Hydrodynamic Performance of Positive Surface Textured Patterns under Mixed Lubrication

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Abstract: It is proved that the optimal design of surface textures (positive or negative) improve the lubrication performance by generating extra hydrodynamic pressure at the interface of the sliding pairs. In the present work, the effect of various positive surface textures on hydrodynamic performance of parallel sliding contact under mixed lubrication is researched. The modified Reynolds equation and asperity contact model are solved simultaneously to assess the hydrodynamic pressure and minimum film thickness for different shape of textures. The results depict that elliptical shaped texture generate high hydrodynamic pressure as well as minimum film thickness at the conjunction than the other texture shapes.

Keywords: Hydrodynamic pressure, Minimum film thickness, Mixed lubrication, Surface texture

1. Introduction

Numerous research works have reported the effects of surface texturing on hydrodynamic and frictional performance under sliding lubrication conditions. In this context, Henry et al. [1] examined the influence of square-shaped dimple textures on load carrying and frictional performance of parallel thrust pad bearing. The results depicted a friction reduction of 30% compared to un-textured surfaces. In contrary, Zhang et al. [2] considered rectangular and circular shaped dimples on piston ring surfaces to assess the frictional performance at top and dead centers of an engine. The rectangular textured surface has shown lower average friction co-efficient than circular shape textures. Furthermore, Rahmani et al. [3] derived the optimal geometrical conditions of a parallel slider bearing with square shaped dimples to evaluate the tribological properties along with the load support, frictional force and friction coefficient. The study concluded the dimple length ratio (the ratio of dimple length to distance between two consecutive dimples) of 1 and 1.5-1.7 exhibit a superior tribological performance. The authors have further extended their work by considering rectangular, equilateral as well isosceles triangular shaped textures. The rectangular shaped textures have shown superior tribological performance than the other. The optimal area density of dimple shaped textures in reducing the friction at different lubrication regimes was reported by Gachot et al. [5]. The authors have found the optimal results at area densities of 10-30%, 7-12% and 5-7% under fluid-film, mixed and boundary lubrication conditions respectively.
The research works reported on positive textures (protrusions) displayed a remarkable influence in lowering the friction at the interface under lubricating conditions. In this regard, Sawant et al. [6] examined the spot textures (protrusions) on machining performance of high speed steel (HSS) tools. The protrusions on rake face of the tool were fabricated through plasma transferred arc powder deposition technique. The authors have proved the reduction in cutting and thrust forces during machining with the aid of spot textured cutting tools than un-textured tools. Furthermore, Obikawa et al. [7] introduced the square shaped protrusions on rake face of carbide cutting tools that yield a better lubricating conditions in orthogonal machining. More recently, Venkateswara Babu and his research group [8-11] carried out an extensive work on positive surface texturing, which exhibited a significant impact on tribological performance. The authors conducted sliding experiments by introducing square-shaped protrusions on the sliding surfaces, which reduced the friction under mixed lubrication conditions [8]. In continuation of their research work, the authors have developed mixed lubrication model to validate their experimentation results as cited in [9].

However, conducting experiments for various textured shapes involves high cost and time. Keeping in view, the present work considers a theoretical model of textured parallel sliding contact as mentioned in [9] to study the effect of positive texture geometry on hydrodynamic pressure generation and minimum film thickness.

2. Numerical Model
To evaluate the hydrodynamic performance of various texture shapes (protrusions), the parallel sliding contact is mathematically modeled by including textures with square, triangular, circular and elliptical shapes. The configuration of sliding contact with square-shaped textures is shown in Fig.1.

![Fig.1 Square-shaped textured parallel sliding contact configuration](image-url)
The symbols in Fig. 1 are defined as follows;

\(B, L\) are axial width and length of the stationary surface respectively, \(h\) is average oil film thickness, \(h_0\) is minimum oil film thickness, \(h_t\) is height of texture, \(2r_p\) is base length of texture, \(U\) is sliding velocity of moving surface and \(x, y, z\) is global coordinate system.

2.1 Film thickness equation

The oil film thickness \((h)\) can be estimated by the following equation [9].

\[
h(x, z) = \begin{cases} h_b & \text{above texture} \\ h_t + h(x, z) & \text{elsewhere} \end{cases}
\]

The oil film thickness profiles for various texture shapes are shown in Fig. 2, corresponding to an area density (ratio of area of texture to the area of unit cell), \(S_p\) of 0.1 and \(h_t\) of 10 μm.

Fig. 2 Oil film thickness profiles for texture shapes of (a) square, (b) triangular, (c) circular, (d) elliptical

2.2 Modified Reynolds equation

To evaluate the hydrodynamic pressure \((p)\), the modified Reynolds equation, which includes the fluid flow factors is applied [12, 13].

\[
\frac{\partial}{\partial x} \left( \phi \frac{h^3}{\mu} \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial z} \left( \phi \frac{h^3}{\mu} \frac{\partial p}{\partial z} \right) = 6U\phi \frac{\partial h}{\partial x} + 6U\sigma \frac{\partial \phi}{\partial x}
\]

The symbols in Eq. (1) are defined as follows;

\(p\) is hydrodynamic pressure, \(\mu\) is viscosity of the lubricant, \(\sigma\) is the composite roughness of interacting surfaces and the factors \(\phi_c, \phi_s, \phi_x, \phi_z\) are factors of contact, shear and pressure flows respectively. These are calculated based on the relations given in [12-14].

The boundary conditions to estimate \(p\) in \(x\)- and \(z\)-directions are considered as follows;

\[
p_{(x=0)} = p_{(x=L)} = 0 \text{ and } p_{(z=0)} = p_{(z=B)}
\]

Eq. (2) is solved by satisfying above boundary conditions using finite difference method to obtain hydrodynamic pressure, \(p\).
2.3 Pressure balance criteria
In mixed lubrication, the external load \( W \) acting on the contact is responsible for generation of both hydrodynamic pressure \( p \) and asperity pressure \( p_{asp} \), which can support the total load. Therefore, total pressure is summation of hydrodynamic and asperity pressures. However, it is required to obtain greater values of \( p \) than \( p_{asp} \) as it enhances the total oil film load carrying capacity and decreases the frictional force. Furthermore, the higher minimum film thickness values are beneficial to minimize friction as well as wear. Therefore, the present work aims on obtaining the optimal values of \( p \) and minimum film thickness for various shapes of texture under mixed lubrication.

2.4 Hydrodynamic and asperity pressure equations
The oil film pressure is obtained by estimating Eq. (2). The asperity pressure \( p_{asp} \) can be obtained by solving Greenwood-Tripp model [15], which is given as;

\[
p_{asp} = K' E' F_{2.5}(h/\sigma)
\]

\[
K' = \frac{16\sqrt{2}}{15} \pi (\eta \beta \sigma)^2 \sqrt{\beta}
\]

\[
E' = \frac{1}{\frac{1}{E_1} + \frac{1}{E_2}}
\]

Herein, the parameters \( \eta, \beta, F_{2.5} \) \((h/\sigma)\) are related to asperity distribution of the surfaces, which depends on the ratio of film thickness, \( h/\sigma \). All these parameters are adopted from the previous studies [9].

3. Results and discussion
It is well known that, the higher values of hydrodynamic pressure and minimum film thickness are beneficial to obtain lower friction and wear of the sliding pairs. In this context, the simulations are performed to obtain optimal values of hydrodynamic pressure and minimum film thickness for different texture patterns. To analyze the different shapes of textured patterns on the hydrodynamic performance, the parameters included in numerical simulations are given in Table 1.

| Table 1 Parameters of numerical analysis [9] |
|---------------------------------------------|
| **Material parameters**                     |
| Elastic modulus of moving surface, \( E_1 \) | 215 GPa          |
| Elastic modulus of stationary surface, \( E_2 \) | 200 GPa          |
| Poisson’s ratio of moving surface, \( \nu_1 \) | 0.3               |
| Poisson’s ratio of stationary surface, \( \nu_2 \) | 0.3               |
| Composite roughness, \( \sigma \)          | 0.546 \( \mu \)m |
| **Texture parameters**                      |
| Area density, \( S_p \)                     | 0.1, 0.2, 0.3    |
| Texture height, \( h_t \)                   | 10 \( \mu \)m     |
| **Operating parameters**                    |

Dynamic viscosity of fluid, $\mu = 0.121$ Pa s
Axial length of the textured surface, $L = 3$ mm
Unit cell size, $B = 1$ mm
Sliding velocity, $U = 1.67$ m/s
External load, $W = 0.5$ N

The hydrodynamic pressure percentage of various shapes of surface texture at different area densities are shown in Fig.3. The higher area densities generated the more hydrodynamic pressures for all the shapes of texture patterns. However, elliptical patterns generate maximum hydrodynamic pressure percentage among all the texture patterns. Because, the hydrodynamic pressure developed in elliptical textures occupy more area under the curve as shown in Fig.4 (d), due to which hydrodynamic pressure is maximum when compared to other texture shapes (can be seen in Fig.4). It confirms that elliptical textures are superior in generating additional fluid pressure, thereby increasing the ability to support the total load. Moreover, among all the shapes, circular shape of texture exhibited low performance in generating extra fluid film pressures at all area densities.

![Fig.3 Hydrodynamic pressure percentage of various texture shapes](image-url)
Fig. 4 Hydrodynamic pressure distribution of various texture shapes of (a) square, (b) triangular, (c) circular, (d) elliptical at $S_p=0.1$ and $h_I=10 \, \mu m$

Fig. 5 Minimum film thickness variation for different texture shapes

The minimum oil film thickness for various texture shapes is described in Fig. 5. For all texture shapes, the area density has shown predominant effect on minimum film thickness. In particular, larger area densities exhibited greater minimum film thickness values. Furthermore, as for the shape of texture’s concern, the elliptical patterns exhibited the superiority in generating higher minimum film thickness values. This is due to the domination of elliptical textures in generating more fluid film pressures (see Fig. 4), which resulted in higher minimum film thickness values as an extra hydrodynamic lift at the junction separates the sliding surfaces.
4. Conclusion

The concluding remarks of the present study are as follows:

- The texture area density has shown a prominent impact on hydrodynamic pressure generation and minimum film thickness. In this context, the higher area densities are superior to lower area densities.
- As for texture shape’s concern, elliptical shape of texture exhibited better performance by generating extra hydrodynamic pressure and minimum film thickness at the conjunction.

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