794	0.289
0.85	0.2454	0.248
0.9	0.1920	0.1965
0.95	0.1129	0.1363
1.0	0	0

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IX. CONCLUSION

A comparative study of the pressure distribution and load capacity of infinitely wide parabolic and inclined slider bearings has been presented. Finite element method was used to discretize the governing equation with the associated boundary conditions. It has been shown that when Reynolds equation is modeled to include couple stresses, finite difference method fails to produce convergent solution to the governing equation. The infinitely wide parabolic slider bearing has been shown to be superior in terms of bearing load as a result of the greater pressure generation resulting from increased wedge effect.

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