Low friction and high load-carrying capacity of a slider bearing with an inhomogeneous surface affinity

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Abstract. In fluid lubrication systems, lower friction means less energy consumption, whereas higher film thickness means higher load-carrying capacity and lower probability of wear. Traditionally, friction coefficient increases with oil film thickness, which cannot meet the design requirements of modern equipment, such as MEMS. In this study, we attempt to tackle this challenge by introducing an inhomogeneous surface affinity on a static slider bearing surface. A model combining the limiting shear stress and slip length is adopted to analyse the effect of boundary slip on the hydrodynamic performance of a slider bearing. The model is firstly verified by comparing the calculated results with the experimental data, and then parameter study is conducted. Results indicate that lower friction and higher film thickness can be realised simultaneously by a specific design of the inhomogeneous surface affinity.

1. Introduction

The boundary condition of fluid flow is one of the most important factors determining the behaviour of hydrodynamics. In 1738, Bernoulli [1] was the first to propose the no-slip boundary condition of fluid flow, which was confirmed by a large number of experiments. Almost all classical mechanics textbooks adopt this hypothesis, which is widely used in the analysis of fluid flow in engineering. However, with the development of micro/nanotechnology, experimental studies [2-7] have shown that the boundary slip at the solid/liquid interface exists in some cases. The boundary slip is small and has little effect on the macro flow characteristics of the fluid. However, for micro/nanoelectromechanical systems (MEMS), the effect of boundary slip cannot be ignored.

The theoretical analysis of boundary slip mainly focuses on the development of the mathematical model [8-13]. Three main slip models currently exist, namely, linear slip length model [8], critical shear stress model [9] and a complex model combining slip length and critical shear stress [10, 11]. The slip length model is widely used in physics for its simplicity. However, the critical shear stress model is more popular in engineering. In this model, slip occurs only when the shear stress at the solid/liquid interface reaches a critical value. On the basis of experimental results, Spikes and Granick [10] proposed a complex model by combining the two previous models. In this new model, the slip behaviour is governed by two parameters, namely, critical shear stress and slip length. The boundary slip occurs when the shear stress at the interface exceeds a critical value, and the shear stress is proportional to the liquid viscosity and slip velocity under slip conditions and inversely proportional to slip length. This model is verified through experiments by Guo et al. [14] later. Salant et al. [11] recently proposed a similar model with that of Spikes and Granick [10] but using a new parameter, slip coefficient, instead of slip length.

In this study, the slip model developed by Spikes and Granick [10] was applied. The model was
firstly verified by comparing the calculated results with the experimental data. Subsequently, a parameter study was conducted to show how to achieve lower friction and higher film thickness design simultaneously by a specific design of inhomogeneous surface affinity.

2. Mathematical model
In the current slip model, a limiting shear stress $\tau_{c0}$ is assumed to exist at the solid/liquid interface, and slip occurs if the shear stress at the solid/liquid interface reaches the limiting shear stress and the slip velocity is proportional to any additional shear stress according to traditional slip length model [15].

$$\tau_s = \tau_{c0} + \frac{\eta}{b} u_s$$  (1)

In the region without slip, $\tau_{c0}$ is infinite. As shown in Figure 1, Surface 1 is stationary, whereas Surface 2 moves in the X direction at speed $U$. The limiting shear stress of Surface 1 is assumed to be much smaller than that of Surface 2, indicating that slip only occurs on Surface 1. Therefore, the velocity boundary conditions in X direction can be described as follows:

$$\text{at } z = 0, \quad u = U$$

$$\text{at } z = h, \quad u = 0 \quad \text{for } \tau < \tau_{c0}$$  (2)

$$\text{at } z = h, \quad u = \frac{b}{\eta} (\tau_s - \tau_{c0}) \quad \text{for } \tau \geq \tau_{c0}$$

![Figure 1. Schematic representation of slider bearing with boundary slip](image)

Based on the conservation of mass, the modified Reynolds equation can be obtained as [6] :

$$\frac{\partial}{\partial x} \left( h^3 \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial y} \left( h^3 \frac{\partial p}{\partial y} \right) = 6\eta U \frac{\partial h}{\partial x} + 6\eta \frac{\partial (u_s h)}{\partial x} + 6\eta \frac{\partial (v_s h)}{\partial y}$$  (3)

3. Results and discussion
3.1. Verification of the model
To verify the slip model, the calculated results were compared with the published experimental data [14]. Figure 2 illustrates the measured glycerol solution film thickness under different speeds. In their experiments, a slider-on-disc test rig was used, and optical interferogram was applied to obtain film thickness. The length and width of the applied slider are 4 and 9 mm, respectively. To generate the boundary slip on the slider surface, an oleophobic coating (EGC) was coated on the original steel surface. Glycerol solutions were used as lubricants, and the properties of the applied lubricant are shown in Table 1. It can be seen the contact angle is greater than 90 degrees and the contact angle hysteresis is quite small, which means the adhesion force between the glycerol solution and the EGC is weak and it is possible to generate boundary slip at the solid/liquid interface. By adjusting the values of slip length $b$ (50 µm) and $\tau_{c0}$ (200 Pa), the theoretical values could be fitted with the experimental data exactly, which verified the new slip model.
Table 1. Properties of glycerol solutions

| Lubricant          | Viscosity (21 °C, mPa s) | Refractive index | CA with EGC (°) | CAH with EGC (°) |
|--------------------|--------------------------|-----------------|----------------|-----------------|
| 90% Glycerol solution | 157.5                    | 1.46            | 106.21         | 15.8            |

Figure 2. Change in the thickness of glycerol solutions with speed

Figure 3. Change of film thickness and friction coefficient with speed

Figure 4. Change of film thickness and friction coefficient with load

3.2. Boundary slip on the whole slider surface (homogeneous surface affinity)

To verify the boundary slip effect on hydrodynamic lubrication, additional calculations were conducted in a relatively large range. Figures 3 and 4 display the change of film thickness and friction coefficient with speed and load, respectively. In the calculation, the whole slider surface is assumed to be homogeneous oleophobic, and the boundary slip is allowed to occur on the whole slider surface. The properties of 90% glycerol listed in Section 3.1 were used here. The value of slip length $b$ and limiting shear stress $\tau_{c0}$ calculated in Section 3.1 were also applied here. Compared with the no slip condition, the friction coefficient decreases significantly in the whole speed and load range because of boundary slip, which is beneficial to energy saving. However, the traditional film thickness (no boundary slip) is much higher than that of the boundary slip condition, which may lead to the wear of bearing surfaces, especially under low speeds and high load conditions. How to realise higher film thickness and lower friction coefficient by surface affinity designing will be discussed later.
3.3. Partial slip of inhomogeneous surface affinity

Figures 3 and 4 prove that the requirements of lower friction and higher load ability cannot be realised through a homogeneous oleophobic surface. Therefore, the partial slip was designed using an inhomogeneous surface, which is realised by manipulating surface wettability on specific areas. Figure 5 shows the design of the inhomogeneous surface affinity of Surface 1. Different patterns were adopted, and the specific parameters of the design are:

(a): \( c_1 = 0, c_2 = 0.7L, c_3 = 0.12B, c_4 = 0.88B \).
(b): \( c_1 = 0.12L, c_2 = 0.88L, c_3 = 0.24B, c_4 = 0.76B \).
(c): \( c_1 = 0.5L, c_2 = L, c_3 = 0, c_4 = B \).

In the following calculations, the properties of 90% glycerol were also adopted, that is, \( \tau_{c0} = 200 \text{ Pa}, b = 50 \mu m \).

![Figure 5. Inhomogeneous surface affinity of Surface 1](image)

Figures 6 and 7 depict the change of film thickness and friction coefficient with speed under different patterns. The load is 2 N, and the inclination is 1/1745. Apparently, for all the cases, film thickness and friction coefficient increase with speed, which is consistent with the idea of traditional hydrodynamic lubrication theory. However, the three designed patterns show different behaviours, especially compared with the no-slip case. The lubrication behaviour of Pattern (c) is similar to that of the full slip case (Figure 3), and film thickness and friction coefficient drop compared with the traditional hydrodynamic lubrication (no slip). However, Patterns (a) and (b) show their advantages in terms of film thickness and friction coefficient. Film thickness increases apparently and is much higher than that of no-slip one. Their corresponding friction coefficients are much lower than those in the traditional no-slip case, thereby satisfying the requirement of modern bearing design and is beneficial to the environment. Specifically, Pattern (a) performs better than Pattern (b) because of its higher film thickness and lower friction coefficient.

![Figure 6. Change of film thickness with speed](image)  
![Figure 7. Change of friction with speed](image)
Figure 8 and 9 illustrates the normalised 3D pressure distribution \( P = \frac{p(h^2 / \eta UL)}{X=x/L, \ Y=y/B} \) of conventional slider bearing and slider bearing with Pattern (a). The working conditions are identical to that of Figure 2. Evidently, boundary slip behaviour affects the pressure distribution significantly. The maximum pressure of slider with Pattern (a) is around 2 times higher than that of conventional slider bearing. The location of the maximum pressure of slider with Pattern (a) locates at the boundary of slip and no-slip areas.

![Figure 8. 3D pressure distribution of conventional slider bearing](image1)

![Figure 9. 3D pressure distribution of slider bearing with partial slip](image2)

4. Conclusion
The boundary slip behaviour was investigated in this study, and a complex slip model developed recently was adopted. The predicted results coincided with the experimental data, which verified the slip model. Parameter studies were conducted, and two main conclusions were obtained as follows:

1) Film thickness and friction coefficient decrease compared with traditional hydrodynamic lubrication if boundary slip occurs on the whole slider surface.

2) Higher film thickness and lower friction coefficient can be achieved simultaneously through partial slip on the specific area of the slider surface.

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