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A Study of Static Performance of Fixed Inclined Slider Bearings – A Power Law Model

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ABSTRACT: In this paper, the performance of a fixed inclined slider bearing whose surfaces are lubricated by a non Newtonian power law lubricant is investigated numerically. Based on the power law model, the modified Reynolds equation is derived and solved using the finite element method. The effect of flow index on pressure distribution, load capacity, maximum pressure is presented graphically. @JASEM

Keyword: power law, slider bearing, finite element, load capacity, pressure

Traditionally, most lubricants are Newtonian in nature, but with the development of modern machines, polymer oil, especially engine oils for vehicles is usually added to the lubricating oil in order to reduce friction loss of the machine parts.

These additives impart non Newtonian behavior on the lubricant. Since the conventional micro – continuum theory cannot accurately describe the flow of these kinds of fluids, various micro – continuum theories have been proposed. (Lin and Yu, 2004). A number of models have been used to characterize the non Newtonian behaviour of lubricants. They include power law, visco-plastic and pseudo – plastic models. Stokes (1966) proposed the simplest micro - continuum theory which permits the presence of couple stresses, body couples and non symmetric tenors

A number of researchers have studied the hydrodynamic lubrication problem with power law fluid model. Li et al (2006) studied partially wetted bearings lubricated with non Newtonian power law fluids as well as the Navier –slip boundary condition. They showed the variation of load capacity for various flow indices and slip lengths. Nagaraju et al (2003) studied a hole entry hybrid journal bearing system and considered the combined influence of surface roughness and various power law indices on the performance of the bearing.

The effects of stepped film-thickness on the various characteristics of squeeze films conical bearings and hydrostatic step seals, using a power law fluid as lubricant was investigated by Shukia and Isa (1974) They showed that the load capacity of squeeze film bearings decrease and those of conical bearings and step seals increase, with the increase in the step height. With a hydrostatic step seal, the load capacity increases as the flow behaviour index of the power law fluid increases. Lin et al (2006) presented the rheological effects of an isothermal incompressible non-Newtonian laminar lubricating film on the steady and dynamic characteristics of finite slider bearings in the absence of fluid inertia and cavitation. They concluded that the effects of non-Newtonian power-law lubricants on the bearing characteristics (load capacity, dynamic stiffness and damping coefficients) are more pronounced when the bearing width becomes large. Mongkolwongrojn et al (2007) compared the performance characteristics of journal bearings lubricated with non-Newtonian Carreau lubricants with the characteristics of journal bearing with non-Newtonian lubricant using the Power-law model.

The lubrication of a conventional, finite width plane bearing using a power-law, non-Newtonian lubricant, was studied by Buchoiz (1986). He obtained analytical expressions for the bearing pressure, load and friction. He compared numerical solutions for the bearing performance metrics with analytical solution using a range of bearing aspect ratios and power law indices.

In this paper, we present the effect of power law fluids on the static performance characteristics of inclined slider bearings. Based on the power law model, the modified Reynolds equation accounting for non-Newtonian power law fluid is derived and solved numerically using the finite element method. The variation of some performance metrics with flow index is presented.

MATHEMATICAL EQUATIONS

As depicted in figure 1, the slider bearing is composed of a bottom plane surface which moves with a velocity u_b, and a fixed inclined upper surface. The oil film thickness at any point is given by equation (1)

$$h = h_0 + s_h\left(1 - \frac{x}{L}\right) \qquad (1)$$

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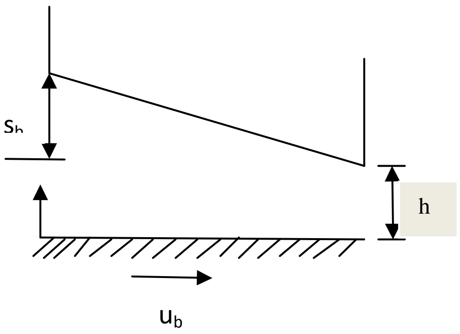

Fig 1: Geometry of a plane slider

Introducing the following dimensionless quantities $H = \frac{h}{s_b}$, $H_0 = \frac{h_0}{s_b}$ and $X = \frac{x}{L}$, equation (1) can be written as shown in equation (2). H_0 is the dimensionless film thickness at the outlet of the bearing, L is the length of the bearing, h_0 is the outlet film thickness, X is the dimensionless coordinate, h is the film thickness and H is the dimensionless film thickness.

$$H = H_0 + 1 - X \qquad (2)$$

The lubricant is assumed to be a non Newtonian power law lubricant. In addition, the lubricant flow is assumed to be steady, laminar and the viscosity and density are assumed to be pressure independent. The fluid inertia and body forces are assumed to be negligible as well as the pressure across the film thickness. The viscosity equation of the power law fluid is expressed as

$$\eta = m\left(\frac{\partial u}{\partial z}\right)^{n-1} \qquad (3)$$

In equation (3), m is known as the consistency index, $\frac{\partial u}{\partial z}$ is the shear rate and n is the flow index. The conditions of $n > 1$, $n = 1$ and $n < 1$ correspond to a dilatants fluid, Newtonian fluid and pseudoplastic fluid respectively. Based on the hypothesis made above, Navier Stokes equation can be written in simplified form as

$$\frac{\partial p}{\partial x} = \frac{\partial}{\partial z}\left(\eta \frac{\partial u}{\partial z}\right) \qquad (4)$$

The boundary conditions can be stated as

$$z = 0,\ u = u_b \text{ and } z = h,\ u = 0 \qquad (5)$$

The velocity distribution can be obtained by using a perturbation technique (Chu et al, 2007) and is given as shown in equation (6).

$$u = u_b\left(1 - \frac{z}{h}\right) - \frac{1}{2nm}\left(\frac{h}{u_b}\right)^{n-1}\frac{\partial p}{\partial x} z(h-z) \qquad (6)$$

The continuity equation is given by

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial z} = 0 \qquad (7)$$

Substituting the value of u from equation (6) into equation (7) and integrating across the film, the steady state modified Reynolds equation for power law fluid can be written as shown in equation (8). It is imperative to mention that when $n = 1$ and $m = \eta$, equation (8) reduces to the classical Reynolds equation considered by Oladeinde and Akpobi [9]

$$\frac{d}{dx}\left(\frac{h^{n+2}}{mn}\frac{\partial p}{\partial x}\right) = 6u_b^n \frac{dh}{dx} \qquad (8)$$

In order to present equation (8) in non dimensional form, we introduce the following dimensionless quantities

$$H = \frac{h}{s_b}, \qquad H_m = \frac{h_m}{s_b}, \qquad P = \frac{ps_b^2}{\eta u_b L}$$

Equation (8) becomes

$$\frac{d}{dX}\left(\frac{H^{n+2}}{\xi^{n-1}}\frac{dP}{dX}\right) = 6n\frac{dH}{dX} \qquad (9)$$

where

$$\xi = \left(\frac{m}{\eta}\right)^{\frac{1}{n-1}}\frac{u_b}{s_b}$$

The boundary conditions now become

$$P = 0 \text{ at } H = H_0 + 1 \qquad (10a)$$

$$P = 0 \text{ at } H = H_0 \qquad (10b)$$

Solution Methodology: The finite element model of the governing equation (9) is first obtained. The finite element model is shown in equation (11)

$$\sum_{j=1}^{2}\int_e\left(\frac{d\phi_i}{dX}\frac{H^{n+2}}{\xi^{n-1}}\frac{d\phi_j}{dX}\right)P_j dX = \int_e 6n\frac{dH}{dX}\phi_j dX + \left[W_i\frac{H^{n+2}}{\xi^{n-1}}\frac{dP}{dX}\right] \qquad (11)$$

The bearing domain is divided into a mesh of one dimensional quadratic element. Stiffness matrices are computed for each element in the domain. The element stiffness matrices are then assembled and boundary condition subsequently imposed. The resultant system of equations is solved sing Gauss seidel iteration method with a convergence criterion of 0.00001 to obtain the pressure solution.

The dimensionless load capacity (W) per unit length is obtained by integration of the stream wise pressure as shown in equation (12)

$$W = \int_0^1 PdX \qquad (12)$$

The volumetric flow rate per unit width is computed by using the expression shown in equation (13). The pressure gradient in each element is obtained by using a forward finite difference method on two consecutive nodal pressures.

$$Q = \frac{H}{2} - \frac{H^{n+1}}{12n} + \xi^{1-n}\frac{dP}{dX} \qquad (13)$$

RESULTS AND DISCUSSION

Based on the formulations described above, the effect of the bearing structural and rheological parameters on the steady state performance of the bearing under consideration was investigated. Figure 2 shows the influence of different flow indices, on the pressure distribution at $\xi = 12.0$ and $H_o = 1.0$. Figure 2 shows that the greater the flow index, the greater the pressure distribution in the oil film of the slider bearing. This finding is attributable to the dependence on the flow index of the equivalent viscosity. The lesser the flow index, the lesser the resultant equivalent viscosity which causes decreased normal load on the bearing. It can also be deduced from figure 2 that the maximum pressure increases with flow index. Figure 3 shows the variation of the maximum pressure with flow index with ξ fixed at two different values and H_o fixed at 1.0. Figure 3 shows that the maximum pressure increases with flow index in an exponential fashion irrespective of the value of ξ. However, it can be observed that at low flow index regimes $(n < 1)$ which describe pseudo – plastic fluid flow that are characterized by linearity at extremely low and extremely high shear rates in inclined slider bearings, there is no significant difference in the maximum pressure with increase in ξ. At $n = 1$, corresponding to Newtonian fluid flow, the maximum pressure appears to be independent of the value of ξ. Specifically, a maximum dimensionless pressure equal to 0.26 is developed in the inclined slider at $\xi = 10$ and $\xi = 12$ respectively, with H_o fixed ar 1.0. Increasing the flow index beyond unity characterizes dilatants fluids which exhibit an increase in apparent viscosity with increasing shear rate. Numerical experiments demonstrate as shown in figure 3 that maximum pressure generated is greater for higher dimensionless outlet film thickness H_o

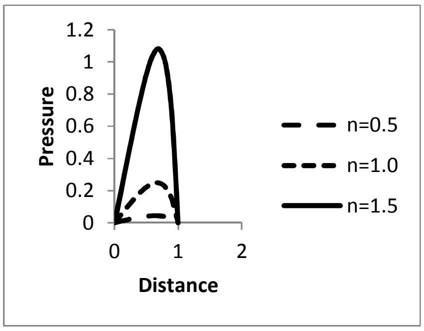

Fig 2: Variation of pressure with distance for different flow indices.

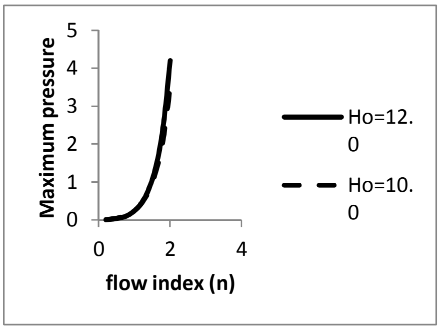

Fig 3: Variation of maximum pressure with flow index

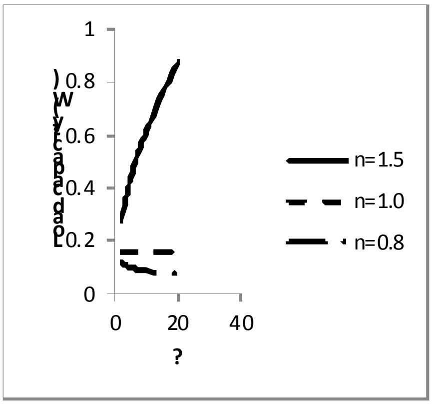

Fig 4: Variation of load capacity with ξ for different flow indices (n)

The influence of ξ on dimensionless load capacity (W) is shown in figure 4 for different values of n with H_0 fixed at 1.0. The plot shows that for $n > 1$, greater bearing load is developed as ξ increases. On the contrary for $n < 1$, the load capacity decreases with increase in ξ. The magnitude of the load capacity appears to be independent of ξ for $n = 1$ which represents the Newtonian lubricant case.

The effect of flow index on load capacity is depicted in figure 5 for different values of H_0. The graph illustrates that in general, the load capacity increases with decrease in H_0 for a given flow index n. The increase in load capacity with decrease in H_0 is more pronounced at higher flow indices. The increase in load capacity with decrease in H_0 is attributable to increased wedge effect. Numerical experiments reveal that the volumetric flow rate increases with H_0 with n and ξ fixed. Computations also put forth that with increase in the flow index, the volumetric flow rate decreases for a fixed value of H_0

Conclusion: The effect of power law fluid on the performance of a fixed inclined slider bearing has been investigated using the finite element method. It has been concluded that the pressure distribution, maximum pressure and load capacity increases as the flow index increases. Computations also put forth that with increase in the flow index, the volumetric flow rate decreases for a fixed value of H_0.

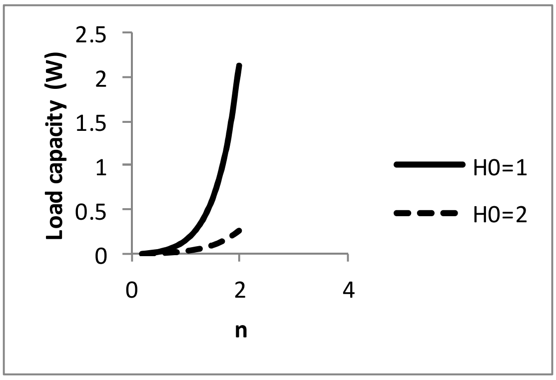

Fig 5: Variation of Load capacity with flow index for different values of H_0 with ξ fixed at 10.0

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