Efficiency of Laser Surface Texturing in the Reduction of Friction under Mixed Lubrication

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Abstract

The tribological behavior of steel sliding pairs with dimples ranging from 15 to 800 µm in diameter was characterized in the mixed lubrication regime in a pin-on-disk experiment. The pin was textured, keeping the total dimpled area constant at 10% and the depth-to-diameter ratio at 0.1. Polyalphaolefin (PAO) was used as a model lubricant. Experiments were carried out under unidirectional sliding conditions at 50 and 100°C. At constant depth-to-diameter ratio the results showed a significant non-linear dependence of the friction coefficient on the texture diameter, sliding speed and the oil temperature (viscosity). A friction reduction of up to 80% was possible with the optimal diameter for certain sliding speeds. The dimple diameters leading to the highest friction reduction significantly depend on the oil temperature. By reducing the oil temperature from 100 to 50°C the dimple diameter resulting in the highest friction reduction changed from 40 to 200 µm.

Keywords: Surface engineering, laser surface texturing, mixed lubrication, size effects

1. Introduction

Energy used to overcome unwanted friction is causing millions of tons of CO_2 emission per year. In passenger cars, for example, about a third of the entire fuel consumption is spent to overcome friction [1]. Since the early 1990s it is realized that one potential way to reduce frictional losses is the improvement of the surface topography of sliding surfaces by micro texturing. The aim of micro texturing is either to increase or to decrease friction. Communicating textures, like channels or crossed channels are reported to do both, increase [2-4] or decrease [5] friction, depending on the tribological conditions. Furthermore, for non-communicating textures like dimples the effect to reduce friction is reported, especially under hydrodynamic lubrication conditions [6, 7]. Four mechanisms are postulated in the literature, which might lead to the reduction in friction and wear by micro dimples: the trapping of wear debris [8], changes in the contact angle [9], or the storage of lubricant [10]. In case of hydrodynamic lubrication the most widely accepted explanation for the effect of dimples is a micro hydrodynamic pressure build-up [11, 12]. Such pressure build-up can be estimated either by solving the Reynolds [11] or the Navier-Stokes equation [13]. Studies of dimple textured surfaces in the literature unfortunately are not conclusive about the effect of dimples on the friction in the mixed lubrication regime and therefore do not give a clear indication on how to optimize dimple geometry.

A first important parameter concerning dimple structure is the depth-to-diameter ratio of the dimples. Shinkarenko et al. [14] showed that the depth-to-diameter ratio is decisive for whether the dimples are building up pressure and reduce friction or not in the elastohydrodynamic regime. They interpret their results such that the depth-to-diameter ratio is the most important controlling parameter at constant dimple area under elastohydrodynamic conditions [15]. We now wanted to investigate if this finding is also valid under mixed lubrication regime by inverting their argument. One can derive a first hypothesis that, by keeping the depth-to-diameter ratio constant and though varying the dimple diameter and thus the dimple depth, the same pressure build-up should be created, resulting in the same amount of friction reduction for a given depth-to-diameter ratio.

Following [16], the pressure build-up is created by the dimple edge and so a second hypothesis is that the pressure build-up depends on the total number of edges in the contact. Indeed, unidirectional sliding experiments of textured steel surfaces with 7 and 14% dimpled area density [17] showed lower friction in the sample with a higher packing density. For a higher textured area, more edges are in contact and so a higher pressure build-up should occur. For a constant textured area one then expects that decreasing dimple diameter will increase the pressure build-up as the number of edges increases.

Additionally, one may expect and formulate as a third hypothesis that a certain fluid flow inside the dimples has to be established so that dimples can build up pressure. This means that dimples have to have a certain size (depth and diameter) to create a pressure build-up and that smaller sizes do not contribute much. Two examples from the literature hint in this direction. Dimpled SiC-bearings tested under unidirectional sliding conditions in water under hydrodynamic conditions [18] showed a twofold increase of the critical load compared to the untextured one at an optimal dimple diameter of 350 µm and a depth of 3 µm. Even if the main focus presented in the work here is not hydrodynamic but mixed lubrication, it is still very interesting since mixed lubrication combines hydrodynamic and solid friction. A second investigation under mixed lubrication in unidirectional sliding conditions (85W-140 gear oil), found the optimum at 100 – 200 µm diameter dimples and a depth of 105 µm [19]. Comparing these two results it seems that shallow dimples are favorable for low viscosity fluids like water, whereas deeper dimples are needed for high viscosity media.

In general, the literature is not even consistent whether surface texturing has a beneficial effect on friction at all. For example Ref. [20] reported that under unidirectional sliding conditions with contact pressures ranging from 0.2 to 1 MPa, sliding speeds of 100 and 500 mm/s, and CD15W-40 oil lubrication, dimples with 100 µm diameter, 10 µm depth and 5% dimpled area density led to the lowest friction. Under the same conditions, dimples with 50 µm diameter, 15 to 20 µm depth and 15 to 20% dimpled area density led to an increase in friction compared to the untextured reference. It therefore appears necessary to conduct a detailed investigation of the effect of dimple geometry and density on the frictional properties of textured surfaces under mixed lubrication. The aim of this paper is to systematically analyze the size effects in micro laser textured steel tribocontacts. Keeping the depth-to-diameter ratio and the dimpled area constant, we vary the sliding speed and the oil temperature to test the above hypotheses.

2. Material and methods

2.1 Sample material

The tribological behavior of laser textured and polished reference surfaces were studied using a pin-on-disc configuration. The pins, with a diameter of 8 mm, were made out of the normalized steel C85 (Stahlbecker, Heusenstamm, Germany) with a homogeneous pearlitic grain structure. The hardness of the material was 400 HV. The surface of the pins were grinded (200 mesh) and afterwards polished using 3 µm and 1 µm diamond suspension (Cloeren Technology, Wegberg, Germany).

The discs, with a diameter of 70 mm, were made out of the hardened and tempered (190°C) steel 100Cr6 (AISI 5210, Eisen Schmitt, Karlsruhe, Germany) with a hardness of approximately 800 HV. They were grinded (200 mesh) to a roughness ranging from $R_a = 0.08$ to 0.12 µm measured by a tactile roughness measurement device. The waviness (2nd order according to DIN 4760) along the frictional radius was below 1 µm measured by a white light profilometer. SEM pictures showing the microstructure of both materials can be found in supporting information (see Figure SI1)

2.2 Laser Surface Texturing

The laser surface texturing (LST) was performed with the Yb-doped fiber laser system Piranha II from Acsys (Kornwestheim, Germany). As a laser source a YLP-1-100-20-20 from IPG Photonics (Oxford, MA, USA) with 1064 nm wavelength was used. Depending on the texture, the power and the frequency of the source was varied between two and six Watt and between 20 to 80 kHz respectively. The laser beam was moved with a velocity of up to 3000 mm/s over the surface with a SCANcube 10 from Scanlab (Puchheim, Germany). Round dimples were generated on the surface of the C85 pin ranging from 15 to 800 µm in diameter. The depth-to-diameter ratio was kept constant at 0.1. The dimpled area density was kept at 10%. The shapes of the dimples were self-similar for the whole diameter range and resemble the shape of a sphere indented in the surface. The orientation of the dimples on the surface was in a hexagonal array.

The debris created from laser texturing was carefully polished away with 3 µm and 1 µm diamond suspension prior to all tribological tests. Optical, digital and 3D microscopy images were taken to make sure the debris was completely removed and the sample is suitable for the tribological test. Exemplary 3D microscopy and scanning electron microscopy images of 40 µm diameter dimples are shown in Figure 1.

2.3 Tribological testing

The tribological tests were carried out on a Plint TE-92 HS tribometer from Phoenix Tribology (Kingsclere, UK). The machine was set to a pin-on-disc configuration with a self-aligning pin-holder (as can be seen on the sketch of the configuration in Figure 2 a). The frictional radius (the distance from the center of the disc to the center of the pellet) was 30 mm. A large distance from the center of the disk was used, to limit the influence of the velocity gradient over the pin diameter. The influence of this parameter on the optimum dimple parameter was investigated in a separate study [21]. The dimpled surfaces were all aligned such that the dimples are along the sliding direction, as demonstrated in Figure 2 b). A normal force of 150 N was applied to the 8 mm diameter pin, resulting in a nominal contact pressure of 3 MPa. The entire set-up was heated to a stabilized oil temperature of 50°C or 100°C in order to mimic realistic engine conditions. Lubrication was realized through continuous delivery of additive-free poly-alpha-olefin (PAO), Klüber Lubrication (Munich, Germany), with a viscosity of 18 mm^2/s at 40°C and 4 mm^2/s at 100°C according to the manufacturer. The viscosities for other oil temperatures were calculated using MobilCalc (ExxonMobil, Irving, TX, USA), a software for calculating viscosity-temperature dependencies for Newtonian fluids. For 50°C a viscosity of 13 mm^2/s was calculated. The oil flow rate was 125 ml/h.

During the tribological tests, the sliding speed was decreased stepwise from the highest (2000 mm/s) to the lowest (40 mm/s) value, holding each of the twelve steps for 300 s (see Figure c) to generate Stribeck curves, following [22]. Each full velocity ramp was repeated five times. For the calculation of the mean value at one sliding speed, only the last three ramps were considered to avoid any possible influence of a running-in process. An example for the third, the fourth and the fifth ramp is given in the supporting information (see Figure SI 3). It can be seen that there are no significant differences between the curves. Every test was carried out at least three times with completely new pellets and discs for each test. The reported data for every surface texture are the mean values of these tests.

3. Results

3.1 Influence of the dimple diameter

The experiments presented in Figure 3 (a and b) were carried out under 3 MPa contact pressure and 100°C system temperature. Fig. 3 a) gives an overview of the Stribeck curves for a polished surface compared to the dimpled ones with a logarithmic sliding speed axis.

The polished surface and the surface with 800 µm dimple diameter showed the highest friction coefficients. The transition between hydrodynamic and mixed lubrication for the polished surface is between 500 and 1000 mm/s, in case of the 40 µm dimples it is in the range of 400 to 500 mm/s. At a speed of 2000 mm/s, the friction of samples with 40 µm dimples is in the same region as that of the polished reference whereas the others are above. All dimpled samples other than the 800 µm dimples, show lower friction coefficients for sliding speeds below 1000 mm/s. The surface with 40 µm dimples generates the lowest friction coefficients in the mixed lubrication regime at sliding speeds between 1000 mm/s and 40 mm/s. In Fig 3 b), the friction coefficient is plotted linearly against the sliding speed for a selection of textures. It can be seen, that the friction coefficient increases linearly with decreasing sliding speed for the polished reference in the range of 40 to 500 mm/s. For the 40 µm diameter dimples, a linear trend is found between 40 to 250 mm/s. A slope of the curves can be determined when one assumes a linear dependence of the data for sliding speeds in the mixed lubrication regime. The surface with the 40 µm diameter dimples displays the highest negative slope.

To better visualize the influence of the dimple diameter on the reduction of the friction coefficient in the mixed lubrication regime, the friction coefficient is plotted against the dimple diameter in Figure 4 for sliding speeds of 200 mm/s, 100 mm/s and 50 mm/s. At all three sliding speeds a reduction of friction can be noted with a local minimum for a dimple diameter of 40 µm. For dimple diameters larger than 50 µm no significant dependence of the friction coefficient on dimple diameter is found. For lower sliding speeds the effect of the dimples is smaller, but the minimum in friction coefficient is still at dimples with 40 µm diameter.

One can calculate an efficiency of friction reduction of the laser textures according to:

$$\text{efficiency} = \frac{\mu_{polished} - \mu_{textured}}{\mu_{polished}} \cdot 100\ \% \qquad (1)$$

In Figure 5, the efficiency is plotted against the sliding speed for samples with 15, 30, 40 and 45 µm diameter dimples. A maximal friction reduction of more than 80% is found for the 40 µm diameter dimples at 500 mm/s sliding speed. The maxima of the efficiency of friction reduction are at a speed of 500 mm/s for all textures plotted in Fig. 5. Below a sliding speed of 250 mm/s the efficiency of texturing decreases and more or less vanishes below sliding speeds of 50 mm/s.

3.2 Influence of the oil temperature

Decreasing the oil temperature from 100°C to 50°C increases the viscosity from 4 to 13 mm^2/s. With an increase in viscosity, the Reynolds number changes and so the flow conditions change. The contact pressure was kept constant at 3 MPa. In Figure 6, the friction coefficient is plotted against the sliding speed for different laser textured surfaces and the polished reference. Comparing the 50°C results with the experiments at 100°C, the friction coefficient in the mixed lubrication regime is generally found to be lower. The point where the contact transits from hydrodynamic to mixed lubrication is shifted from about 1000 mm/s (for 100°C) to lower sliding speeds in the range of 300 mm/s for 50°C due to the lubricants higher viscosity. In the hydrodynamic regime, we measured no significant differences between the different sliding pairs. Below 300 mm/s, in the mixed lubrication regime, the lowest friction coefficient was generated by the surface textured with dimples of 200 µm diameter. The 40 µm diameter dimples, the most efficient texture at 100°C, now have no significant influence on the friction coefficient.

In Figure 7, the efficiency of the surface texturing in reducing the friction coefficient is plotted against the sliding speed for the 50°C experiments for 80, 150 and 200 µm diameter dimples. A maximal reduction of the friction coefficient of over 80% is found for the 200 µm diameter dimples. For these dimples, the efficiency stays at a high level in a sliding speed range from 75 to 200 mm/s. For the other dimple diameters the high efficiency plateau is smaller. Compared to the 100°C experiments (Fig. 5) the maximum efficiency has shifted from a sliding speed of 500 mm/s to 150 mm/s at 50°C.

4. Discussion

Tribological properties of laser textured steel surfaces were characterized under unidirectional sliding conditions in order to investigate the size effects of round dimples in the mixed lubrication regime. A decrease in friction can be realized by laser surface texturing under certain conditions. To elucidate the mechanisms by which dimples influence sliding conditions three hypotheses about the dependence of friction coefficient on surface texturing are tested in this publication. The number of dimples edges and the importance of the dimple diameter were investigated directly.

The proposition that the depth-to-diameter ratio is the most important parameter for the dimples to build up pressure was tested indirectly.

Keeping both the depth-to-diameter ratio and the total area of the dimples constant at 0.1, we have investigated the dependence of the friction coefficient on sliding speed in the mixed lubrication regime for spherical dimples with diameters ranging from 15 µm to 800 µm at two different temperatures. At both temperatures, the friction coefficient significantly depends on velocity and dimple diameter as displayed in Fig. 3 and 6. An optimal diameter with respect to friction reduction reaching an efficiency of more than 80% was found at 40 µm in case of the tests at 100°C oil temperature (see Fig. 5) and at 200 µm for 50°C (see Fig. 6).

The first hypothesis is formulated based on published work measured under elastohydrodynamic lubrication conditions [14, 15]. There it was reported that the depth-to-diameter ratio should be the main influencing parameter. In our investigation under mixed lubrication conditions we found a strong effect of the dimple size at constant depth-to-diameter ratio (Fig. 3, 4 and 6). Thus the depth-to-diameter ratio is certainly not the main influencing parameter under mixed lubrication conditions.

Also the second hypothesis formulated after the work in [16] that friction reduction should depend on the density of edges perpendicular to the sliding direction is clearly shown to be insufficient. It would imply a monotonous decrease of friction coefficient with decreasing dimple diameter. On the contrary, comparing large dimples with successively smaller dimples we find (Fig. 4) an insignificant dependence of friction coefficient on dimple diameter for dimple sizes between 500 µm and 50 µm. Thereafter the friction coefficient decreases sharply around a dimple diameter of 40 µm and rises again towards lower dimple diameters. Consequently, we conclude here that hydrodynamic lift from edges normal to the sliding direction may well contribute to the observed dependence of friction on dimple size but it is clearly not sufficient to explain the experimentally observed behavior.

The third hypotheses investigated in this paper is that a certain fluid flow inside the dimples has to be established in order to build up sufficient additional pressure. The facts, that we find an optimal dimple size and that the oil temperature and so the oil viscosity is affecting the optimum dimple diameter are hints that a specific fluid flow inside the dimples has to be established in order to generate a micro hydrodynamic lift. In case of the higher viscosity, larger dimples are needed to get this fluid flow within the dimples. Hence there is a dependence of the optimum diameter on the oil temperature. With decreasing oil temperature and therefore increasing viscosity, an increasing optimum dimple diameter was found.

Following [22] where the Stribeck parameter is introduced as (sliding speed * viscosity) / contact pressure, the friction coefficient μ for a selection of results is plotted against this parameter in Figure 8. For the polished reference specimen, the curves for 50 and 100°C experiments blend perfectly into each other as predicted by Stribeck. The curves for the different temperatures and dimples of the same size do not blend into each other. However, the curve for the optimal dimple size (40 µm dimples at 100°C and 200 µm dimples at 50°C) do blend again very well. This is an additional hint that for a given viscosity, a specific size of dimples is necessary for the maximum build up of pressure.

Three examples from the literature are also supporting this hypothesis. An optimum texture at 20-30% textured area and a diameter of 100-200 µm was reported for a 85W-140 gear oil [19]. For the oil CD15-40 an optimum texture was found at 100-200 µm dimple diameter, 5-10 µm depth and 5% textured area [20]. Both, the 85W-140 and the CD15-40 have higher viscosity than the PAO at 50 and 100°C used in our investigation. For the low viscosity of water much shallower dimples of only 3 µm in depth were reported as optimal [18]. To summarize the literature mentioned here and the results presented in this paper, an increase of the optimum dimple geometry can be noticed with increasing oil viscosity.

The combination of the two competing mechanisms (number of dimple edges affecting the friction reduction and the influence of the dimple size on the possibility to build up a flow in the dimples) leads to the formation of a maximum in friction reduction at a specific dimple diameter (40 µm @ 100°C and 200 µm @ 50°C). The third hypothesis also explains why the optimum is dependent on the oil viscosity. The higher the oil viscosity, the larger the dimples have to be in order to build up a sufficient fluid flow inside the dimples.

In this investigation, no or only small effects of the dimples under hydrodynamic lubrication conditions were found, as it is shown in Fig. 3 (100°C oil temperature) for sliding speeds of 1000 and 2000 mm/s and in Fig. 6 (50°C oil temperature) for sliding speeds above 300 mm/s. Under these conditions, a full fluid film is present between the two surfaces. The models for micro hydrodynamic pressure build-up show that an additional pressure is created due to the textures [23]. Hence, there should be more lubricant between the two sliding surfaces and in our case no or only slight changes in the friction coefficient would be expected.

In addition to the hypotheses discussed above, there are further findings, which will be considered in the following. The largest decrease in friction coefficient was observed for dimples with 40 µm diameter in case of 100°C oil temperature. For every sliding speed in this lubrication regime the lowest friction coefficient is created by the 40 µm dimple diameter surface (see Fig. 5). This leads to the assumption that the hydrodynamic effects separating the two surfaces are at their maximum for this geometry and viscosity for every sliding speed. However, the strength of the lift obviously depends on the sliding speed. With lower speeds there is a smaller reduction of the friction coefficient. This can also be seen in the efficiency curves in Fig. 5 and Fig. 7. The maximum of the efficiency is found right after the point of the transition from hydrodynamic to mixed lubrication. Since there is no offset of the efficiency curves with respect to sliding speed, there does not seem to be a direct connection between the optimum dimple geometry and the film thickness.

The data presented above clearly indicated the potential of laser surface texturing when it comes to reducing friction forces; a maximum reduction by over 80% was achieved. At the same time, the data also highlight that the search for *one* set of texturing parameters being optimum in all applications and for all operation points most likely will be futile. Rather, in each new case, the textures need to be optimized. In real life applications, laser surface texturing already has proven its worth [7] and with our data we can definitely state that applications like compressors, pumps or range extenders in automobiles could dramatically benefit from a morphological texturing of their surfaces under a tribological load.

5. Conclusion

Laser surface texturing with spherical dimples with diameters ranging from 15 to 800 µm, a depth-to-diameter ratio of 0.1 and a textured area of 10% was applied to C85 steel surfaces in order to investigate the size effects on friction reduction. Tribological tests under mixed lubrication were carried out against the bearing steel 100Cr6. Three hypotheses were tested in this publication in order to investigate the mechanisms by which dimpled surfaces influence the sliding conditions. It could be shown that the depth-to-diameter ratio is not the only important parameter. A severe dependence of the dimple diameter on the friction reduction was found in our investigation. Friction reduction by laser surface texturing is at least also influenced by the dimple diameter. Two competing mechanisms are occurring. There is a dependency of the number of the dimples edges and there is an influence of the dimple size on the possibility to build up pressure with the dimples. In case of the optimum texture over 80% of friction reduction can be achieved. The optimum texture is found for certain operation conditions. This suggests that a specific flow pattern must be achieved within the dimples. The optimal diameter depends on the oil temperature and the oil viscosity. For lubrication with the PAO used in our investigation the optimal dimple geometry is found at 40 µm dimple diameter at 100°C, and at 200 µm dimple diameter at 50°C. The sliding velocity does not affect the optimal dimple diameter but it has a clear effect on the efficiency of friction reduction.

6. Acknowledgement

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Figure 1: Scanning electron microscopy image of an array of dimples with 40 µm diameter (a) and 3D microscope image of one dimple with 40 µm diameter (b)

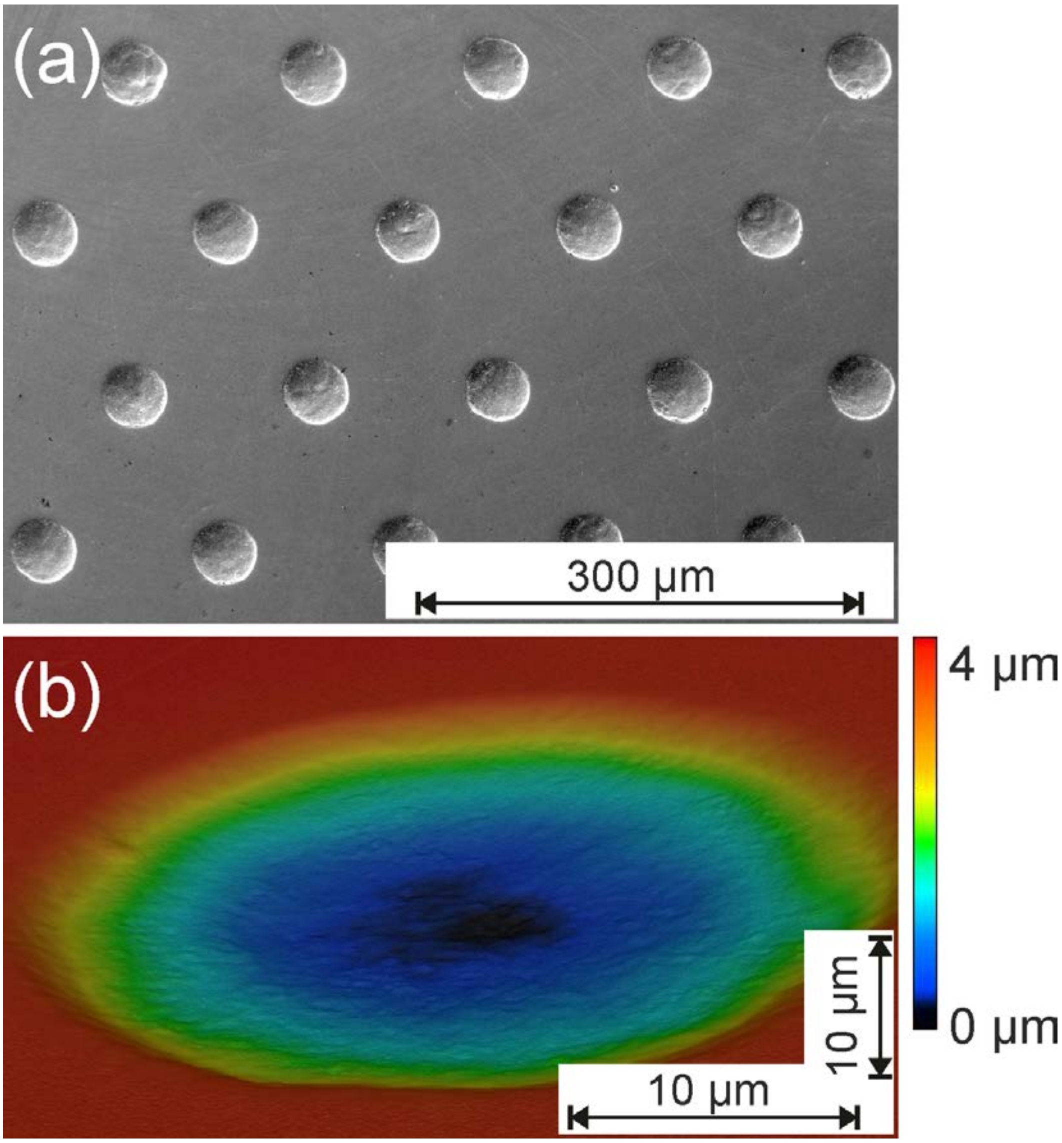


Figure 2: Experimental set-up of the Plint TE-92 HS tribometer from Phoenix Tribology (a), alignment of the dimples with respect to the sliding direction (b) and sliding speed against time (c)

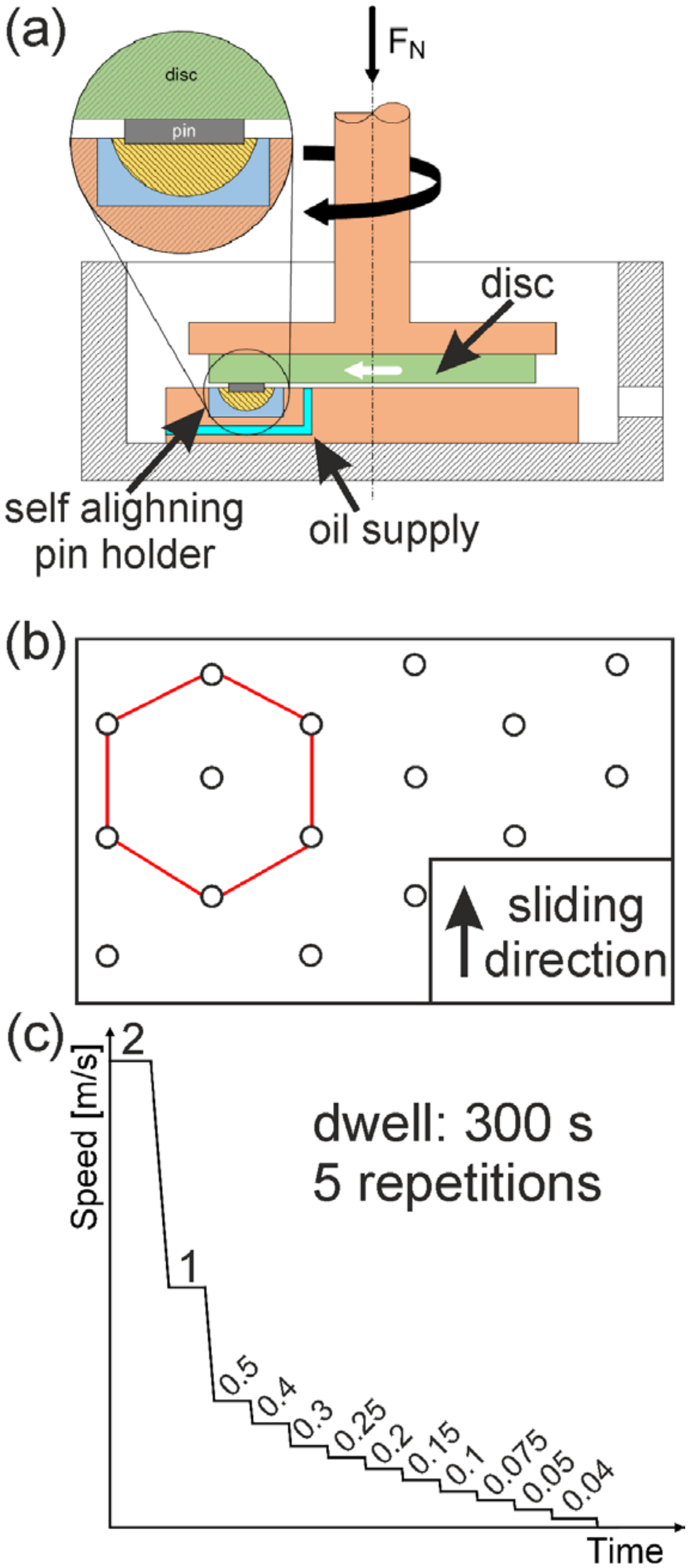


Figure 3: Friction coefficient plotted against the sliding speed for the polished reference and the dimpled surfaces with a logarithmic sliding speed axis (a) and with a linear sliding speed axis (b) for a selection of dimpled surfaces (T = 100°C; p = 3 MPa; PAO)

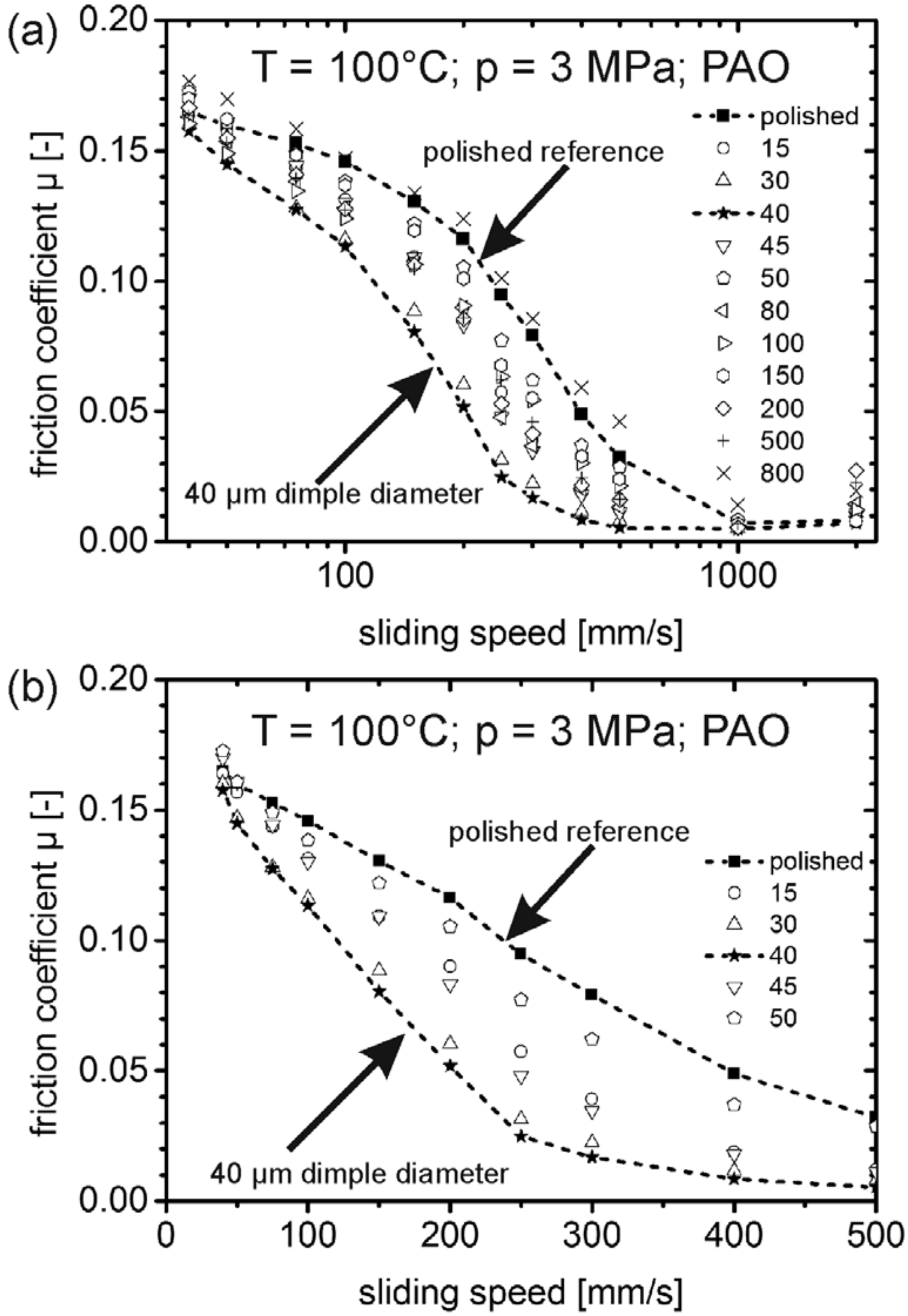


Figure 4: Friction coefficient plotted against the dimple diameter for the discrete sliding speeds 50, 100 and 200 mm/s (T = 100°C; p = 3 MPa; PAO)

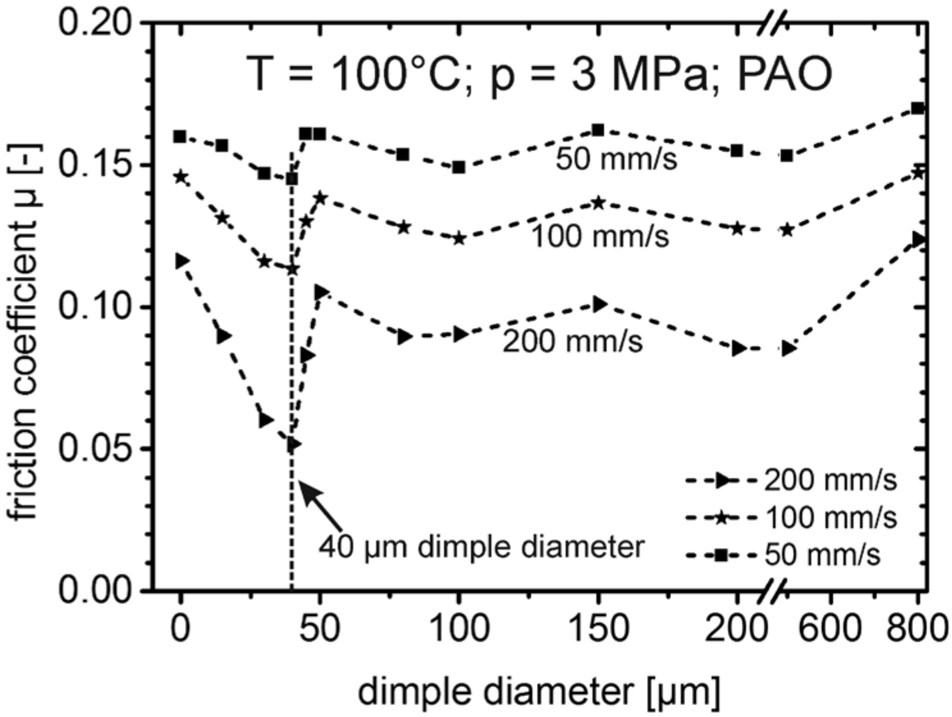


Figure 5: Efficiency (defined as: (($\mu_{polished}$-$\mu_{textured}$) / $\mu_{polished}$)*100%) plotted against the sliding speed for textures with 15, 30, 40 and 45 µm dimple diameter (T = 100°C; p = 3 MPa; PAO)

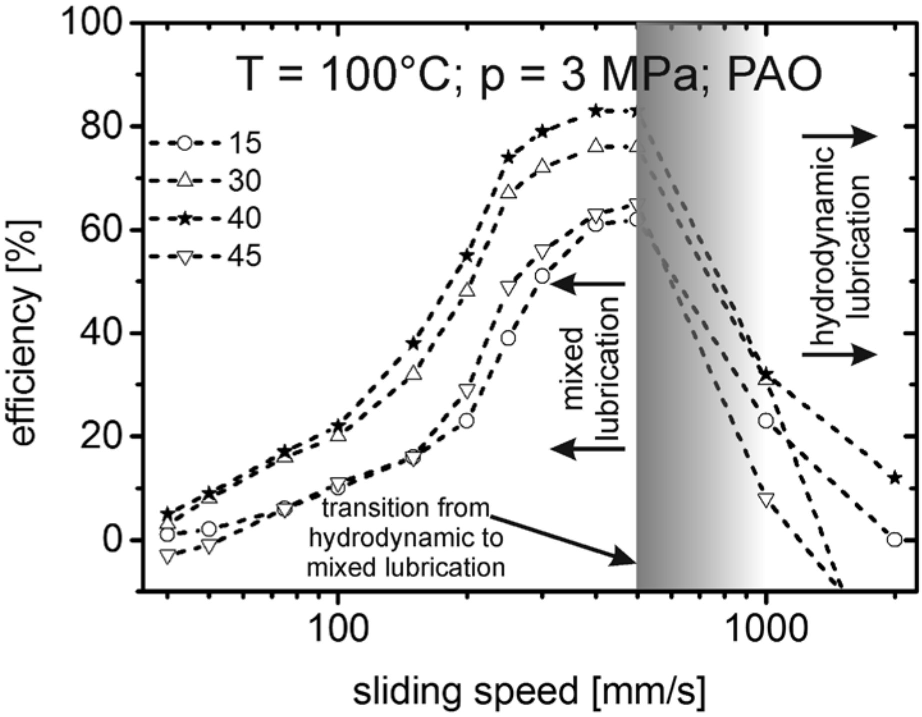


Figure 6: Friction coefficient plotted against the sliding speed for the polished reference and the dimpled surfaces (T = 50°C; p = 3 MPa; PAO)

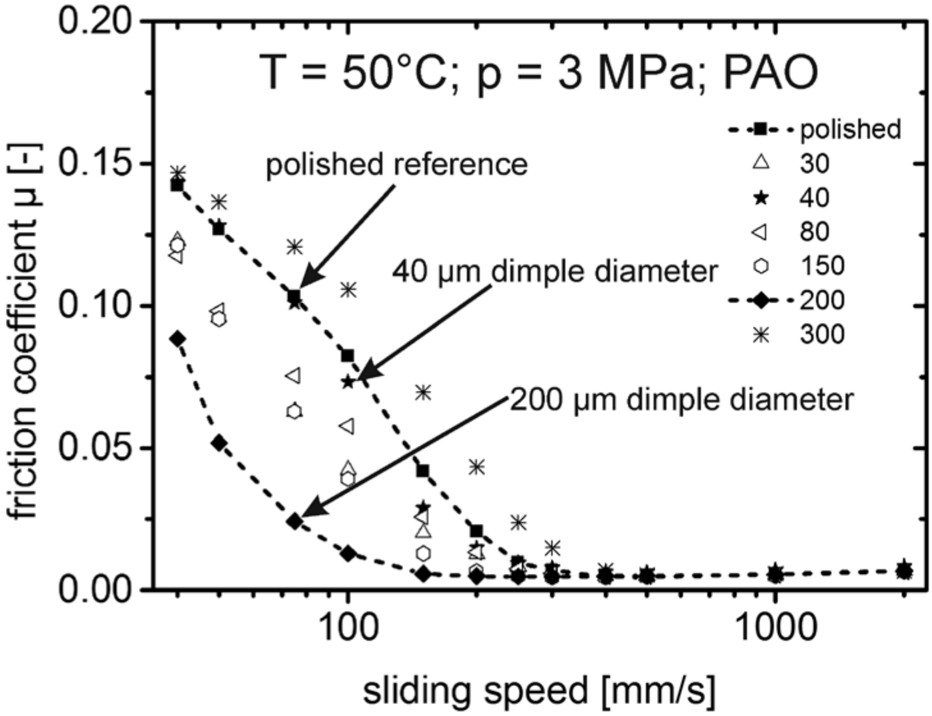


Figure 7: Efficiency (defined as: (($\mu_{polished}$-$\mu_{textured}$) / $\mu_{polished}$)*100%) plotted against the sliding speed for textures with 80, 150 and 200 µm diameter dimples (T = 50°C; p = 3 MPa; PAO)

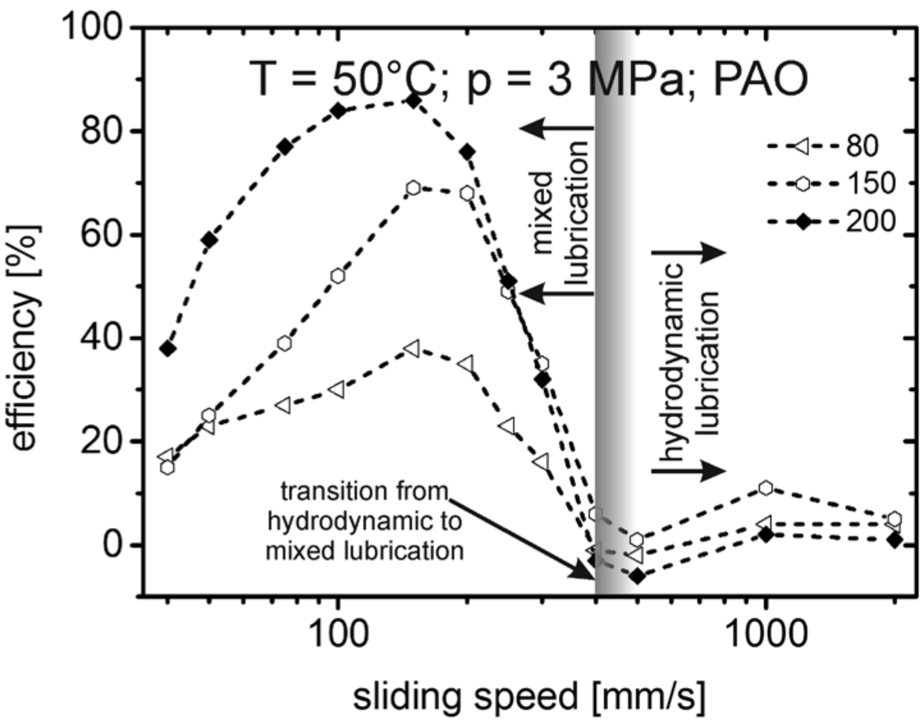
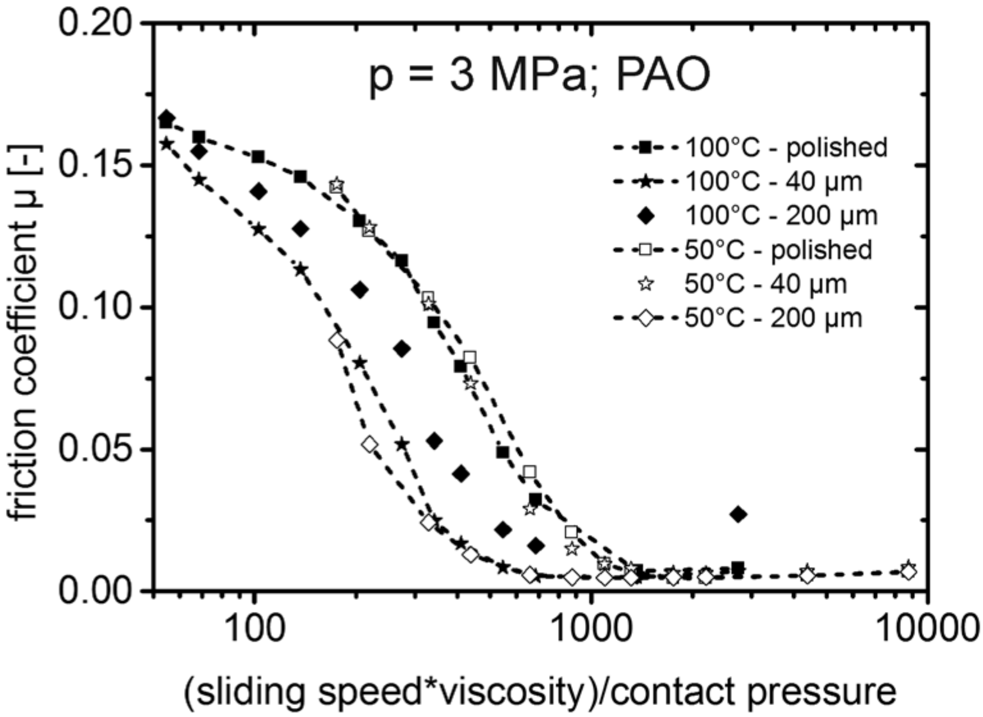


Figure 8: Friction coefficient plotted against the Stribeck parameter ((sliding speed * viscosity)/contact pressure) for a selection of results (polished, 40 µm and 200 µm dimple diameter) performed at 50 and 100°C (p = 3 MPa; PAO)



Supporting Information

Laser surface Texturing

Three different manufacturing methods were used depending on the diameter of the dimples. For small dimples, singles pulses, for medium sized “Archimedes spirals” were used as it is described in [24]. Large dimples were manufactured by filling a circle with crossed lines. Three final optics with different focal width of 56, 100 and 254 mm were used.

Additionally an exemplary cross section of one dimple with 40 µm diameter is shown in Figure SI 2 a. An overview of 4 dimples is shown in Figure SI 2 b. The roughness was measured in proximity and far away from the micro-machined areas.

(Area 1: Sa = 6,5 nm; Area 2: Sa = 5,0 nm; Area 3: Sa = 5,2 nm; Area 4: Sa = 5,8 nm; Area 5: Sa = 4,4 nm)

Results

In Figure SI 3 the friction coefficient of the third, the fourth and the fifth ramp is plotted against time. It can be seen that there are no significant differences between these three ramps.

Figure SI 1: SEM images of the 100Cr6 (a) and of the C85 (b) steel

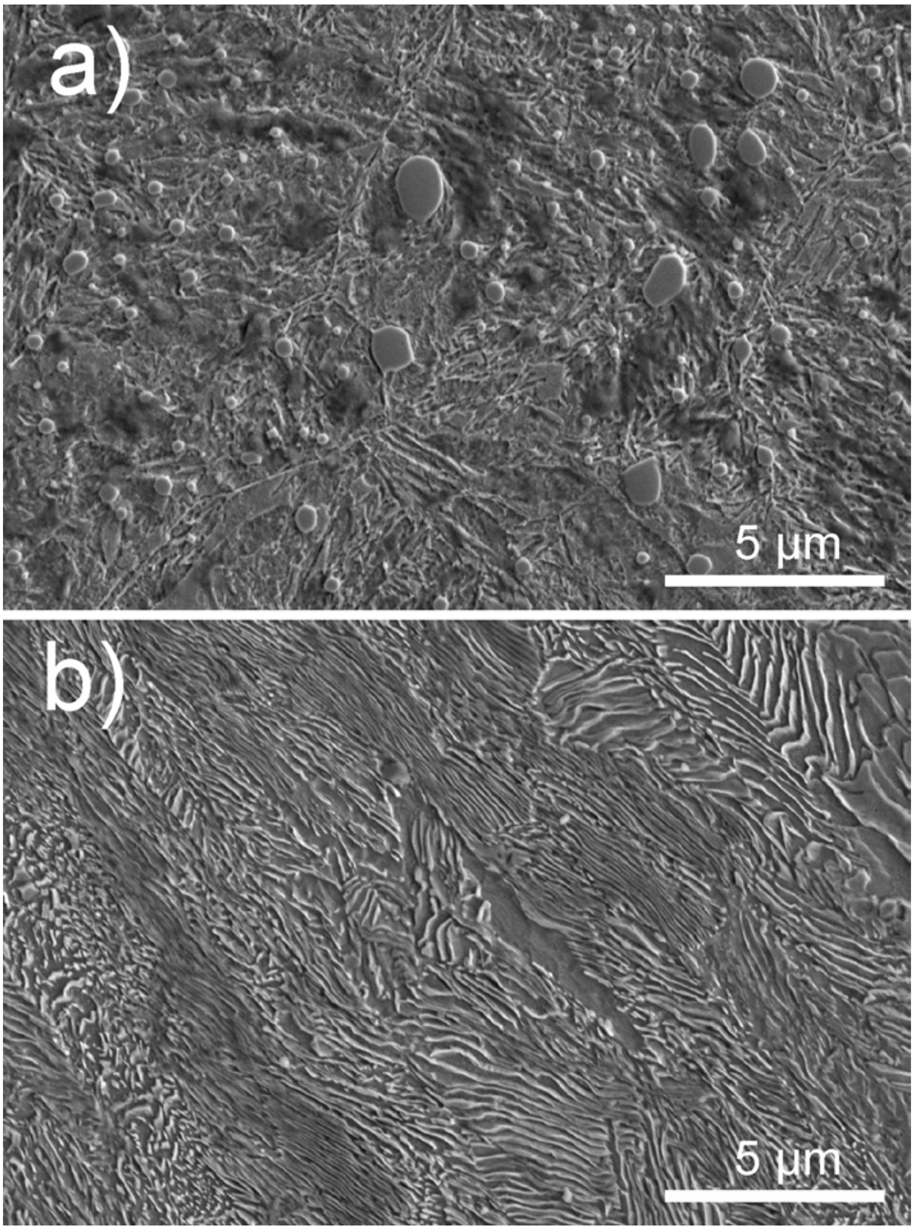


Figure SI 2: Exemplary cross section of a dimple with 40 µm diameter (a), a dimple with 200 µm diameter (b) and an overview of 4 dimples (c); additionally the roughness was measured in proximity and far away from the micro-machined areas (Area 1: S_a = 6,5 nm; Area 2: S_a = 5,0 nm; Area 3: S_a = 5,2 nm; Area 4: S_a = 5,8 nm; Area 5: S_a = 4,4 nm)

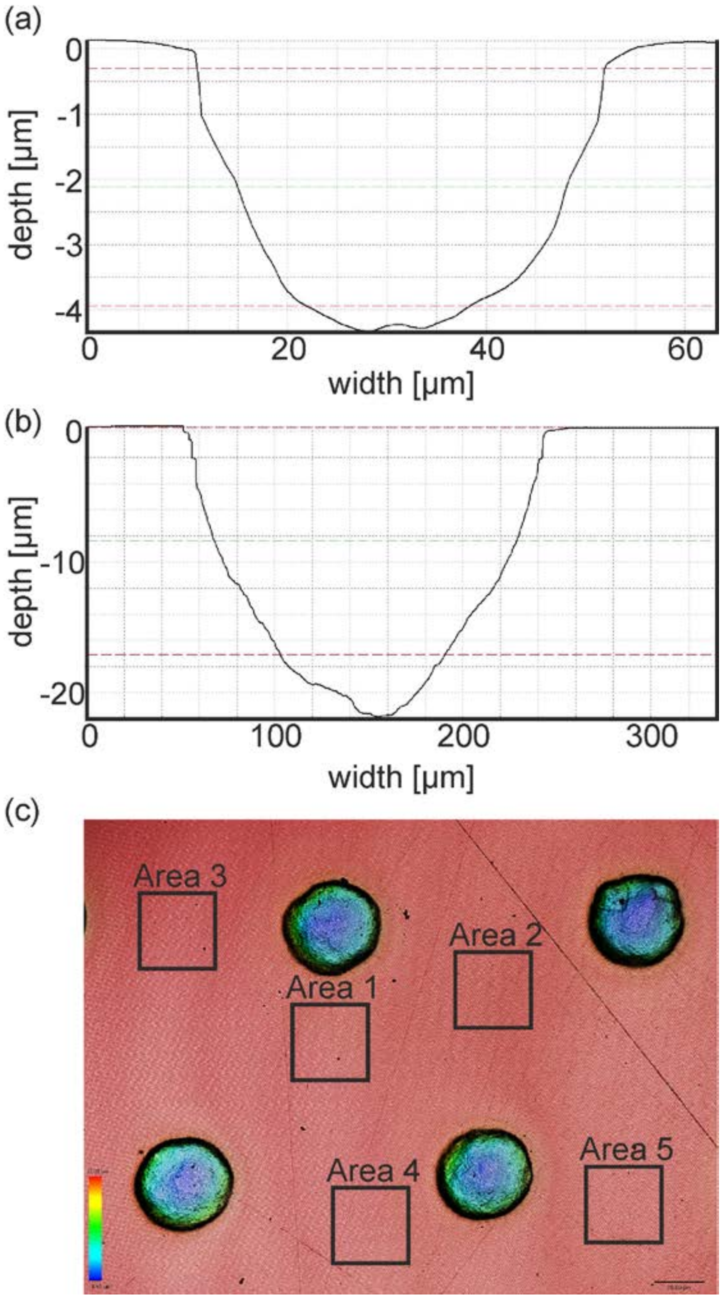


Figure SI 3: Friction coefficient for a polished reference surface plotted against time for the third, fourth and the fifth Stribeck-ramp (T = 50°C; p = 3 MPa; PAO)