Steady State Performance of Sliding Contact Bearings with Non-Newtonian Fluids

Devendra Kumar Patel[1*], Shoma Mitra[2], Babu Chellappachetty[1]
[1]School of Mechanical Engineering, VIT University, Vellore - 632014, Tamilnadu, India.
[2]Department of Mechanical Engineering, CSIT, Durg - 491001, Chattisgarh, India.
***Corresponding author: E-Mail: devendra21082@gmail.com**

ABSTRACT

The aim of present study is to investigate and predict theoretically the effects of Non Newtonian couple stress fluids on the steady state performance of infinitely wide slider bearings of three different wedge geometries viz. inclined, parabolic and exponential. A mathematical model using Finite Difference Method is obtained for the hydrodynamic lubrication of infinitely wide inclined, parabolic and exponential slider bearings with couple stress lubricants is presented. Together with the continuity equation, the Stokes motion equations have been used and modified Reynolds equation is derived to account for the couple stress effects arising from a Non Newtonian lubricant. Various optimized bearing characteristics such as maximum pressure, load carrying capacity, frictional force, frictional parameter, volume flow rate and adiabatic temperature rise have been computed using MATLAB for the different wedge geometries, respectively. The results are compared and it has also been shown that when load carrying capacity is the yardstick for comparison, the parabolic slider bearing is superior among all the wedge geometries.

KEY WORDS: slider bearings, couple stress fluid, finite difference method, non-Newtonian lubricant.

1. INTRODUCTION

The rapid development of science and technology has reflected on the innovative design and development of bearings with high precision and of optimum performance, even in adverse working environment. Many of the challenging applications of bearings now-a-days can be found in space technology, electronics, cryogenics, nuclear engineering, computer science etc. A bearing is a mechanical element that permits or allows constrained relative motion between two or more parts, typically rotation or linear movement, with minimum friction. Most of the bearings are lubricated with non-Newtonian fluids. Depending upon the type of friction, bearings are classified into two main groups; 1) Sliding contact bearing or journal bearing. 2) Rolling contact bearings. Sliding contact bearings also known as fluid film bearing as the opposing or mating surfaces are completely separated by a fluid film of lubricant. Different wedge geometries in the slider bearings are used e.g. inclined, parabolic, exponential etc. There are many ways of lubricating the bearings, Newtonian and non-Newtonian both kinds of fluids are used for lubricating the hydrodynamic bearings.

The present numerical study has been focused on hydrodynamic lubrication of sliding contact bearing with non-Newtonian couple stress lubricant. The viscosity of non-Newtonian fluids is effectively dependent on the shear rate. Since the conventional continuum theory cannot accurately describe the flow behaviour of these kinds of fluids, various micro-continuum theories are proposed (Stokes, 1966). The Stokes micro-continuum theory allows for polar effects such as the presence of couple stresses, body couples, and non-symmetric tensors. The effects of couple stresses are quite large for large values of dimensional material constant l. It varies for different liquids depending upon the molecular dimensions of the liquid. Therefore, couple stresses are in-noticeable magnitudes in liquids with very large molecules. These fluids with effective magnitudes of material constant l are known as couple stress fluids (Stokes, 1966).

The concept of couple stress fluids was also applied for porous bearings by Hwang (1996). They concluded that as the value of the porous wall thickness increases, it increases the load capacity of the bearing. Also, when the inclination of the slider increases, the load carrying capacity increase with decrease in friction parameter. Based on the micro continuum theory, Lin (1997) investigated theoretically the rheological effects of couple stress fluids on the lubrication performance of finite journal bearings. According to the results, the effects of couple stresses enhance the load-carrying capacity, as well as reduce the friction parameter and the attitude angle. In continuation, Lin (1998), has theoretically studied squeeze film behaviour for a finite journal bearing lubricated with couple stress fluids. On the basis of the Brinkman model, a theoretical study of the optimal load-carrying capacity and friction coefficient for one-dimensional curved porous slider bearings with the gap width varying slowly is presented by Lin (2001). A numerical study of thermo-hydrodynamic performance of a high-speed slider bearing with conduction to a stationary pad has been analysed by Rathish Kumar (2001). Lin (2003), have derived a general dynamic Reynolds equation of sliding-squeezing surfaces with non-Newtonian fluids for the assessment of dynamic characteristics of a lubricating system. From the results obtained, it was found that the effects of couple stresses provide an improvement on both the steady-state performance and the dynamic stiffness and damping characteristics especially for the bearing with a higher value of aspect ratio means the quantitative effects of couple stresses are more pronounced when the bearing tends to be wide. Other than couple stress fluids, effects of some other non-Newtonian fluids were also analysed, Ferro-fluid lubrication of a porous secant shaped slider bearing was analysed considering slip velocity at the interface of film and porous regions by Shah (2003), in their work.

In the field of inclined slider bearing, different wedge geometries were considered for analysis of bearing characteristics. A bearing with its slider in exponential form and stator with a porous facing of uniform thickness was analysed using a Ferro-fluid lubricant also considering slip velocity is presented by Shah (2003). A slider bearing consisting of connected surfaces with another non-Newtonian fluid known as Powell-Eyring fluid as a lubricant is analysed by Yurusoy (2003). Lin and Lu (2004); Yan-Yan Ma and Cheng (2004); Bayrakceken and Yurusoy (2006); Devakar and Iyengar (2008); and Montazeri (2008), have also analysed the performances of the hydrodynamic bearings using different non-newtonian fluids. A few studies have been concentrated on the effects of surface waviness on the lubrication process numerically. An optimization study was done by Ozalp and Umur (2006), to propose an innovative surface profile design by implementing a wavy form on the upper surface, without varying the physical limits of the complete slider-bearing structure. A mathematical model for the hydrodynamic lubrication of parabolic slider bearings with couple stress lubricants was presented by Humphrey and John (2009). The rheological effect of couple stress fluids on the static and dynamic characteristics of squeeze film lubrication in finite porous journal bearings were studied by Naduvinamani and Patil (2009), by using the finite difference technique. The effects of couple stress fluid on the stability of three lobe hydrodynamic journal bearing were analysed by Mehta (2010).

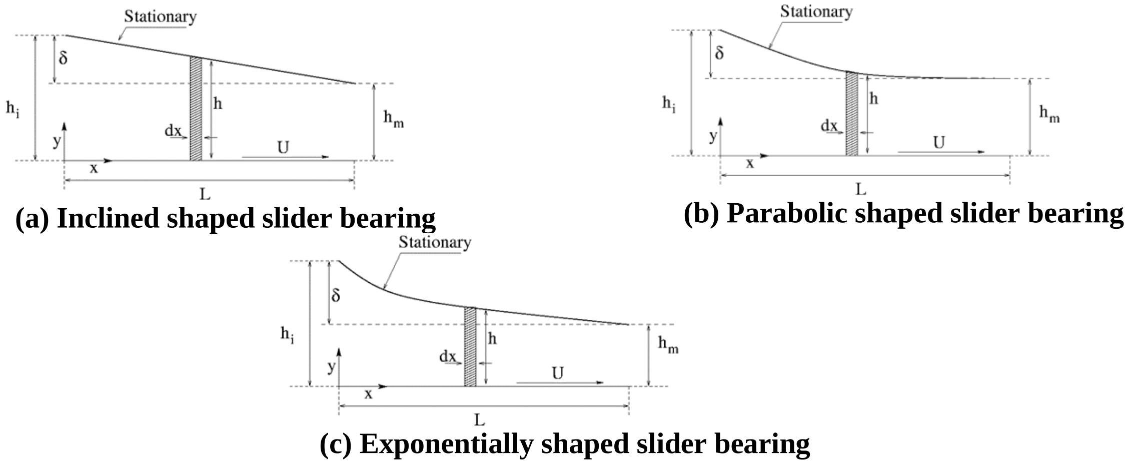
In the present study three different wedge geometries (i.e. inclined, parabolic and exponential) of slider bearings have been considered. Where, the non-Newtonian couple stress fluids are used for lubrication. The steady state performance of all the geometries have been studied using finite difference method on uniform grid point distribution. The iterative procedure of solutions have been performed using MATLAB code. Initially, the parabolic wedge profile has been analysed for four values of couple stress material constant with varying profile parameter. Results are compared with Lin and Lu (2004). Further the optimized value of profile parameter have been obtained for different wedge geometries. Where, the optimized value of profile parameter has been decided based on maximum load carrying capacity with Newtonian fluid lubrication. Further, based on the optimized values of profile parameter for different wedge geometries, the effect of couple stress on various properties have been presented. The variation of dimensionless pressure over the different wedge geometries have also been compared for their optimized working condition.

2 MATHEMATICAL FORMULATIONS

The geometries of inclined, parabolic and exponential slider bearings under consideration are shown in Figs. 1a, 1b and 1c respectively. The slider bearing has a length L and moves with a velocity U. Where h_i and h_m are the maximum and minimum oil film thickness respectively. δ represents the profile parameter of the bearing which is basically the difference between two thickness ($\delta = (h_i - h_m)/h_m$). The one-dimensional hydrodynamic lubrication is expressed mathematically as,

$$\frac{d}{dx}\left[\left(\frac{h^3}{\mu U}\right)\frac{dp}{dx}\right] = 6\frac{dh}{dx} \tag{1}$$

Which is also known as "Reynolds equation governing the hydrodynamic film pressure". Where μ is dynamic viscosity of fluid and p is the pressure. The Reynolds equation requires some simplified assumptions those are as follows; fluid inertia is neglected as viscous force is much greater than the inertia force, body forces are neglected, the lubricating fluid behaves as a Newtonian fluid, fluid density is constant, viscosity is constant throughout the generated fluid film.



(a) Inclined shaped slider bearing **(b) Parabolic shaped slider bearing**

(c) Exponentially shaped slider bearing

Figure.1. Bearing geometry

The dimensionless form of the governing equation has been obtained by following dimensionless parameters as defined by Hwang (1996); Rathish Kumar (2001); Lin (2001).

$$x^* = \frac{x}{L}, y^* = \frac{y}{h_m}, u^* = \frac{u}{U}, v^* = \frac{lv}{h_m U},$$

$$p^* = \frac{p h_m^2}{\mu U L}, l^* = \frac{l}{h_m} = \frac{\left(\frac{\eta}{\mu}\right)^2}{h_m}, h^* = \frac{h}{h_m}$$

In these equations u^* and v^* represents the dimensionless velocity components in x and y directions respectively. Here p^* is the dimensionless pressure and η is a new material constant with the dimension of momentum. The material constant (η) is responsible for the couple stress property of fluid. The length is identified as the characteristic material length which is given by $l = (\eta/\mu)^{1/2}$. So the dimensionless form of Reynolds equation (Eq. 1) governing the hydrodynamic film pressure is given by Eq. 2, which is only for Newtonian lubricant.

$$\frac{d}{dx^*}\left[(h^*)^3 \frac{dp^*}{dx^*}\right] = 6\frac{dh^*}{dx^*} \tag{2}$$

The above non dimensional Reynolds equation governing the hydrodynamic film pressure, is modified for couple stress lubricant as suggested by Lin and Lu (2004), given in Eq. 3.

$$\frac{d}{dx^*}\left[f(h^*,\ l^*) \frac{dp^*}{dx^*}\right] = 6\frac{dh^*}{dx^*} \tag{3}$$

Where, the function $f(h^*, l^*)$ is defined by

$$f(h^*,\ l^*) = (h^*)^3 - 12(l^*)^2\left[h^* - 2l^* \tanh\left(\frac{h^*}{2l^*}\right)\right] \tag{4}$$

And the effects of couple stress are therefore dominated through the dimensionless couple stress parameter $l^* = l/h_m = (\eta/\mu)^2/h_m$. If $\eta = 0$, therefore $l^* = 0$, and the classical form of the Newtonian lubricant is obtained, shown in Eq. 2. The dimensionless form of the u-velocity profile at any cross section obtained by Eq. 5 Lin and Lu [11].

$$u^* = 1 - \frac{y^*}{h^*} + \frac{dp^*}{dx^*}\left\{y(y^* - h^*) + 2l^{*2}\left\{1 - \frac{\cosh[(2y^* - h^*)/2l^*]}{\cosh[(h^*)/(2l^*)]}\right\}\right\} \tag{5}$$

The oil film profile for the inclined, parabolic and exponential slider bearing in dimensionless form is given in Eq. 6, Eq. 7 and Eq. 8 respectively.

$$h^* = h_m^* + \delta(1 - x^*) \tag{6}$$

$$h^* = h_m^* + \delta(1 - 2x^* + x^{*2}) \tag{7}$$

$$h^* = (h_m^* + \delta)e^{-x^* \ln\left(\frac{\delta}{h_m^*}+1\right)} \tag{8}$$

Where h_m^* m is the minimum film thickness (dimensionless) at the exit of the slider bearing.

Bearing characteristics: The dimensionless load carrying capacity per unit width (W^*) of a slider bearing is obtained by integrating the dimensionless pressure distribution over the specified area (Lin and Lu, 2004).

$$W^* = \frac{W h_m^2}{6\mu U L^2 B} = \int_{x^*=0}^{1} p^* dx^* \tag{9}$$

Where W is the load carrying capacity, B is the width and L is the length of bearing. The non-dimensional frictional force on the lower surface (F_L^*) and on the upper surface (F_U^*) are given by [11]

$$F_L^* = \frac{F_L h_m}{\mu U L B} = \int_{x^*=0}^{1} \left[\tau_{yx}^*\right]_{y^*=0} dx^* = \int_{x^*=0}^{1} \left[-\frac{1}{h^*} - \frac{h^*}{2}\frac{dp^*}{dx^*}\right] dx^* \tag{10}$$

$$F_U^* = \frac{F_U h_m}{\mu U L B} = \int_{x^*=0}^{1} \left[-\tau_{yx}^*\right]_{y^*=h^*} dx^* = \int_{x^*=0}^{1} \left[+\frac{1}{h^*} - \frac{h^*}{2}\frac{dp^*}{dx^*}\right] dx^* \tag{11}$$

Which is obtained by integrating the dimensionless shear stress (τ_{yx}^*) over the bearing area. Where τ_{yx}^* is obtained by differentiation of Eq. 5 with respect to y. The friction coefficient is evaluated from the load carrying capacity and friction forces as shown in equation 12 as suggested by [11].

$$\mu_p = -\frac{F_L^*}{W^*} \tag{12}$$

The dimensional form of volume flow rate Q_x^* in the x direction is given by equation 13 as defined by [11].

$$Q_x^* = \frac{q_x}{U h_0 B} = \int_0^{h^*} u^* dy^* \tag{13}$$

The dimensionless form of adiabatic temperature rise (t_m^*) due to the work done against the shearing stresses is obtained by Eq. 14 as suggested by Hamrock (1994).

$$t_x^* = \frac{\rho g J C_p h_m^2}{\mu U L} t_m = \frac{F_L^*}{Q_x^*} \tag{14}$$

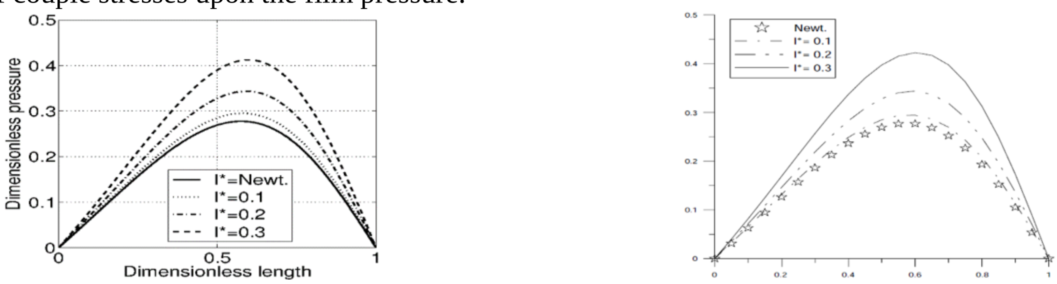
Where g gravitational acceleration, J Joule's mechanical equivalent of heat, C_p specific heat of the lubricant at constant pressure.

The uniform one-dimensional grid point distribution has been considered for discretization of equations. The non-dimensional Reynolds equation governing the hydrodynamic film pressure (Eq. 3) with couple stress lubricant has been discretized using central difference scheme. Dirichlet boundary condition have been considered to solve the Eq. 3. Here the dimensionless pressure at the inlet and at the outlet have been considered zero. Ones getting the distribution of dimensionless pressure, the dimensionless velocity profile at particular location of x has been obtained by the Eq. 5. The load carrying capacity has been calculated by the integration of pressure distribution over the length of bearing as given in Eq. 9. Further, the frictional force on the lower and upper surface of the bearing have been calculated by integrating the Eq. 10 and Eq. 11 over the length of bearing. The frictional parameter have been obtained by the values of force on the surface and load carrying capacity (Eq. 12). In steady state the volume flow rate for each cross section of the thin film will be constant. The volume flow rate has been calculated using the velocity profile at the region where pressure is maximum or pressure gradient is zero ($dp/dx = 0$). At this position the velocity profile integrated over the thickness of the thin film ($0 - h$) as shown in Eq. 13. The dimensionless form of adiabatic temperature rise t_m^* due to the work done against the shearing stresses has been calculated by the Eq. 14.

3. RESULTS AND DISCUSSION

Steady state performance of parabolic slider bearing: In the present section, computational results are obtained for parabolic profile slider bearing which have been compared with the results of Lin and Lu (2004). The results show the effects of dimensionless couple stress parameter upon the steady state performance of wide parabolic-shaped slider bearing. As the value of couple stress parameter (l^*) approaches zero, the effects of couple stress vanishes and the dimensionless non-Newtonian modified Reynold's equation reduces to the Newtonian-lubricant Reynolds equation case. In order to validate the present codes with the results of Lin and Lu (2004), the numerical computations are performed by choosing the reference film thickness$h_0 = h_m$, the profile parameter $\delta = 0.0\ to\ 3.0$ and couple stress parameter $l^* = 0.1, 0.2, 0.3$ respectively.

Pressure distribution: Figure 2 shows the dimensionless film pressure as a function of x^* for different values of couple stress parameter (l^*). It is observed that the effects of couple stresses ($l^* = 0.1$) increase the values of p^* as compared to the Newtonian-lubricant case. Increasing values of the couple stress parameter ($l^* = 0.2, 0.3$) increase the effects of couple stresses upon the film pressure.



(a) Present computation **(b) Lin and Lu (2004)**

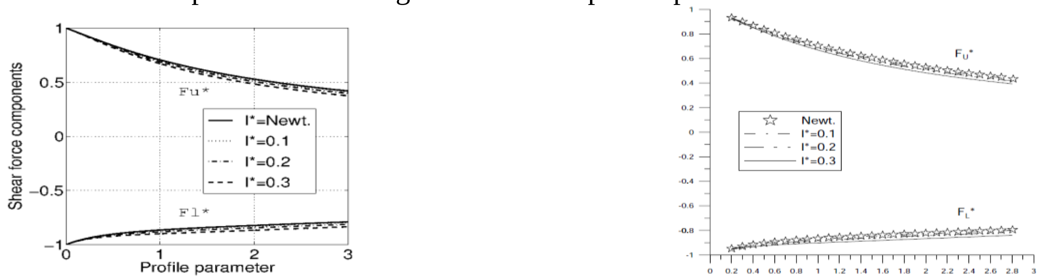
Figure.2.Dimensionless film pressure (p^*) as a function of x^* for different l^*

(a) Present computation **(b) Lin and Lu (2004)**

Figure.3.Dimensionless load capacity (W^*) as a function of profile parameter (δ) for different l^*

Load carrying capacity: The dimensionless load-carrying capacity (W^*) as a function of profile parameter (δ) for different values of couple stress (l^*) are presented in Fig. 3 Compared to the Newtonian-lubricant case, the effects of couple stresses are observed to increase the load-carrying capacity. The increase in dimensionless load-carrying capacity (W^*) is more accentuated at a higher value of the couple stress parameter ($l^* = 0.3$), for the parabolic slider bearing with medium profile parameter (about $\delta = 1.1$).

Shear force component: Figure 4 describes the dimensionless shear force components (F_U^* and F_L^*) as a function of profile parameter (δ) for different values of dimensionless couple stress (l^*). It is shown that the shear force component F_U^* decreases with increasing values of profile parameter (δ), but the shear force component F_L^* increases with profile parameter. Comparing with the Newtonian-lubricant case, the effects of couple stresses signify a decrease in the shear force components under large values of the profile parameter.



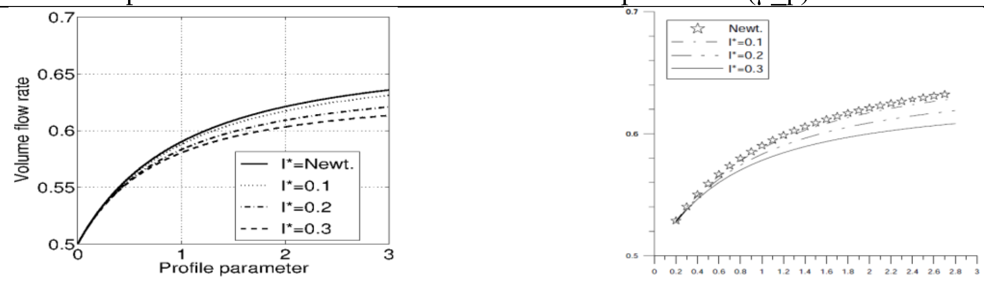
(a) Present computation **(b) Lin and Lu (2004)**

Figure.4.Shear force components (F_U^* and F_L^*) as a function of profile parameter (δ) for different l^*

(a) Present computation **(b) Lin and Lu (2004)**

Figure.5.Friction parameter (μ_p) as a function of profile parameter (δ) for different l^*

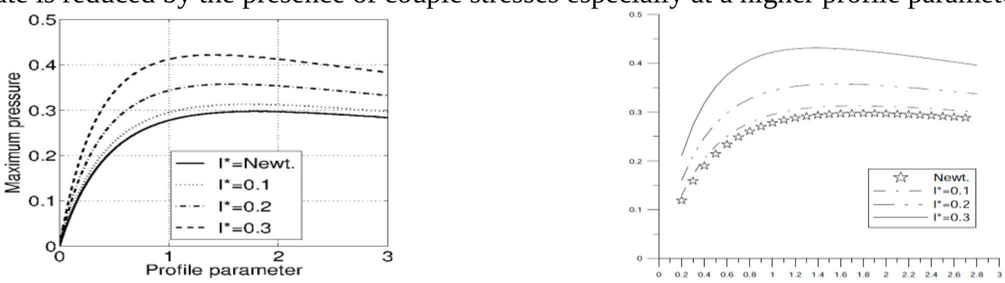
Friction parameter: The friction parameter (μ_p) as a function of profile parameter (δ) for different values of l ¤are presented in Fig. 5 The value of friction parameter (μ_p) is observed to decrease with profile parameter (δ) to attain a minimum, and thereafter increases with profile parameter (δ). Comparing with the Newtonian-lubricant case, the effects of couple stresses provide a decrease in the value of friction parameter (μ_p).



(a) Present computation **(b) Lin and Lu (2004)**

Figure.6. Volume flow rate (Q_x^*) as a function of profile parameter (δ) for different l^*

Volume flow rate: Figure 6 shows the dimensionless volume flow rate (Q_x^*) as a function of profile parameter (δ) for different values of dimensionless couple stress (l^*). It is found that the value of dimensionless volume flow rate (Q_x^*) increases with the value of profile parameter (δ). Compared to the Newtonian-lubricant case, the required volume flow rate is reduced by the presence of couple stresses especially at a higher profile parameter.

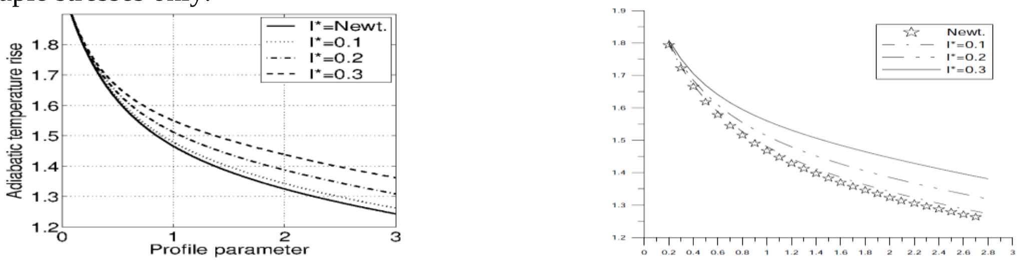


(a) Present computation **(b) Lin and Lu (2004)**

Figure.7.Maximum pressure (p_M^*) as a function of profile parameter (δ) for different l^*

Maximum pressure: Figure 7 shows the dimensionless maximum pressure (p_M^*) as a function of profile parameter (δ) for different values of dimensionless couple stress (l^*).

Adiabatic temperature rise: Figure 8 shows the dimensionless adiabatic temperature rise (t_m^*) as a function of profile parameter (δ) for different dimensionless couple stress (l^*). Higher value of delta yield lower values of dimensionless adiabatic temperature rise (t_m^*), since it requires larger amount of flowrate (Q_x^*). It also found that the effect of couple stress increase the temperature rise because of the reduced volume flow rate which is also due to the presence of couple stresses only.



(a) Present computation **(b) Lin and Lu (2004)**

Figure.8.Adiabatic temperature rise (t_m^*) as a function of profile parameter (δ) for different l^*

The effects of couple stresses characterized by the couple stress parameter signify an improvement in the steady-state performance as compared to the traditional Newtonian-lubricant case. Increasing values of the couple stress parameter increases the bearing load carrying capacity, and reduces the required volume flow rate and the friction parameter.

It is seen that all the bearing characteristic like load capacity, shear force components, volume flow rate, frictional parameters and maximum pressure obtained in the present work have the similar ranges of values as obtained by Lin and Lu (2004). Also, the variation pattern of the bearing characteristics in the present work are identical to the results of Lin and Lu (2004). They have presented their work mainly on four values of couple stress material constant (l^*), those values are 0.0 (Newtonian fluid), 0.1, 0.2 and 0.3 respectively.

Now, we have modified the work by considering two more wedge profiles for slider bearings, those are inclined and exponential (Fig. 1a and 1c). Our first aim is to calculate the optimized value of δ for each profile where the load carrying capacity is maximum respectively, by keeping $l^* = 0.0$ Then after we will discuss the various bearing characteristics with respect to dimensionless couple stress (l^*) for each profile.

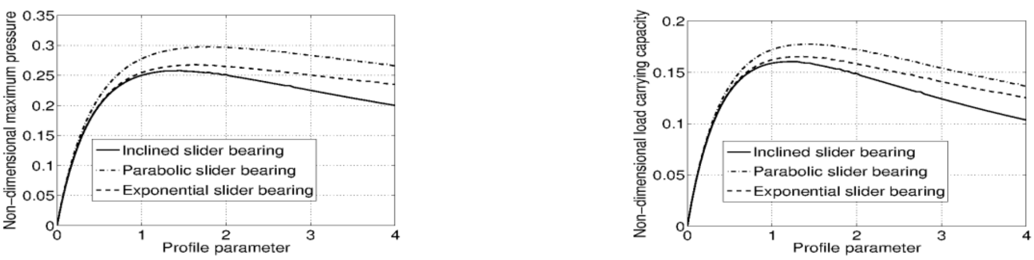
Optimizing the profile parameter (δ) for different wedge geometries: The dimensionless modified Reynolds equation 3 has been solved to get the equation of pressure and is further discretized using finite difference method. The central difference scheme have been used in uniform grid structure, the number of grid point chosen is 80. The obtained pressure profile has been integrated over the length of bearing to get the value of load carrying capacity as given in equation 9.

Figure 9a shows the Variation of dimensionless maximum pressure (p^*) w.r.t profile parameter (δ) for inclined, parabolic and exponential slider bearings considering Newtonian fluid as a lubricant. This figure shows, as the value of profile parameter (δ) increases from 0 to 1.5 approximately the value of dimensionless maximum pressure P_M^* increases then it decreases continuously for each profile.

The value optimized profile parameter (δ) for three different wedge profiles have been found are as follows: For inclined $(p^*)_{max} = 0.2580$ at $\delta_i = 1.45$, For parabolic $(p^*)_{max} = 0.2976$ at $\delta_i = 1.7$, For exponential $(p^*)_{max} = 0.2678$ at $\delta_i = 1.65$.

Similarly, figure 9b shows the variation of dimensionless load carrying capacity (W^*) w.r.t delta profile parameter (δ). Here we found almost same situation that the load carrying capacity increases with profile parameter in the range of 0 to 1.5 approximately, then it decreases continuously. The value optimized profile parameter (δ) for three different wedge profiles have been found are as follows: For inclined $(W^*)_{max} = 0.1604$ at $\delta_i = 1.25$, For parabolic $(W^*)_{max} = 0.1774$ at $\delta_i = 1.45$, For exponential $(W^*)_{max} = 0.1653$ at $\delta_i = 1.35$.

As the load carrying capacity is considered to be the most important parameter in bearing design, we have finalized the three optimized δ values which are found according to the maximum load carrying capacity of the three profiles.



(a) Variation of dimensionless maximum pressure (P_M^*) w.r.t delta (δ). **(b) Variation of dimensionless load carrying capacity (W^*) w.r.t delta (δ).**

Figure.9.Variation of dimensionless maximum pressure and load carrying capacity for $l^* = 0.0$.

Variation of various properties with material constant for inclined, parabolic and exponential wedges:

Effect of couple stress on maximum pressure: Figure 10 shows the variation of dimensionless maximum pressure $((p^*)_{max})$ w.r.t dimensionless material constant (l^*). It depicts, as the value of material constant increases from 0.0 to 0.43 approx., the value of maximum pressure increases. Further increasing the couple stress material constant results a decreases in the values of maximum pressure for all the three profile. It has been noticed that the profile of $(p^*)_{max}$ for all bearing geometries starts merging together after increasing the value of dimensionless material constant (l^*) from 0.43 approx. At the value of $l^* = 2.0$, $(p^*)_{max}$ attends the similar value for all the profile.

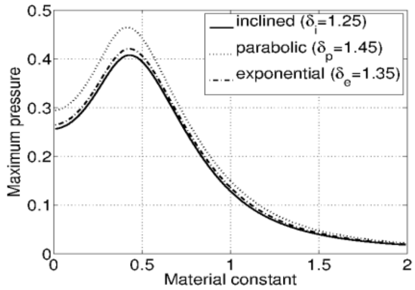


Figure.10.Variation of dimensionless maximum pressure $((p^*)_{max})$ w.r.t dimensionless material constant (l^*)

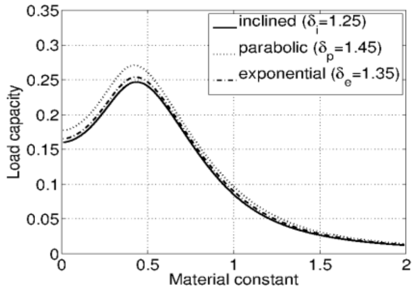


Figure.11.Variation of dimensionless load carrying capacity (W^*) w.r.t dimensionless material constant (l^*)

Effect of couple stress on load carrying capacity: A plot of dimensionless load carrying capacity (W^*) for different values of couple stress material constant (l^*) is shown in Fig. 11. Since the effect of increase in couple stress (till $l^* = 0.43$ approx) provides an increase in the lubricant film pressure, the load carrying capacity is similarly influenced. In general, with increase in couple stress (till $l^* = 0.43$ approx) the load carrying capacity of all the three different wedge geometries slider bearings is increased. The load carrying capacity is the most important bearing characteristic to decide the bearing geometry. Also, the Fig. 11 shows that the load carrying capacity for a parabolic slider bearing is highest among all the three different wedge geometry slider bearing for the same couple stress parameter at its optimized profile parameter.

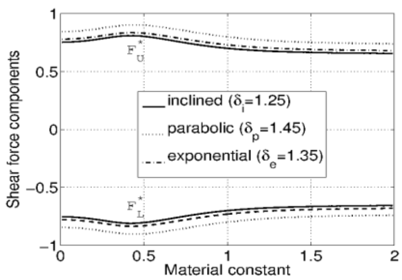


Figure.12.Variation of dimensionless shear force components (F_U^* and F_L^*) w.r.t dimensionless material constant (l^*)

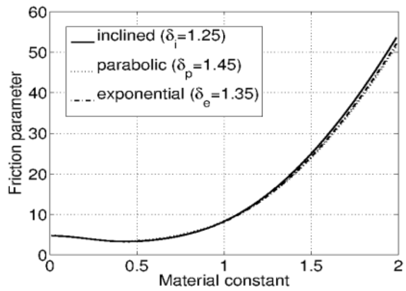


Figure.13.Variation of dimensionless frictional parameter (μ_p) w.r.t dimensionless material constant (l^*)

Effect of couple stress on shear force components: Figure 12 describes the dimensionless shear force components F_U^* and F_L^* as a function of couple stress material constant (l^*) for different values of optimized profile parameter (δ) of three slider bearing wedge geometries. It is shown that the shear force components F_U^* and F_L^* both increases up till the value of $l^* = 0.43$ (approx.) then gets decreases with increasing values of dimensionless material constant (l^*), but the shear force component F_L^* is in the negative direction. The higher values of the shear force at $l^* = 0.43$ (approx.) also shows higher load carrying capacity at the same value of dimensionless material constant (l^*).

Effect of couple stress on frictional parameter: Figure 13 presents the friction parameter (μ_p) as a function of couple stress material constant (l^*) for different values of optimized profile parameter (δ) of three slider bearing wedge geometries. The value of μ_p is observed to decrease with increase in l^* from 0.0 to 0.43 (approx.) to attain a minimum value, and there after increase with increase in l^*. As shown in Eq. 12, μ_p depends upon the values of frictional force and load carrying capacity, but the range of load carrying capacity is much more higher than variation in frictional force, which effects more the value of μ_p.

Effect of couple stress on volume flow rate: Figure 14 shows the variation in dimensionless volume flow rate (Q_x^*) as a function of (l^*) for different values of optimized profile parameter (δ). It is found that the volume flow rate decreases as the non-Newtonian behaviour of lubricant increases. It attains the minimum $l^* = 0.43$ approximately; thereafter it increases rapidly till the value of $l^* = 1.0$ approximately. It starts decreasing slowly afterwards.

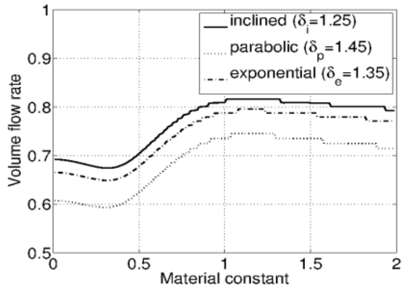


Figure.14.Variation of dimensionless volume flow rate (Q_x^*) w.r.t dimensionless material constant (l^*)

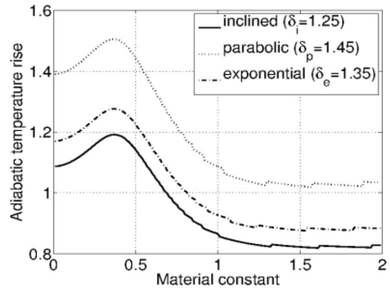


Figure.15.Variation of dimensionless adiabatic temperature rise (t_m^*) w.r.t dimensionless material constant (l^*)

Effect of couple stress on adiabatic temperature rise: The variation in dimensionless adiabatic temperature rise (t_m^*) as a function of (l^*) for different values of optimized profile parameter (δ) depicted in Fig. 15. It is found that the effect of couple stress increase the temperature rise in the range of 0.0 to 0.5 approximately, because of continuous reduction of volume flow rate in the same range.

Variation of pressure for optimized conditions: In the section 3.2 optimize values of δ have been calculated for all three bearing profiles by assuming the Newtonian fluid as lubricant. These optimize values of δ have been used to find optimize values of l^* for all respective profiles, to get the situation of high load carrying capacity. Figure 16 shows the dimensionless pressures distribution for all three infinitely wide slider bearings for their respective optimum load carrying capacity.

4. CONCLUSION

The effects of couple stresses upon the steady state performance of infinitely wide slider bearings of three different wedge geometries are studied according to the Stokes micro-continuum theory. Using the Stokes motion equations together with the continuity equation, the non-Newtonian modified Reynolds-type equation is derived to account for the couple stress effects arising from a Newtonian lubricant blended with various additives. From the results obtained and discussed, conclusions can be drawn as follows.

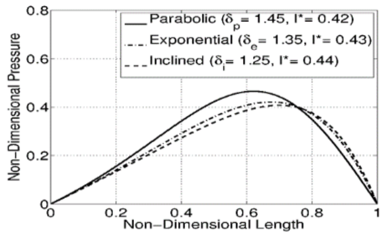


Figure.16. Dimensionless pressures distribution for infinitely wide inclined, parabolic and exponential slider bearing for its optimum load carrying capacity

The effects of couple stresses characterized by the couple stress parameter signify an improvement in the steady-state performance as compared to the traditional Newtonian-lubricant for all three wedge profiles. Increasing values of the couple stress parameter increases the bearing load carrying capacity, and reduces the required volume flow rate and the friction parameter up to $l^* = 0.42$ (approx.). In comparison among the three wedge profiles the parabolic shaped slider bearing should be preferred due to high load carrying capacity. The parabolic-shaped slider lubricated bearing with couple stress $l^* = 0.42$ (approx.) fluids results in a higher load-carrying capacity i.e. $W^* = 0.27084$ (Ref. Fig. 11 and Table 1) and a smaller required volume flow rate (Q_x^*) at its optimized profile parameter $\delta_p = 1.45$.

Table.1.Values of various parameters at optimum load carrying capacity

Bearing Profile	W_{max}^*	p^*	$F_U^* = -F_L^*$	μ_p	Q_x^*	t_m^*
Inclined ($\delta_i = 1.25,\ l^* = 0.44$)	0.24690	0.40761	0.80869	3.27535	0.68805	1.17534
Parabolic ($\delta_p = 1.45,\ l^* = 0.42$)	0.27084	0.46523	0.90278	3.33332	0.60425	1.49404
Exponential ($\delta_p = 1.35,\ l^* = 0.43$)	0.25391	0.42099	0.83490	3.28795	0.66061	1.26383

The only drawback of parabolic shaped slider bearing is the maximum rise of adiabatic temperature i.e. $t_m^* = 1.49404$ (Ref. Fig. 15 and Table 1) at the value of couple stress $l^* = 0.42$ (approx). This rise of adiabatic temperature shows the parabolic shaped slider bearing requires more cooling during its operation. The present study also shows that the performance of inclined and exponential slider bearing are almost similar, so due to manufacturing ease inclined profile is much more popular than exponential. The present study provides engineers useful information in bearing designs.

REFERENCES

Bayrakceken H, Yurusoy M, Comparison of pressure distribution in inclined and parabolic slider bearings, Mathematical and Computational Applications, 11, 2006, 65-73.

Devakar M, Iyengar TKV, Stokes problems for an incompressible couple stress fluid, Nonlinear Analysis: Modelling and Control, 1, 2008, 181-190.

Hamrock BJ, Fundamentals of fluid film lubrication. Mcgraw-hill Inc., New York, 1994.

Humphrey OM, John A, Parametric characterization of load capacity of infinitely wide parabolic slider bearing with couple stress fluids, World Academy of Science, Engineering and Technology, 57, 2009, 386-390.

Hwang CC, Lin JR, Yang RF, Lubrication of long porous slider bearings, (use of the brinkman-extended darcy model), Jsme International Journal, Series B: Fluids And Thermal Engineering, 39, 1996, 141-148.

Lin J.R, Squeeze film characteristics of finite journal bearings couple stress fluid model, Tribology International, 31 (4), 1998, 201-207.

Lin JR, Effects of couple stress on the lubrication of finite, Journal bearing wear, 206, 1997, 171-178.

Lin JR, Lu RF, Chang TB, Derivation of dynamic couple-stress Reynolds equation of sliding-squeezing surfaces and numerical solution of plane inclined slider bearings, Tribology International, 36 (9), 2003, 679-685.

Lin JR, Lu YM, Steady-state performance of wide parabolic-shaped slider bearings with a couple stress fluid, Journal of Marine Science and Technology, 12, 2004, 239-246.

Lin JR, Optimal design of one-dimensional porous slider bearings using the brinkman model, Tribology International, 3, 2001, 57-64.

Mehta NP, Rattan SS, Verma R, Stability analysis of three lobe hydrodynamic journal bearing: couple stress fluid effects, International Journal of Engineering Science and Technology, 2 (10), 2010, 5736-5745.

Montazeri H, Numerical analysis of hydrodynamic journal bearings lubricated with ferrofluid, Journal of Engineering Tribology, 222, 2008, 51.

Naduvinamani NB, Patil SB, Numerical solution of finite modified Reynolds equation for couple stress squeeze film lubrication of porous journal bearings, Computers and Structures, 87, 2009, 1287-1295.

Ozalp A, Umur H, Optimum surface profile deign and performance evaluation of inclined slider bearings, Current Science, 90, 2006, 1480-1491.

Rathish Kumar BV, Srinivasa Rao P, Sinha P, A numerical study of performance of a slider bearing with heat conduction to the pad, Finite Elements In Analysis And Design, 37, 2001, 533-547.

Shah, Rajesh C, Bhat MV, Lubrication of a porous exponential slider bearing by ferrofluid with slip velocity, Turkish Journal Of Engineering And Environmental Sciences, 27, 2003, 183-187.

Shah, Rajesh C, Bhat MV, Porous secant shaped slider bearing with slip velocity lubricated by ferrofluid, Industrial Lubrication And Tribology, 55, 2003, 113-115.

Stokes VK, Couple stresses in fluids, Physics of Fluids, 9, 1966, 1709-1715.

Yan-Yan Ma WHW, Cheng XH, A study of dynamically loaded journal bearings lubricated with non-newtonian couple stress fluids, Tribology Letters, 17 (1), 2004, 69-74.

Yurusoy M, A study of pressure distribution of a slider bearing lubricated with powelleyring fluid, Turkish Journal of Engineering and Environmental Sciences, 27, 2003, 299-304.