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Effect of slip/no-slip on finite slider bearing using non-Newtonian micropolar fluid

Pentyala S. Rao [1,*], Santosh Agarwal [1,2]

[1] Department of Applied Mathematics, Indian Institute of Technology, 826001, Dhanbad, INDIA.
[2] Department of Mathematics, Gaya College of Engineering, 823002, Gaya, Bihar, INDIA.
[*]Corresponding author: pentyalasrinivasa@gmail.com

KEYWORD

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ABSTRACT

In this paper, the modified Reynolds equation of finite slider bearing lubricated with micropolar fluid is numerically solved for the computational aspects of the bearings. The finite difference scheme has been employed to solve the governing equations. The effect of micropolar parameter and slip parameter is investigated on slider bearing. For investigating the effect of slip boundary on the pressure distribution in sliding surface is numerically presented in the Reynolds model. The two-dimensional modified Reynolds equation can predict the performance of lubrication process with boundary slip in sliding contact which can be seen by the obtained results. The pressure and load capacity are displayed graphically. The pressure and load carrying capacity is lesser for slip case as compared to no slip case.

1.0 INTRODUCTION

The Reynolds lubrication theory has become an efficient tool in both analysis and design of lubricated contacts. In the derivation of the Reynolds equation, where it has been assumed that there is no boundary slip at the liquid-solid interface. So, it is called as no-slip boundary condition. On micro-scale, however, due to the progress in micrometer measurement technology, it is possible to observe boundary slip of fluid flow over a solid surface, and therefore the conventional boundary no-slip condition can split down. Under such conditions, the classical Reynolds equation is no more applicable. In lubricated-MEMS, proper lubrication is a key issue in reducing the liquid stiction and hence has received a great deal of attention in the relevant literature (Henck, 1997; Spikes, 2003). In bearing analysis and design the classical Reynolds equation is a powerful tool. Therefore, in order to make a good design and analysis of the fluid film lubricated-contact, researchers have extended the classical Reynolds equation by taking into account boundary slip.

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A number of researchers have carried out studies on hydrodynamic lubrication considering velocity slip at the surfaces. Rao (2017) studied the Reynolds equation of two symmetrical surfaces considering slip at the bearing surfaces. They have focused on the effects of velocity slip and developed expressions for pressure in the thin film, load capacity and coefficient of friction which he analyzed numerically for various parameters and found out that due to high viscous layer at the periphery the position of the cavitation moves towards the centre and moves away from the centre due to slip. Rao and Prasad (2004) investigated the effect of velocity slip on load capacity on journal bearings. It was shown that load capacity decreases with slip. It was also shown that the coefficient of friction decreases with a high viscous layer and increases with slip. Beavers and Joseph (1967) discussed the effect of slip for an incompressible fluid.

A lot of work has been carried out to study the influence of boundary slip on hydrodynamic lubrication with the emphasis on the (slightly) parallel sliding configuration. The earliest work on artificial complex slip/no-slip, lubricated parallel sliding devices was reported (Salant and Fortier, 2004; Fortier and Salant, 2005). The effect of velocity slip with porous inclined slider bearing lubricated with a ferrofluid was studied theoretically by Shah and Bhat (2002). Expressions for pressure, load capacity, friction and coefficient of friction were obtained. The increase in slip parameter was found to cause a decrease in the load capacity. The effect of surface roughness on porous journal bearing with heterogeneous surface is examined by Kalavathi et al. (2014). The Christensen stochastic theory of hydrodynamic lubrication have been used to derive the generalized Reynolds equation by considering porosity. They observed that the load carrying capacity increases and coefficient of friction decreases in heterogeneous pattern of slip/no-slip surface of porous narrow journal bearing. Oladeinde and Akpobi (2010) investigated the effect of slip surfaces on finite slider bearings using couple stress fluid. Recently, Kalavathi et al. (2016) have studied numerically the influence of roughness on finite porous journal bearing considering slip/no-slip surface. This has been shown that the bearing with slip/no-slip surface with surface roughness increases the pressure and load distribution.

In all the above studies, none of the researchers have studied the influence of slip on micropolar fluid numerically. So, the aim of the present work is to analyze the effect of the slip on the finite slider bearing using micropolar fluids.

2.0 MATHEMATICAL MODEL

The basic equations governing the flow of micropolar lubricants under the usual assumptions of lubrication theory for thin films are (Henck, 1997; Spikes, 2003):

$$\frac{1}{2}(2\mu+\chi)\frac{\partial^2 u}{\partial y^2}+\chi\frac{\partial v_3}{\partial y}-\frac{\partial p}{\partial x}=0 \tag{1}$$

$$\frac{\partial p}{\partial y}=0 \tag{2}$$

$$\frac{1}{2}(2\mu+\chi)\frac{\partial^2 w}{\partial y^2}+\chi\frac{\partial v_1}{\partial y}-\frac{\partial p}{\partial z}=0 \tag{3}$$

$$\gamma \frac{\partial^2 v_1}{\partial y^2} - \chi \frac{\partial w}{\partial y} - 2\chi v_1 = 0 \tag{4}$$

$$\gamma \frac{\partial^2 v_3}{\partial y^2} - \chi \frac{\partial u}{\partial y} - 2\chi v_3 = 0 \tag{5}$$

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0 \tag{6}$$

Where u, v and w describes the velocity components in the x y and z directions respectively, p is the pressure, v_3 represents the micropolar velocity component, μ is the traditional viscosity coefficient, γ is the additional viscosity coefficient and χ is the additional spin viscosity coefficient of micropolar fluids.

Figure 1 shows the geometry and physical configuration of the slider bearing. It consists of two surfaces separated by a lubricant film. The lower surface of the bearing moving in its own plane with a constant velocity, U_s and the upper solid surface is at the rest. h_1 is the inlet film thickness and h_0 is the outlet film thickness.

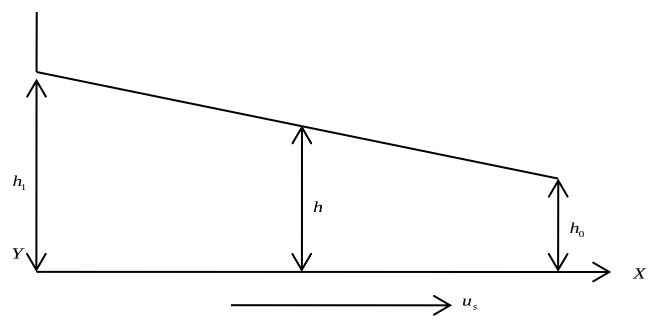

Figure 1: Schematic of bearing configuration.

The relevant boundary conditions for the velocity and micro rotational velocity components are:
(a) At the lower surface (y = 0)

$$u = u_s, v = w = 0, v_1 = v_3 = 0 \tag{7}$$

(b) At the upper surface (y = h)

$$u = -\alpha\mu \frac{\partial u}{\partial y}, v = w = 0, v_1 = v_3 = 0 \tag{8}$$

Under these boundary conditions slip occurs. If the slip length is set equal to zero, the above boundary condition reduces to the traditional no-slip case. The solution of the equations (1), (3), (4) and (5) subject to the boundary conditions given in equations (7) and (8) reduced in the dimensional modified Reynolds equation:

$$\frac{\partial}{\partial x}\left[f(N,l,h,\alpha\mu)\frac{\partial p}{\partial x}\right]+\frac{\partial}{\partial y}\left[f(N,l,h,\alpha\mu)\frac{\partial p}{\partial y}\right]=6\mu u_s\frac{\partial g(N,l,h,\alpha\mu)}{\partial x} \tag{9}$$

where,

$$h=1+k-kx \tag{10}$$

$$A_{11}=-4h^3N^2+24N^2l^2h-4N^2h[2h^2+6l^2+3h(1-N^2)\alpha\mu]cosh\left(\frac{Nh}{l}\right)$$
$$+\frac{Nh}{l}sinh\left(\frac{Nh}{l}\right)[h^3+12hl^2(1+N^2)+4(h^2+3l^2)(1-N^2)\alpha\mu] \tag{11}$$

$$A_{22}=2N^2\left(cosh\left(\frac{Nh}{l}\right)-1\right)-\frac{N}{l}[h+(1-N^2)\alpha\mu]\,sinh\left(\frac{Nh}{l}\right) \tag{12}$$

$$A_{33}=2N^2h\left(cosh\left(\frac{Nh}{l}\right)-1\right)-\frac{Nh}{l}[h+2(1-N^2)\alpha\mu]\,sinh\left(\frac{Nh}{l}\right) \tag{13}$$

$$f(N,l,h,\alpha\mu)=\frac{A_{11}}{A_{22}},g(N,l,h,\alpha\mu)=\frac{A_{33}}{A_{22}} \tag{14}$$

Equation (9) is non-dimensionalized by defining the following dimensional variables and parameters:

$$X=\frac{x}{l_x},Y=\frac{y}{l_y}P=\frac{p}{p_a},H=\frac{h}{h_0},A=\frac{\alpha\mu}{h_0},U=\frac{6\mu u_s l_x}{h_0^2p_a},L_1=\frac{l_x}{l_y}$$

The dimensionless modified Reynolds equation is as follows:

$$\frac{\partial}{\partial X}\left[F(N,L,H,A)\frac{\partial P}{\partial X}\right]+\frac{\partial}{\partial Y}\left[F(N,L,H,A)\frac{\partial P}{\partial Y}\right]=U\frac{\partial G(N,L,H,A)}{\partial X} \tag{15}$$

Where

$$A_{11}^*=-4H^3N^2+24N^2L^2H-4N^2H[2H^2+6L^2+3H(1-N^2)A]cosh\left(\frac{NH}{L}\right)$$
$$+\frac{NH}{L}sinh\left(\frac{NH}{L}\right)[H^3+12HL^2(1+N^2)+4(H^2+3L^2)(1-N^2)A] \tag{16}$$

$$A_{22}^*=2N^2\left(cosh\left(\frac{NH}{L}\right)-1\right)-\frac{N}{L}[H+(1-N^2)A]\,sinh\left(\frac{NH}{L}\right) \tag{17}$$

$$A_{33}^*=2N^2H\left(cosh\left(\frac{NH}{L}\right)-1\right)-\frac{NH}{L}[H+2(1-N^2)A]\,sinh\left(\frac{NH}{L}\right) \tag{18}$$

$$F(N,L,H,A)=\frac{A_{11}^*}{A_{22}^*},G(N,L,H,A)=\frac{A_{33}^*}{A_{22}^*} \tag{19}$$

In order to solve the modified dimensionless Reynolds equation (15) to obtain film pressure distribution, the following boundary conditions are used:

$P = 0 \; at \; X = 0{,}1 \; and \; Y = 0{,}1$

The load per width unit is carried by lubricant film and the calculation is simply an integration of the lubricant film pressure. The load carrying capacity is defined as the integral of the pressure profile over the surface area, and in terms of dimensionless quantities:

$$W = \int_0^1 \int_0^1 P dx dy \tag{20}$$

3.0 RESULTS AND DISCUSSION

The dimensionless Reynolds equation 15 is solved numerically using finite difference method subjected to boundary condition (11). By employing the discretization scheme, the computed domain is divided into a grid with uniform mesh size, ΔX and ΔY . The computational domain is $0 \leq X \leq l_x$and $0 \leq Y \leq l_y$. The mesh size is $\Delta X = \frac{l_x}{n}$ and $\Delta Y = \frac{l_y}{n}$, and n+1 is the number of grid points. The uniform grid is applied on slip face surface. The 40×40 meshes are employed in the computational domain which happens to be grid independent. The grid independency was validated by various numbers of size meshes. But obviously the computational cost increased. Considering the processing time limitation, 40×40 meshes were adopted for all simulation cases. To discretize the modified Reynolds equation first order central difference has been used.

Characteristics of a finite slider bearing with hetero- generous slip/no-slip surfaces are obtained on the basis of various non-dimensional parameters such as the slip parameter$A$and coupling parameter N. These characterize the coupling of the linear and rotational motion arising from the micro motion of the fluid molecules or the additives molecules, L is called the characteristic length where $L\left(= (\frac{l}{h_0}\right)$is the ratio of microstructure size to the minimum film thickness. Hence L gives the mechanism of interaction between the bearing geometry and the fluid. The non-dimensional Reynolds equation reduces to the Newtonian case as the value of L approaches zero.

In order to find the better solution of the numerical model, grid independent study plays a significant role. The significant change comes in practice as the grid size goes on increasing from (10×10) to (50×50). Various 3D meshes of non-dimensional pressure distribution are shown in Figures 2, 3 and 4. The pressure distribution is shown in Figs. 5 and 6 for different value of A with fixed N, L and U of mesh grid 41x41. Fig 5 shows the pressure distribution for the no-slip surface while Fig. 6 shows the pressure distribution for slip surface. It is observed from the figure that the bearing surface with the slip having less pressure as compared to no slip case. Figure 7 shows the variation of load with sliding velocity U. It is found that the load carrying capacity has linear variation with velocity. The load support for two configurations is zero when sliding velocity is absent (U = 0). The load carrying capacity supported by the surface with slip boundary shows a lesser load than those without slip.

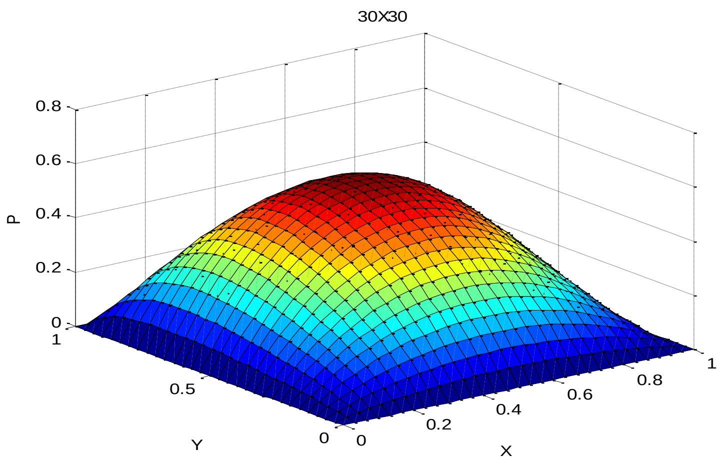

Figure 2: Non-dimensional pressure distribution with grid size (30 ×30).

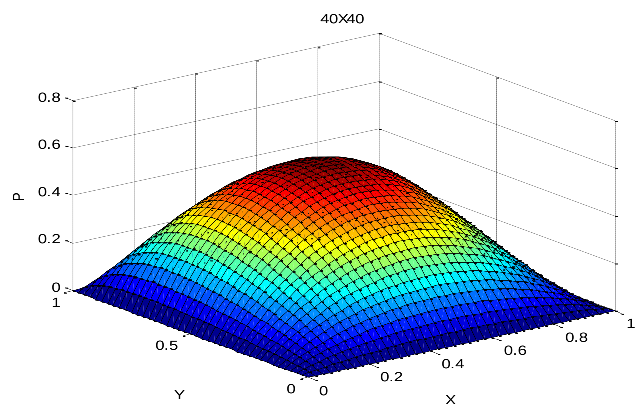

Figure 3: Non-dimensional pressure distribution with grid size (40 ×40).

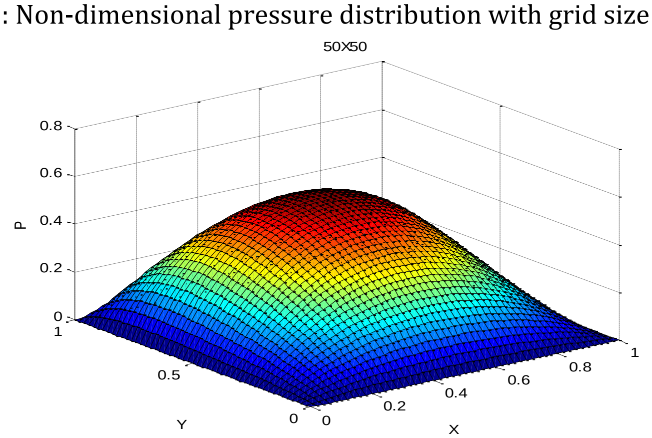

Figure 4: Non-dimensional pressure distribution with grid size (50 ×50).

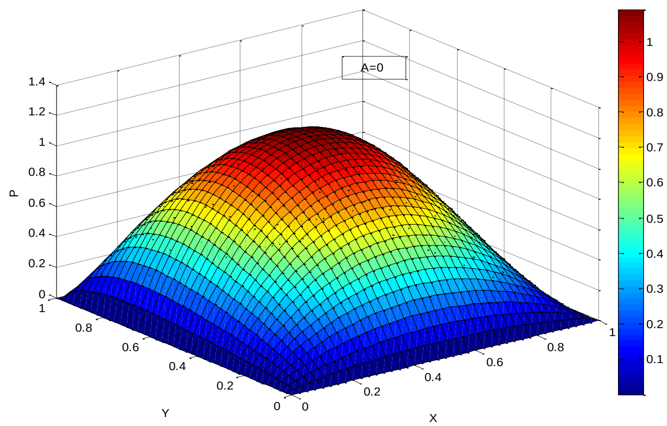

Figure 5: Variation of Pressure distribution for $N = 0.3, L = 0.2, U = 50K = 1.25$ and $A = 0$.

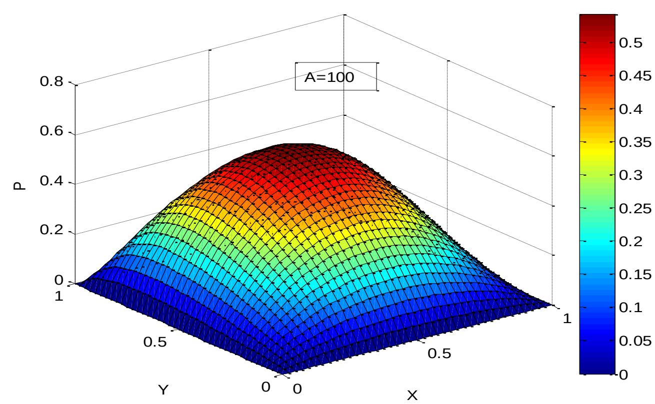

Figure 6: Variation of Pressure distribution for $N = 0.3, L = 0.2, U = 50, K = 1.25$ and $A = 100$.

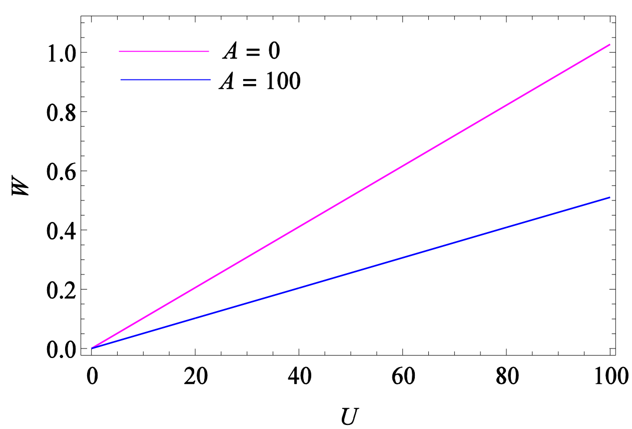

Figure 7: Variation of W with U for different values of $N = 0.3, L = 0.2, K = 1.25$.

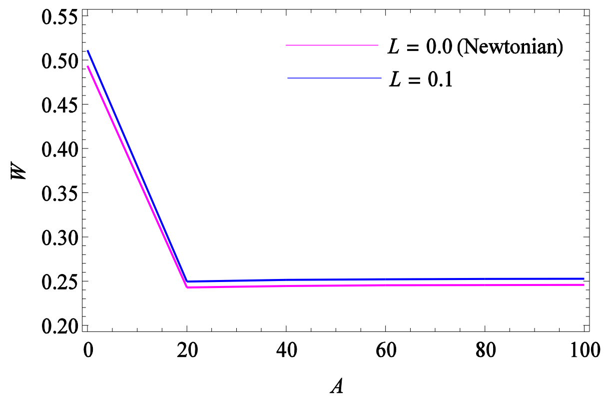

Figure 8: Variation of W with A for different values of $N = 0.3, U = 50, K = 1.25$

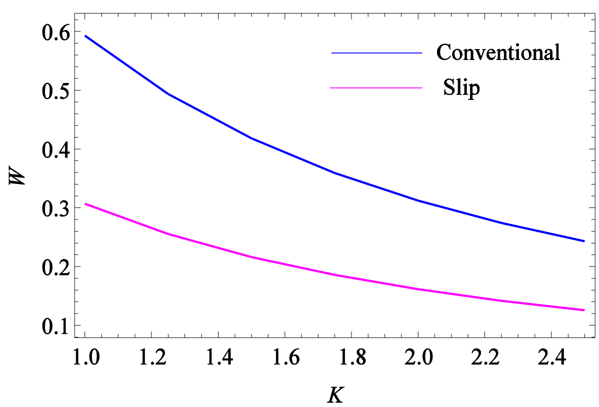

Figure 9: Variation of W with A for conventional bearing and finite slider bearing with slip using micropolar fluid.

Figure 8 shows the variation of load capacity W with dimensionless slip parameter A, different values of characteristic length L with coupling number $N = 0.3$. It indicates that bearing load increases with increasing value of L as compared with the Newtonian case ($L = 0$).The figure shows that for a Newtonian lubricant the load capacity first decreases with increase in dimensionless slip parameter up to a slip parameter of 20. Further increase in the slip parameter does not bring any major improvement on the load capacity. Fig. 9 shows the variation of load capacity with aspect ratio for finite slider bearing with and without slip. It is observed from this figure that load carrying capacity is maximum when aspect ratio is 1. The graph shows that load capacity decreases with aspect ratio in the range from 1.0 to 2.5.

4.0 CONCLUSIONS

This work analyzes the hydrodynamic performance characteristics of finite slider bearing with effect of slip/no slip surface using micropolar fluid. The governing equations which are expressed in the non-dimensional form are discretized using FDM which finally have been solved for pressure distribution using appropriate boundary conditions. According to the results obtained the following conclusions are drawn:

(a) As mesh size changes it has been observed that improvements in accuracy of the results were significant.
(b) The pressure values have been changed considerably with the form of grid refinement analysis.
(c) The load carrying capacity for a bearing with no slip surface produce higher load support as compared to slip surface.
(d) The effect of micropolar fluid is to increase the load carrying capacity as compared to Newtonian case.

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