Effects of friction reduction of micro-patterned array of rough slider bearing

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Abstract. Complex micro-scale patterns have attracted interest because of the functionality that can be created using this type of patterning. This study evaluates the frictional reduction effects of various micro patterns on a slider bearing surface which is operating under mixed lubrication. Due to the rapid growth of contact area under mixed lubrication, it has become important to study the phenomenon of asperity contact in bearings with a heavy load. New analysis using the modified Reynolds equation with both the average flow model and the contact model of asperities is conducted for the rough slider bearing. A numerical analysis is performed to determine the effects of surface roughness on a lubricated bearing. Several dented patterns such as, dot pattern, dashed line patterns are used to evaluate frictional reduction effects. To verify the analytical results, friction test for the micro-patterned samples are performed. From comparing the frictional reduction effects of patterned arrays, the design of them can control the frictional loss of bearings. Our results showed that the design of pattern array on the bearing surface was important to the friction reduction of bearings. To reduce frictional loss, the longitudinal direction of them was better than the transverse direction.

1. Introduction
Technologies to enhance energy efficiency have highlighted the importance of tribology in traditional mechanical systems. Improving tribological performances in terms of reducing friction, wear, and lubricant consumption is one of the important aspects in energy efficiency. To achieve enhanced and cost-effective performance of engineering components, many studies have investigated the effects of surface roughness on the performance of a variety of mechanical systems with the application of stochastic concepts [1]. Especially since Patir and Cheng [2-3] introduced an approach for driving the average flow model applicable to any general 3D roughness structure, the effects of surface roughness upon lubrication have received considerable attention. This model is based on defining empirical pressure and shear flow factors that are derived independently from the mean flow quantities obtained from a numerical solution of the model surface area with a randomly generated surface roughness. Then, the modified Reynolds equation, governing the mean pressure, is derived in terms of these flow factors. Many studies [4-7] evaluating the average flow model have been carried out in this field. However, the average flow model introduced previously did not include the phenomenon of the asperity contacts between two bodies of the bearing under boundary lubrication. The peaks of the surfaces often carry very high loads due to their contact. Shearing strength between the asperities cannot be ignored in a regime in which asperity contacts occur. Thus, the modeling of the contact
between rough surfaces can lead to an improved understanding of loads, friction, wear, and thermal occurrences between surfaces. It is difficult to obtain exact solutions for real contacts due to the complex variables involved. Researchers have expended much effort to model the elastic-plastic behavior of two bodies in contact [8-11]. Their results provided empirical coefficients for the relationship as a function of contact load, real contact area, and mean contact pressure. However, due to the rapid growth of contact area under partial or boundary lubrication, it has become important to study the phenomenon of asperity contact in bearings with a heavy load. Also, recently, complex micro-scale patterns have attracted interests because of the functionality that can be created using this type of patterning. These functional sliding surfaces can lead to reduced frictional forces [12]. In this manuscript, a new analytical technique is proposed to estimate the effects of different patterned arrays, utilizing the modified Reynolds equation and including both the average flow and asperities contact models. Analytical and corresponding experimental investigations of varying micro-patterned arrays are performed to establish the relationship between the geometric parameters of the patterned arrays and the resulting frictional behaviours. The proposed technique provides an accurate method to simulate and design the micro surface patterned arrays optimal to desired characteristics.

2. Theoretical approach

The schematic diagram in Figure 1 shows how the asperity contact between contact bodies can be interpreted in terms of forces in the analysis [13]. An intermittent contact between the sliding surfaces at the top of asperities occurs with mixed or boundary lubrication. The mechanism of mixed lubrication becomes much more complex if surfactants and additives are involved in the lubrication process.

\[
W = W_p + W_c = \int pdA_n + \int \sigma_c dA_r
\]

where \( W \) is the total resultant load, \( W_p \) is the load supported by hydrodynamic pressure \( p \), and \( W_c \) is the load supported by normal stress \( \sigma_c \) of the asperity contact area. \( A_n \) and \( A_r \) are the apparent and the real contact area of bearing surface, respectively. The frictional force is obtained by

\[
F_r = F_p + F_c = \int \tau_p dA_n + \int \tau_c dA_r
\]

where \( F_r \) is the total frictional force, \( F_p \) is the force of fluid friction due to shear stress \( \tau_p \), and \( F_c \) is the shearing force due to shear stress \( \tau_c \), of the asperities in contact between the bearing surfaces.

![Figure 1. Schematic diagram of the asperities contacts between bearing surfaces.](image-url)
To analyze a rough surface bearing, a surface with a 3D surface roughness profile should be defined. In this study, a generated surface pattern is considered. Figure 2 shows 3-Dimensional shapes of asperities of the generated surface.

![Figure 2. Generated rough surface for analysis(mean value: 0, standard deviation, $\sigma$: 1$\mu$m, kurtosis, $ku$: 2.831, skewness, $sk$: 0.037, $h$=8$\mu$m).](image)

$W_p$ and $F_p$ are calculated by the hydrodynamic analysis of fluid film bearing. The Reynolds equation is derived by considering the mean expected flow on a rough bearing, according to Patir and Cheng [2-3]. The average Reynolds equation of the steady state for the finite bearing of Figure 1 is derived as

$$\frac{\partial}{\partial x} \left( \Phi_x \frac{h^3}{12\mu} \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial y} \left( \Phi_y \frac{h^3}{12\mu} \frac{\partial p}{\partial y} \right) = \frac{U \partial h}{2 \partial x} + \frac{U \partial h}{2} \partial y$$

(3)

The flow factors $\Phi_x$, $\Phi_y$, and $\Phi_z$ were obtained in reference [2], [3] through numerical simulation. From the surface profile of Figure 2, the flow factors of the average flow model were calculated by using given equations. The numerical approximation is computed using FDM. Finite difference grids in the X*Y axis are 393*393. After solving for the pressure, the flow factors $\Phi_x$, $\Phi_y$, $\Phi_z$, $\Phi_{fp}$ and $\Phi_{fs}$ were calculated by the generated surface of Kim and Lee [13].

The nominal film thickness of the finite slider is given by

$$h = h_m + S_m \left( 1 - \frac{x}{L} \right)$$

(4)

where $h_m$ is the minimum nominal film thickness, $S_m$ is the slope height, and $L$ is the length of the bearing. The numerical approximation using FDM is performed with grids in X*Y axis, 101*101. The boundary conditions are such that the mean pressure is zero at the boundaries of the slider and the velocities of bearing set $U_{x=0} = U$ and $U_{x=L} = 0$. After that the mean pressure is solved, the mean hydrodynamic load and mean viscous friction force are calculated by

$$W_p = \int p dA_n = \int_0^R \int_0^L p \, dx \, dy$$

(5)

$$F_p = \int \tau_p dA_n = \int_0^R \int_0^L \tau_p \, dx \, dy$$

(6)

To calculate the total load and friction, the asperity load and friction force is calculated by the statistical contact model. When one surface of bearing approaching to the other, the ratio $A_r(d)/A_n$ for the statistical contact model can be obtained by calculating the interference of the asperities between the rough surface and a smooth rigid surface. Thus, as a function of the non-dimensional mean clearance of bearing $h/S_n$, the ratio $A_r(h/S_n)/A_n$ for the statistical contact model is obtained.
The loading force, $W_c$, and the friction force, $F_c$, of multiple asperities’ contact are obtained by the contact model proposed in Kim and Lee [13]. The load and friction components due to asperity contact are obtained by

$$W_c = \int \sigma \, dA_c = \int_0^B \int_0^L 0.417 \nu (2.8Y_0)(-0.0037 + 0.657 e^{-0.75h_{2c}}) \, dxdy \tag{8}$$

$$F_c = \int \tau \, dA_c = \int_0^B \int_0^L 0.5Y_0 (-0.0037 + 0.657 e^{-0.75h_{2c}}) \, dxdy \tag{9}$$

where the $Y_0$ is yield strength of bearing material and $\psi$ is the plastic index defined by Greenwood and Williamson [8].

Thus, the total load and friction on the slider can be calculated by the sum of each part in Eq. (1) and Eq. (2). The friction coefficient of the slider bearing is calculated by

$$C_f = \frac{F_f}{W_f} = \frac{F_f + F_c}{W_f + W_c} \tag{10}$$

### 3. Analytical conditions and Experimental setup

The key to verify the proposed solution is in the accurate control of the experimental parameters and sliding conditions. Considering limitations in both the computational power and the experimental apparatus, parameters and conditions are carefully selected.

Investigation of the effects on micro-patterned arrays was carried out through theoretical and experimental analyses. Among various patterned arrays, four types of distinct patterned arrays were selected for investigation. Figure 3 illustrates the different patterned arrays investigated in this study.

The first pattern (Fig. 2(a)) is a simple pattern of half-sphere dimples. The dimples are distributed evenly in both X and Y axis. The second pattern (Figs. 2(b), Figs. 2(c)) is the pattern of dashed lines. The cross section is in the shape of a half-sphere. The third is a zig-zag pattern of the dashed line (see Fig. 2(d)) used in the second patterns. To find the patterns having friction reducing effects, comparative analysis of the resulting friction coefficient has been performed. This manuscript provides not only the qualitative but quantitative analyses of the resulting friction coefficients from each pattern. The total depth of each patterned dimples were controlled to be 3 $\mu$m in all theoretical analysis.

![Figure 3](image)

**Figure 3.** Schematics of analysed micro pattern arrays.

The given input parameters shown in Table 1 was used in this analysis. The sliding condition of bearing surfaces was the X-axis direction of Figure 1. The slope parameter $h_{m}/S_m$ is also varied in number 10 [13]. The larger slope parameter would emphasize the mixed lubrication conditions due to the asperity contact.
Table 1. Input data in the analysis.

| Parameter                              | Conditions |
|----------------------------------------|------------|
| Surface roughness, $S_\sigma$          | 1 µm       |
| Slope parameter, $h_m/S_m$             | 10         |
| Minimum film thickness, $h_m$          | 0.8~8 µm   |
| Length, $L$                            | 100mm      |
| Width, $B$                             | 100mm      |
| Material yield strength, $Y_0$         | 333 Mpa    |
| Velocity, $U$                          | 0.5 m/s    |
| Lubricant Viscosity, $\mu$             | 0.001 Pa*s |

Experimental apparatus was set up to verify the simulated resulting from the theoretical analyses. A custom sliding friction test apparatus was built in-house to measure the sliding friction. Translation in the X-axis was enabled using an air guide-way. Installing load cells on the left and right enabled measurement of the frictional force opposing translation. The load was applied to the Z-axis with dead weight where the applied weight could be adjusted (see Figure 4). A mating surfaces with various patterns (SKD61) was placed on the stylus tip with its flat surface facing down. Sapphire surfaces were attached on top of the sliding guide.

Micro patterned arrays were fabricated on the SKD61 surface using Laser machining. Each test sample was the 3 mm thick and the 5 mm x 5 mm dimension. The dot dimple had Ø15µm diameter, 0.4 µm depth, and 100 µm pattern pitch. The dash line pattern had 15µm width, 80µm length, 200 µm pattern pitch of length direction, 100µm pattern pitch of dimple direction, and 400nm pattern depth. In experiments, a lubricant was applied to minimize wear; we used commercially available oil (GS Caltex/RANDO DFL 32) with a viscosity of 32.18cSt at 40°C. Sliding velocity was maintained at 20 mm/min. And, input loads were 0.6N, 2.6N, 4.6N, 6.6N, 8.6N, 10.6N, 12.6N and 14.6N.

4. Results & Discussion
To investigate the frictional effects on micro-patterned arrays in mixed lubrication regime, the theoretical analysis utilizing the modified Reynolds equation with the average flow of the rough
surface and including the asperities contact models was performed. The experimental analysis was performed on the condition similar to the theoretical analysis to verify the results. For each analysis, the resulting friction coefficient ($C_f$) was plotted against increasing Stribeck number. Stribeck number is the dimensionless number given by the ratio of lubrication dynamic viscosity, velocity and hydrodynamic pressure ($\mu_0 U/\omega$). Where $\omega$ means $[(W/A) S_{\sigma}]$ in this study. The effects of different types of patterned arrays on the friction coefficients showed different results, which were dependent on the lubrication regime (see Figure 5), while all patterns showed friction coefficients smaller than that of a No pattern surface. In the boundary lubrication regime (i.e., ($\mu_0 U/\omega < 0.2E-8$), the Zig-zag pattern showed the largest friction coefficient, while the Half-sphere dimple pattern showed the smallest. Interestingly, the X-axis dimple pattern showed rapidly decreasing friction coefficients with increasing Strubeck numbers. In the partial lubrication regime, the X-axis dimple pattern showed the least amount of friction. The Half-sphere dimple pattern had the largest friction coefficient in the partial lubrication regime. The hydrodynamic lubrication regime showed different characteristics from other regimes. The effects of contact asperities decreased with increasing film thickness. The X-axis dimple pattern resulted in the lowest friction coefficient. The Half-sphere dimple pattern had the largest friction coefficient in this regime.

![Figure 5. Effects of micro-patterned arrays in the theoretical analysis.](image)

The experimental analysis of various micro-patterned arrays showed similar trends to the theoretical analysis (see Figure 6). The results showed that the existence of patterns decreased the friction coefficient in all cases. The friction coefficient decreased the most with the X-axis dimple pattern. The Zig-zag pattern had the second most friction reduction effect, followed by the Y-axis dimple pattern. The Half-sphere dimple pattern had the least effect on friction reduction. Although the friction coefficients from the experiments did not match those of the theoretical analyses, the trends were identical. It can be concluded that the model provides an adequate method to investigate the effects of different surface patterns and that it can be further utilized to find the optimal pattern arrays for desired friction coefficients.
5. Conclusion
The effects of micro-surface-patterned arrays were investigated using a new analytical model that considers asperity contacts in the mixed lubrication regimes. A theoretical analysis using the newly proposed model was performed for various surface and surface pattern conditions. The validity of the model was verified with an experimental analysis. The contribution of this manuscript can be summarized as follows:
(1) The proposed analytical model showed that surface roughness and asperity contacts have large effects on the friction coefficient, regardless of surface patterns, especially in mixed lubrication regimes.
(2) Various surface patterns were analyzed using the proposed model. Different surface patterns, including Half-sphere dimples, dashed lines in two different directions and Zig-zags were analyzed to investigate their effects on resulting friction coefficient. Notably, the X-axis dimple pattern had the lowest friction coefficient in all regimes, while other patterns showed different behaviors, depending on the lubrication regimes.
(3) Experimental analysis was also performed in accordance with the analysis performed with theoretical model. The experimental results show the same trend as the model.

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7. References
[1] Tzeng S T and Saibel E 1967 Surface Roughness Effect on Slider Bearing Lubrication ASLE Transactions Vol. 10, pp 334-338
[2] Patir N and Cheng H S 1978 An Average Flow Model for Determining Effects of Three Dimensional Roughness on Partial Hydrodynamic Lubrication ASME Journal of Lubrication Technology, Vol. 100, pp 12-17
[3] Patir N and Cheng H S 1979 Application of the Average Flow Model to Lubrication between Rough Sliding Surfaces, *ASME Journal of Lubrication Technology*, Vol.101, pp 220-229
[4] Teale J L and Lebeck A O 1980 An Evaluation of the Average Flow Model for Surface Roughness Effects in Lubrication, *ASME Journal of Lubrication Technology*, Vol. 02, pp 360-367
[5] Tripp J H 1983 Surface Roughness Effects in Hydrodynamic Lubrication: The Flow Factor Method, *ASME Journal of Lubrication Technology*, Vol.105, pp 58-465
[6] Lunde L and Tonder K 1997 Pressure and Shear Flow in a Rough Hydrodynamic Bearing, Flow Factor Calculation, *ASME Journal of Tribology*, Vol. 119, pp 549-555
[7] Susan R H and Salant R F 2001 An Average Flow Model of Rough Surface Lubrication with Inter-Asperity Cavitation, *ASME Journal of Tribology*, Vol.123, pp 134-143
[8] Greenwood J A and Williamson, J B P 1966 Contact of Nominally Flat Surfaces, *Proc. R. Soc. London, Ser. A*, Vol. 295, pp 300–319
[9] Chang W R, Etsion I and Bogy D B 1988 Static Friction Coefficient Model for Metallic Rough Surfaces, *ASME Journal of Tribology*, Vol.110, pp 57–63
[10] Kogut L and Etsion I 2003 A Finite Element Based Elastic-Plastic Model for the Contact of Rough Surfaces, *ASME Journal of Tribology*, Vol. 46, pp 383–390
[11] Robert L J and Itzhak G 2006 A statistical model of elasto-plastic asperity contact between rough surfaces, *Tribology International*, Vol. 39, pp 906–914
[12] Kitamura K et al. 2016 Tribological effects of punch with micro-dimples in blanking under high hydrostatic pressure *CIRP Annals-Manufacturing Technology*
[13] Kim M R, Lee D W, Lee S M, Park S and Kim S H 2017 Tribological effects of a rough surface bearing using an average flow analysis with a contact model of asperities, *International Journal of Precision Engineering and Manufacturing*, Vol.18, Issue 1, pp 99-107