Tribological Characterization of Smooth and Artificially Textured Coated Surfaces Using Block-on-Ring Tests

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Tribological behaviour of sliding surfaces under boundary/mixed and fully lubrication conditions is an important aspect of tribo pair design. At the same time, artificially textured surfaces have a worth noted tribological behaviour in piston ring tribology. In this paper, experiments were performed in a custom made block-on-ring test rig calibrated according to ASTM D2714. Smooth and textured species were tested for 30 min experimental time. Thereafter, the block specimens have been treated with a chromium layer of 10 μm thickness. Repeatability tests were performed for the artificially textured and coated samples. The minimum oil thickness was measured using the capacitance method, since the friction coefficient is calculated in the computational manner. The block-ring tribo pair was lubricated with Newtonian oil SAE 30. Optical images of the blocks were illustrated after the tests. Results show that the textured surfaces improved the oil distribution generated in a hydrodynamic lift, since the friction coefficient was reduced. Textured coated stainless steel (TCSS) has shown a substantial improvement in wear resistance compared to the textured stainless steel sample.

Keywords: *block-on-ring test rig, capacitance technique, friction coefficient, minimum oil thickness, chromium layer, rectangular texturing, wear.*

1. INTRODUCTION

In recent years, attempts to reduce the friction force between sliding surfaces have been investigated to improve the tribological performance of mechanical parts (such as piston rings, thrust journal bearings, etc.). The literature on the friction characteristics of micro-textured surfaces is vast. These studies have shown potential results, such as the load carrying capacity increment in fully flooded conditions using micro dimples in one of the two tribo surfaces. However, the dimples contribution in the boundary/mixed regimes is an ongoing research concerning the piston ring-cylinder friction problem. At the same time, the micro scale surface irregularities and its manufacturing is another challenge for engineering technology.

Ramesh et al. [1] presented an experimental and analytical study for micro-textured surface contact. A pin-on-disk test rig was used and a range of patterns was examined. Numerical results were reported and compared with the experimental ones. Furthermore, Mishra and Polycarpou [2] discussed the effect of laser surface texture (LST) patterns in air-conditioning compressors. As expected, the type of lubricant oil plays an important role in surface texturing wear. Etsion [3,4] investigated the tribological performance of laser surface texturing (LST) in several mechanical parts. A substantial reduction of friction was observed using a piston ring with surface texturing [4]. Spherical micro dimples were examined for different textured positions. Numerical and experimental results were obtained and compared showing a good agreement. In the paper by Vladescu et al. [5] a valuable experimental work is obtained for different textured patters and operation conditions. Cavitation contribution was also investigated, which is related to the oil film distribution on the surface textured liner.

Smazalová et al. [6] presented some experimental results on the block-on-ring device for continuous and discontinuous tests. In their paper the friction coefficient, wear and the contact temperature of the sliding tribo pair were illustrated and discussed. Later on, an important investigation has been done in piston ring-cylinder dry contact using a block-on-ring test ring by Bihlet et al. [7]. Their work examined different sliding pairs, showing that the ceramic coating on piston ring profile decreases the dry wear while the friction coefficient is maximized. A range of loads was examined and the wear scars of test cylinder liners were measured. Since then the wear situation of the piston ring and cylinder liner has been estimated through scuffing. More recently, Morales-Espejel et al. [8] determined the influence of lubrication conditions on rolling contact. A numerical model developed and compared with the experimental results showed a good agreement. In fact, the basic surface parameters (such as surface roughness) were examined by the authors, hence the friction and wear of sliding moving surfaces can be investigated.

Currently, many numerical and simulation models describe the benefits of micro dimples at mechanical

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components. A 2-D axisymmetric simulation model of the top piston ring-cylinder contact using micro-textures was developed by authors Zavos and Nikolakopoulos [9,10]. Solving the Navier Stokes equations, the basic tribological characteristics (friction, hydrodynamic pressure and oil film thickness) were reported in elastohydrodynamic conditions. Spherical and rectangular micro dimples were examined for different textured positions showing a substantial friction reduction [9].

The measurement of the oil film thickness using laboratory tests is an important process, in order to predict the friction and wear between mating surfaces. Several methods, such as capacitance, ultrasound and laser induced fluorescence methods, were developed by many researchers. Sherrington [11] offered a review article about the methods of oil thickness measurement in the piston-piston ring pack conjunction. From this notable study, it was concluded that the monitor of oil distribution on piston ring profile can assess the oil starvation and the cavitation conditions. Grice et al. [12] described the capacitive technique related to the oil film distribution. The calibration of sensors for this measurement technique was included. Mills et al. [13] presented the ultrasound method for the oil film thickness measurement of piston ring-cylinder contact. Piezoelectric sensors were used along to cylinder liner, and hence the dynamic behaviour of the piston assembly tribo pair was evaluated. Additionally, a laser induced fluorescence (LIF) technique was used for oil thickness measurement along the piston ring profile by the authors Dhunput et al. [14]. In practical terms, a test rig was used which simulates the piston ring motion across the cylinder inner liner. Therefore, cavitation conditions were predicted around the piston ring liner using the LIF method.

The objective of the current paper is to measure the friction coefficient and the oil film thickness of sliding surfaces. For this reason, a custom made block-on-ring test rig was used for the experiments that took place in the Machine Design Laboratory of the University of Patras. An aluminium ring and a stainless steel block were examined. A series set of tests has been done for smooth and textured specimens. Afterwards, the artificially textured and coated species were compared with the same operating conditions. Chromium coating was performed using the electro-deposit method. Friction tests were carried out at 5 N for 30 min at 15 °C, using SAE 30 as lubricant oil. To this point, the capacitance technique was applied for the minimum oil film thickness measurement, and hence the friction coefficient has been calculated. Optical images of the blocks are illustrated before and after the experimental tests.

2. DESCRIPTION OF EXPERIMENTAL TESTS

2.1 Block-on-ring set-up

This method describes the procedure for the determination of the friction coefficient of sliding materials using a block-on-ring test ring. A stationary block specimen is pressed with a constant load (F_N = 5 N) against a rotating ring specimen at 90° to the ring's axis of rotation. Friction between the sliding surfaces of the block and ring results in loss of material from both specimens. Figure 1 shows the main function of the block-on-ring test rig. Tribological characteristics were obtained using a standard test method for friction coefficient measurement according to ASTM D2714 [15].

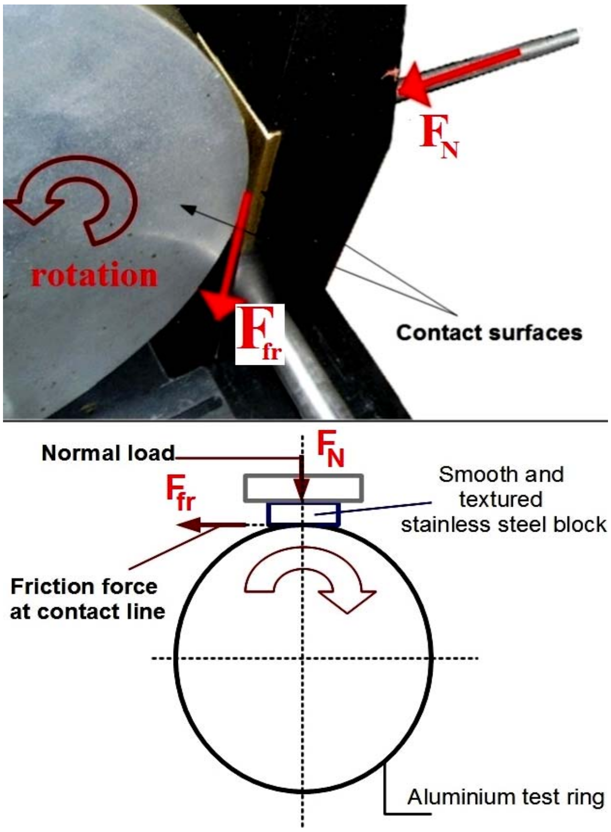

Figure 1. The block-on-ring test rig function

The block was made from stainless steel 307 and the ring was made from an aluminium alloy. The basic dimensions of block specimens were $B \times L \times d$ = $50 \times 70 \times 5$ mm and the ring radius was 13 cm. The average roughness (Ra) of the block was measured in 0.8 μm and the relevant ring was measured at 0.1 μm before the experiments. The surface parameters were picked up using the mobile roughness measuring instrument HOMMEL TESTER T500.

Similar and repeatable tests were also performed using artificially textured and coated samples. Chromium layer of 10 μm was achieved by an electro-deposition method. The average roughness parameter of coated specimens was also measured, as Ra = 0.1 μm.

2.2 Surface texturing geometry

As it can be obtained, from the introduction section, various forms of surface texturing can be used, since the tribological performance is promoted. In this experimental study a rectangular texturing geometry is taken into account. For the rectangular shape, the parameter W is the textured width; H_d is the rectangular depth; while L_c is the textured cell length. Table 1 shows the numerical dimensions of the dimples.

In fact, a partial texture portion was selected according to the literature [3,4,9]. Figure 2 shows the typical dimensions of rectangular shape and its position on the steel block surface. The dimple array patterns were specified using CATIA software. An Electro discharge machining (EDM) oriented in micro drilling was utilized to fabricate a defined square micro dimple array (width 1000 μm, length 1000 μm and depth of 4 μm) on stainless steel (SS) hip prosthesis heads. The dimples density γ is 0.54 as obtained from the expression $\gamma = \pi W^2 / 4L_c^2$ [1].

Table 1. Basic dimensions of rectangular texturing geometry

Parameters	W	H_d	L_c
Rectangular geometry	1000 μm	4 μm	2200 μm

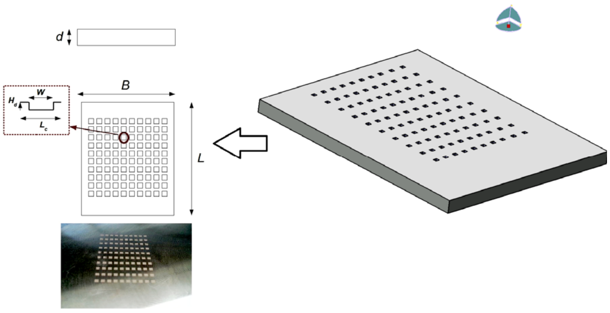

Figure 2. Schematic of rectangular geometry of micro-dimple by EDM

2.3 Experimental procedure

The specimens were cleaned and dried in order to remove all dirt and foreign matters. The ring was immersed in a pool of lubricant oil. The system was operated at room temperature and atmospheric pressure. The experimental parameters are summarized in Table 2.

Table 2. Experimental parameters

Test Parameters	Value
Rotational ring velocity	0.35 m/s
Normal load	5 N
Experimental time	30 min
Dynamic viscosity	0.35 Pas
Ambient temperature	15 – 17 °C

The block-ring system was lubricated with Newtonian oil SAE 30. The oil temperature was measured with a thermocouple type K. As follows, the main specifications of the test oil are presented in Table 3.

Table 3. Test oil SAE 30 specifications

Density	885 kg/m^3
Specific heat	1.985 kJ/kgK
Thermal conductivity	0.143 W/mk
Dynamic viscosity at 15 °C	0.35 Pas

3. OIL FILM AND FRICTION COEFFICIENT

3.1 Oil film thickness and capacitance technique

There are important applications of a fluid film at the rolling motion of a test ring and a flat block. Gears, cams and cylindrical bearings are some interesting examples. A thin oil film that separates the contact surfaces is illustrated in ZOOM A of Figure 3. In this case, the block is stationary and the test ring has a rotational velocity u in the x axis. For a small ratio of x/R, the thickness of the oil h (x, y) can be calculated as:

$$h(x,y) = \\ = \begin{cases} h_1 = h_{\min} + \dfrac{x^2}{2R} \Rightarrow h_1 = h_{\min} \to \text{smooth case, } x=0 \\ h_2 = h_{\min} + H_d(x,y) \to \text{textured case} \end{cases} \quad (1)$$

where $h_{\min}$ is the minimum oil film and $H_d\,(x, y)$ is the contribution of rectangular texture amplitude in the position coordinate (x, y). Here, the capacitance sensor is located at x = 0, since the minimum oil film $h_{\min}$ is measured for smooth cases, and sequentially for the textured cases the contribution of the rectangular micro scale irregularities have been taken into account (see ZOOM B of Figure 3).

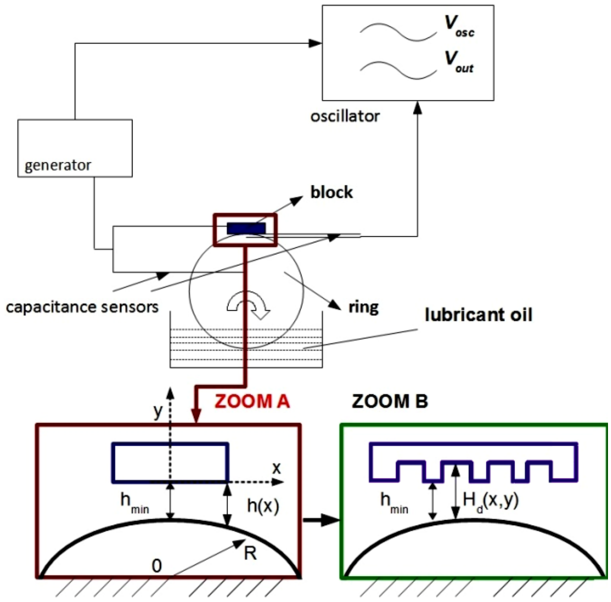

Figure 3. Oil film thickness between the ring and the test block (ZOOM A and ZOOM B); Signal procedure using capacitance method

There are different methods to measure the oil thickness between sliding surfaces. Capacitance measurement [12], ultrasound method [13] and laser induced fluorescence (LIF) concept [14] are some interesting techniques. In this case, the capacitance of a parallel plate capacitor was used. A draft of signal processing from the block-ring test rig is presented in Figure 3.

The basic equations of oil film thickness are presented below:

$$C_T = \frac{\varepsilon_o \varepsilon_r A_s}{h(x,y)}, \quad (2)$$

$$V_{out} = -\frac{V_{osc} C_{osc}}{C_T}, \quad (3)$$

where C_T is the transducer capacitance, ε_o is the value of the permittivity for air which is 8.84 · 10^{-12} F/m, ε_r is the relative permittivity (1.8 in oil), As is the cross section area of sensor (As = 0.16 · 10^{-6} m^2), $h(x, y)$ is the oil film thickness, V_{osc} is the oscillator input rms voltage, V_{out} is the output voltage and C_{osc} is the oscillator capacitance (C_{osc} = 28 ± 5 % pF). A sine wave with constant frequency was applied using a wave form generator, as an input in the oscillator. The deviations of input and output voltage were below ± 1 %.

Using the combination of (2) and (3), the measured oil film thickness $h(x, y)$ could be defined as:

$$h = h(x, y) = -\frac{V_{out}\varepsilon_o\varepsilon_r A_s}{V_{osc}C_{osc}}. \quad (4)$$

Laboratory experimentation remains the only practical method available for the accurate identification of the minimum film thickness for arbitrary material pairs. However, accurate and repeatable minimum film thickness measurement remains challenging due to the dependence of the capacitance of the oil material, surface, environment, and measuring equipment. When reporting any measured quantity, it is also necessary to provide a quantitative statement regarding the quality of the reported value, so that those who need to use the data can have an indication of its reliability. The dispersion of the values that could reasonably be attributed to the measurement uncertainty [16] can be calculated as follows:

$$u_c(y) = \\ = \sqrt{\sum_{i=1}^{N}\left(\frac{\vartheta f}{\vartheta x_i}\right)^2 u^2(x_i) + 2\sum_{i=1}^{N-1}\sum_{j=i+1}^{N}\frac{\vartheta f}{\vartheta x_i}\frac{\vartheta f}{\vartheta x_j}u(x_i, x_j)}, \quad (5)$$

where $u_c(y)$ is the combined standard uncertainty of output estimate, $u^2(x_i)$ is the estimated variance associated with input estimate x_i that estimates input quantity x_i, $u(x_i, x_j)$ is the estimated covariance associated with two input estimates x_i and x_j that estimates input quantities x_i, x_j, and $\vartheta f/\vartheta x_i$ is the partial derivative with respect to input quantity x_i of functional relationship f between measurement y and input quantities x_i on which y depends.

In (5) there is not any correlation between the input parameters, so $u(x_i, x_j) = 0$. Therefore, the expression for the uncertainty of the minimum oil film thickness result is provided by (6):

$$u^2(h) = \left(\frac{\vartheta h}{\vartheta V_{out}}\right)^2 u^2(V_{out}) + \\ + \left(\frac{\vartheta h}{\vartheta V_{osc}}\right)^2 u^2(V_{osc}) + \left(\frac{\vartheta h}{\vartheta C_{osc}}\right)^2 u^2(C_{osc}) \Rightarrow \\ \frac{u(h)}{h} = \\ = \sqrt{\left(\frac{u(V_{out})}{V_{out}}\right)^2 + \left(\frac{u(V_{osc})}{V_{osc}}\right)^2 + \left(\frac{u(C_{osc})}{C_{osc}}\right)^2} \Rightarrow \\ \Rightarrow \frac{u(h)}{h} = 5.19\,\%. \quad (6)$$

Consequently, the combined standard uncertainty of the minimum lubricant film measurement $u(h)/h$ is 5.19 %.

3.2 Calculation of friction coefficient

It is well known that friction is a major problem because it deteriorates the moving surfaces of many mechanical components. The instantaneous coefficient of friction μ is obtained as the ratio of the measured friction force (F_{fr}) to the measured normal force (F_N), as shown in (7).

$$\mu = \frac{F_{fr}}{F_N}. \quad (7)$$

Actually, the friction and the normal force at the contact of moving surfaces are separately measured. In fact, the applied normal force F_N is constant 5 N and the friction force on the block surface can be expressed as:

$$F_{fr} = \begin{cases} \mu_{oil} A u\left(\frac{1}{h_1}\right) \rightarrow \text{smooth case} \\ \mu_{oil} A u\left(\frac{1}{h_2}\right) \rightarrow \text{textured case} \end{cases}, \quad (8)$$

where μ_{oil} is the dynamic viscosity of monograde oil SAE 30, A is the contact area, u is the rotational ring velocity and h_1, h_2 is the measured minimum lubricant thickness for each examined case. Typically, the minimum oil film in the experiments was estimated using the (4), and the friction force (F_{fr}) is calculated using the (8).

4. RESULTS AND DISCUSSION

4.1 Oil film thickness and friction coefficient

The interface between the block and ring is important for the understanding of elastohydrodynamic conditions on contacting moving surfaces. Surface damage modes can be predicted and examined.

Figure 4a shows the minimum oil film for uncoated and coated stainless steel (SS) spaces under the same operation conditions. Each test was 30 min, and the oil film was measured using a capacitor sensor. It is evident that using the coated block the lubrication performance is improved 5 – 28 % in comparison with the uncoated stainless steel specimen. In fact, the chromium coated block has a smoother surface (Ra = 0.1 μm) than those of the stainless steel (Ra = 0.8 μm). Therefore, as observed in Section A (Fig. 4a), the minimum oil thickness is substantially increased concerning the stainless steel case. In practical terms, the surface roughness of the stainless steel (SS) is reduced to 0.5 μm as it is measured from the mobile roughness measuring instrument. Hence, the oil film enhanced the hydrodynamic lubrication ($h_1/\sigma > 3$) between block-ring interface, where h_1 is the minimum lubricant film thickness and σ represents the root mean square surface finish of contacting bodies respectively.

In Figure 4b the oil film distribution is presented when the rectangular textured geometry was examined. Coated and uncoated surfaces were used using a pattern with 100 dimples (Fig. 2). At constant operating mode, the textured samples have larger minimum oil thickness 27 – 40 % compared to untextured sample. This oil film increment can be attributed to the added hydrodynamic lift provided by the dimple pattern. Furthermore, the measured film thickness regarding the textured case has a good agreement with the theoretical expression $H_{\text{eff}} = f(h_{\min}, H_d, W)$ by the authors Ramesh et al. [1]. In Table 4, comparisons between the experimental and numerical calculated oil film thickness are presented for the textured stainless steel (TSS) sample.

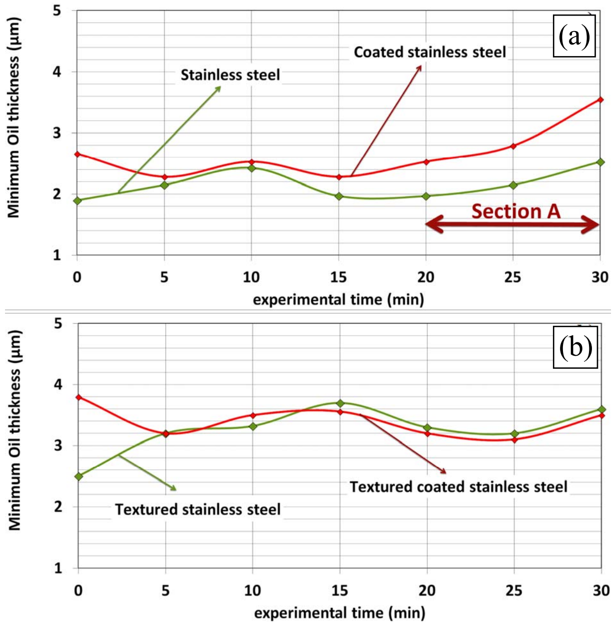

Figure 4. Minimum oil film thickness vs. experimental time: (a) smooth block specimens, and (b) textured block specimens

Table 4. Comparisons of experimental and numerical lubricant film thickness of TSS case

Measured oil thickness	3.3 μm
Numerical calculated oil film thickness	2.9 μm
Difference	~ 10 %

Therefore, it is important to refer that in the textured stainless steel (TSS) sample the minimum oil film is moderately increased nearly 15 – 30 min of the test duration. In fact, very small debris could be led to oil increment after the experimental test in (SS) space (Fig. 7). At the same time, in the coated textured block any wear scar has not been identified.

In Figure 5 the friction coefficient is illustrated for smooth and textured coated/uncoated specimens under the same operating test conditions. The calculation of the friction force under the mixed-hydrodynamic lubrication conditions is performed following the (8). The normal load $F_N = 5$ N is constant for each examined case.

Surface texturing of the block surface (textured density $\gamma = 54$ %) was done by EDM technique, resulting in a significant improvement of friction performance in comparison with untextured species. In summary, micro-textures work as micro-bearings, improve the load capacity and increase the oil film thickness, which leads to slighter friction force. In Section B (Fig. 5b), coated and uncoated textured stainless steel specimens, have similar tribological behaviour, and the stainless textured space appeared with more scratches on the contact area block-ring (Fig. 7). For coated textured sample, there is no wear keeping the same operation time.

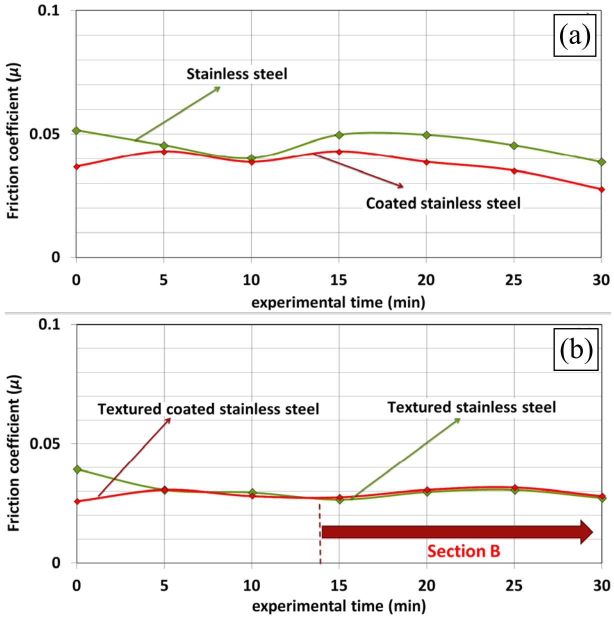

Figure 5. Friction coefficient performance for: (a) smooth block specimens, and (b) textured block specimens

4.2 Optical images of the test blocks

Friction force and the oil amount between the contact moving surfaces play an important role in identifying the surface damage. Scratches and scuffing are some basic aspects of surface wear. Figure 6 shows the images of smooth blocks before and after the 30 min duration of the test. In stainless steel (SS) scratches appeared in sliding direction as it is observed in the Area A. At the same time, the chromium coated block shows slighter scars in the Area B. It is concluded that the chromium plated block has better resistance to the abrasive and adhesive wear.

Figure 6. Block species before and after the tests: (a) stainless steel (SS) and (b) chromium coated stainless steel (CSS)

Simultaneously, the photographs of the textured coated/uncoated samples are illustrated in Figure 7. Here, the oil film motion is provided and the scratches produced from the metal to metal contact are shown for textured stainless steel (TSS). Comparison of the images between the coated textured sample before and after the experiment (Fig. 7b) has shown no visible scratches, indicating that the block did not contact with the ring. Indeed, due to this operating condition a significant contact is not expected. Note that the asperity contact is not enhanced, as the load/velocity is indicated around the mixed lubrication.

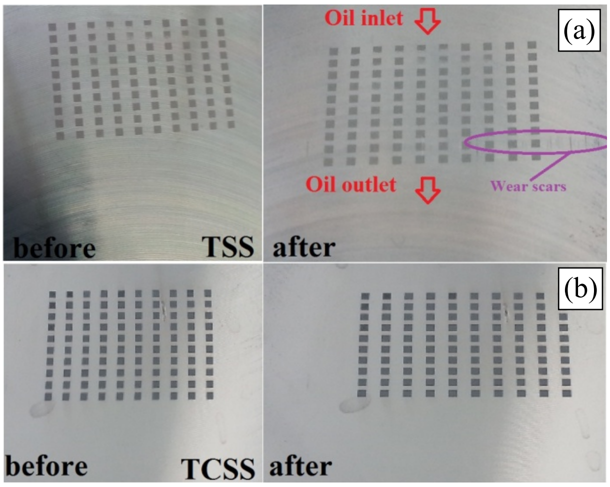

Figure 7. Block species before and after the tests: (a) textured stainless steel (TSS) and (b) textured coated stainless steel (TCSS)

5. CONCLUSION

In this paper, the minimum oil thickness was measured by a block-ring system located at the Machine Design Laboratory of the University of Patras using the capacitive technique. Experimental measurements and numerical calculations of the oil film thickness are in good agreement with relevant literature results. Artificially textured patterns with rectangular dimples were examined and compared with untextured cases. The duration of each test was 30 min using Newtonian oil SAE 30 under constant normal load of 5 N. Chromium coated and uncoated spaces were examined and optical images are presented before and after of each experiment.

The minimum oil film and the friction coefficient of the smooth/coated block were improved by 5 – 28 % in comparison with uncoated case. Textured surface blocks have shown a substantial oil increment in the order of 27 % in relation with the untextured case. The hydrodynamic lift of textured pattern increases the oil film, and hence the friction force is reduced.

Coated smooth and textured stainless steel shows a significant resistance without scratches in the contact area. However, the stainless steel blocks illustrate a larger area of scratches in sliding direction.

A numerical model should be further developed for a block-ring tribo pair, showing the pressure and thermal field of the surface contact. A scanning electron microscope (SEM) images can be compared after the tests, indicating more clearly the metal to metal contact and the dimples wear. This is a point for further investigation.

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ТРИБОЛОШКА КАРАКТЕРИЗАЦИЈА ГЛАТКЕ ПОВРШИНЕ ПРЕВЛАКЕ И ПОВРШИНЕ ПРЕВЛАКЕ СА ВЕШТАЧКОМ ТЕКСТУРОМ ПОМОЋУ ТРИБОМЕТРА ТИПА „БЛОК НА ДИСКУ“

Анастасиос Завос, Пантелис Г. Николакопулос

Триболошко понашање клизних површина у условима граничног/мешовитог и потпуног подмазивања је важан аспект при конструисању кинематског пара. Примена површина са вештачком текстуром код клипних прстенова омогућава побољшање триболошких карактеристика. Испитивања приказана у раду су изведена на посебно прилагођеном трибометру типа „блок на диску“, калибрисаном према стандарду ASTM D2714. Време испитивања свих узорака је износило 30 минута. Прво су испитивани узорци (блокови) без превлаке (глатки и са вештачком текстуром), а након тога узорци са превлаком (глатки и са вештачком текстуром). Дебљина електролитички нанете превлаке хрома је износила 10 μm. Испитивани кинематски пар блок-диск је подмазиван моноградним уљем са ознаком SAE 30. Минимална дебљина слоја мазива је одређивана мерењем промене електричног капацитета, а на основу ње је помоћу рачунара одређиван коефицијент трења. Изглед површине блокова пре и после испитивања је такође приказан. Резултати испитивања показују да је применом текстуре побољшано подмазивање, с обзором да је добијен мањи коефицијент трења, а да је примена текстуре на узорке са превлаком допринела значајном смањењу хабања.