A study of deformable rough surfaces on finite slider bearing

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Abstract:

A numerical solution to the modified Reynolds' equation for a finite slider bearing lubricated with micropolar fluid, focusing on the computational aspects of bearing performance has been presented. Governing equations are solved using a finite difference scheme.This study examines the effects of micropolar and slip parameters on the slider bearing. The impact of slip boundary conditions on the pressure distribution along the sliding surface is analyzed using the Reynolds' model. Two – dimensional modified Reynolds' equation effectively predicts the lubrication performance with boundary slip in sliding contact, as evidenced by the results. Pressure and load capacity are provided, demonstrating that both pressure and load carrying capacity are lower in the presence of slip compared to the no slip case. The study of deformable rough surfaces in finite slider bearings provides valuable insights into improving the design and functionality of such systems. By combining theoretical models (like contact mechanics), numerical methods (FEM), and experimental approaches, engineers can better understand and predict the behavior of these bearings in real-world applications. Addressing surface roughness, deformation, lubrication, and wear will ultimately lead to more efficient, durable, and reliable bearing systems.

Keywords:

Reynolds' Expression, Pressure Distribution, Load Carrying Capacity, Micropolar fluid, SlipParameter, Deformable Roughness

Nomenclature:

LCC	Load Carrying Capacity
MPF	Micropolar Fluid
DRS	Deformable Rough Surfaces

SP	Slip Parameter
PD	Pressure Distribution
MEMS	Micro Electro Mechanical System is a miniature machine that has both mechanical and electronic components. (mm to micro meter, hair)
FDM	Finite Difference Method
ϕ	Permeability of porous facing; The common unit is the Darcy (D) [about 10^{-12} m^2];
H_0	Minimum film thickness of porous facing (m)
σ	Standard deviation (Standard deviation in units of feet.)
ε	Skewness (Skewness has no units: it's a pure number, like a z-score.).
α	Variance (Variance always has squared units.).
δ	Deformation (Deformation is the change in the shape or size of an object. It has dimension of length with SI unit of meter (m).)
σ^*	Non-dimensional standard deviation $=\sigma / c$
ε^*	Non-dimensional skewness $=\varepsilon / c^3$
α^*	Non-dimensional variance $= \alpha / c$
δ^*	Non-dimensional deformation $= \delta / c$
ψ	$12\phi H_0 / c$ [The porosity is conventionally given the symbol φ, and is expressed either as a fraction varying between 0 and 1, or a percentage varying between 0% and 100%. Sometimes porosity is expressed in 'porosity units', which are the same as percent (i.e., 100 porosity units (pu) = 100%).]

Introduction and literature review:

The Reynolds lubrication theory has become an efficient tool in both the analysis and design of lubricated contacts. In the derivation of the Reynolds equation, it is assumed that there is no boundary slip at the liquidsolid interface, known as the no–slip boundary condition. However, on a microscale, advancements in micrometer measurement technology have made it possible to observe boundary slip of fluid flow over a solid surface, challenging the conventional no-slip boundary condition. Surface roughness is one of the important properties that must be considered in any material to determine tribological properties. Researchers have studied various techniques to determine these properties using various principles, Liu Jian et.al., studied the surface roughness measurements using a color distribution statistical matrix. This method is based on the overlap degree of the color image which has relatively high accuracy and a relatively wide measurement range to a certain degree of the brightness to the light source and the texture direction. In a different study F. Luk et. al., studied another method based on a microcomputer-based vision system to analyze the pattern of scattered light from the surface to derive a roughness parameter. The roughness parameters were obtained for a few tool-steel samples which were ground to different roughness. Under these conditions, the classical Reynolds equation is no longer applicable. In lubricated MEMS, proper lubrication is crucial in reducing liquid stiction and has received significant attention in the literature (Henck, 1997; Spikes, 2003). Despite the powerful utility of the classical Reynolds equation in bearing analysis and design, researchers have extended it to account for boundary slip in order to improve the design and analysis of fluid film lubricated contacts.Numerous studies have examined hydrodynamic lubrication considering velocity slip at the surfaces. Rao (2017) studied the Reynolds equation for two symmetrical surfaces with slip at the bearing surfaces, focusing on the effects of velocity slip and developing expressions for pressure in the thin film, load capacity, and coefficient of friction. Numerical analysis revealed that a high viscous layer at the periphery moves the cavitations position towards the center, while slip moves it away from the center. Rao and Prasad (2004) investigated the effect of velocity slip on load capacity in journal bearings, finding that load capacity decreases with slip and that the coefficient of friction decreases with a high viscous layer but increases with slip. Beavers and Joseph (1967) discussed the effect of slip for an incompressible fluid. Significant research has explored the influence of boundary slip on hydrodynamic lubrication, particularly in slightly parallel sliding configurations. Salant and Fortier (2004) and Fortier and Salant (2005) conducted early work on artificial complex slip/no−slip lubricated parallel sliding devices. Shah and Bhat (2002) theoretically studied the effect of velocity slip on a porous inclined slider bearing lubricated with ferrofluid, deriving expressions for pressure, load capacity, friction, and coefficient of friction, and finding that increased slip parameter decreases load capacity. Kalavathi et al. (2014) examined the effect of surface roughness on porous journal bearings with heterogeneous surfaces, using the Christensen stochastic theory of hydrodynamic lubrication to derive a generalized Reynolds equation considering porosity. They observed that load carrying capacity increases and coefficient of friction decreases in heterogeneous slip/noslip surface patterns of porous narrow journal bearings. Oladeinde and Akpobi (2010) investigated the effect of slip surfaces on finite slider bearings using couple stress fluid.Kalavathi et al. (2016) numerically studied the influence of roughness on finite porous journal bearings considering slip/ no slip surfaces, showing that these conditions increase pressure and load distribution. Despite extensive research, none have numerically studied the influence of slip on micropolar fluids. Recently, Pentyala and Agarwal, (2018) discussed the problem based on effect of slip / no slip on finite slider bearing using non − Newtonian micropolar fluid. In this article they found that the effect of micropolar fluidis to increase the load carrying capacity as compared to Newtonian case. Hence, the aim of the present work is to analyze the effect of deformable rough surfaces on finite slider bearing with the help of micropolar fluid.

Mathematical formulation:

The fundamental equations that describe the flow of micro polar lubricants, considering the standard assumptions of lubrication theory for thin films, are provided by Henck (1997) and Spikes

(2003).

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

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

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

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

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

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

In this context, u, v, w represent the velocity components in the x, y, z directions respectively. The variable p denotes the pressure. The term v_3 indicates the micro polar velocity component. Here, μ is the traditional viscosity coefficient, γ represents an additional viscosity coefficient and χ is the additional spin viscosity coefficient for micropolar fluids.

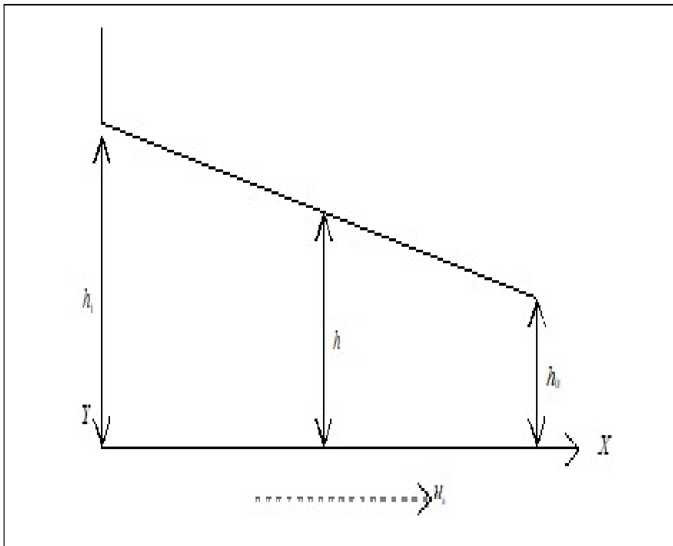

Figure (1): Schematic of bearing configuration.

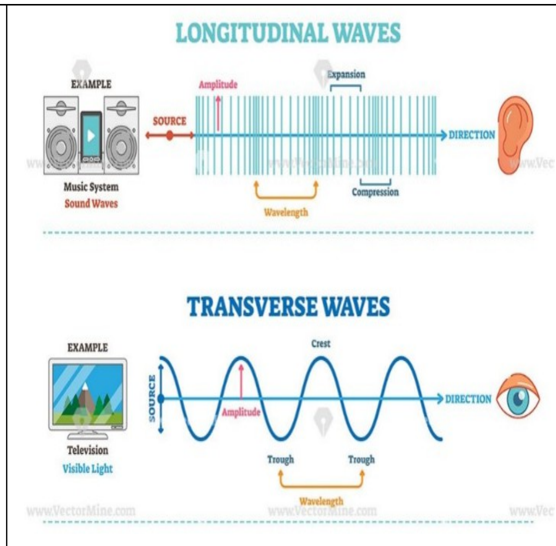

Figure (2): Roughness pattern.

Figures (1) and (2) depict the geometry and physical configuration of a slider bearing. It comprises two surfaces separated by a lubricant film. The lower surface moves in its own plane at a constant velocityU_s, while the upper solid surface remains stationary. The inlet film thickness is denoted as h_1, and the outlet film thickness is denoted as h_0.

The relevant boundary conditions for the velocity and micro rotational velocity components are as follows:

(a) At the lower surface (i. e. y=0)

$u = u_s,\ v = w = 0,\ v_1 = v_3 = 0$ (7)

(b) At the upper surface (i.e. y=h)

$u = -\alpha\mu\ (\partial u/\partial y),\ v = w = 0,\ v_1 = v_3 = 0$ (8)

Under these boundary conditions, slip occurs. When the slip length is set to zero, the boundary conditions simplify to the traditional no-slip case. Solving equations (1), (3), (4), and (5) with the boundary conditions specified in equations (7) and (8) leads to a reduction of the problem to the dimensional modified Reynolds' equation.

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

where

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

$g(N, l, h, \alpha u) = A_{11} / A_{22}$ and $f(N, l, h, \alpha u) = A_{33} / A_{22}$ (11)

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

$$A_{22} = 2N^2\cosh\left(\frac{N\{a(h)\}^{1/3}}{l}-1\right)-\frac{N}{l}\left[\{a(h)\}^{1/3}+(1-N^2)\alpha\mu\right]\sinh\left(\frac{N\{a(h)\}^{1/3}}{l}\right) \tag{13}$$

$$\begin{aligned}A_{33} &= 2N^2\{a(h)\}^{1/3}\cosh\left(\frac{N\{a(h)\}^{1/3}}{l}-1\right)\\ &-\frac{N\{a(h)\}^{1/3}}{l}\left[\{a(h)\}^{1/3}+2(1-N^2)\alpha\mu\right]\sinh\left(\frac{N\{a(h)\}^{1/3}}{l}\right)\end{aligned} \tag{14}$$

where

$$a(h) = (h + p_a p'\delta)^3 + 3(\sigma^2 + \alpha^2)(h + p_a p'\delta) + 3(h + p_a p'\delta)^2\alpha + 3\sigma^2\alpha + \alpha^3 + \varepsilon + 12\phi H_0 \quad (15)$$

List of dimensionless terms:

$$X^* = \frac{x}{l_x} \qquad Y^* = \frac{y}{l_y} \qquad \sigma^* = \frac{\sigma}{h_0} \qquad \alpha^* = \frac{\alpha}{h_0} \qquad \varepsilon^* = \frac{\varepsilon}{h_0^{\,3}}$$

$$p^* = \frac{p}{p_a} \qquad H = \frac{h}{h_0} \qquad \delta^* = \frac{\delta}{h_0} \qquad p_1^* = p_a p' \qquad U = \frac{6\mu u_s l_x}{h_0^{\,2} p_a}$$

$$L_1 = \frac{l_x}{l_y} \qquad P^* = \frac{h_2^{\,3}\, p}{\mu_1\, u\, B^2} \qquad W^* = \frac{h_2^{\,3}\, w}{\mu_1\, u\, B^4} \qquad \overline{\psi} = \frac{D_c^{\,2}\, l_1}{h^3} \qquad A = \frac{\alpha\mu}{h_0}$$

$$A^* = (1 + p^*\delta^*)^3 + 3(1 + p^*\delta^*)^2\alpha^* + 3(1 + p^*\delta^*)(\sigma^{*2} + \alpha^{*2}) + 3\sigma^{*2}\alpha^* + \alpha^{*3} + \varepsilon^* + 12\,\overline{\psi}$$

Employing the discussion for the stochastic modeling of transverse roughness given by Christensen and Tonder, the expression for the film thickness h(x) of the lubricant film is considered to be,

$$h(x) = \overline{h}(x) + h_s$$

where $\overline{h}(x)$ denotes the mean film thickness and h_s is the deviation from the mean film thickness characterizing the random roughness of the bearing surfaces. The h_s derived by the probability density function deviation

$$f(h_s) = \begin{cases} 1.09375c^{-7}\left(c^2 - h_s^{\,2}\right)^3; -c \le h_s \le c \\ 0, elsewhere \end{cases}$$

where c is the maximum deviation from the mean film thickness. The details of the mean α, the standard deviation σ and the parameter ε which is the measure of symmetry of the random variable, h_s are considered from the discussion of Christensen and Tonder. With making use of above all parameters and non – dimensional terms in equation (9) one gets the modified form of deformable Reynolds' equation as:

$$\frac{\partial}{\partial X}\left[g(N,L,H,A)\frac{\partial P^*}{\partial X}\right] + \frac{\partial}{\partial Y}\left[g(N,L,H,A)\frac{\partial P^*}{\partial Y}\right] = U\,\frac{\partial f(N,L,H,A)}{\partial X} \quad (16)$$

where

$$g(N, L, H, A) = A^*_{11} / A^*_{22} \text{ and } f(N, L, H, A) = A^*_{33} / A^*_{22} \quad (17)$$

$$A^*_{11} = -4A^*N^2 + 24N^2L^2\{A^*\}^{1/3} - 4N^2\{A^*\}^{1/3}\left[2\{A^*\}^{2/3} + 6L^2 + 3\{A^*\}^{1/3}(1-N^2)A\right]\cosh\left(\frac{N\{A^*\}^{1/3}}{L}\right)$$
$$+\frac{N\{A^*\}^{1/3}}{L}\sinh\left(\frac{N\{A^*\}^{1/3}}{L}\right)\left[\{A^*\} + 12\{A^*\}^{1/3}L^2(1+N^2) + 4\left(\{A^*\}^{2/3} + 3L^2\right)(1-N^2)A\right]$$

(18)

$$A^*_{22} = 2N^2\cosh\left(\frac{N\{A^*)\}^{1/3}}{L} - 1\right) - \frac{N}{L}\left[\{A^*\}^{1/3} + (1-N^2)A\right]\sinh\left(\frac{N\{A^*\}^{1/3}}{L}\right) \tag{19}$$

$$A^*_{33} = 2N^2\{A^*\}^{1/3}\cosh\left(\frac{N\{A^*\}^{1/3}}{L} - 1\right)$$

$$-\frac{N\{A^*\}^{1/3}}{L}\left[\{A^*\}^{1/3} + 2(1-N^2)A\right]\sinh\left(\frac{N\{A^*\}^{1/3}}{L}\right) \tag{20}$$

To solve the modified dimensionless Reynolds' equation (16) and determine the film pressure distribution, the following boundary conditions are applied:

$$P^* = 0 \text{ at } X = 0, 1 \text{ and } Y = 0, 1 \tag{21}$$

The load per unit width is supported by the lubricant film, and its calculation involves integrating the lubricant film pressure. The load-carrying capacity is defined as the integral of the pressure profile over the surface area, expressed in dimensionless quantities.

$$W^* = \int_0^1\int_0^1 P^*\, dX\, dY \tag{22}$$

Findings:

- The dimensionless Reynolds' equation (16) is numerically solved using the FDM, subject to the boundary conditions specified in (21). The computational domain, defined by $0 < X < l_x$ and $0 < Y < l_y$, is divided into a grid with a uniform mesh size, ΔX and ΔY. The mesh sizes are given by $\Delta X = (l_x / n)$ and $\Delta Y = (l_y / y)$, where (n+1) represents the number of grid points.
- A uniform grid is applied on the slip face surface. For the simulations, a 40×40 mesh, which has been validated as gridindependent, was used despite the computational cost associated with increasing mesh sizes. To discretize the modified Reynolds' equation, a first-order central difference scheme was employed.
- The characteristics of a finite slider bearing with heterogeneous slip/noslip surfaces are

analyzed based on various non-dimensional parameters, such as the slip parameter A and coupling parameter N. These parameters characterize the interaction between linear and rotational motion due to the micro-motion of fluid or additive molecules.

- The parameter L represents the characteristic length, defined as L= (l / h_0), where L is the ratio of microstructure size to the minimum film thickness. As L approaches zero, the non-dimensional Reynolds equation simplifies to the Newtonian case.
- A grid independence study was crucial for obtaining an accurate numerical model. Significant changes in results were observed when increasing the grid size from 10×10 to 50×50.
- The pressure distribution for different values of A with fixed N, L, and U using a 41×41 mesh grid, suggests that the bearing surface with slip exhibits lower pressure compared to the noslip case.
- Variation of load with sliding velocity U, showing a linear relationship and hence load carrying capacity is zero when U = 0 and the surface with slip boundary supports a lower load than the no slip surface.
- For a Newtonian lubricant, the load capacity initially decreases as the dimensionless slip parameter increases, reaching a minimum at a slip parameter of 20. Beyond this point, further increases in the slip parameter do not result in significant changes in load capacity.

Conclusions:

This work analyzes the hydrodynamic performance characteristics of finite slider bearing with effect of slip / no slip surface using micro polar 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:

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

Declaration of Competing Interest:

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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Materials to read:

A study of deformable rough surfaces in the context of finite slider bearings involves exploring how the microscopic and macroscopic surface characteristics affect the overall performance, efficiency, and wear of the bearing system. These surfaces, commonly encountered in mechanical systems such as sliding contacts or journal bearings, can significantly influence load distribution, friction, and lubrication dynamics. Here's an outline of key concepts and areas typically examined in such a study:

1. Introduction to Slider Bearings and Surface Roughness

- Slider Bearings: These are bearings in which a flat surface (the slider) moves over another surface (the bearing). They are designed to support a load and reduce friction during motion.
- Surface Roughness: In real-world applications, surfaces are never perfectly smooth. They exhibit roughness, which affects the contact mechanics between the sliding surfaces. This roughness can be modeled as periodic or stochastic variations, and it influences the load-bearing capacity and frictional forces.

2. The Impact of Surface Deformability

- Elastic Deformation: As the slider moves over the bearing, the contact points experience deformation. This deformation can be elastic or plastic depending on the material properties and the contact pressures.
- Plastic Deformation: If the applied load is high, the surface may undergo plastic deformation, which can alter the surface roughness and influence the friction and wear characteristics.
- Micro-Scale and Macro-Scale Deformation: Both micro-scale (on the level of individual asperities) and macro-scale (overall bearing surface) deformations are considered in such studies. They impact the load distribution and the performance of the bearing.

3. Contact Mechanics and Friction Models

- Hertzian Contact Theory: In the elastic regime, the theory of Hertzian contact is often used to describe the deformation and contact mechanics between the surfaces. It provides a relationship between the applied load and the contact area.
- Amontons-Coulomb Law of Friction: This is commonly used to describe the relationship between the normal and tangential forces at the contact interface, though it can be modified for rough, deformable surfaces.
- Asperity Interaction: For rough surfaces, the interaction between individual surface asperities (microscale protrusions or depressions) must be considered. The deformation and contact area between asperities can significantly affect the overall friction and load-bearing characteristics.

4. Finite Element Method (FEM) Simulation

- Modeling the Rough Surface: A numerical approach using FEM is often employed to simulate the deformable rough surface interactions in slider bearings. FEM allows the analysis of complex geometries and varying roughness profiles.
- Load Distribution: FEM can be used to calculate how the load is distributed across the contact surface, considering both the roughness and the deformation of the surfaces under different operating conditions.

5. Lubrication Effects

- Hydrodynamic Lubrication: If a lubricant is present, the roughness and deformation of the surfaces will influence the formation and behavior of the lubricant film. In this case, the lubricant film can be affected by both the surface asperities and the pressure distribution across the bearing.
- Elastohydrodynamic Lubrication (EHL): This theory models the interaction between the lubricant film and the deformed surfaces. It is particularly relevant in high-pressure conditions where both surface roughness and deformability play a significant role in the lubrication effectiveness.

6. Wear and Friction

- Tribological Effects: Deformable rough surfaces can lead to increased wear rates, as repeated sliding of asperities leads to material removal or surface smoothing. A detailed study would explore how the roughness and material properties influence the wear rate and lifetime of the bearing.

- Frictional Behavior: Friction is typically higher in rougher surfaces, and deformation can either increase or decrease friction depending on factors like contact pressure, lubrication, and the sliding velocity.

7. Experimental Methods and Validation

- Surface Profiling: Advanced tools like atomic force microscopy (AFM) or profilometers are used to characterize the roughness of the surfaces.
- Tribometers: Tribological testing devices, such as pin-on-disk or block-on-ring, can be used to experimentally assess the friction and wear behaviors of the bearing under controlled conditions.
- Correlation with FEM: Experimental results are often compared with numerical simulations (such as FEM models) to validate the proposed models.

8. Applications

- Mechanical and Aerospace Engineering: Finite slider bearings with rough, deformable surfaces are commonly used in applications such as engines, turbines, and machinery where load-bearing, friction, and wear are critical factors.
- Design Optimization: Understanding the behavior of deformable rough surfaces helps in optimizing bearing design for better performance, longer lifespan, and reduced energy consumption.