The simulation of asymmetric micro-textures on the lubrication characteristics of sliding friction pairs

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Abstract. In order to study the effects of micro-texture lubrication characteristics of sliding friction pairs with asymmetric cross section, a theoretical model based on the Navier-Stokes equations was established, and numerical simulations were performed to investigate the influence of the morphology parameters of asymmetric triangular micro-texture on the pressure distribution, cavitation contours, flow field, bearing capacity, friction coefficient etc. The results show that during friction process, a negative pressure and vortex region is formed inside the texture, and a high pressure zone is generated near the outlet of the texture. The morphology parameters of asymmetric micro-texture affect the magnitude of bearing capacity and friction coefficient by changing the distribution characteristics of pressure and vortex of the lubricant. With the increase of the slope at outlet of asymmetric triangular texture, the original vortex center gradually shifts downwards to the right; meanwhile, a prominent vortex region and a high pressure region are generated near the inlet of the texture. As a result of aforementioned redistribution of pressure and flow field, the bearing capacities gradually increase, while the friction coefficients continuously decline.

Key words: Sliding friction pair; Asymmetric micro-texture; Vortex; Lubrication characteristics; Cavitation

Introduction
The surface micro-texture has a significant influence on the lubrication characteristics of sliding friction pairs, many scholars have carried out a large number of researches and a vast amount of progress was achieved. The idea of improving the tribological characteristics through artificial texture is originated from the 1960s. Hamilton [1] et al. proposed the idea of using surface texture to generate additional dynamic pressure lubrication effect, and proved additional carrying capacity on relative sliding friction surface through experiments. In recent years, surface micro-texture technology has been successfully applied in bearing [2] and other friction pairs [3,4]. This technique effectively improves the lubrication characteristics of friction pairs. At the same time, there are some new achievements on the mechanism of texture. Ashwin Ramesh et al.[5] researched the effect of surface texture on friction, confirmed that in the same experimental conditions, the friction coefficient of textured samples is
much smaller than that no textured samples, by about 80 percent. C Chouquet et al.\cite{6} fabricated partially distributed micro-textures on the circuit board and conducted the pin-on-disc experiment which showed that the friction coefficient reduced significantly. S Kango et al.\cite{7} solved partially distributed micro-textures on the circuit board and conducted the pin-on-disc experiment which showed that the friction coefficient reduced significantly. S Kango et al.\cite{7} solved incompressible Navier-Stokes equation and simulated the influence of symmetric circular and square textures on lubrication properties, finding that both types can effectively reduce the friction force and friction coefficient, and the circular texture has a smaller friction force than the square texture in the same experimental conditions. Yan Lu et al.\cite{8} researched the influences of micro-texture on the surfaces of sliding friction pairs through theoretical and experimental study. The results show that the friction coefficient decreases with the increase of the texture parameter (length) in the dynamic pressure lubrication. When the texture parameter (length) is greater than 0.45, the friction coefficient reaches the optimum value. Jonathon k Schuh\cite{9} investigated the lubricant properties of asymmetric surfaces of sliding friction pairs through theoretical and experimental study. The results show that the friction coefficient decreases with the increase of the texture parameter (length) in the dynamic pressure lubrication. When the texture parameter (length) is greater than 0.45, the friction coefficient reaches the optimum value. Jonathon k Schuh\cite{9} investigated the lubricant properties of asymmetric micro-textures on the friction pairs, and the results show that asymmetric surface texture can reduce the shear stress and generate additional load, thus it can effectively reduce the friction coefficient. However, in that essay, the texture’s depth and angle change simultaneously, which makes impossible to distinguish how the single variable influences the lubrication property.

To sum up, the surface micro-texture can significantly improve the lubrication characteristics on the sliding friction pairs\cite{10}, and the shapes and distributions of surface textures have different effects on lubrication characteristics\cite{11}. However the existing researches mainly focused on the influences of symmetrical micro-textures of the friction pairs; the study on the effects of asymmetric textures is rare, the principles of the structure parameters of asymmetric textures on the lubrication characteristics have not fully disclosed. Therefore, in our study, based on the Navier-Stokes equations, a theoretical lubrication model of asymmetric micro-texture on the sliding friction pairs surface is established, and numerical simulations are performed to reveal the influences of angle parameters on the pressure distribution, flow field distribution, bearing capacity and friction coefficient of the friction pairs. Finally, this paper also discusses the mechanism of promoting tribological properties by adopting the asymmetric micro-textures. The results of this study have certain significance for the development and improvement of the lubrication theory based on the surface micro-texture.

1. Theoretical model

Existing researches showed that the lubrication theory based on Navier-Stokes equations can accurately simulate the pressure and flow field of the micro-texture lubrication. Sceri et al.\cite{12} pointed out that when the oil-film thickness rapid changes in the dimple, the influence of the inertia of fluid should not be neglected. Because Reynolds equation does not consider the inertia term, the model based on Reynolds equation is not quite suitable for simulation for micro-textures. So this paper establishes a simulation model with the Navier-Stokes equations which considers inertial role of fluid to investigate the lubrication in asymmetric micro-textures. To simplify the model, a two-dimensional model has been adopted. The model ignores the role of the body force and the time effects (the steady state model). In addition, lubricant is incompressible, and the viscosity is constant.

The simplified Navier-Stokes equation in the x direction is

\[
\rho(u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y}) = -\frac{\partial P}{\partial x} + \eta(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2})
\]  

(1)

In the y direction, the equation is

\[
\rho(u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y}) = -\frac{\partial P}{\partial y} + \eta(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2})
\]  

(2)

\(\rho\) indicates the density of lubricating oil; \(u\) and \(v\) represent the flow velocity of \(x\) and \(y\) direction respectively; \(P\) represents oil-film pressure; \(\eta\) represents the viscosity of the lubricating oil. The last terms on the right-hand side in equation (1) and (2) are the inertia terms. By using Fluent software, the shear stress and pressure values of each mesh point are obtained, and friction coefficient of the friction
pair is solved\(^{13}\). The pressure \(P(x,y)\) integral can be obtained in the whole lubrication oil-film (along the upper wall) to get the lubricating film bearing capacity.

The formula for bearing loading capacity \(W\) is

\[
W = \iint P \, dx \, dy
\]  

(3)

The formula for the friction factor \(\mu\) is

\[
F = \int \eta \, \frac{\mu}{\gamma} \, dx
\]

(4)

\[
\mu = \frac{F}{W}
\]

(5)

In the formula (4), \(F\) is the frictional force, \(\eta\) is the dynamic viscosity, \(\dot{\gamma}\mu\partial \gamma\) is the velocity gradient along the direction of the oil-film thickness of lubricating oil, \(\dot{\gamma}\mu\partial \gamma\) is lubricating oil-film shear force. In the formula (5), \(\mu\) is the friction coefficient, which is the ratio of friction to carrying capacity.

2. Computational area and simulation parameters
In this article, the micro-textures with triangular section were selected as the simulated object, and the simulation domain of friction pairs of asymmetric micro-texture is shown in Figure 2.1. In the Figure 2.1, the simulation domain length \(L=1\) mm and the oil-film thickness \(H=0.3\) mm; The viscosity of lubricant is 0.028 Pa.s with a density of 920 kg/m\(^3\). According to Fredrik Sahlin\(^{15}\), the optimal ratio range of surface micro-texture depth \(h\) to film thickness \(W\) \((0.5~0.75)\) was selected as a constant, \(h/W = 0.6\), in this article.

![Figure 2.1 Diagram of simulation domain](image)

Table 2.1 The slope at outlet of the asymmetric triangular texture

| The angle of the triangle slope(°) | A  | B  | C  | D (symmetric) | E  | F  | G  | H  | Flat plate |
|-----------------------------------|----|----|----|---------------|----|----|----|----|------------|
|                                   | 30.94 | 33.93 | 40.60 | 50.19         | 63.43 | 71.57 | 90  |    |            |
The physical parameters of the lubricating oil and the motion parameters in the simulation are shown in Table 2.2. According to the above model and parameters, numerical simulations are performed by using Fluent software. K-epsilon (2 eqn) is chosen as the steady-state turbulence model, and the periodic boundary conditions are used to the inlet and outlet region. With respect to the numerical simulation, the number of grid nodes is more than 94,000, residual error is selected as $10^{-5}$. The iterative computation is calculated until the convergence condition is satisfied. According to the Fluent simulation results, the shear stress and pressure values of each grid point were obtained, and the MatLab software was introduced to calculate the friction factor and bearing capacity of the friction pair according to Formula (3) and Formula (4).

### Table 2.2 Physical parameters and running condition

| Physical Parameter                      | Value  |
|----------------------------------------|--------|
| Viscosity of lubricating oil (Pa.s)   | 0.028  |
| Density of lubricating oil (kg/m3)    | 920    |
| Temperature (°C)                       | 20     |
| Oil-film thickness (mm)               | 0.3    |
| Depth of micro-texture (mm)           | 0.3    |
| Width of micro-texture (mm)           | 0.5    |
| Length of micro-texture (mm)          | 1      |
| Speed of upper wall (m/s)             | 10     |

#### 3. The simulation results and analysis

Through simulation according to Table 2.2 parameters, the stress contours with different surfaces are obtained as shown in Figure 3.1. The black lines in the Figure are dividing lines between different pressure areas, and the pressure specific values are marked. Through comparing with flat plate with not texture (Figure 3.1 h), in each triangle texture (Figure 3.1 a ~ g) a high pressure zone are formed at the tail of the micro-texture exports surface, and the maximum stress value appears near the micro-texture outlet area; simultaneously a low pressure zone is formed inside micro-texture, and the center of the low pressure zone is slightly close to the outlet slope. The positions of the high-pressure zone hardly change when the outlet slope $\gamma$ increases, but the range of it gradually reduces. With gradually increase of the outlet slope $\gamma$, the range of Low-pressure zone increases at first; while when the outlet slope is $40.60^\circ$, low-pressure range reaches maximum value; then the low-pressure range decreases with further increase of the outlet slope. When outlet slope $\gamma$ reaches $40.60^\circ$, other two new high pressure areas appear in the entrance and bottom areas of the triangle texture. With the increase of angle $\gamma$, both ranges of these high pressure areas increase, especially obvious for that in the entrance area. When angle $\gamma$ is greater than or equal to Angle $71.57^\circ$, the entrance and the outlet high-pressure areas are connected into one high pressure region (Figure 3.1 f ~ g).
In the same conditions in Table 2.2, Figure 3.2a-g are Simulation contour of lubricating oil cavitation, the percentage of cavitation area in the whole texture model under different texture conditions is shown in the Figure 3.2 h. Cavitation occurs in all triangular texture (Figure 3.2a-g), and cavitation occurs within the texture.

In the figure 3.2 a-d, the cavitation section can run through the entire triangle textures, and With the increase of the outlet Angle of texture, the cavitation area gradually shrinks. The cavitation area appears only in the texture inlet area. Figure 3.2h shows when the outlet Angle is larger, the proportion of cavitation area to the whole texture model decreased, the downward trend from violent to flat and then to violent.

Figure 3 contours of Volume fraction (cavitation contours)

Figure 3.2 shows the distribution of flow field between friction pairs. The main vortex area center coordinates are marked in Figure 3.2a-h, and the streamlines show the flow directions in the simulation domain. With respect to Figure 3.2a-g, fluid flow direction at upper region between two
surfaces is less affected by the texture; while near the texture area the flow direction slightly turns to the texture; within the triangle texture a vortex area is formed. Figure 3.3 shows the relationship between centers coordinates of the main vortex area and the outlet slope $\gamma$. With the increase of the outlet slope $\gamma$, the center positions of the vortex area move from center to the lower right direction, and the shapes of vortex correspond to the shapes of texture areas. When the outlet slope $\gamma$ is greater than or equal to 50.19° (Figure 3.2 e ~ g), a part of the flow deviates from the main vortex area and produces protuberant shape flow at the entrance area. The flow direction in the protuberant vortex is clockwise, having which is the same direction of the main vortex, so this protuberant vortex is considered as a special portion of the main vortex.

Figure 3.2 Streamline charts and vortex center coordinates

![Streamline charts and vortex center coordinates](image)

Figure 3.3 The center coordinates of the main vortex region

![The center coordinates of the main vortex region](image)

The Figure 3.4 shows the influence of the viscosity of lubricating oil on the pressure distribution. Other boundary conditions and physical parameters remain the same as for Figure 3.1.

When the viscosity of the lubricating oil is 0.5 Pa. s (Figure 3.4a), the fluid forms a low-pressure zone in the entrance area and a high-pressure zone at outlet area; When the viscosity of the lubricating oil is 0.028 Pa. s (Figure 3.4b), a small high pressure zone is formed in the entrance area to the left of the low-pressure zone. Comparing with Figure 3.2, the position of the small high-pressure area is consistent with that of the protruding vortex. When the oil viscosity has a lower value, a small high-pressure zone in the entrance area is formed due to joining of the high velocity of backflow liquid and entrance fluid in the vortex region. In the case of adopting high viscosity lubricating oil, due to the low
reflux velocity, the interaction between backflow fluid and the inlet fluid is weak, making it difficult to form a similar high-pressure zone in the entrance area.

Table 3.1 gives the bearing loading capacity values which were obtained by substituting unit pressures on the upper wall into Formula (3). Based on the values in Table 3.1, a bar chart is produced to show the influence of different texture on bearing capacity of upper wall in Figure 3.5. As from Table 3.1 and Figure 3.5, the bearing capacities of all triangular texture surfaces are greater than that of the flat plate with no texture, which implies textures on surface can promote bearing loading capacity of friction pairs. With the increase of export incline angle $\gamma$, the bearing loading capacity increase gradually. When $\gamma = 90^\circ$, the bearing loading capacity reaches the maximum (90.17 Pa), which is 87% larger than that without texture on surface condition.

When triangle micro-texture export angle $\gamma$ changes from 33.93 $^\circ$ to 40.60 $^\circ$, the bearing capacity forms first significant increase. By comparing with Figure 3.1 b ~ c, a new high pressure area begins to form at the entrance area of the texture, which leads to the rapid increase of the loading capacity.

When triangle micro-texture export angle $\gamma$ changes from 33.93 $^\circ$ to 40.60 $^\circ$, another distinct increase of bearing loading capacity happens. Referring to Figure 3.1 e ~ f, when triangle micro-texture export angle $\gamma$ reaches 71.57 $^\circ$, with the expansions of high pressure areas and contraction of low pressure area, the high pressure zones in entrance area begins to contact with that in outlet area. The combination of two high pressure areas reduces the influence of low pressure area, which attributes to the obvious increase of the loading capacity.

Table 3.1 Influence of the slope at outlet of the asymmetric triangular texture on the bearing loading capacity of upper wall

| Number       | A   | B   | C   | D   | E   | F   | G (flat plate) |
|--------------|-----|-----|-----|-----|-----|-----|----------------|
| Not thinking about cavitation | 39.12 | 40.68 | 42.40 | 42.53 | 48.59 | 59.53 | 69.97         |
| Thinking about cavitation      | 60.56 | 61.5 | 71.64 | 73.04 | 76.76 | 85.17 | 90.17 | 48.57         |

Figure 3.5 Comparison of the bearing loading capacity of cavitation and no cavitation

Figure 3.6 Influence of different textures on friction coefficients of upper wall

Table 3.2 lists the averages of the upper wall friction coefficients calculated according to Formula (4). Figure 3.6 is a bar graph showing the frictions with different texture surfaces; the frictions
correspond to those in Table 3.2. As from Table 3.2 and Figure. 3.6, the friction coefficients of all triangular texture surfaces are smaller than that of the flat plate with no texture, which implies textures on surface can reduce friction coefficient of friction pairs. With the increase of export incline angle $\gamma$, the friction coefficient decreases gradually. When $\gamma = 90^\circ$, the friction coefficient reaches the minimum ($1.50 \times 10^{-3}$), which is 30% smaller than that without texture on surface condition. The fluid flows clockwise in the vortex area of the triangle texture; the clockwise flow acts a role as rolling bearing, which is conducive to reducing the shearing action between the fluid and the wall surfaces of the friction pair, so the friction coefficient can be effectively reduced.

When triangle micro-texture export angle $\gamma$ changes from 33.93 ° to 40.60 °, the friction coefficient occurs first significant decrease. By comparing with Figure. 3.1 b ~ c, a new high pressure area begins to form at the entrance area of the texture, which leads to the rapid decrease of the friction coefficient. Due to the fluid action on the upper wall, the normal compressive stress $\tau$ increase significantly while fluid shear force $\sigma$ change slightly, so the friction factor (proportional to the ratio $\tau/\sigma$) decreased significantly. Another significant reduction of the friction coefficient occurs when $\gamma$ changes from 63.43° to 71.57° (refer Figure. 3.1e-f and 3.2e-f). In these conditions, the high pressure zones in entrance area begins to contact with that in outlet area, which reduces the influence of low pressure area; the appearance of protruding vortex area expands the contact area between the vortex and the interfacial fluid, so the friction coefficient can be effectively reduced.

According to the above research on the effect of the triangle micro-texture export angle $\gamma$ of the triangle textures on the frictional load bearing capacities and friction forces of friction pairs under the condition of fluid lubrication, it can be concluded that the increase of $\gamma$ is beneficial to increasing the bearing capacity of the friction pair and reducing the friction; the formation of the high pressure zone in the inlet and its distribution forms and the appearance of the protruding vortex zone are important reasons for the increase of frictional bearing capacity and the reduction of friction coefficients.

### Table 3.2 Influence of different textures on the friction coefficients of upper wall

| Number | A    | B    | C    | D    | E    | F    | G    | H (flat plate) |
|--------|------|------|------|------|------|------|------|----------------|
| friction coefficient ($\times 10^{-5}$) | 7.191 | 5.647 | 5.418 | 5.228 | 4.169 | 3.448 | 2.345 | 34             |

4. Conclusion

In this paper, the influence of triangular asymmetric micro-texture shape on the lubrication state of sliding friction pair was explored by numerical simulation, and principle of the triangle micro-texture export angle $\gamma$ influencing on fluid pressure distribution of texture area, flow field forms, friction coefficient and bearing capacity of upper surface was obtained.

a. With triangular asymmetric surface micro-texture friction pairs, during the lubrication and frictional motion of the fluid, a negative pressure zone is formed inside the texture, while a high pressure zone is formed in the outlet zone of the texture. When the triangle micro-texture export angle $\gamma$ gradually increases, the area of high pressure zone gradually narrows and the low pressure zone area first increases and then decreases gradually. When the triangle micro-texture export angle $\gamma$ increases to 40.60 °, two new high pressure zones begin to appear in the entrance area of the texture and the bottom of the triangle, then gradually expand with the increase in the triangle micro-texture export angle $\gamma$. When $\gamma$ is greater than or equal to 71.57 °, the entrance area and exit of the high-pressure zone begin to combine with each other.

b. During fluid lubrication and frictional movement, a vortex zone is formed in the negative pressure zone inside the triangular texture. With the increase of angle $\gamma$, the center of the vortex moves to the right. When $\gamma$ is greater than or equal to 50.19°, a protruding vortex region extending from the main vortex is generated at the entrance of the texture.
c. The increasing of the triangle micro-texture export angle has a positive effect on increasing the bearing capacity and decreasing friction under fluid lubrication. With the increase of angle $\gamma$, the high-pressure area at the entrance and the exit region of the texture gradually combine gradually, and a protruding vortex flowed region appears, thereby the bearing capacity significantly increases and the friction coefficient significantly reduces. When the triangle micro-texture export angle $\gamma$ is equal to 90°, the improvement of bearing capacity improvement and reduction of friction coefficient are most obvious.

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