Wear of a rough disc in dry sliding contact with a smooth ball: experiment and modeling

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Abstract

The main objective of this work is to model wear of a disc which was subjected to dry contact with a ball in unidirectional sliding. Tribological tests of sliding pairs were carried out using a tribological tester T-11 in a ball-on disc configuration. Stationary balls made of 100Cr6 steel with a hardness of 62 ± 2 HRC co-acted with rotating discs with 42CrMo4 steel with a hardness of 40 ± 2 HRC. Discs were machined by lapping, grinding, milling, and vapour blasting. The values of the Sq parameter of disc surfaces were between 0.1 and $5.86 \mu\text{m}$. Wear volumes of the discs were lower for bigger roughness heights. The simulation of disc wear was conducted on the basis of the repetitive contact between sliding surfaces. Strong correlation was achieved between the modeled and measured volumetric wear levels.

Keywords Friction · Wear · Surface texture · Contact · Modeling

List of symbols

d	Diameter of the wear track
DSS	Sliding distance after which the friction force obtained the steady-state condition
F_{av}	The average value of friction force
F35	The value of the friction force obtained for the sliding distance of 35 m
S	The cross-sectional area of the wear track
Sal	Fastest decay autocorrelation length
Sk _u	Kurtosis of height distribution
Sp _d	Density of peaks
Sq	Root mean square height of the surface
Ssk	Skewness of height distribution
Str	Texture aspect ratio
Sz	Maximum height of the surface
VD	Wear of the disc
Zo	Calculated wear volume
Zo _{sim}	Simulated wear volume

1 Introduction

Wear modeling is an important issue in tribological research. Wear studies are mainly focused on identifying the most probable destruction mechanisms under different operating conditions of the friction couples and at searching for methods enabling the control of these processes. Two major types of wear modeling can be distinguished: experimental models, defining the relationships between several factors and the resulting factor, and theoretical models, derived from classical mechanics (mainly contact mechanics) or fracture mechanics.

Change in the shape of surface topography due to wear was predicted. First, the truncation model was used [1]. According to this model peak surface part above the truncation level was removed. This model was proposed by Thomas [2] who simulated surface profile after abrasive wear. Stout et al. [3] simulated wear of phosphor bronze using simple truncation model. This model was also used in three dimensions [4]. Mathia et al. [5] modeled changes in surface roughness of cylinder liners as the results of the co-action with piston rings. However, the surface topography obtained after using this model differed quite significantly from the real surfaces. For example, Rosen et al. [6] found big discrepancies between of the truncation model and the measurement of cylinder liner surface topography.

An alternative approach to the truncation model takes into account the plastic deformations [7]. In this model,

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originated from the work of Sugimura et al. [8], the change in the ordinate distribution during wear was considered. Wear of profiles perpendicular to the sliding direction was analysed. The results were compared with those obtained using truncation model. Similar trends of changes in the skewness R_{sk} and rms. roughness height R_q parameters during wear in both models were found.

Two-process surface topography formed in a low wear (within the limit of the original surface topography) consists of the original surface and the smoother upper plateau surface of Gaussian ordinate distribution. During mild wear, this surface became smoother with time [9]. Therefore, the imposition of smooth Gaussian surface topography on the machined surface seems to be a better solution than the truncation model [10–17]. The imposed surface is typically modeled. Trends of changes in surface topography parameters were similar to the real ones. Pawlus presented the method of two-process surfaces simulation [10]. Nonogaki et al. [11] related the parameters of two-process profiles to skewness and kurtosis. Hu et al. used the imposition method for simulation of worn seal surfaces [12]. They modified this method to obtain a correct autocorrelation function of a modeled surface [13]. The extension of this approach can be used when the number of processes is higher than two [14]. This method was used to simulate a low wear of the cylinder surface [15], of piston skirt surface [16], and of surface with isolated oil pockets [17]. In the last case, the modeled surfaces were correctly matched to the measured surfaces in 90% of analyzed cases.

Simulation of wear higher than the initial surface height is typically based on contact mechanics. Jiang and Arnell [18] carried out one of the first work in this field. They used the Archard equation in which the volumetric wear at a given surface point was proportional to the contact pressure at this point. Bortoleto et al. [19] and Ilincic et al. [20] presented similar approaches. A slightly different methodology was described in the publication [21], in which the wear level was calculated on the basis of elastic deformation. The distribution of contact pressures for the analysis of wear in ball-on-disc configuration was used by Kucharski and Mroz [22]. Changes in the roughness height during running-in as a result of plastic deformation were determined by Akbarzadeh and Khonsari [23].

The main aim of this work is to model wear of disc which was subjected to dry contact with ball in unidirectional sliding. In the first part of this paper, the results of tribological tests will be described. Then an idea of wear modeling will be shown. In the last part of the modeling, results will be compared with the results of tribological tests. Wear of surfaces with very high ranges of rms. height was not modeled yet. Similar research was not found in the technical literature.

2 Experimental process

2.1 Tribological tests

Tribological tests of sliding pairs containing discs characterized by different values of rms. height of surface S_q were carried out using a tribological tester T-11 in ball-on-disc configuration. The frictional pairs consisted of stationary balls made of 100Cr6 steel with a hardness of 62 ± 2 HRC and discs rotating at a given rotational speed. The discs were made of 42CrMo4 steel. The hardness of the discs, regardless of the applied finishing treatment, was 40 ± 2 HRC. The finishing treatments of the discs were made by lapping (sample Z1), grinding (sample Z2), milling (samples Z3–Z4), and vapour blasting (samples Z5–Z14). Tribological tests were performed in dry sliding conditions at an ambient temperature of 20–22 °C and at a sliding speed of 0.2 m/s. The sliding distance was set to 282.6 m (test duration was 30 min), while the normal load was set to 9.81 N. The same operating parameters were applied in previous research [24]. Test duration of 30 min took into account the stage of running-in of the working surfaces of the friction pair. During tests, the friction force was monitored as a function of time. Wear levels of the discs and balls were measured by a white light interferometer Talysurf CCI Lite. The measuring area of a disc surface was 3.3 mm × 3.3 mm. Then the disc profiles perpendicular to wear tracks were extracted to obtain cross-sectional areas. The last step was to average the values of these areas and calculate the wear volume according to formula 1. Wear levels of discs and balls were measured by a white light interferometer Talysurf CCI Lite. The measuring area of a disc surface was 3.3 mm × 3.3 mm. Then the disc profiles perpendicular to wear tracks were extracted to obtain cross-sectional areas. The last step was to average the values of these areas and calculate the wear volume according to the formula 1.

$$VD = \pi dS \quad (1)$$

where: d diameter of the wear track, S the cross-sectional area of the wear track. Wear volumes of balls were assessed after removals of forms.

2.2 Surface textures of disc samples

Table 1 presents selected surface topography parameters of the tested discs and Fig. 1 shows isometric views of the discs surfaces. The machining treatments of the discs were carried out to obtain different texture heights, characterized by the S_q parameter (between 0.5 and 5.86 μm). Table 1 also presents values of the following parameters characterizing disc surface topography: S_{sk} skewness of height distribution, S_{ku} kurtosis of height distribution,

Table 1 Selected surface topography parameters of tested discs

	Z1	Z2	Z3	Z4	Z5	Z6	Z7
Sq, μm	0.101	0.387	0.626	0.834	1.39	2.04	2.47
Ssk	− 0.61	− 0.71	− 0.105	0.059	− 0.322	− 0.427	− 0.641
Sku	4.39	4.61	3.25	2.41	4.76	5.31	4.99
Sz, μm	1.4	5.29	6.56	5.61	13.9	31.9	26.9
Sal, mm	0.026	0.012	0.0504	0.0293	0.0276	0.0236	0.0221
Spd, $1/\text{mm}^2$	1022	659	633	172	46.8	270	402
	Z8	Z9	Z10	Z11	Z12	Z13	Z14
Sq, μm	3.04	3.56	4.11	4.86	5.1	5.51	5.86
Ssk	− 0.3	− 0.301	− 0.434	− 0.403	− 0.328	− 0.3	− 0.323
Sku	3.71	3.88	3.91	4.7	3.66	3.98	4.35
Sz, μm	31.4	27.4	38.8	57.7	48.1	63.8	56.5
Sal, mm	0.0251	0.0693	0.068	0.0638	0.0669	0.0627	0.0618
Spd, $1/\text{mm}^2$	418	116	123	86.1	102	91.8	106

Fig. 1 Isometric views of discs surfaces: Z1—Sq=0.101 μm (a), Z2—Sq=0.387 μm (b), Z9—Sq=3.56 μm (c), Z13—Sq=5.51 μm (d)

Sz maximum height of the surface, Sal fastest decay auto-correlation length and Spd density of peaks. Typically surfaces except of the Z4 disc sample were characterized by negative values of skewness Ssk. However, the values of skewness were not very low, therefore one-process surfaces were tested. The value of kurtosis of the Z4 sample is characteristic for milled surfaces. Most of the surfaces, except for milled were random. Surfaces after milling and grinding were anisotropic, while surfaces after lapping and vapour blasting were isotropic. The correlation length Sal had a tendency to increase with roughness height, the opposed tendency occurred for the density of summits

Spd. The observed trends are characteristic for random surfaces.

Figure 2 presents roughness heights of the tested disc samples.

3 Results of experimental tests and discussion

Table 2 and Fig. 3 present the results of the conducted tests (average values). For each type of surface, volumetric wear of the discs (VD), the average value of the friction force

Fig. 2 Root mean square height of the surface Sq tested samples

(Fav), the value of the friction force obtained for the sliding distance of 35 m (F35) and the sliding distance after which the friction force obtained a steady-state condition (DSS) were calculated.

At the sliding speed of 0.2 m/s, the lowest values of volumetric wear were achieved for the Z13 and Z14 discs with the surface rms. height Sq of 5.51 and 5.86 μm , respectively. The sliding pair containing Z13 disc was also characterized by the highest value of DSS parameter (212.4 m), the lowest value of the friction force after the distance of 35 m F35 (1.116 N) and the lowest average value of the friction force Fav (7.771 N). The highest values of wear volume were observed for the samples with the lowest value of Sq parameter—Z1 and Z2: 0.655 and 0.561 mm^3 , respectively. The sliding pair containing Z1 disc was also characterized by the lowest value of the friction force F35. The average coefficients of friction were in the range: 0.79 (Z13 disc)—0.85 (Z1 disc). Sedlacek et al. [25, 26] obtained similar results in dry sliding: a higher surface roughness resulted in lower friction but longer distance to obtain the steady-state condition. Figure 4 presents the relations between the selected surface

Table 2 Results of tribological tests

	Z1	Z2	Z3	Z4	Z5	Z6	Z7
VD, mm^3	0.655	0.561	0.559	0.542	0.519	0.528	0.4875
DSS, m	31	32	34	40	40	48.8	85.8
Fav, N	8.388	8.221	8.219	8.103	8.012	8.363	8.109
F35, N	7.782	7.852	7.812	7.676	7.872	7.866	4.68

	Z8	Z9	Z10	Z11	Z12	Z13	Z14
VD, mm^3	0.431	0.313	0.259	0.267	0.231	0.222	0.224
DSS, m	78.6	102	83.9	102.5	142.1	212.4	197
Fav, N	8.125	7.915	8.047	8.256	7.802	7.771	7.848
F35, N	4.236	4.346	2.76	2.958	2.232	1.116	1.506

Fig. 3 Selected results of tribological tests

Fig. 4 Dependencies between the Sq parameter (a), Sz parameter (b) and Sal parameter (c) and disc volumetric wear VD

texture parameters Sq, Sz, Sal and disc volumetric wear. The highest coefficient of determination was achieved for the dependence between the Sq parameter and volumetric wear, because the Sq parameter is more robust than the maximum height Sz. The relation between wear volume and the Sal parameter resulted from larger main wavelength of a rougher surface. Figure 5 shows cross-sectional areas of wear tracks of the selected discs and Fig. 6 presents examples of the friction coefficient runs.

To determine the wear intensity of the sliding pairs in some cases the tribological system was stopped and disc wear was measured at the same place (the relocation method was used). Tribological pairs containing lapped (Sq=0.101 µm), ground (Sq=0.387 µm) and vapour-blasted (Sq=3.56 µm and 5.51 µm) discs were stopped after the distance of 47, 94, 188 and 288 m, which corresponded to 5, 10, 20 and 30 min. Figures 7,8,9,10 show the cross-sectional areas of the wear tracks versus time for the selected sliding pairs.

The highest wear volume was obtained for the lapped disc (Fig. 7), which was a consequence of large wear in the initial period of the running-in of 5 min duration. After this time, the wear intensity decreased significantly. The volumetric wear of the lapped disc after 5 min reached the value of 50% of the total wear. After 10 min, wear of the disc increased to 85% of its total value, and the depth of wear reached 89% of the total value. The wear profiles could be approximated by a circle with a radius close to the ball radius. The Z2 disc (Sq=0.387 µm) was characterized by slightly lower volumetric wear in comparison to the lapped disc (Fig. 8). Again, the highest wear intensity was observed in the first 5 minutes. After the distance of 47 m, the volumetric wear

achieved 56% of its total value. After 10 min, 70% of the total wear volume was observed, with the wear intensity significantly lower. The volumetric wear of the Z9 disc (Fig. 9) with a higher value of the Sq parameter than previously analyzed disc samples in the initial period was invisible, only plastic deformation was found. When the sliding distance reached 94 m, the volumetric wear was 0.026 mm³. In the initial phase of the test, the wear intensity of the rough surface was also significantly lower than those of the smooth surfaces. This can be explained by the fact that for the lapped and ground discs the running-in process was completed after a distance of 47 m (or less), while for the Z9 the disc—after 188 m.

The wear volume of the sample Z13 (Sq=5.51 µm) was the lowest of all analyzed discs (Fig. 10). Due to the big maximum height of the surface, in the initial periods of the tests (for the friction distances of 47 and 94 m), the wear level was difficult to determine. When the sliding distance reached 188 m, the volumetric wear was 0.122 mm³, which corresponded to 58% of its total value. A precise approximation of the wear track of the Z13 disc by a circle with a ball radius was not possible.

The obtained wear intensities of the discs were similar to the runs of the friction force in time. When the disc surfaces were smooth, the friction forces were stabilized very quickly. This behavior can be explained by removal of the original height of the roughness created during machining. When rough disc surfaces were tested (Z9, Z13), sliding distances necessary to achieve a stable value of the friction force were longer. Dzierwa et al. [24] obtained similar results in earlier tests. They can be explained by the higher pressing of the

Fig. 5 Cross-sectional areas of worn discs: Z1 (a), Z2 (b), Z9 (c), Z13 (d)

ball to the smooth disc compared to rough disc in the initial part of test. For the contact of the ball and the smooth disc surface bulk deformation occurred, however, when the ball contacted the disc of a big height, only asperity deformation happened. This behavior led to higher wear and friction of the assembly containing a smoother disc. This analysis suggests that disc wear can be modeled on the basis of the

repetitive contact between sliding surfaces. Disc of a big height, only asperity deformation happened. This behavior led to higher wear and friction of the assembly containing smoother disc. This analysis suggests that disc wear can be modeled on the basis of the repetitive contact between sliding surfaces.

Fig. 6 Coefficients of friction of selected frictional pairs co-acted with the discs: Z1 (a), Z2 (b), Z9 (c), Z13 (d) versus sliding distance

The analysis of worn surface topography revealed the formation of a one-way texture. The obtained values of the Str parameter (texture aspect ratio) ranged from 7.4 to 12.8%. Figure 11 shows isometric views, while Fig. 12 presents SEM images of Z5 and Z13 discs after tribological tests and details of the worn.

For the Z5 disc, the value of the Str parameter was 8.7%, and for the Z13 disc it was 10.8%. As a result of the co-action of the sliding elements, new anisotropic disc surface textures were created. On the surfaces of the discs, one can see in Fig. 11 elongated wear tracks in the sliding direction as well as smoothed areas with longitudinal grooves resulting from plastic deformation. The SEM images in Fig. 12 clearly show the wear scar patterns of the 42CrMo4 steel under dry sliding. Extensive wide and deep surface furrows indicate the predominant abrasive character of wear. Moreover, the presence of wear debris indicates a more complex abrasive-adhesive character of wear, with plastic deformations of the asperities.

Wear of balls after the tribological tests was much lower than wear of discs. This was due to the higher hardness of the ball compared to that of the disc.

3.1 Wear process modeling

Simulation of the wear process of the disc surfaces was performed for all tested variants. The modeling process was conducted using the following methodology. The ordinates of generated spheres with a radius of 3.175 mm were added to the ordinates of a disc surface. Then, the contact analysis of the obtained equivalent surface with the smooth flat surface was carried out using the numerical method developed by Sahling et al. [27]. They developed an efficient algorithm

to calculate the elastoplastic deformation of the contact between rough surfaces. Rough surface deformation was calculated with an improved algorithm based on DC-FFT [28]. Plastic δ_p and elastic δ_e displacements were obtained. The contact pressures ranged from 0 to p_p , which is the hardness of the softer surface (disc) H . Figure 13 shows the relation between the deformed height of the equivalent surface h and the contact pressure p . The sum of the shaded areas is equal to the normal load P . The plastic deformation was taken into account by truncation of the contact pressure to the H value (right side of Fig. 13).

Then, the obtained deformations were subtracted from the ordinates of each points of the total initial equivalent surface (sum of the rough surface of the disc and of the surface of the ball). For the new surface obtained in this way, the contact with the flat surface was analyzed again using the procedure developed by Sahling et al. [27]. The described procedure was repeated until the assumed number of cycles was achieved (120). Then, the ordinates of the generated ball surface were subtracted from the ordinates of the obtained (deformed) surface. The ball was assumed to be smooth. The calculations were performed in the C program. To speed up the calculations, the normal load increased to 60 N and the sampling interval of disc surface topography increased four times (from 3.3 to 13.2 μm). The procedures described in publications [18, 21, 23] were previously used to model wear by subtracting the deformations from the original surface ordinates. The use of the model of rough surface deformation for repeated contact of rough surfaces is the new idea of wear simulation.

Figure 14 shows an example of the modeled surfaces after 120 cycles. Figure 15 presents the profiles of selected surfaces with tracks created during simulated wear. From

Fig. 7 Cross-sectional areas of Z1 disc ($S_q=0.101 \mu\text{m}$) after tribological tests for the sliding distance of 47 m (a), 94 m (b), 188 m (c) and 288 m (d)

profiles passing through the simulated wear tracks, cross-sectional wear areas were calculated using a TalyMap software. On the basis of the surface areas of the formed holes, a simulated disc wear volumes Zosim were obtained. Figure 16a shows the relation between the volumetric wear

of disc Zo and the simulated volumetric wear Zosim. The linear coefficient of correlation was very high and amounted to 0.94. In addition, non-linear models were also analyzed. Figure 16b shows the relationship between Zo and Zosim

Fig. 8 Cross-sectional areas of Z2 disc ($S_q=0.387 \mu\text{m}$) after tribological tests for the sliding distance of 47 m (a), 94 m (b), 188 m (c) and 288 m (d)

obtained after using a polynomial of the 3rd degree. The coefficient of correlation was 0.97.

The main aim of this model was not to achieve the exact wear volumes of the discs, but the attempt was done to explain the obtained trends of wear changes. Several

simplifications were done. The model did not take into consideration the mechanical resistances occurring during the movement of the ball on the rough flat disc, only the normal contact between these surfaces was analyzed. The model also did not take into account the slight displacements

Fig. 9 Cross-sectional areas of Z9 disc ($S_q=3.56 \mu\text{m}$) after tribological tests for the sliding distance of 47 m (a), 94 m (b), 188 m (c) and 288 m (d)

between the sliding paths on the disc surface which occurred during disc rotation. Despite these simplifications, a very strong correlation between the volumetric wear from experimental tests (Z_o) and from simulation ($Z_{o_{sim}}$) was obtained, however, the value of the ratio of the maximum

to the minimum real wear values of Z_o (3: 1) was higher than that obtained during modeling $Z_{o_{sim}}$ (2: 1). Probably, the contact of the ball with the smooth disc surface was concentrated in its center, and bulk deformation occurred. Therefore the contact track of the ball with the smooth disc

Fig. 10 Cross-sectional areas of Z13 disc ($S_q=5.51\ \mu\text{m}$) after tribological tests for the sliding distance of 47 m (a), 94 m (b), 188 m (c) and 288 m (d)

appeared from the beginning of the friction process, and during sliding the wear scar increased. On the other hand, the contact of the ball with the disc of big surface height occurred only at the highest summits and asperity deformation happened. The authors of the publication [29] obtained

a similar finding. Therefore, the loss of the very rough disc was unnoticeable at the beginning of the friction process, there were only traces of the plastic deformation. Small displacements between the sliding paths, not considered in the wear model, promoted additionally low wear of the

Fig. 11 Isometric views of the discs Z5 (a) and Z13 (b) after tribological tests and details of their worn surfaces Z5 (c), Z13 (d) surfaces

Fig. 12 SEM images of the worn discs Z5 (a) and Z13 (b) after tribological tests

Fig. 13 Relation between the contact pressure p and surface deformation h , after [27]

Fig. 14 Examples of the modeled surfaces after 120 cycles

rough disc. This analysis explains why the larger wear of surfaces with low roughness height was obtained compared to surfaces with significant surface height in the initial non-conformal contact.

Similar to the experimental studies, the intensity of the modeled wear versus time was analyzed for selected friction pairs. The simulated tests were carried out for 20, 40, and 80 cycles, which corresponded to 5, 10, and 20 min of the real test.

The obtained results are presented in Figs. 17,18,19,20. For surfaces with a significant surface height (Fig. 20), no substantial change in profile was observed in simulation, and wear occurred within the initial roughness height.

A similar tendency was observed for the vapour blasted sample with the parameter $S_q = 3.56 \mu\text{m}$, except for the last profile after 120 cycles (Fig. 19). Then the wear exceeded the initial range of disc amplitude.

A different tendency of changes in profiles was observed for discs with lower surface roughness heights. In these cases

(Figs. 17 and 18), increases in heights of the surface profiles from the beginning of the wear process were noticed, which was caused by the formation of holes. The agreement was found between modeled and experimental results.

Summarizing, larger wear levels of smooth disc surfaces were probably caused by higher maximum pressures in the centers of the contacts with balls. The situation was different for discs with significant roughness heights. In that case, contact pressures occurred in the places of the highest summits. In that case, contact pressures occurred in the places of the highest summits.

4 Conclusions

An experimental investigation was carried out to study the effects of disc roughness height on dry friction and wear in unidirectional sliding. The values of the S_q parameter

Fig. 15 Profiles of disc surfaces after simulated wear with areas of holes: **a** Z1, **b** Z2, **c** Z9, **d** Z13

Fig. 16 Relation between measured wear volumes Z_o and simulated wear volumes Z_{osim} of discs: **a** linear, **b** after using a polynomial of the 3rd degree

of disc surfaces were between 0.1 and 5.86 μm . Disc wear modeling was performed based on the contact between sliding surfaces. There are the main conclusions of this work:

1. Wear volumes of one-process rough discs were lower for bigger roughness heights. For smooth disc surfaces, wear intensity was the highest in the first part of the test. After this time, the wear intensity decreased significantly. In contrast, in the initial phase of the test, the wear intensity of the very rough surface was also significantly lower than the wear intensities of the smooth surfaces.
2. Wear intensities of the discs were similar to the runs of the friction force in time. When the disc surfaces were smooth, the friction forces were stabilized very quickly. Higher surface roughness resulted in longer distance to obtain the steady-state condition and resulted in lower coefficient of friction.
3. As a result of the co-action between the sliding elements, new anisotropic disc surface textures were formed. Elongated wear tracks in the sliding direction were created. Disc wear had mainly abrasive character with addition of the plastic deformation. Wear of ball was significantly smaller than wear of disc.
4. An algorithm to model disc wear was developed. It is based on the calculation of the elastoplastic deformations of the contact between rough surfaces based on DC-FFT. The simulation of disc wear was conducted on the basis of the repetitive contact between sliding surfaces. The disc ordinates in each step were decreased by the values of the calculated deformations.
5. The agreement was found between modeled and experimental results. A very strong correlation was achieved between the modeled and measured volumetric wear levels. The conducted simulations also confirmed the possibility of wear modeling versus time.
6. On the basis of experiment and wear modeling, it was found that various wear levels and friction forces of assemblies with different height were the consequences of the contact between the ball and the disc. The contact of the ball with the smooth disc surface was concentrated in its center, and bulk deformation occurred leading to high wear and friction from the beginning of the test. The contact of the ball with the disc of big surface height occurred at the highest summits, and only asperity deformation initially occurred. Therefore, the material loss was very small at the beginning of the friction process, causing low total wear and friction.

Fig. 17 Profile of lapped disc ($Z1-Sq=0.101 \mu\text{m}$) after machining (a) and after simulated wear for the number of cycles of 20 (b), 40 (c), 80 (d) i 120 (e)

Fig. 18 Profile of vapour blasted disc ($Z_2-S_q=0.387 \mu\text{m}$) after machining **(a)** and after simulated wear for the number of cycles of 20 **(b)**, 40 **(c)**, 80 **(d)** i 120 **(e)**

Fig. 19 Profile of vapour blasted disc (Z9—Sq=3.56 μm) after machining (a) and after simulated wear for the number of cycles of 20 (b), 40 (c), 80 (d) i 120 (e)

Fig. 20 Profile of ground disc ($Z_{13}-S_q=5.51 \mu\text{m}$) after machining (a) and after simulated wear for the number of cycles of 20 (b), 40 (c), 80 (d) i 120 (e)

Declarations

Conflict of interest The authors declare that they have no conflict of interest.

Ethical approval The article was prepared according to the rules of good practice.

Ethical statement I ensure that the work described has been carried out in accordance with Publishing Ethics.

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