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Orientation effects of micro-grooves on sliding surfaces

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ABSTRACT

Based on a brief review of the researches related to the patterns of micro-grooves for tribological purpose, experiments were designed and carried out to study the orientation effects of grooves on the friction performance. The experimental data indicated that the grooves perpendicular or parallel to the sliding direction have a strong impact on the friction performance of sliding surfaces, and the merits of perpendicular or parallel orientation may swap under different contact conditions. The results were then discussed from the aspects of hydrodynamic effect, lubricant supply effect and contact stress effect.

1. Introduction

The quality of the surface is often of the utmost importance for the correct functioning of the mechanical components since the most physical phenomena involving the exchange of energy and/or signal transmission take place on surfaces [1].

In the past decade, there has been a growing interest on the designing of surface texture to improve the tribological performances of interface. Progresses in the understanding of surface phenomena, particularly at a micro- and nano-meter scale, have played a fundamental role in offering designers additional freedom to create novel functions or combinations of functions [2]. As a result, large numbers of products have emerged where the surface has been specifically textured to provide low friction [3], high durability [4], anti-seizure [5], and high load carrying capacity [6]. Such examples include cylinder liner [7], piston ring [8,9], hard disk [10], sliding bearing [11], mechanical seal [12], and even lip seal, etc. [13].

Grooves and dimples at the micro-meter scale are the most common geometric features used for the pattern on tribo-surfaces so far. They could be arranged evenly or randomly distributed on the surface. Efforts have been made to compare the effects of the patterns of grooves and dimples on tribological performances. It has been reported that independent and closed texture cells such as dimples are better to obtain hydrodynamic effect than connected structures such as grooves [14–16], which would lead to channel the lubricant away from the contact [17]. However, micro-grooves might be the most successful pattern of surface texture historically by the example of cross-hatch pattern for the cylinder liner of combustion engine. Low machining cost and effective anti-seizure ability make the honing technique irreplaceable for combustion engine as of today. It can be found that researchers are still trying to improve honing process and pattern of grooves, which includes combing a honing structure with laser spots [18], new laser grooving method [19], modifying honing tools to obtain a better feature [20], quantification method of honed features, etc. [21,22].

This paper will study the effect of micro-grooves on the frictional performance of sliding surfaces. We would like to have a brief review about the researches on the groove patterns for the purpose to improve tribological performance.

As an early and successful pattern of the surface texture, the micro-grooves on the surface of cylinder liners manufactured by the honing process has been used for more than half a century [20]. As indicated in a precious review by Willis, the development of the surface finish for cylinder liners is not only an important milestone that people had recognized a suitable texture could be beneficial for the sliding friction and the wear of piston/cylinder pair, but also a progress that people understood how to measure, and how to evaluate surface geometry with proper parameters [7].

45° cross-hatched deep valleys as lubricant reservoir and plain plateau without folded metals are the necessary features of honing surface for cylinder liners. It is obvious that the regular roughness parameters, such as center-line average R_a, root mean square R_q could not provide sufficient information to describe a suitable surface finish for cylinder liners. Instead, "Abbott curve" or "bearing area curve" is the most useful graph to evaluate the honing surface by engine builders as of today [23].

Parallel and perpendicular to the sliding direction are two extreme cases of groove orientation. It has been reported that the effect of

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grooves in other directions are expected to lie between these two extreme cases [24]. That is possibly the reason people use the 45° cross-hatch grooves for the cylinder liners of combustion engine.

Perpendicular or parallel? It is not only just an interesting question, but may also concern different fundamental mechanisms since it is obvious that different models and different experimental settings might bring about different results.

Cheng [25] developed a model by modifying Reynolds equation to solve the lubrication problem in thin-film regime associated with one-dimensional roughness. It was found that transverse (perpendicular) roughness generated a higher average pressure than the longitudinal (parallel) roughness based on a same mean film thickness. But the author indicated that this theoretical results contradicted to the experimental results obtained by Hata et al. [26], which showed that the longitudinal grooves had appeared to give a higher film thickness, larger hydrodynamic lubrication range and lower friction.

Currently, although the grooves in the commercial sliding bearings are parallel to the sliding direction [11], it seems that the data supporting the perpendicular design are more than that for parallel.

Moronuki and Furukawa [27] compared the effect of groove orientation by the frictional tests between planar Si and textured Si in a normal environment. Compared to the mirror polished Si, the friction decreased more in the case of perpendicular than that in the case of parallel in light pressure condition. The reason was considered that the grooves perpendicular to sliding could minimize the adhesion force due to water capillary, which became essential in the situation of very light load.

At starved lubrication, Pettersson and Jacobson [28] results also showed that the performance of surface texture was strongly dependent on the orientation. For the patterns of two small grooves (5 and 20 μm), when the grooves were oriented along the sliding direction, high-level friction and wear rates were shown, comparatively, when the grooves oriented perpendicular to the sliding direction, the friction and wear behavior were excellent. The reason suggested was that for the ball sliding along the grooves, a very small circular contact area is less likely to cross an oil reservoir.

Costa and Hutchings carried out reciprocating sliding tests of line contact between a patterned plane and a cylinder lubricated by mineral oil [17]. For the lowest load of 12 N, the sample with grooves perpendicular to the sliding direction showed worse behavior than that in the other orientation. However, for the high normal load, the sample with perpendicular groove gave the greatest film thickness. The author explained that higher load would induce larger Hertz contact area, and the groove parallel to the sliding direction would channel the lubricant away from the contact.

Ren et al. [29] improved Hu and Zhu's full-scale mixed-elastohydrodynamic model to simulate the effects of micro-groove in a slide-to-roll contact. The result showed that texture orientation could be a dominant factor, and straight grooves perpendicular to the motion direction offered the strongest hydrodynamic lift.

Zum Gahr et al. [30] investigated the friction coefficient and film thickness of 100Gr6/sapphire pairs as function of the orientation of microchannels. It is shown that the microchannels perpendicular to the sliding direction generated greater film thickness and lower friction coefficient than that of parallel orientation.

Beside the extreme cases of groove orientation, just like the cylinder liner, there are experimental results showing that the pattern of micro-grooves could be optimized further to obtain better tribological performance through the mesh design [31], and through the angle of cross-hatch and groove aspect ratio [32].

As reviewed above, although documented results contribute a lot for us to understand the effect of micro-grooves, there are still many uncertainties during the design of groove patterns. Efforts are still needed to reveal the mechanisms of the lubrication associated with micro-grooves, particularly, concerning the type of contact, materials of the interface, and most importantly, operating conditions.

2. Experimental

2.1. Specimens and surface texturing

The objective of this study is to investigate the orientation effects of micro-grooves in conformal contact condition under different loads and speeds.

As shown in Fig. 1, both upper and lower specimens were cut from a commercial cylinder of combustion engine made of boron–copper alloy cast iron. Each of the specimen was firstly milled to generate a planar surface, and grinded to obtain the final surface roughness *Ra* in the range of 0.4–0.5 μm. The dimension of the planar area of the upper specimen was 20 mm × 20 mm, which would generates a relatively larger contact area than that in previous documented studies. Then, micro-grooves were fabricated on the plane surface of the upper specimen by photolithography combined with electrolytic etching process, which is detailed in Ref. [33]. This process has the advantages that the dimensions of micro-features could be controlled accurately, and the surface need not to be polished again to remove the undesired rim up, which is usually induced by other fabrication techniques such as laser, etc.

Table 1 lists the geometric parameters of the patterns used in this research. All textured specimens have the same groove width of 100 μm, the same area density of 10%, which is the preferable value for dimple pattern [33], but different depth and orientation. Fig. 2 shows a microscope image of a groove pattern, and a 3D profile of a groove constructed by the depth from the defocus method. Fig. 3 shows the measurement of the micro-grooves obtained by non-contact optical interferometer.

2.2. Test rig and testing procedure

A schematic view of the test rig is shown in Fig. 4. The lower specimen 1 is fixed in the oil box 10, which is driven by the slider-crank mechanism to reciprocating slide in the *Y* direction. The

Fig. 1. Lower (left) and upper (right) specimens.

upper specimen 3 is held by a jig 2, which is fixed with the arm 6 and keeps stationary during the test. The jig provides the upper specimen freedoms to rotate around *X* and *Y* axes to ensure the correct alignment between the upper and the lower specimens. The arm 6 also has freedoms to rotate around the pivots 4 and 5 so that load could be applied through dead weight 9, and friction force could be measured by force sensor 8. By applying a pre-force to the arm through a screw-ball mechanism 7, the plus and minus friction force in the *Y* direction could be measured by only one force sensor.

Table 1
Specimens list.

Specimen no.	*Width B* (μm)	*Depth h* (μm)	*Pitch A* (μm)	*Area density r*(%)	α (deg.)
0	0	0	0	0	
1	100	7	1000	10	0
2	100	7	1000	10	45
3	100	7	1000	10	90
4	100	19	1000	10	0
5	100	19	1000	10	30
6	100	19	1000	10	60
7	100	19	1000	10	90

The sliding tests were carried out with normal loads from 49 to 196 N, corresponding to the contact pressure from 0.12 to 0.5 MPa, and the rotational speeds of the crank from 50 to 500 rpm. The length of an entire stroke was set to 80 mm, so that the maximum sliding speed within a stroke is 0.21 m/s corresponding to the rotational speed of 50 rpm, and 2.1 m/s corresponding to 500 rpm. All the experiments were conducted at room temperature. A commercial diesel engine oil CD 15W-40 was used as the lubricant. At the beginning of each test, 0.3 ml oil was dropped on the contact area. At each load, a running-in procedure was carried out for 15 min at 200 rpm, and then the friction force was recorded continuously by a PC through a data acquisition card.

2.3. Results

Fig. 5 shows the friction coefficient curves of the specimens nos. 0 and 5 during reciprocating tests. The sign of friction coefficient reverses while the movement direction changed. The friction coefficients on the different side of each reversal point have almost the same absolute value but different sign. For the friction during every half stroke, it usually has two high values at the start and the end positions, and relative low value in the center where the sliding speed is higher. It is more obvious for the

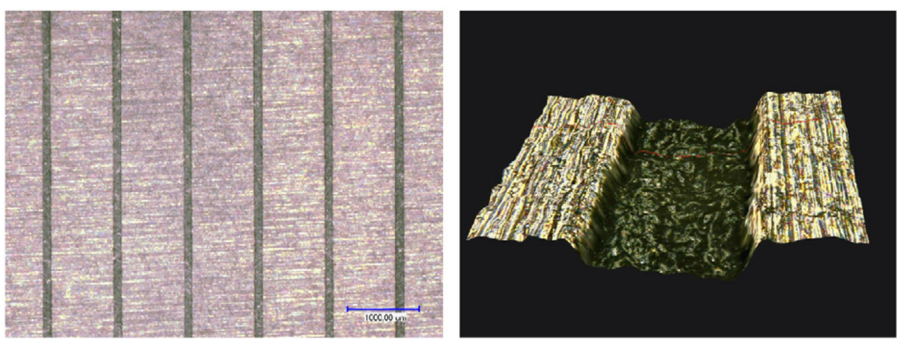

Fig. 2. Optical microscope image and 3D profile of the micro-grooves on the upper specimen.

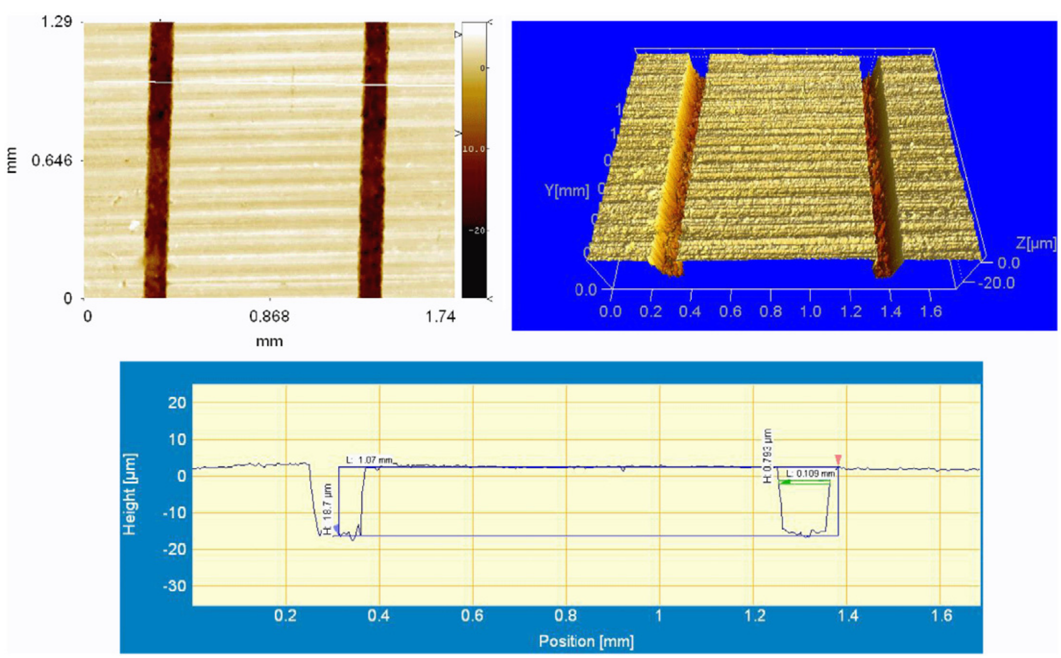

Fig. 3. Surface profile of the micro-grooves on the upper specimen no. 7 obtained by a non-contact optical interferometer.

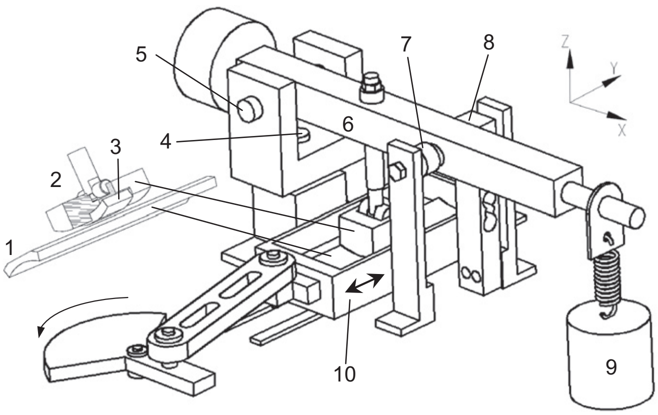

Fig. 4. Schematic view of the test rig.

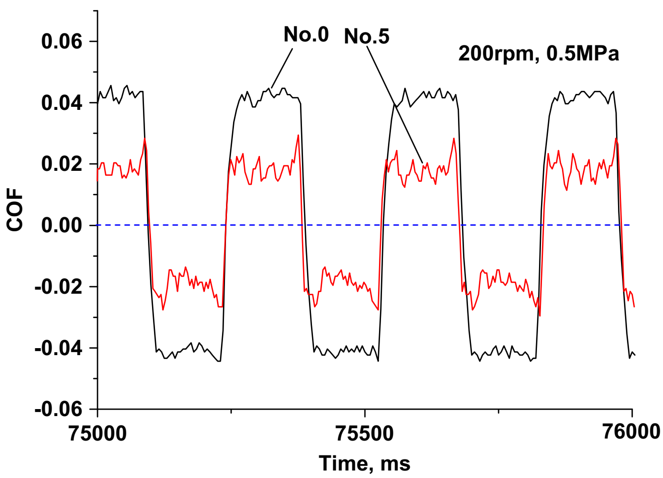

Fig. 5. Friction coefficient curves of the specimens nos. 0 and 5 obtained during reciprocating tests.

textured surface, implying that hydrodynamic effect is enhanced by surface texture.

In order to evaluate the representative frictional properties of the entire stroke, average friction coefficient was used in the following figures. It is the mean of absolute friction coefficients calculated with the data of 1000 cycles for each condition.

Fig. 6 shows the average friction coefficients of reciprocating sliding of the specimens with groove depth around 7 μm at different load and speed conditions. In the low contact pressure case shown in Fig. 6(a), the grooves with the angles 45° and 90° (perpendicular) to the sliding direction have obvious friction reduction effect compared to the untextured specimen. The grooves perpendicular to the sliding direction resulted in the best friction reduction effect in most rotational speeds except 500 rpm. At the low rotational speeds from 50 to 200 rpm, the friction reduction rate achieves maximum, which could be up to 38.2% of the friction coefficient of the untextured specimen. On the other hand, the specimen with grooves parallel ($\alpha=0°$) to the sliding direction has no better effect in friction reduction compared to the untextured specimen. Fig. 6(b) and (c) shows the test results obtained at relatively high contact pressures 0.25 and 0.5 MPa. Compared to the low contact pressure condition of 0.12 MPa, all patterned surfaces have lower friction coefficient than that of untextured surface no matter how grooves are oriented. However, the difference between different orientations decreases. It is particularly under the contact pressure of 0.5 MPa that the difference of average friction coefficient between parallel and perpendicular direction is really small. The best result of

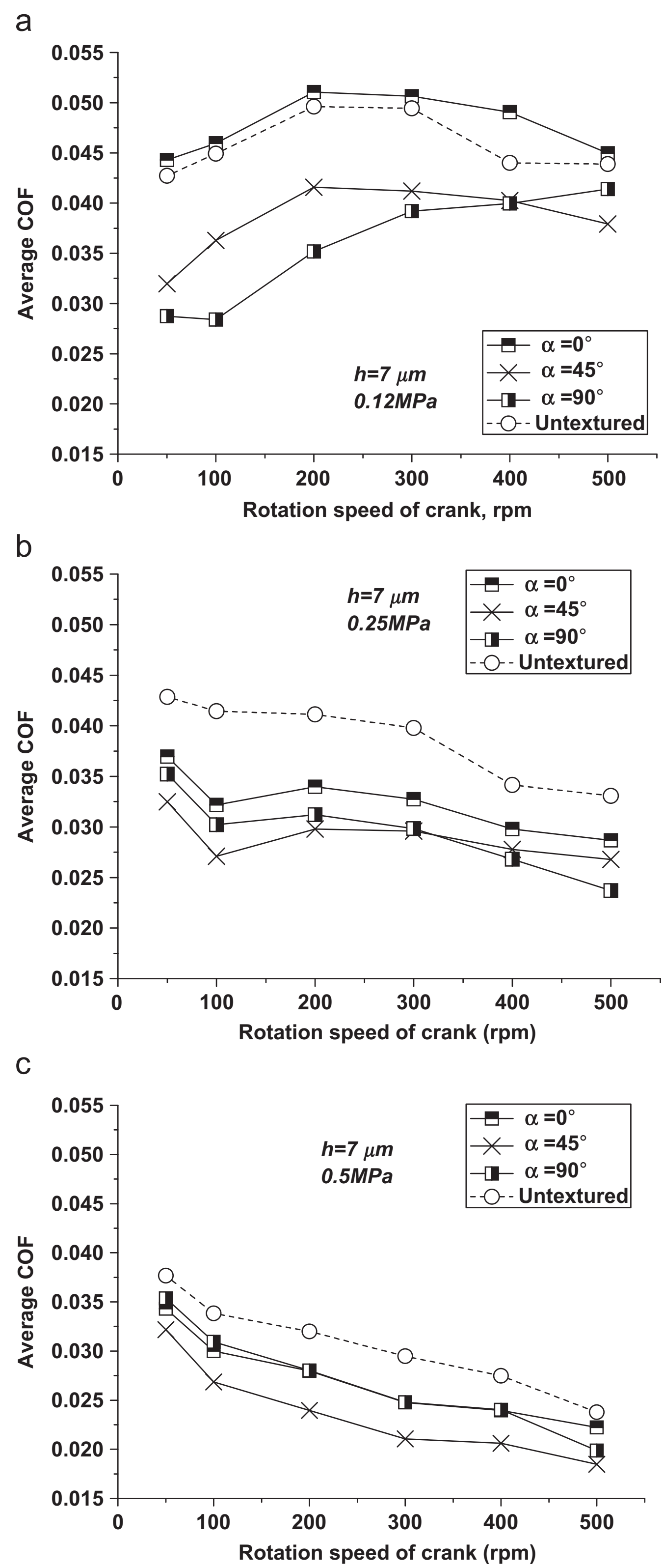

Fig. 6. Average friction coefficients of the specimens with the groove depth around 7 μm at different load and speed conditions.

friction reduction was obtained by the grooves 45° oriented to the sliding direction at the contact pressure of 0.5 MPa.

Fig. 7 shows the average friction coefficient of the specimens with groove depth of 19 μm at different load and speed conditions. In the low contact pressure case shown in Fig. 7(a), it is hard

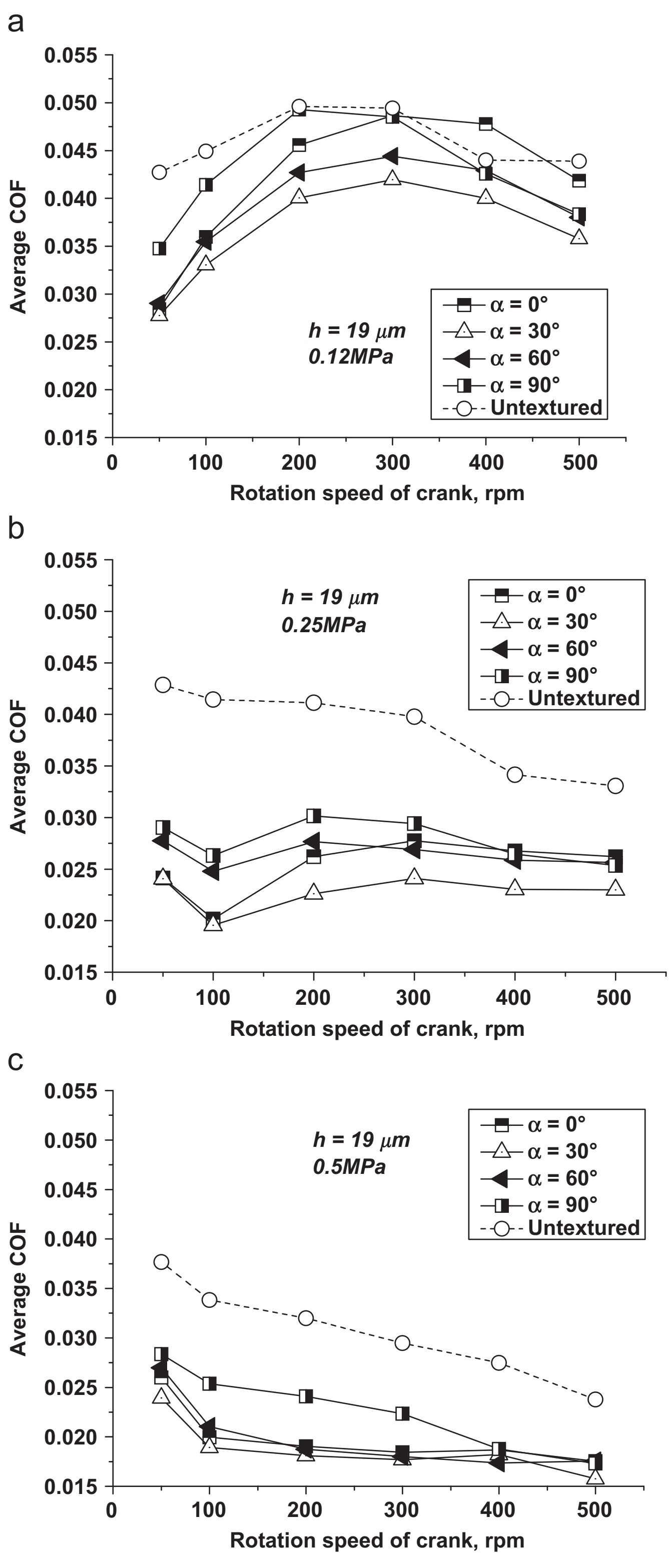

Fig. 7. Average friction coefficients of the specimens with the groove depth around 19 μm at different load and speed conditions.

to say the grooves perpendicular or parallel, which is better for friction reduction. Except the conditions of low rotational speeds, both of these patterns have no obvious friction reduction effect compared to the untextured specimen. Relatively, the grooves with angles of 30° or 60° show better effect of friction reduction. While the contact pressure increased to 0.25 and 0.5 MPa as shown in Fig. 7(b) and (c), clearly, all the textured specimens presented obvious friction reduction effect compared to the untextured specimen. And in most cases, the pattern with grooves parallel to the sliding direction is better than that of the perpendicular orientation. Furthermore, the best result of friction reduction was obtained by the grooves with angle between parallel and perpendicular to the sliding direction. The maximum reduction rate of the average friction coefficient could be up to 44%.

2.4. Discussion

The above data show that the influence of groove orientation on the friction performance is different at different groove depths and contact pressure conditions. For the purpose of comparing conveniently, Fig. 8 is drawn with the data in Figs. 6 and 7 at a rotational speed of 200 rpm. The dash lines represent the friction coefficient of untextured specimen.

It is clear that the orientation of micro-grooves has a significant effect on the friction behavior. At a low contact pressure 0.12 MPa as shown in Fig. 8(a), the grooves with depth 7 μm increased friction while they were parallel to the sliding direction, but gave the best results of friction reduction while they were perpendicular to the sliding direction. However, the grooves with depth 19 μm performed in a different way. Parallel was a little better than that of the perpendicular orientation. When the

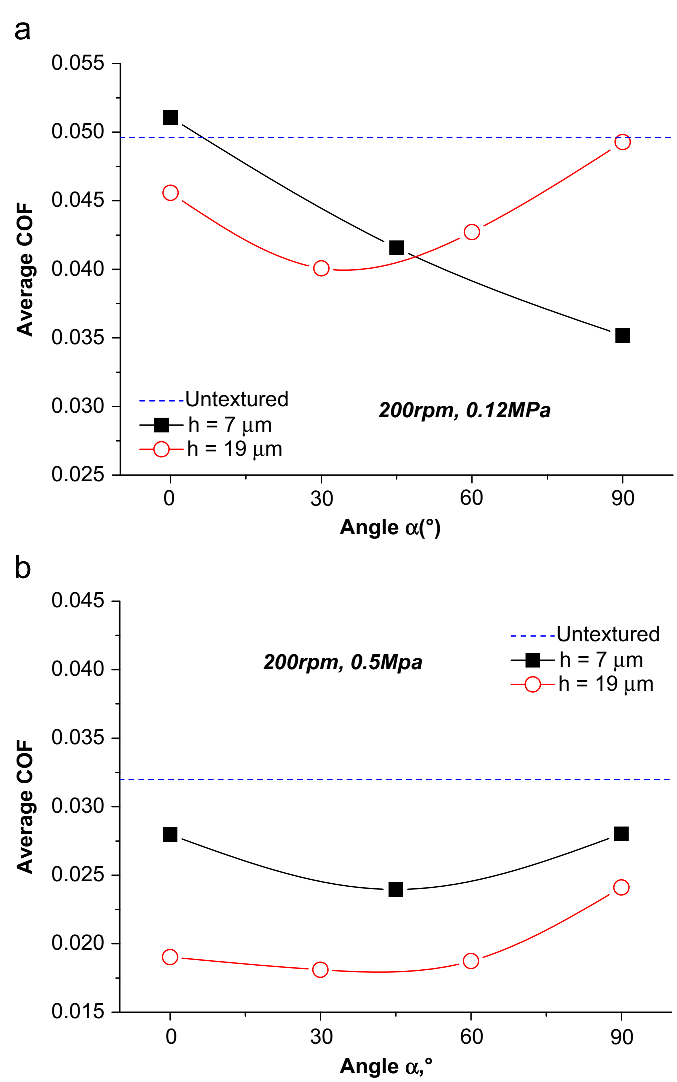

Fig. 8. Orientation effect of micro-grooves at different contact pressures.

contact pressure was increased to 0.5 MPa as shown in Fig. 8(b), the grooves with depth 19 μm became better than 7 μm, and parallel to the sliding direction was better than perpendicular orientation.

It is still a difficult question that how micro-groove influences the friction of sliding surfaces. We would like to analyze the influence of micro-grooves from the aspects of hydrodynamic effect, lubricant supply effect, and the contact stress on the surface.

2.4.1. Hydrodynamic effect

In previous studies on the patterns of micro-dimple on conformal sliding surface, hydrodynamic effect was regarded as the most important effect of surface texture. To optimize the geometry and distribution of micro-dimples to maximize additional hydrodynamic pressure was adopted as the most important process of surface texture design. While micro-groove is perpendicular to the sliding direction, it also works like a step bearing with certain width. Therefore, hydrodynamic effect of micro-grooves should be taken into account.

Based on the studies on the pattern of micro-dimples, there are several well accepted conclusions listing as follows.

- The hydrodynamic effect would become obvious at low-load high-speed condition of conformal contact [34].
- For the geometric feature such as micro-dimples, the aspect ratio (depth over diameter) is the most important factor influencing the hydrodynamic effect. Theoretical and experimental researches indicate that the value of aspect ratio at the range of 0.01–0.05 is preferable [6,35].
- For non-circular dimple, the strongest hydrodynamic lift is offered while its long axis is perpendicular to the sliding direction [29,36].

As shown in Fig. 8(a), perpendicular orientation of the grooves with depth of 7 μm give the best results of friction reduction compared to other orientations at low contact pressure. This could be well explained by the theories above. For the grooves with depth of 19 μm, its aspect ratio is 0.19, so far from the preferable range that hydrodynamic effect should not as significant as that with the depth of 7 μm. This also agreed with the opinion that shallow grooves can lead to a substantial increase in film thickness, while deep grooves can result in a local decrease or even collapse of the lubrication film [30].

While the contact pressure is increased to 0.5 MPa, the hydrodynamic effect is not as critical and effective as that at low contact pressure. Hence, perpendicular orientation did not show any good result in Fig. 8(b).

2.4.2. Lubricant supply effect

Working as lubricant reservoir to improve the boundary lubrication performance is considered as another important lubrication mechanism of surface texture. Additionally, for the connected textures like the pattern of micro-grooves, it has better ability to channel lubricant from high pressure region to low pressure region than the pattern of dimples. That should be the reason why the lubricant flow rate of the sliding bearing with micro-grooves could be 8% more than normal bearing [11].

In order to understand the behavior of lubricant flow in the grooves, an FEM analysis was carried out to simulate the flow of lubricant during sliding. Fig. 9 shows both the model and the simulation results.

A smooth plane surface is sliding against another surface, which has a groove with a depth of $h=4$ μm. There is a constant clearance $h_c=3$ μm between the two surfaces. An incompressible Newtonian liquid with a viscosity $\eta=0.0228$ Pa s is filled in the space between the two surfaces. The sliding direction is parallel to the groove, and the sliding velocity is 1 m/s.

Of course, the liquid is mainly driven by virtue of viscous drag force to flow along the sliding direction. However, by noticing the cross-section perpendicular to the sliding direction as shown in Fig. 9, the liquid also has a flowing direction from the groove region into the small clearance between surfaces. It is because the velocity gradient in the region of clearance is much larger than that in the region of groove. The difference of the velocity gradient in these two regions would generate a difference in fluid pressure, which enables the flow from the groove getting into the contacting area.

Therefore, while the grooves are parallel oriented to the sliding direction, the movement would drive the lubricant in the grooves to the surrounding area. That would be one of the reasons why parallel orientation also shows some good results, particularly with deeper groove and under high contact pressure as shown in Fig. 8(b).

2.4.3. Contact stress

Besides the advantages such as additional hydrodynamic lift and lubricant supply effect, surface texturing may also bring disadvantages to the sliding surfaces. As a simple example, surface texture

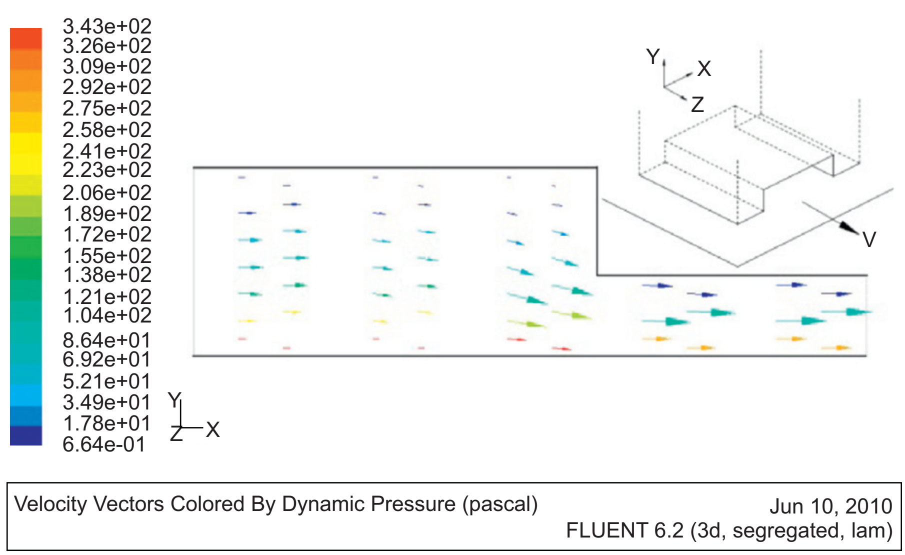

Fig. 9. Simulation results of lubricant flow on the cross-section perpendicular to the sliding direction.

increases the surface roughness, which is usually undesirable for building up full fluid lubrication. Theoretical optimum area density for micro-dimples based on hydrodynamic simulation could reach as high as 30–40% of the contacting area. But maybe due to the above reason, the optimum area density obtained by experiment is only in the range of 5–15%, lower than the theoretical results.

For the surfaces sliding in mixed lubrication regime, there is always a portion of the area keeping in solid contact. Hence, the geometric contour of surface texture would definitely influence the contact pressure contour.

An FEM analysis was carried out to simulate the contact situation of a groove during sliding. Fig. 10 shows both the model and the simulation results.

The contact problem was simplified as a block sliding against a plane with contact. So the two edges of the block are like the edges of groove perpendicular and parallel to the sliding direction, respectively. Both contacting surfaces are smooth ignoring surface roughness. The calculation conditions are set as follows:

- Contact pressure $P=0.5$ MPa.
- Young's modulus $E=120$ GPa.
- Poisson's ratio $\varepsilon=0.25$.
- Friction coefficient $\mu=0.10$.

For the purpose of easy understanding, the analysis results are illustrated by Fig. 10(b) and (c). It is obvious that there is relatively high stress at the edge while the groove is perpendicular to the sliding direction. Comparatively, there is almost no stress while the groove is parallel to the sliding direction.

This understanding could be helpful to explain the experimental results in Fig. 8. The contact stress problem may not be serious under low contact pressure while the portion of contact area is low. However, high contact stress would generate at the edge of grooves perpendicular to the sliding direction under high contact pressure, this is another reason that parallel grooves could be better than perpendicular orientation.

Therefore, the grooves with the angle between perpendicular and parallel to the sliding direction would take advantages of both perpendicular and parallel orientations, that is, proper hydrodynamic effect, proper lubricant supply ability, and not very high contact stress at the edge. Consequently, they usually have good frictional performance in most cases of this study.

3. Conclusions

Experiments were carried out to study the orientation effects of micro-grooves on the friction performance. Groove patterns with width of 100 μm, area density of 10%, and depth of 7 and 19 μm were fabricated on the surface of cast iron by lithography and the electrolytic etching technique. The orientation effects of micro-grooves on friction were evaluated by reciprocating tests of conformal contact with area of 20 mm × 20 mm at different normal load and speed conditions. Based on the experimental results, the orientation effects of micro-grooves were analyzed from the aspect of hydrodynamic effect, lubricant supply effect, and contact stress effect. The conclusions are summarized as follows:

(1) The orientation of micro-grooves has a strong effect on the friction performance of sliding surfaces. The merits of perpendicular and parallel orientation may swap under different contact conditions.
(2) Under a relatively low contact pressure in this research, the grooves with depth around 7 μm, and perpendicular to the sliding direction, have better effect on friction reduction than that of parallel orientation. Compared to the untextured surface, it could reduce friction up to 38.2%.
(3) Under a relative high contact pressure in this research, the grooves with depth around 19 μm, and parallel to the sliding direction, have better effect on friction reduction than that of perpendicular orientation. The ability to channel lubricant to

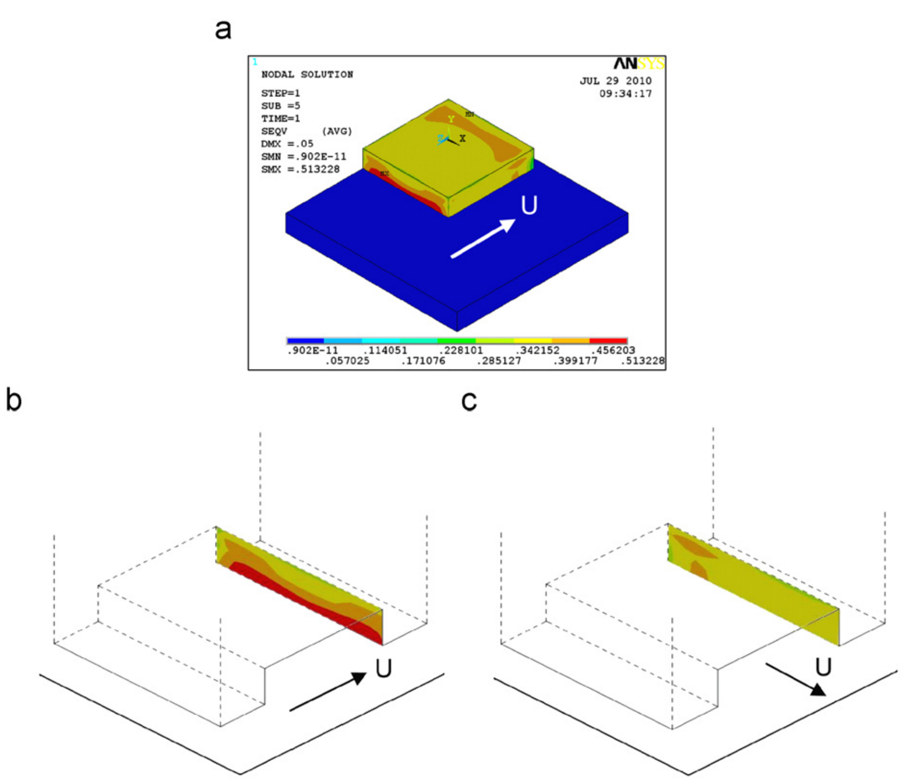

Fig. 10. Contacting stress caused by different sliding directions.

the frictional surface, and low additional contact stress are supposed to be the reasons.

(4) The grooves with an angle between parallel and perpendicular to the sliding direction takes advantage of both parallel and perpendicular orientations, as a result, it is usually the best at most cases. Compared to the untextured surface, it could reduce friction up to 44% at the contact pressure of 0.5 MPa.

Acknowledgments

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