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The Tribological Behavior of Cast Iron by Laser Surface Texturing under Oil-Lubricated Initial Line Contact for Rotary Compressor

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Abstract: The tribological behaviors of cast iron by laser surface texturing were experimentally compared with the behavior of untextured by unidirectional rotary sliding friction and wear tests under oil-lubricated initial line contact. The friction coefficient and temperature rise were analyzed with the increasing load applied by block-on-ring tests. In addition, the wear loss and wear mechanism were also investigated through the surface topographies analysis. The results showed that the tribological improvement strongly depended on the contact form. For the oil-lubricated initial line contact in this work, the textured surface showed a better frictional advantage with a lower friction coefficient and lower temperature rise. The hydrodynamic effect enhanced the load-carrying capacity of the oil film and increased the film thickness. The friction coefficients were 11~64% lower than those on the untextured one. Meanwhile, the textured surface deteriorated the wear behavior due to the coupling effect between the micro-cutting effect of the texture edges and the material deformations of the counter surface. The material loss induced by abrasive wear and fatigue wear was the dominant wear mechanism. Namely, the laser surface texturing improved the friction properties but reduced the wear resistance.

Keywords: textured surface; hydrodynamic effect; friction and wear; line contact; rotary compressor



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1. Introduction

Improvements in mechanical efficiency and wear reliability remain important for the rolling piston rotary compressors used in the refrigeration and air conditioning fields. In particular, with more attention being paid to the low global warming potential (GWP) refrigerants, such as carbon dioxide (R744) and propane (R290), as well as the development need for high speed and high pressure for refrigeration compressors, the typical common materials such as cast iron are unsuitable for a lower lubrication condition. Considering the self-lubricating property, friction-reducing performance, and low cost, the cast irons are widely applied and play an irreplaceable role. Laser surface texturing (LST) provides an achievable and potential means to improve the tribological properties. But the tribological benefits of LST highly depend on the lubrication conditions (for example, rich-oil [1,2], poor-oil [3–5] lubrication, or dry friction [6,7]) and the contact types (for example, face [8,9], line [10,11], or point contact [12]) to increase or decrease the friction. This is particularly difficult to achieve for rotary compressors due to their many sliding parts, including the different line and face contacts. So, the applicability of LST in cast iron for the rotary compressors needs further confirmation and analysis.

For refrigeration compressors, considerable efforts have been made to reduce the friction loss and elevate the wear resistance. Therein, the protective surface coating [13,14]

and structural optimization [15] are the current main means. For example, for the protective surface coating, He et al. [16] experimentally verified the feasibility of diamond-like carbon film (DLC) for scroll compressors and showed the DLC had a low friction coefficient and a high bonding strength due to its excellent wear-reducing and self-lubricating property. By reciprocating the pin-on-disk tests under dry contact, Solzak and Polycarpou [17] analyzed the influence of various chamber temperatures on the tribological behaviors of WC/C coatings in R134a ambient. The results showed that the steady-state friction coefficients decreased with the increasing temperature from 23 to 120 °C, while the wear rate first decreased and then increased due to the wear mechanism transforming from abrasive to adhesive wear. With the similar test rigs, subsequently, De Mello et al. [18] compared the DLC coating between three different refrigeration ambients (air, CO₂, and R600a) to find that the R600a chamber atmosphere presented the lowest friction coefficient (56.7% and 31.6% reduction, compared to CO₂ and air, respectively) and wear rate (25.2% and 34.6% reduction, compared to CO₂ and air, respectively). And, for the structural optimization, Liu et al. [19] developed a design optimization procedure for lower frictional losses with 14.1~18.1% reduction under specified conditions. Effective means included the narrower and shorter dimensions of the lower bearing and the smaller inner and outer diameter of the thrust surface. However, these effects increase the technical difficulty and require a higher cost, which makes applications of them difficult to implement.

Laser surface texturing (LST) by artificially fabricating the regular micro-pattern on surfaces has been applied to modulate the lubrication regime and tribological behavior. Its benefits also have been validated in many fields, including the mechanical seals, bearings, cylindrical face rings or piston rings [20–22]. The main mechanism can be concluded as follows. The micro-patterns provide a hydrodynamic effect for the greater load-carrying capacity of oil film and serve as the micro-containers for lubricant oil to guarantee lubrication or the micro-traps for wear debris to prevent further abrasion [23].

For the LST, up to now, the vast majority of efforts have focused on the face-to-face sliding contacts. The main focus has been on the geometrical optimization [24–26], the pattern comparison [27–29], and the coupling effect of the surface texturing and protective coating [30,31]. Ding et al. [32] optimized the geometric characteristics of a micro-dimpled texture to obtain the best friction reduction, including the dimple area density, hole depth, inclination angle, and slender ratio. The geometrical parameters generated significant influence on the friction coefficient. With the increased hole depth, the frictional coefficient decreased first, and then increased with a minimum at 7.1 µm. By a reciprocating sliding pin-on-disk test under dry friction, Zhan et al. [33] compared the tribological properties of six different single- or multi-shape textured surfaces. The sinusoidal textured surfaces had the lowest friction coefficient than other patterns, such as the multi-shape or dimpled one. Liu et al. [34] coupled the elliptical dimpled textures and DLC coating for a gas seal and found the average friction coefficient had a 46.2% reduction compared with the smooth sample.

Elliptical micro-dimples, as the most typical textured pattern, have been verified under oil-lubricated face contact. A stronger hydrodynamic effect generates a 26.3% increase in the load-carrying capacity, compared to a normal circular dimple pattern [8], due to the fluid's cumulative effect. And its friction coefficient can be reduced by 10~20% compared with other dimple patterns [32]. However, under the same oil lubrication, ambiguous and conflicting results have been presented for line and point contacts, which lead to difficulty in industrial application. Kovalchenko et al. [12] stated that the LST, by dimpling under lubricated initial point ball-on-flat contact, facilitated the transition from the boundary to mixed lubrication and reduced the friction coefficient and, at the same time, accelerated the face wear due to the reduced oil film thickness. Nevertheless, Andersson et al. [27] found that the LST by dimpling significantly reduced the friction and wear of the point counterface. In addition, Hao et al. [35,36] performed an experimental investigation to exhibit that the surface textures with a lower dimple area density and larger dimple diameter provided a better friction performance in line contacts. Da Silva et al. [37] investigated the

tribological behaviors of cast iron by MECT (Maskless Electrochemical Texturing) using lubricated block-on-ring tests. The textured surface showed a larger reduction in friction (up to 60%) and wear (higher than 80%) compared the untextured sample. Nevertheless, Meylan et al. [38] investigated the effect of the geometrical parameters of micro-textures on cast iron reciprocating against steel. The dimple depth and diameter were the most significant factors. The textures showed a negative effect on the tribological properties due to oil trapping or an increase in local pressure.

Despite its new role in refrigeration compressor applications, the LST is gradually attracting more attention for friction reduction and anti-wear enhancement, with verified feasibility in some cases. Nagata et al. [39] applied the micro-textures to a thrust bearing of a reciprocating compressor. The friction loss saw a 20~60% reduction, and the performance coefficient had a 1.4% increase. And similar results were also confirmed in rotary compressors by Lyu and Polycarpou [40]. The thrust bearings with optimized geometric parameters significantly reduced the contact force and wear depth at high frequency. Mishra et al. [41] investigated the tribological behaviors of LST for a scroll compressor within different refrigeration ambients, and the surface texturing showed significant tribological improvements, largely independent of the type of lubricant or refrigerant.

According to the above descriptions, for certain applications of LST in compressors, the effects focus on the face-to-face sliding contact such as the thrust bearing [42]. There are few publications on the tribological performance of line contacts. However, this is the main source of friction power consumption in the rotary compressors. And its applicability and the advantage of the tribological properties under line contact remain unclear and need to be investigated, which is the main topic of this article.

So, the tribological behavior of cast iron by laser surface texturing under oil-lubricated initial line contact for rotary compressor was analyzed experimentally by a block-on-ring test. The friction coefficients, wear depth, and wear width were compared with untextured cast iron under different operating loads. The wear topographies were also analyzed for the wear mechanism.

2. Tribological Experiments

2.1. Samples

The schematic diagram of a rolling piston rotary compressor was shown in Figure 1. The main structural units included the rolling piston, sliding vane, crankshaft, cylinder, upper bearing, and bottom bearing. Therein, the cylinder, upper bearing, and bottom bearing enclosed a working chamber for suction and compression. Then, the rolling piston, sliding vane, and crankshaft were installed inside the chamber, and made the relative sliding.

In this manuscript, a block-on-ring friction couple with initial line contact (as shown in Figure 2) was selected based on the two contact geometries: (1) the rolling piston sliding against the sliding vane, and (2) the crankshaft sliding against the upper bearing, or bottom bearing. In the analysis of wear issues, the initial state was regarded as a line contact based on the actual contact stress of rotary compressor, far exceeding a face contact. With the further development of wear and tear, the initial line contact made a transition towards a face contact.

In Figure 2, the block, namely a static one with the dimensions of $15.75 \times 6.35 \times 10.16$ mm (length, width, and height, respectively), was made of HT250 cast iron coming from the typical bearing material. A hole for mounting a solid temperature sensor with a diameter of 3 mm and depth of 5 mm was machined on the non-contacting face.

The elliptical micro-dimpled textures were processed by laser surface texturing (LST) on the contacting face of block specimen. The major diameter was 0.4 mm, the minor one, 0.2 mm, and the depth was designed a certain value of 6 μ m [32]. The center distances between adjacent micro-dimples were 1 mm and 0.5 mm in the major and minor axes, and the major axis was parallel to sliding rotating direction of the ring specimen. By calculation, the dimple area density, namely the percentage of the summation of the dimpled area to

the lubrication area, was 10.36%. The detailed meaning and dimension of each variable is listed in Table 1.

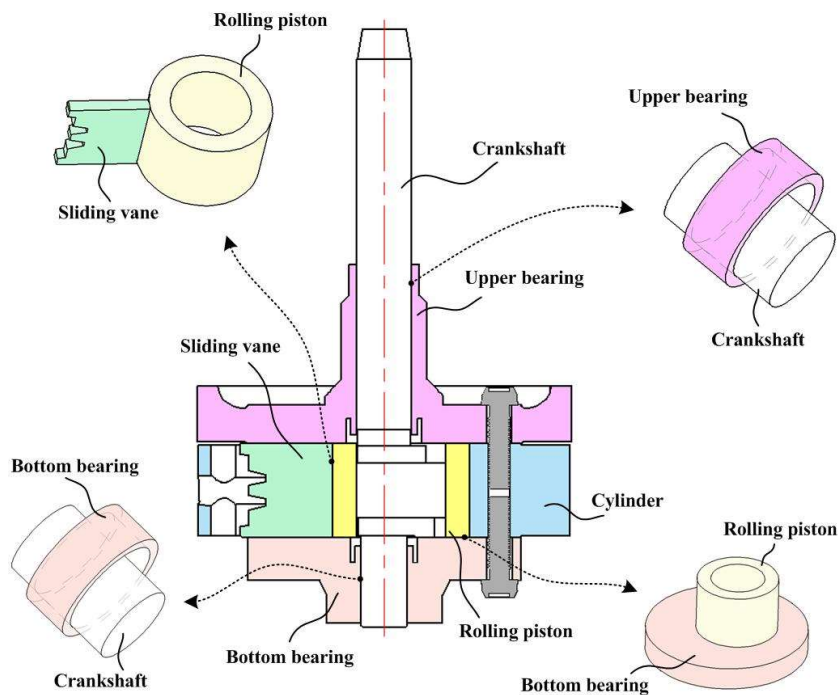


Figure 1. The schematic diagram of the rolling piston rotary compressor.

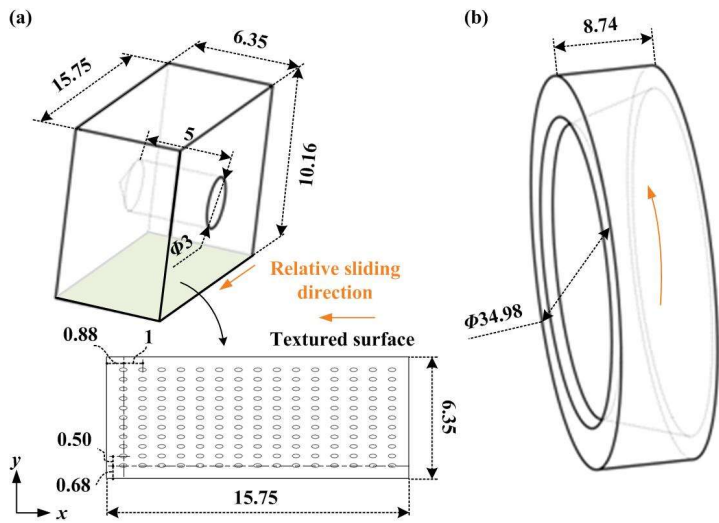


Figure 2. Dimensions of the test samples: (a) block specimen; (b) ring specimen.

Table 1. Meanings and dimensions of the different geometrical parameters of elliptical dimpled textures.

No.	Designed Depth/ μm	Dimple Diameter/mm		Dimple Area Density	Dimple Numbers		Dimple Patterns
		Major	Minor		x-Direction	y-Direction	
1#	--	--	--	--	--	--	Untextured
2#	6.0	0.4	0.2	10.36%	15	11	Ellipse

The ring, namely the rotating one with an external radius of 34.98 mm and thickness of 8.74 mm, was made of QT600 cast iron, a common crankshaft material in a rolling piston rotary compressor. Its inner surface was designed with a conical surface fastened on the rotating shaft driven by a motor.

Before laser processing, the original surfaces of the test blocks and rings were polished, and the roughness Ra was controlled with a range of 0.10~0.15 μm . The topographies were measured by a white-light-interfering 3D profilometer supported by BRUKER Contour GT-K, shown in Figure 3. Taking the ring specimens as an example, three measuring points evenly distributed along the circumference were selected for multiple specimens. Finally, the average value of the Ra was obtained within 0.126~0.137 μm , as shown in Table 2.

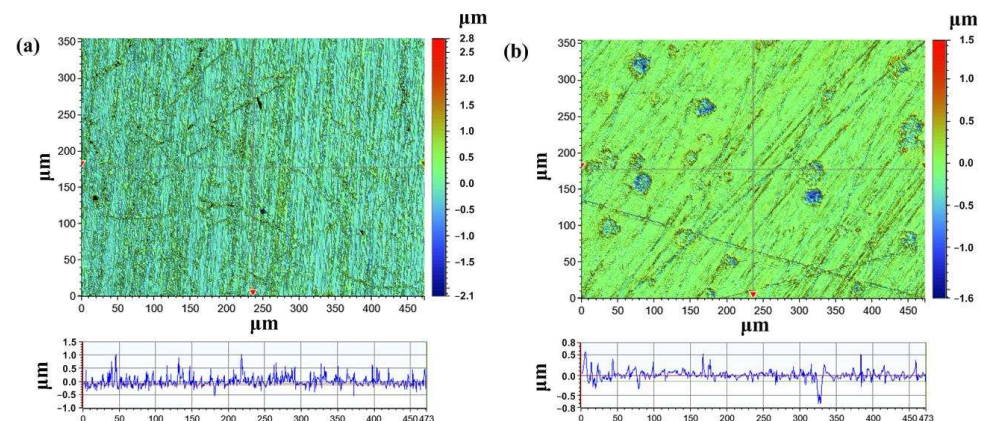


Figure 3. Topographies of the test samples: (a) block specimen of HT250; (b) ring specimen of QT600.

Table 2. Roughness tests with Ra (μm).

Specimen Numbers	Test 1#	Test 2#	Test 3#	Average Value
No. 1#	0.136	0.127	0.138	0.134
No. 2#	0.139	0.143	0.128	0.137
No. 3#	0.122	0.135	0.133	0.130
No. 4#	0.130	0.121	0.127	0.126
No. 5#	0.114	0.138	0.129	0.127

Figure 4 analyzes the surface roughness influences on the frictional coefficient noted as f . The roughness of both two friction surfaces was changed synchronously from 0.025 to 0.7. At least three tests were used to average the friction coefficient in the steady wear phase. The applied load was 200 N and the rotational speed was 1000 rpm. Each group test was conducted for 60 min. After testing, the friction coefficients were analyzed. As the Ra decreased, the f first decreased and gradually steadied. By fitting the curve with an equation of $f = 0.03371 + 0.01995 \cdot Ra + 0.08341 \cdot Ra^2$, the error influence of the roughness difference on f was restricted to 1.2%.

After finishing polishing, all test specimens were cleaned in an ultrasonic cleaner with acetone and alcohol and dried in an oven. The micro-dimpled textures were fabricated by a fiber optical laser from HGTECH LSF20 with a wavelength of 1064 nm. The processing parameters contained the laser power of 7 W, a scanning speed of 800 $\text{mm} \cdot \text{s}^{-1}$, a frequency of 80 kHz, and three overscan. The obtained textured topographies shown in Figure 5 had a measured hole depth of 5.95 μm and 6.03 μm in the x - and y -directions, respectively, instead of the designed value of 6 μm . The textured surfaces were polished and cleaned repetitively to remove ridges or bulges around the dimples, owing to the metal melting by thermal diffusion. The roughness Ra on the non-textured surface was also controlled at 0.10~0.15 μm .

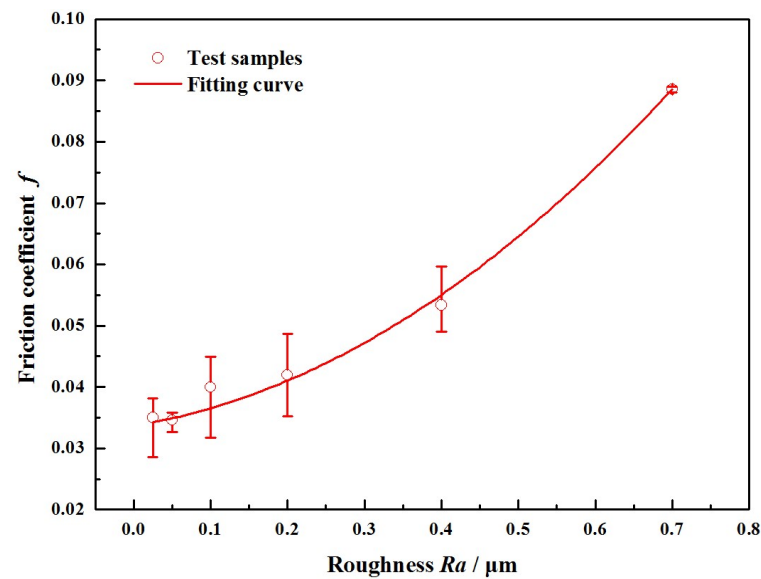


Figure 4. Surface roughness influences on the frictional coefficient.

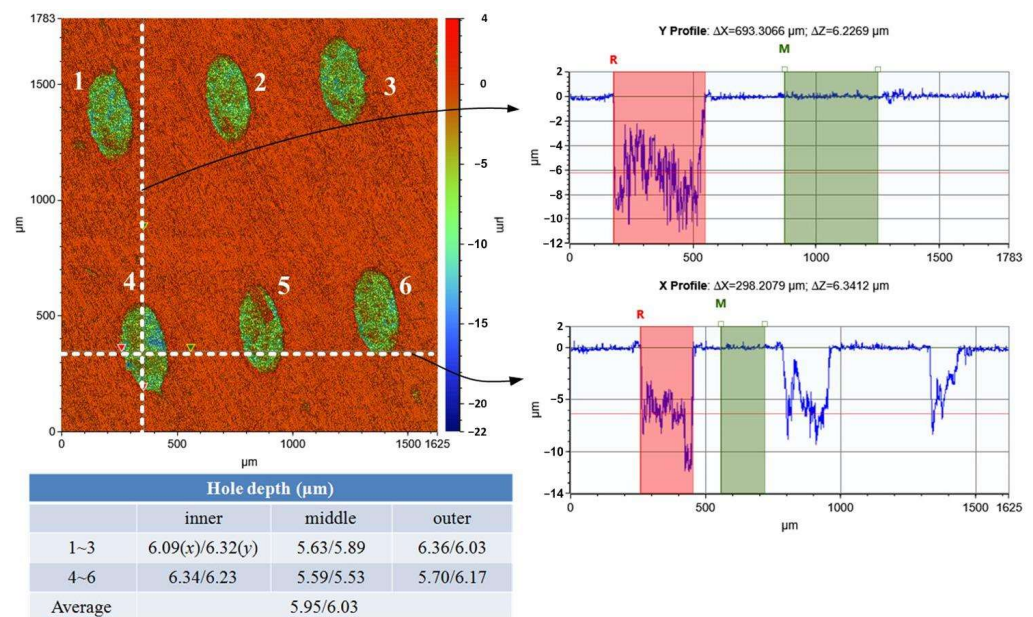


Figure 5. Topographies of the elliptical dimpled surface.

2.2. Testing Parameters

The tribological behaviors of the laser textured specimens were compared with the untextured one under oil-lubricated initial line contact conditions with different operating loads by a FALEX block-on-ring test machine, shown in Figure 6. The block-on-ring friction pairs were submerged into the lubricant (FV50S) with a set volume of 100 mL under ambient pressure. The ring specimen with smooth surfaces was pressed tightly on the rotating shaft by the conical surface at the inner diameter with a set rotation speed of 1000 rpm. The block specimen with textured or untextured surfaces was installed directly above the ring and endured the applied load vertically to form the initial line contact. A specimen temperature sensor was inserted into the mounting hole of the block specimen to measure the solid temperature, and another oil temperature sensor was mounted at the top of the oil pool reaching below the oil level.

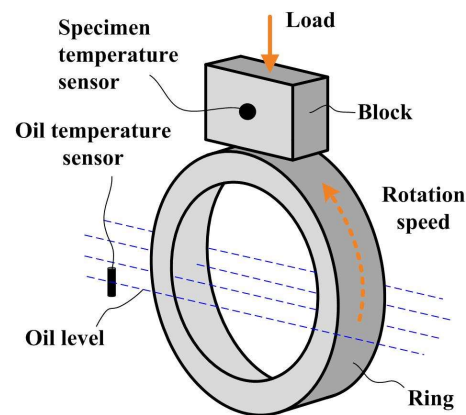


Figure 6. Schematic of the test rig.

Table 3 lists the operating conditions. Each group test for the different applied loads was conducted for 60 min. The rotational speed was limited by the tribometer. The applied load was based on the actual contact stress of rolling piston rotary compressor during operation. The friction coefficient, solid temperature, and oil temperature were measured. After completing the testing, the wear topographies of the contacting surfaces on the block specimen were analyzed by SEM analysis (FEI Quanta 250).

Table 3. Operating conditions during the tests.

Item	Symbol	Dimensions and Data
Rotational speed	ω	1000 rpm
Applied load	W	356, 445, 556, 667, 778, 889 N
Running time	t	3600 s
Lubrication condition	/	oil lubrication with 100 mL

To determine the oil volume and ensure oil lubrication, a visualization test was used to observe the lubrication status. PC (Polycarbonate) material was used to machine the cover plate of the oil pool. Figure 7 presents the oil level change of the pre- and post-test for 100 mL of lubricating oil. Before testing, the static oil level only reached half of the ring height. However, with the steady rotation of the test ring, the lubricating oil was sucked into the frictional interface between block and ring, which finally provided a sufficient oil lubrication condition. Meanwhile, the 100 mL oil volume also guaranteed that the oil temperature sensor was always below the oil level.

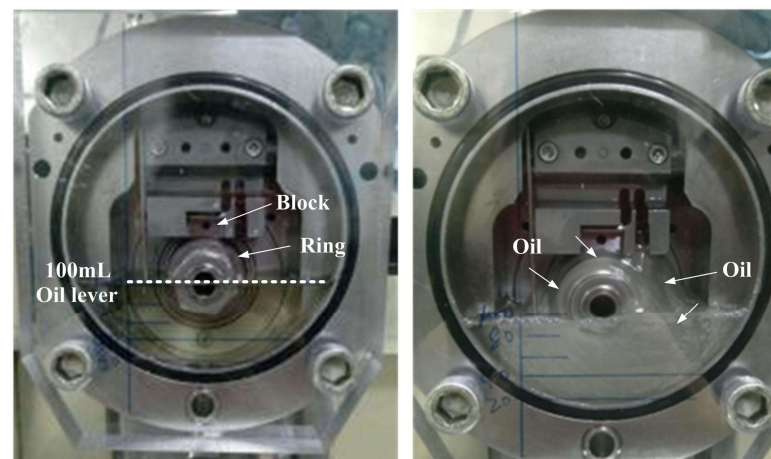


Figure 7. Visualization of the lubrication status.

3. Results and Discussion

3.1. Friction Characteristics

Figure 8 shows a comparison of the tribological behaviors of the textured surface with the untextured one with the increase in the running time under different applied loads, including the friction coefficient f , the oil temperature T , and the specimen temperature T_s . At the applied load of $W = 356$ N, as shown in Figure 8a, with the running time increasing from 0 to 60 min, the f on the untextured surface slowly increased in the running-in wear phase of 1204 s, then remained steady and entered a steady wear phase. For the textured surface, the longer running-in time lasted up to 2821 s, and the f increased rapidly up to ~ 0.025 and then decreased slowly and steadied with a larger fluctuation. The reason for this can be attributed to the fact that, despite the polishing, the residual ridges or bulges around the dimples owing to the metal melting during laser processing extended the running-in process. But, when both surfaces entered the steady wear, the friction coefficients had no significant difference and maintained at ~ 0.010 .

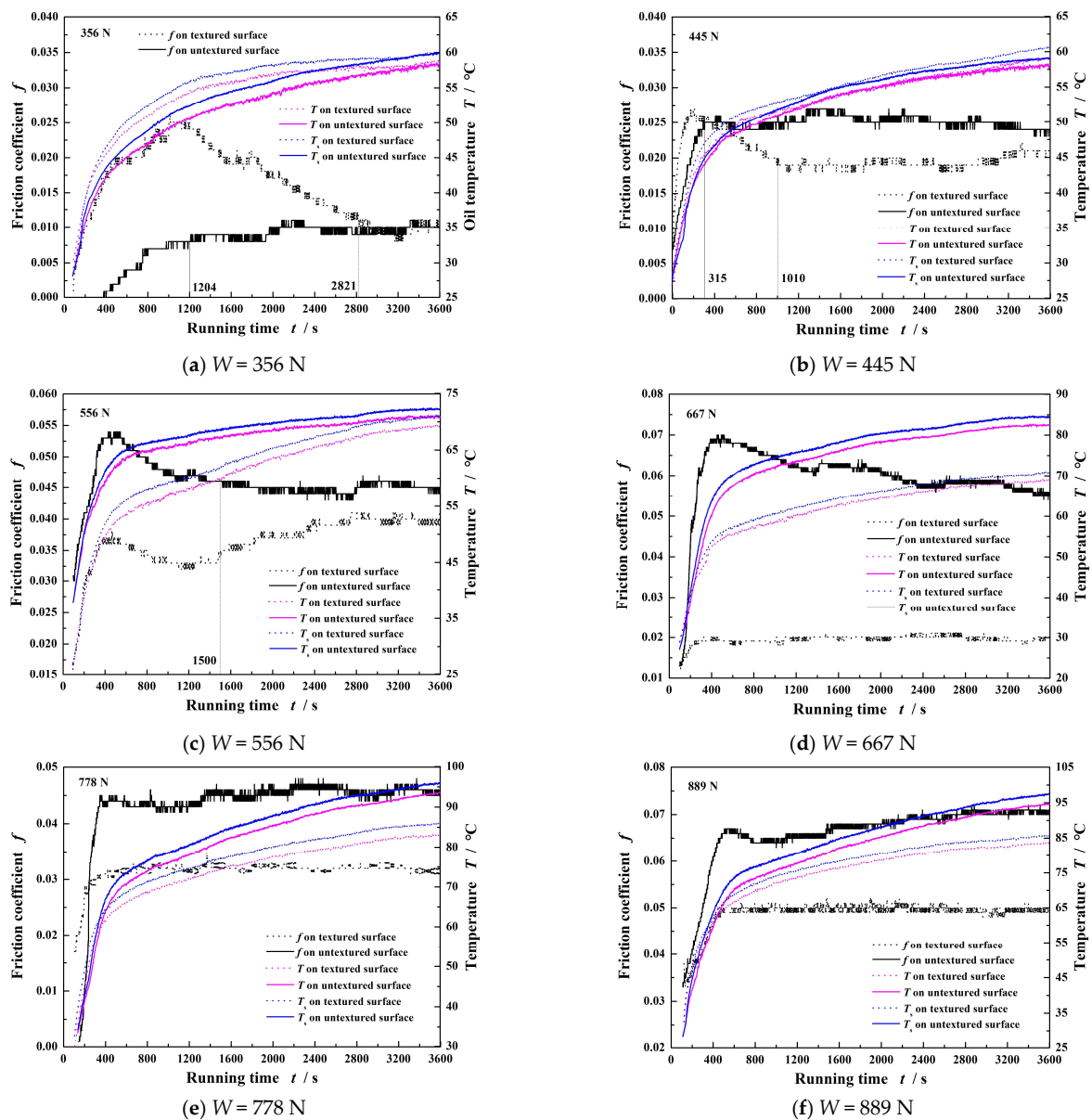


Figure 8. Friction coefficients and temperature changes of the untextured surface and textured surface with the increase in the time and loads.

In addition, the T and T_s gradually increased. The running-in wear phase presented a larger increase in amplitude, but during the steady wear phase, the increase in the amplitude decreased. Compared with the untextured surface, the textured one generated a higher temperature rise due to more friction heat coming from the larger friction coefficient. However, after reaching the steady wear state, the T and T_s of the two surfaces were similar. The T and T_s were 58.8 °C and 60.0 °C for the textured specimen and, 58.4 °C and 60.1 °C for the untextured specimen, respectively. The specimen temperatures T_s were 1.2 °C and 1.7 °C higher than the oil temperatures T for the textured and untextured specimens, respectively.

When the W continuously increased to 445 N, the textured surface started to gradually present a better friction reduction, as shown in Figure 8b. The friction coefficients of the two different surfaces showed a similar tendency, first increasing rapidly and then decreasing slowly during the running-in wear phase with 315 s for the untextured one and 1010 s for the textured one, finally maintain stability during the steady wear phase with $f = 0.024$ and 0.021 for the untextured and textured ones, respectively. The temperature changes were also relatively close. The T of 59.4 °C and T_s of 60.8 °C for the textured specimen were slightly higher than the untextured specimen with 58.1 °C and 59.1 °C, respectively. The specimen temperatures T_s were 1.4 °C and 1.0 °C higher than the oil temperatures T for the textured and untextured specimens, respectively.

With the increase in the W , shown in Figure 8c–f, the advantages of the friction reduction and slower temperature rise for the textured surface were gradually expanded. The running-in time was roughly equivalent to that of the untextured surface, for example, at $W = 556$ N, as shown in Figure 8c, where both were ~1500 s. In addition, the temperature rise for the textured surface was significantly lower than the untextured one, owing to the visibly reduced friction coefficient.

As shown in Figure 8c, at $W = 556$ N, the textured surface in the final steady phase had an f of 0.040, a T of 69.7 °C, and a T_s of 71.7 °C, and the untextured surface had an f of 0.045, a T of 70.8 °C, and a T_s of 72.3 °C.

As shown in Figure 8d, at $W = 667$ N, the f on the textured surface was 0.021, the T was 69.2 °C, and the T_s was 70.9 °C, and for the untextured surface, the f , T , and T_s were 0.056, 82.6 °C, and 84.5 °C, respectively. Compared with the textured surface, the oil temperature and specimen temperature on the untextured surface increased by 13.4 °C and 13.6 °C, respectively.

With the increase in the W from 778 N, as shown in Figure 8e, to 889 N, as shown in Figure 8f, the f on the textured surface increased from 0.032 to 0.050, and the T increased from 83.3 °C to 95.0 °C, and the T_s increased from 86.1 °C to 98 °C. Correspondingly, the f on the untextured surface increased from 0.046 to 0.071, the T increased from 93.6 °C to 99.2 °C, and the T_s increased from 96.2 °C to 101.8 °C.

The friction coefficients of the steady phase (in Figure 8) for the textured and untextured specimens under different applied loads were shown in Figure 9. Generally speaking, as the W increased, the f showed a gradually increasing trend. This is typical friction behavior of the boundary to mixed lubrication modes. Similar experimental results were obtained by Wakuda et al. [43]. It is also noted that the textured surface had a better frictional advantage with a lower friction coefficient, and the advantage difference gradually expanded with the increase in the W . At $W = 356$ N, the friction coefficients had no significant differences for the two surfaces. From $W = 445$ to 889 N, the f on the textured surface was 12.5%, 11.1%, 64.3%, 30.4%, and 29.6% less than that on the untextured one. Hao et al. [36] and Wakuda et al. [43] also verified the beneficial lubrication effects of surface texturing under oil lubricated line contact. This could be due to the micro-dimples providing a hydrodynamic effect leading to the greater load-carrying capacity of the oil film and preserving the lubricant to supply an oil source under the applied load. As a result, the textured dimples are beneficial, improving the lubrication and reducing the friction.

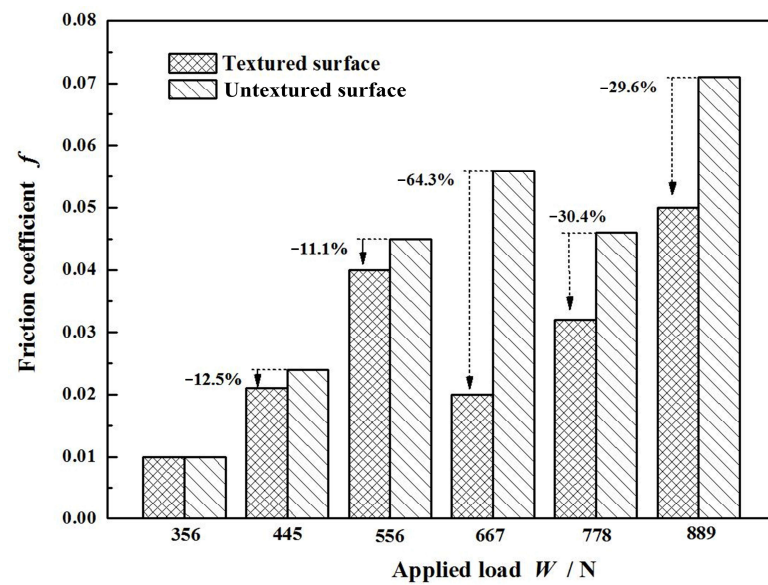


Figure 9. Friction coefficient with the increase in the applied load.

Figure 10 summarizes the temperature changes with the increase in the applied load. The oil temperature and specimen temperature were positively correlated with the friction coefficient change. Similarly, the f , T , and T_s increased especially when $W > 556$ N. The textured surface presented a slower temperature rise compared with the untextured surface, with a decrease of 1.1 °C, 13.4 °C, 10.3 °C, and 4.2 °C for T and a decrease of 0.6 °C, 13.6 °C, 10.1 °C, and 3.8 °C for T_s at $W = 556, 667, 778$, and 889 N, respectively. Furthermore, the T_s was 1.0–2.6 °C higher for the untextured surface and 1.2–3.0 °C higher for textured surface than the T .

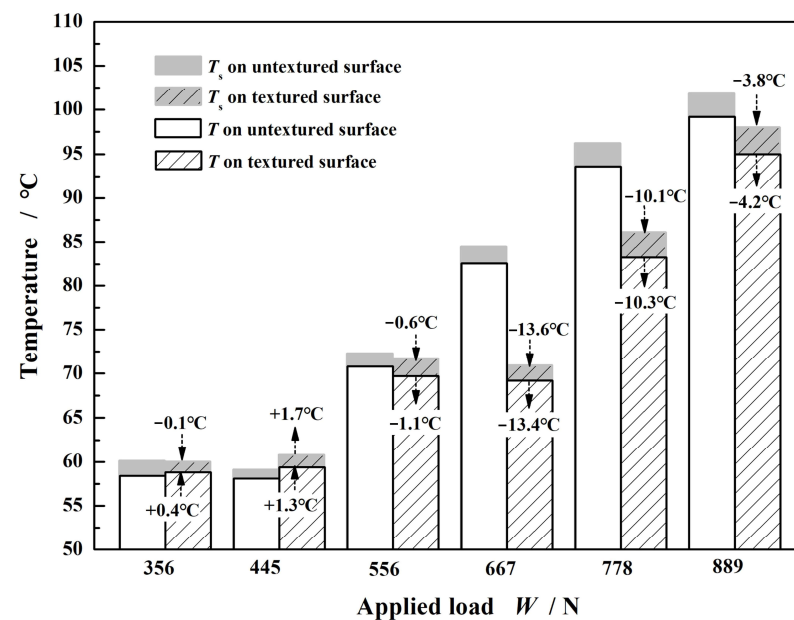


Figure 10. Temperature changes with the increase in the applied load.

3.2. Wear Performances

Wear scars on the block surfaces were measured by a white-light interfering 3D profilometer, as shown in Figure 11. Eight measuring points evenly distributed along the width direction of the block specimen were selected for the wear width and depth analysis. The average values were used to compare the wear characteristics between the textured

and untextured surfaces. Taking $W = 356$ N as an example, for the textured surface, the average wear depth $h = 1.27$ μm was 38.04% greater than that of the untextured one with $h = 0.92$ μm , and the average wear width $L = 620$ μm was 4.91% greater than that of the untextured one with $L = 591$ μm .

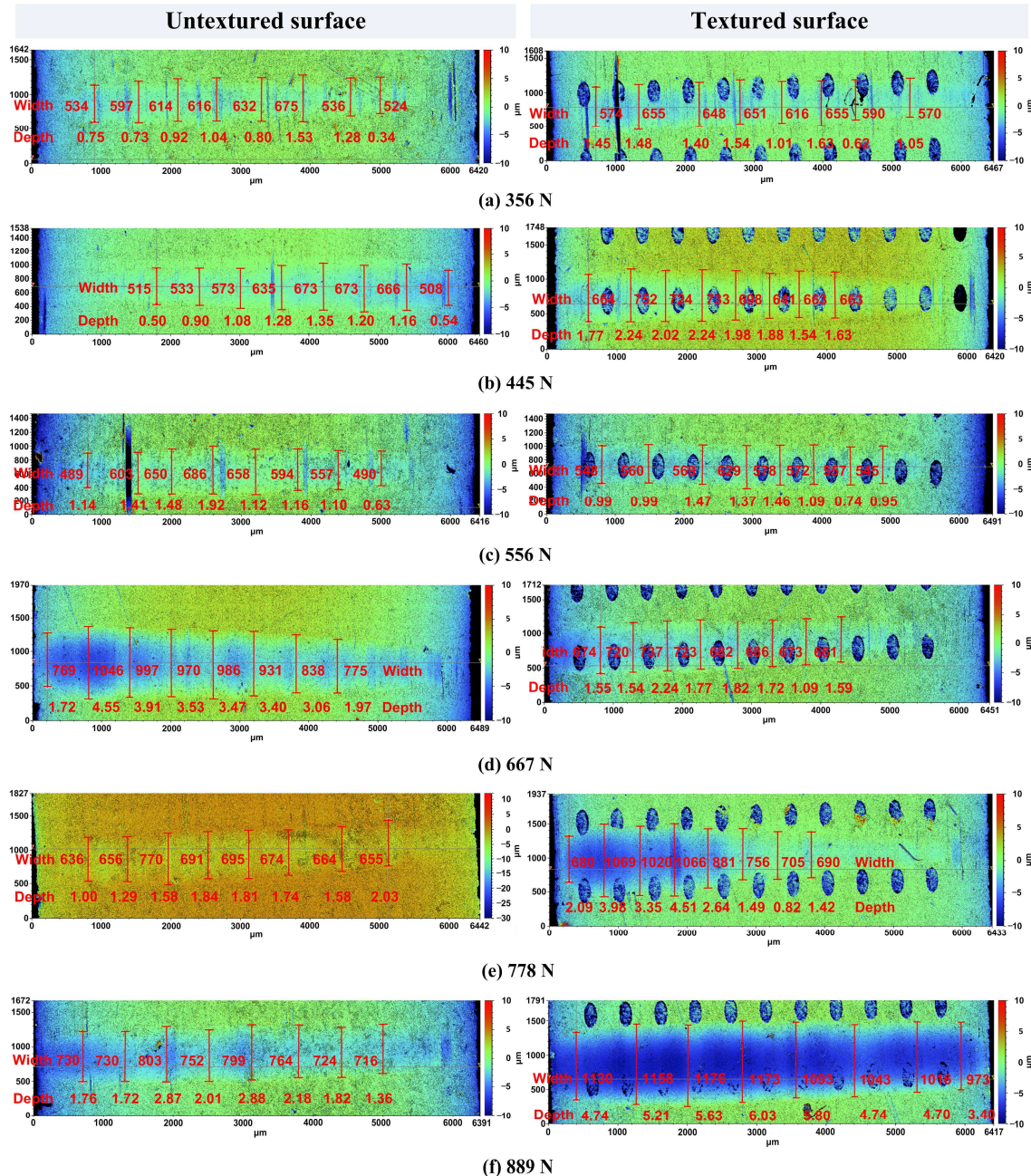


Figure 11. Measurements of the wear scars on the untextured and textured blocks with the increase in the applied loads.

The average wear depth h and wear width L were summarized and compared in Figure 12 for two different specimens. With the increase in the applied load, the h and L exhibited an overall increasing trend, which illustrates that the surface wear further deteriorated. In addition, the wear loss for the textured one was much larger than the untextured one. This is in contrast to the friction coefficient changes. So, the laser surface texturing (LST) did not improve the wear resistance and instead increased the wear loss.

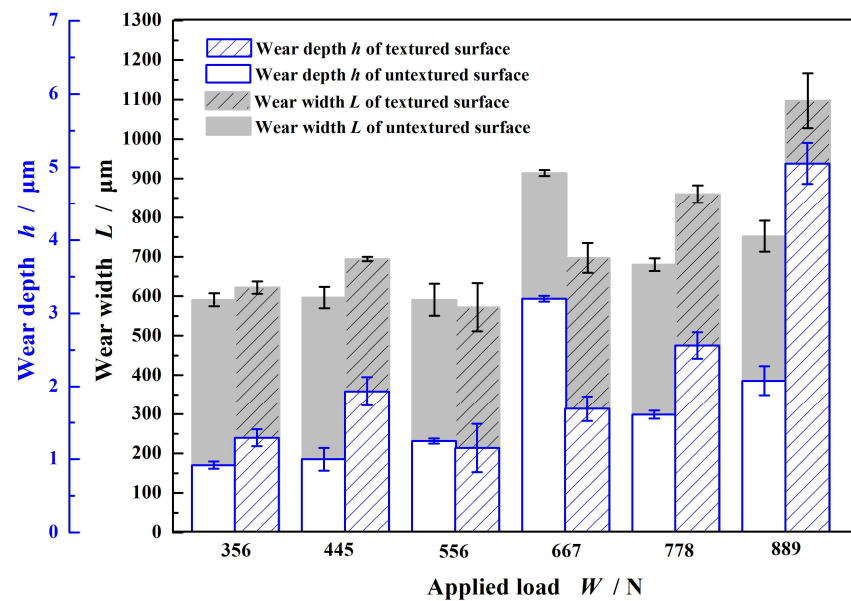


Figure 12. Wear depth and wear width with the increase in the applied load.

By analyzing the friction coefficient and wear loss comprehensively, shown in Figures 9 and 12, differing in oil-lubricated face contact with friction reduction and wear resistance [32], the LST optimized the friction coefficient but deteriorated the wear behavior under the present oil-lubricated initial line contact. Namely, compared with the untextured surface, LST has the potential to reduce the friction power consumption in refrigeration and air-conditioning compressors but needs to avoid direct contact and collision between friction pairs for material reliability and durability.

Figure 13 details the wear phenomenon shown in Figure 11. On the one hand, compared with the untextured face, the present textured contact surface suffered from a greater localized unit linear load due to the smaller real contact area. It then increased the contact width to more wear loss, based on the Hertz contact theory.

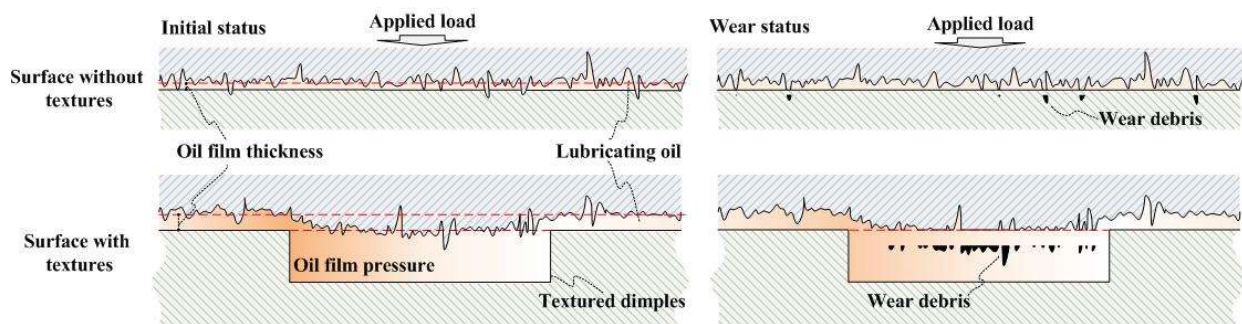


Figure 13. Schematic of the wear mechanism.

On the other hand, during the wear process along with asperity contact, for the untextured specimen, the surface asperities were worn down to form wear debris. However, for the textured surface, the hydrodynamic effect induced by the textured micro-dimples enhanced the load-carrying capacity of the oil film to balance the applied load. It follows that the increasing film thickness reduced the friction coefficient as a benefit. Meanwhile, the textured micro-dimples also broke the continuity of the contact region profile, and the counter surface against the textured region lacked adequate support. As a result, the material deformations forced the counter surface to protrude into the textured cavity. While rotary sliding, the micro-cutting effect by the texture edges generated repetitive cutting wear between the textured region and its counter region. So, the textured surface caused

more severe wear loss than the untextured one. Overall, the LST improves the friction properties but reduces the wear resistance in the present conditions.

After the completion of the tests, the wear topographies on the frictional surfaces of block specimens analyzed by SEM are shown in Figure 14. The wear topographies between textured and untextured surfaces had no significant difference. The wear mechanisms were quite similar. So, the SEM results on the untextured surface without the influence of textured dimples were shown. With the increase in the load, the edges of the worn area were more significant, which illustrates that the wear losses were increasing. And the wear mechanism was also transformed from the scratches induced by abrasive wear and mechanical rubbing on the plastic extrusions. As a result, the scratches on the worn area gradually disappeared, and the surface topographies became flatter and smoother. In addition, the crack propagation by fatigue wear may also be formed and strengthened under cyclic loading. At $W = 778$ N, material spalling occurred.

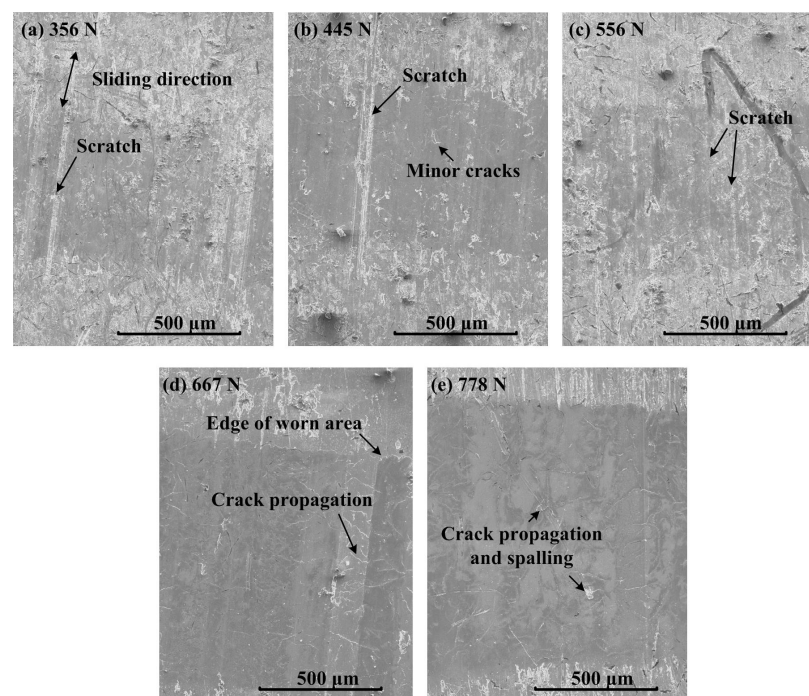


Figure 14. Wear topographies on the untextured block specimens.

4. Conclusions

In this paper, the tribological behaviors of cast iron by laser surface texturing were compared with the behavior of the untextured one under oil-lubricated initial line contact with different operating loads via a block-on-ring test. The friction properties and wear topographies were discussed. The following conclusions can be summarized:

- (1) The tribological improvement brought about by textured surfaces strongly depends on the contact form. For the oil-lubricated initial line contact in this study, the textured surface showed better frictional advantage with a lower friction coefficient and slower temperature rise. The advantage gradually expanded with the increase in the applied load. The friction coefficients on the textured surface were 11~64% lower than the friction coefficients on the untextured one. However, the textured surface deteriorated in terms of the wear behavior with a greater wear depth and width; namely, the laser surface texturing (LST) did not improve the wear resistance.
- (2) The textured micro-dimples generated a hydrodynamic effect to enhance the load-carrying capacity of the oil film and increase the film thickness, thus reducing the friction coefficient. However, the coupling effect between the material deformations of counter surface and the micro-cutting effect of the texture edges generated repetitive

cutting wear between the textured region and its counter region, which caused more severe wear loss than for the untextured one. The abrasive wear and fatigue wear were the dominant wear mechanisms.

- (3) Differing from wide applications in face contact, for the initial line contact, laser surface texturing has the potential to reduce the friction power consumption in refrigeration and air-conditioning compressors. But it needs to avoid direct contact and collision with the high applied load between friction pairs for material reliability and durability. The coupling processing of the textured surface and a protective surface coating is an important way to keep the friction reduction and improve the wear resistance.

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