AN EFFICIENT STUDY OF BEARINGS WITH LUBRICANTS

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Abstract: In tribological studies, the incorporation of partial slip surfaces in single-grooved slider and journal bearings has been analysed to understand its impact on pressure distribution and shear stress. Partial slip refers to regions on the bearing surface where the lubricant experiences reduced frictional resistance, altering the fluid flow dynamics. Research indicates that introducing a single groove immediately followed by a partial slip region on the stationary surface leads to an increase in pressure distribution compared to conventional no-slip bearings. This configuration enhances the hydrodynamic pressure, thereby potentially improving the load-carrying capacity of the bearing. Additionally, the shear stress behaviour is affected; it increases before the slip/no-slip interface and decreases in the no-slip region. These findings suggest that strategically placing partial slip regions in conjunction with grooves can optimize bearing performance by modifying pressure and shear stress distributions.

I. INTRODUCTION

The classical Reynolds equation, which governs the behavior of lubricated bearings, assumes a no-slip condition, meaning that the lubricant adheres completely to the surfaces in relative motion. However, recent advancements have demonstrated that slip can occur on specially prepared hydrophobic surfaces, such as those coated with mica and lubricated with water. Spikes [1] analyzed the effects of wall slip on the hydrodynamic performance of half-wetted bearings and found that slip conditions can significantly alter the pressure distribution and frictional behavior of bearings.

Wall slip is generally characterized using two models: the slip length model, which applies at low shear rates, and the limiting shear stress model, which is relevant at high shear rates. In a half-wetted bearing, where fluid exhibits a no-slip condition on the moving surface but a slip condition on the stationary surface, the result is higher hydrodynamic pressure and lower friction. This phenomenon occurs because fluid entrainment enhances pressure, while reduced shear stress at the slip interface minimizes friction. On the other hand, in bearings where the fluid has a slip boundary condition against the moving surface and a zero critical shear stress, there is no significant fluid motion, leading to an inability to support a load. This condition indicates that slip on the stationary surface can be beneficial, whereas slip on the moving surface may compromise bearing performance.

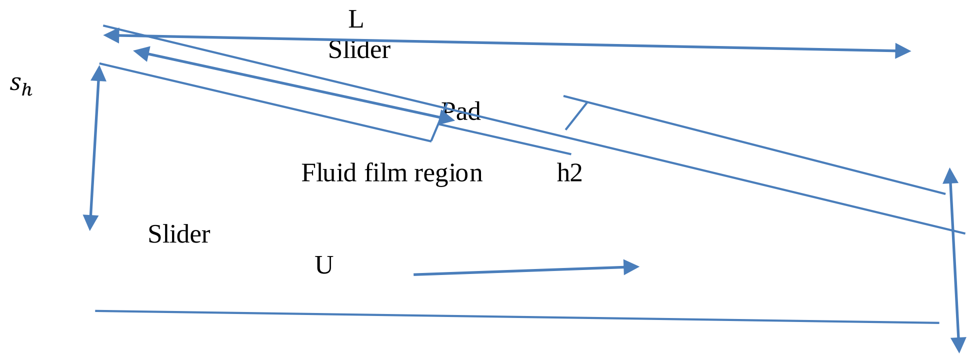
Salant and Fortier [2,3] conducted numerical analyses of slider and journal bearings using a modified slip-length model. Their research demonstrated that bearings with heterogeneous slip/no-slip surfaces exhibit higher load-carrying capacity and lower friction. Similarly, Wu et al. investigated slider bearings with mixed-slip surfaces and found that various wedge configurations (convergent, parallel, and divergent) can contribute to hydrodynamic load support.

In modern tribological applications, hydrodynamic bearings integrated with micro-electromechanical systems (MEMS) offer innovative solutions for lubrication using textured surfaces with partial slip. These developments have opened new avenues for designing high-efficiency bearings with optimized load support and reduced energy dissipation.

In this study, pressure and shear stress in single-grooved slider and journal bearings under steady-state conditions are derived from the governing equations using a quadratic velocity distribution along the sliding direction. The analysis considers partial slip on the stationary surface of the single-grooved slider bearing. This approach allows for a more accurate representation of fluid behavior in practical applications.

The presence of a single groove in conjunction with a partial slip region can enhance the pressure distribution within the bearing. This effect is particularly significant in slider and journal bearings, where the hydrodynamic pressure plays a crucial role in load-carrying capacity. The reduction in friction due to the slip condition on the stationary surface further improves bearing efficiency, making it a promising design strategy for advanced tribological systems.

Overall, this study contributes to a deeper understanding of how slip conditions affect the lubrication performance of bearings. By leveraging slip boundaries strategically, engineers can design bearings with improved efficiency, higher load support, and lower frictional losses, ultimately advancing the field of fluid film lubrication.



Mathematical Formulation:

A Two-dimensional analysis of single grooved partial slip slider bearing are considered in the analysis. Assuming that the pressure are vary along the film thickness and considering that the pressure is a function of sliding direction(x, y) the momentum equations are simplified as

$$\frac{\partial p}{\partial x} = \eta\left(\frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2}\right) \text{ -- --(1)}$$

The shear stress can be written as

$$\tau_{xyz} = -\eta\left(\frac{\partial u}{\partial y} + \frac{\partial u}{\partial z}\right) \text{ ---(2)}$$

The quadratic velocity distribution of the fluid in the x direction is of the form [4,5]

$$u = U\,(C_1 y^2 + C_2 y + C_3 + C_4 z^2 + C_5 z + C_6) - - - - - - - - - - (3)$$

Considering the slip region on the nonrotating surface of the slider bearing .the boundary condition for the steady flow at the sliding surface(z, y=0) and the stationary surface(y=h,z=h) are as follows [2,3]:

At y=0, u=U ,z=0,u =U and at y=h, u=$-\alpha\eta\left(\frac{\partial u}{\partial y} + \frac{\partial u}{\partial y}\right)$ ----------------(4)

Substituting the boundary condition in eq.(4) in Eq.(3) yields the velocity distribution of the fluid in the x direction as

$$u = U\,[C_2\left(y - \frac{y^2(h+\alpha\eta)}{h(h+2\alpha\eta)}\right) + 1 - \frac{y^2}{h(h+2\alpha\eta)} + C_3\left(z - \frac{yz^2(h+\alpha\eta)}{h(h+2\alpha\eta)}\right) + 1 \quad - - - (5)$$

Three boundary conditions are required for u in Eq.(3) since the velocity distribution is quadratic .The third boundary condition or the steady flow is as follows .

$$\int_0^h \int_0^h u\,dy\,dz = q \text{ ------------------------------------- ---- (6)}$$

Substituting the value of u from Eq.(5) in the boundary condition of Eq.(6) and simplifying for constant ,result in

$$C_2 = \frac{q(h-\alpha\eta)}{Uh(1+2\alpha\eta)} - \frac{4(h+\alpha\eta)}{h(1+4\alpha\eta)}\ , C_3 = \frac{q(h+\alpha\eta)}{Uh(h+2\alpha\eta)} + \frac{(h-3\alpha\eta)}{h(1-4\alpha\eta)} \text{ ----------- -(7)}$$

Substituting Eq. (5) and (7) in Eq.(1) results in the pressure gradient distribution in the slider/journal bearing as

$$\frac{\partial p}{\partial x} = \eta U\left(\frac{q(h-\alpha\eta)}{h^3(h+2\alpha\eta)} - \frac{q(h^3+\alpha\eta)}{Uh^6(h-\alpha\eta)}\right) \text{ ----------------------------- -------- (8)}$$

The pressure distribution can be obtained from Eq.(8) as

$$P(x) = pIx{=}0 + \int_0^x \frac{dp}{dx}dx \quad - - - - - - - - - - - - - - - \quad - \quad (9)$$

Substituting Eq. (5) and (7) in Eq. (2) the shear stress in the slider bearing at y=0 is obtained as

$$\tau_{xy}|y = 0 \ = -\eta U\left(\frac{q(h-\alpha\eta)}{h^2(h+2\alpha\eta)} - \frac{4(h^2-3\alpha\eta)}{h(h-4\alpha\eta)}\right) \text{ ------------------------- (10)}$$

Analysis of Bearing:

The Film thickness expression of the plain slider bearing (un grooved) for $0 \le x \le L$ *is* shown in Eq. (11) and the film thickness of the single grooved bearing is h-h_g

$$h = h_2 - s_h \text{ ------------------------------------- ------------ (11)}$$

Equation (11) can be rewritten in non-dimensional form as

$$H = H_0 + (1 - X + Y) \text{ --------------------------------- ------- -(12)}$$

Using the no dimensional parameter for the slider bearing we will get pressure gradient distribution from eq.(8) is simplified as

$$\frac{dp}{dX} = \frac{h_1^2}{L}\left(\frac{(1+2A)}{H^3 s_h^2(1-8A)} - \frac{24Q(1+A)}{H^6 {s_h}^3(1+4A)}\right), \frac{dp}{dY} = \frac{h_1^2}{L}\left(\frac{(1+2A)}{H^6 s_h^2(1-4A)} - \frac{24Q(1+12A)}{H^3 {s_h}^3(1-4A)}\right. \text{- --(13)}$$

The boundary condition for no dimensional pressure at the inlet(X=0) and outlet(X=1) of the slider bearing are

$$P|(X = 0) = P\,|(X = 1) = 0 \text{ ---------------------------------------(14)}$$

For $0 \le X \le 1$ the nondimensional pressure in the single –grooved partial slip slider bearing can be written as

$$P(0 \le X \le X_s) = P|(X, Y = 0) + \int \int_0^X \frac{h_1^2}{L}\left(\frac{(1-2A)}{H^3 s_h^2(1+4A)}\right. - \int \int_0^X \frac{h_1^2}{L}\frac{24Q(1+A)}{H^3 {s_h}^3(1+4A)} \text{ --(15)}$$

$$P(X_s \le X, Y \le X_s + X_g) =$$

$$P|(X = X_s) + \int \int_{X_s}^X \frac{h_1^2}{L}\left(\frac{1}{(H+H_g)^2 s_h^2(1+4A)} - \int_{X_s}^X \frac{h_1^2}{L}\frac{24Q}{(H+H_g)^3 {s_h}^3(1+4A)}\right. \text{ ---(16)}$$

$$P(X_s + X_g \le X, Y \le 1) = P\Big|(X, Y = X_s + X_g) + \int \int_{X_s - X_g}^X \frac{h_1^2}{L}\frac{1}{(H)^2 s_h^2} - \int \int_{X_s - X_g}^X \frac{h_1^2}{L}\frac{24Q}{(H)^3 {s_h}^3} \text{ –(17)}$$

Where H= h/s_h , $H_1 = h_2/s_h$, X= x/L Y=y/L, A= $\frac{\alpha\eta}{s_h}$ $\alpha = Slip\ Coificient\ \eta = fluid\ viscocity$

Substituting the boundary condition from eq.(14) in equations 15,16,&17 we get

$$Q = \frac{\int \int_0^{X_s}\left(\frac{(1+2A)}{H^2 s_h^2(1+4A)}dX + \int \int_{X_s}^{X_s+X_g}\frac{1}{(H+H_g)^2 s_h^2}dX + \int \int_{X_s+X_g}^{1}\frac{1}{(H)^2 s_h^2}dX\right.}{2\int \int_0^{X_s}\left(\frac{(1+2A)}{H^3 s_h^3(1+4A)}dX + 2\int \int_{X_s}^{X_s+X_g}\frac{1}{(H+H_g)^3 s_h^3}dX + \int \int_{X_s+X_g}^{1}\frac{1}{(H)^3 s_h^3}dX\right.} \text{ --------------------(18)}$$

We consider uniform film thickness for the slider bearing (H=1) for $0 \le X, Y \le 1$,the value of Q will be obtained from Eq.(18) Hence

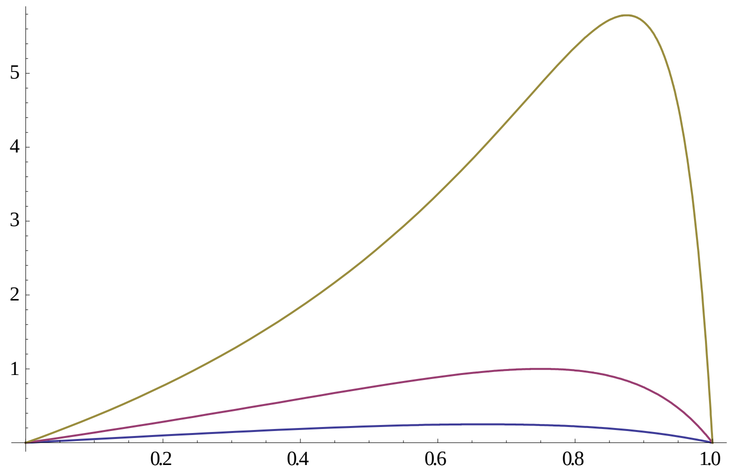
$$Q = \frac{\frac{(1+A)}{s_h^3(1+4A)}(X_s+X_g) + \frac{1}{(1+H_g)^2 s_h^2}(X_g) + \frac{1}{s_h^2}(1 - X_s + X_g)}{2\{\frac{(1++X_s 4A)}{s_h^4(1+6A)}(X_s) + \frac{1}{(1++X_s H_g)^6 s_h^3}(X_g) + \frac{1}{s_h^6}(1\_ + X_s - X_g)\}} \text{ --------------------------------(19)}$$

In the case of partial slip slider bearing with uniform film thickness for $0 \le X \le 1$

Results:

The no dimensional slip length A is zero in no slip regions we assume the value of A are in slip regions are A=1,100 .The parameters used in the analysis of slider bearing are s_h=0.6 and 0.4 the value of $X_s = 0.9$ *and* $X_g = 0.6$.

From Eq.10 we will find the value of P in terms of X and other non-dimensional parameter then by using the value of Q we will find the different pressure distribution.



Nomenclature:
H_g = depth of groove
L= length of slider bearing
τ_{xy} = shear stress
P = pressure distribution

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