 

# Evaluation of Surface Functional Properties of Polymeric Sliding Materials for Lubricated Metal-Polymer Pairs Applications

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**Abstract** *This work presents a comparative study of the surface mechanical properties and tribological wear resistance of polymeric sliding materials. The authors focused on the application of polymeric materials in producing spare parts for vehicles that are still in use but for which original components are no longer manufactured or are disproportionately expensive compared to the vehicle's market value. The subject of the study was frictional interactions between the timing chain and the sliding element. Three commercial materials were tested, designated: PA6G, PA6G+$MoS_2$, PE 1000. A commercial lubricant PMO 5W40 Extreme 100 % (intended for vehicles without a particulate filters) was used. The coefficient of friction and wear were analyzed under dry friction and oil bath. The lowest coefficient of dry friction was obtained for the PA6 G, whereas PE 1000 exhibits the highest. Under oil lubrication, however, PE 1000 demonstrated the lowest coefficient of friction. Microscopic analysis of wear scars was performed. The extent of wear depends on the type of material used and the nature of the interaction (dry friction, oil-bath friction). Wear under dry friction was lower than under oil-bath conditions, possibly due to the influence of the Rebinder effect.*

**Keywords** *tribological wear resistance, polymeric sliding materials, frictional contact, coefficient of friction*

**Highlights**

- *Tests showed two thermoplastics had lower hardness than declared by the manufacturer.*
- *PE 1000 demonstrated low friction under oil condition but high wear due to deformation and the Rebinder effect.*
- *Polyamide with $MoS_2$ improved friction properties more under oil than under dry conditions.*

## 1 INTRODUCTION

Currently, the high cost of new vehicles limits their accessibility for many customers. Many are deterred by an unfavorable price-to-quality ratio or by the proposed drivetrain solutions, which fail to meet customer expectations [1]. As a result, there has been a noticeable rise in the number of repairs involving the reconditioning of camshaft components. Camshaft chain drive systems typically exhibit greater durability but are characterized by higher frictional resistance. A common issue observed is the accelerated wear of timing chain sliders. Furthermore, the unavailability of spare parts poses a significant challenge, particularly for brands whose vehicles remain in use but for which original replacement components are no longer manufactured (e.g. SAAB). The cost of spare parts is often disproportionately high relative to the market value of the used vehicles. Due to the increased emphasis on reducing material consumption and energy use, life-extending repairs are becoming increasingly important [2]. For reproducibility, the widespread availability of suitable materials for repairs is also important. One widely available polymeric material of key technological importance is polyamide. Another popular and relatively inexpensive plastic is polyethylene. Components made from these materials are used extensively in the mechanical engineering and automotive industry [3]. The development of precision computer numerical control (CNC) machining, reverse engineering (RE), computer-aided design (CAD), and three-dimensional (3D) printing enable the rapid production of robust replacement parts. These parts must be characterized by appropriate performance properties [4]. In the automotive industry, there is a high degree of specialization in the production of components, taking into account environmental measures related to the use of materials, components, and energy [5]. In the classification of causes of machine damage, the failure of the components analyzed in this paper is described as a result of fatigue and wear processes and is classified as intermediate-mechanical [6].

The purposes of lubrication in moving friction pairs are to reduce friction, minimize wear, thereby improving durability, enhance heat dissipation from the contact zone and eliminate contaminants and wear particles [7]. The physical essence of lubrication processes is the transformation of unfavorable external friction into friction within the lubricant film [8]. A suitable oil is used to lubricate internal combustion engines, which should be characterized by a low coefficient of friction, the ability to seal mating surfaces and to cool friction pairs, protect against corrosion, to damp vibrations and to reduce negative effects on seals and engine equipment components in contact with oil [9].

The timing system of a passenger car receives drive from the crankshaft using a timing belt or chain. It is more reliable but requires lubrication with engine oil (Fig. 1), and uses chain guide elements. These elements fix the position of the chain, ensuring the desired chain path, and can be used to change camshaft phases, in which a slider modifies the camshaft chain path [10]. Sliding friction occurs at the point of contact between the chain and the sliding element. This is a case of metal-polymer contact, with friction occurring in

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the presence of a lubricant. Despite lubrication, sliding components made of engineered polymers [11] are subjected to frictional wear.

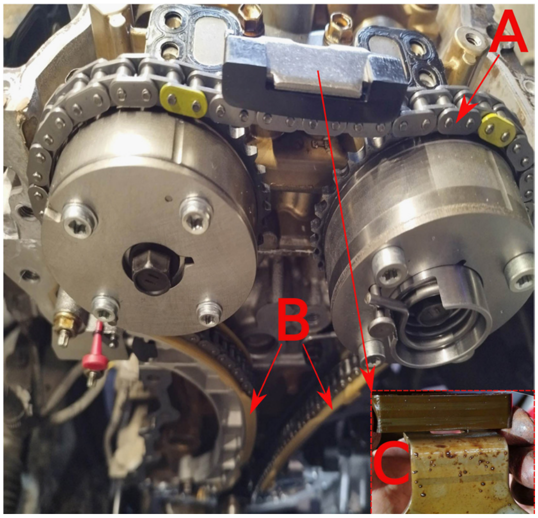


Fig. 1. A photography of the timing system of an internal combustion engine: A timing chain, B timing chain guide slides, and C worn timing chain tensioner slide

Abrasive wear manifests itself in the loss of polymeric material in the surface layer, caused mainly by particle separation resulting from micro-cutting, scratching and furrowing [12]. During the wear process of a sliding element, the surface condition changes unfavorably [13], adversely affecting the performance of the camshaft. Resistance to abrasive wear is a key determinant in the choice of materials before producing replacement parts of suitable operating quality [10]. Typically, thermoplastic polymers (e.g. polyethylene, polyamide), resins and thermosetting polymers (duroplastics e.g. epoxy and polyester resins), elastomers (e.g. polyisoprene, polybutadiene), natural polymers (e.g. compressed wood, cellulose fibers) are used in mechanical engineering [8]. Polymeric materials are applied not only in the accessories of internal combustion engines, but also within the engine block itself. In [14], it was shown that the use of high-strength fiber-reinforced thermosetting polymer components in a modular camshaft design made it possible to significantly reduce the so-called NVH (noise, vibration, harshness) of an automotive vehicle. Thermoplastics are often used in the kinematic nodes of machines, such as sliding bearings and gears. In engineering practice, polymer components exposed to contact loads are selected based on Shore hardness, one of the mechanical parameters quoted by manufacturers. Therefore, the aim of this study is to experimentally verify the tribological properties of selected sliding thermoplastic materials with a specific Shore hardness when paired with a metal component in a lubricated non-conformal contact.

## 2 MATERIALS AND METHODS

### 2.1 Materials

Commercial materials were used in the study, designed to reproduce timing chain slides (skates). Polymeric materials commonly used for this purpose in automotive repair workshops were selected. Table 1 lists the selected materials, namely polyamide (Zatorski, Poland), polyamide with molybdenum disulphide (Zatorski, Poland), and polyethylene (Simona Polska, Poland). They were obtained in the form of rods, from which disks with a diameter of 30 mm were cut on a lathe. Two samples of each material were used in the tribological tests.

PMO 5W40 Extreme 100 % PAO oil was used in the tests. According to the manufacturer's data sheet [16], the oil is described as an engine oil based on high-quality fully synthetic group IV-PAO base oils. The oil's structure is complemented by additives that providing properties such as a stable viscosity index, easy cold starting due to a low pour point, high oxidation protection, wear protection, corrosion protection and sediment formation. The test oil is suitable for petrol and diesel engines without diesel particulate filter, both naturally aspirated and turbocharged in passenger cars and trucks. This oil is particularly recommended for engines that operate under extremely severe conditions. The discs of polymer materials were processed on a Saphir 550 (ATM Gmbh, Germany) laboratory grinder with abrasive papers (grit P600, P1200, P2400) and polished with polishing cloth. The surface was cooled with water during machining.

Table 1. Materials used in the study

| Materials | Label | Manufacturer | Declared shore hardness according to ISO 868 [15] |
|---|---|---|---|
| Polyamide | PA6G | ZATORSKI | 83 |
| Polyamide with molybdenum disulphide | PA6G+$MoS_2$ | ZATORSKI | 83 |
| Polyethylene | PE 1000 | SIMONA | 60 |

### 2.2 Roughness Measurement

Surface roughness of samples machined on a grinding-polishing machine was investigated. A Veeco Dektak 150 (Veeco, Plainview, NY, USA) contact profilometer was used. This profilometer allows 2D topography and 3D surface measurements with a resolution of 0.01 µm in the Z-axis. The measurement resolution was specified as 0.1 µm. The maximal peak profile height (*Rp*), maximum profile valley depth (*Rv*), maximum profile height (*Rz*), the arithmetic everage of the profile hight (*Ra*), the total height of the profile (*Rt*) and the root mean square average of the profile (*Rq*) were analyzed.

### 2.3 Shore Hardness

The hardness test was carried out using a Shore "D" scale apparatus in accordance with the ISO 868 [15], using a Shore HPE II Bareiss durometer mounted on a BS 61 II Bareiss tripod (Bareiss Prüfgerätebau GmbH, Germany) (Fig. 2a). Five samples of each material were used in the shore hardness tests.

### 2.4 Sliding Friction and Wear Test

Sliding friction tests were performed with a 6 mm diameter 100Cr6 steel ball, against a 30 mm diameter polymer disc on a microtribometer (CSM Instruments SA, Switzerland) under a 5 N normal load. For the tribological tests, samples with $Ra$ < 0.5 µm were selected, while those showing surface waviness after cutting on a lathe were excluded. The roughness parameters (including waviness) of the samples were measured on a Veeco Dektak 150 contact profilometer, and visually inspected at 16× magnification to ensure the absence of scratches after grinding and polishing. Consequently, samples with the lowest and most similar roughness were selected for the tribological tests, ensuring repeatability of the surface condition on contact pressure and friction [17].

During a non-conformal contact (Fig. 2b), the sample (disc) performed a rotational motion (360º), with a tangential velocity at the contact point of 48 mm/s. The temperature in the laboratory room was 22 °C. In the first stage, tests were performed under technically

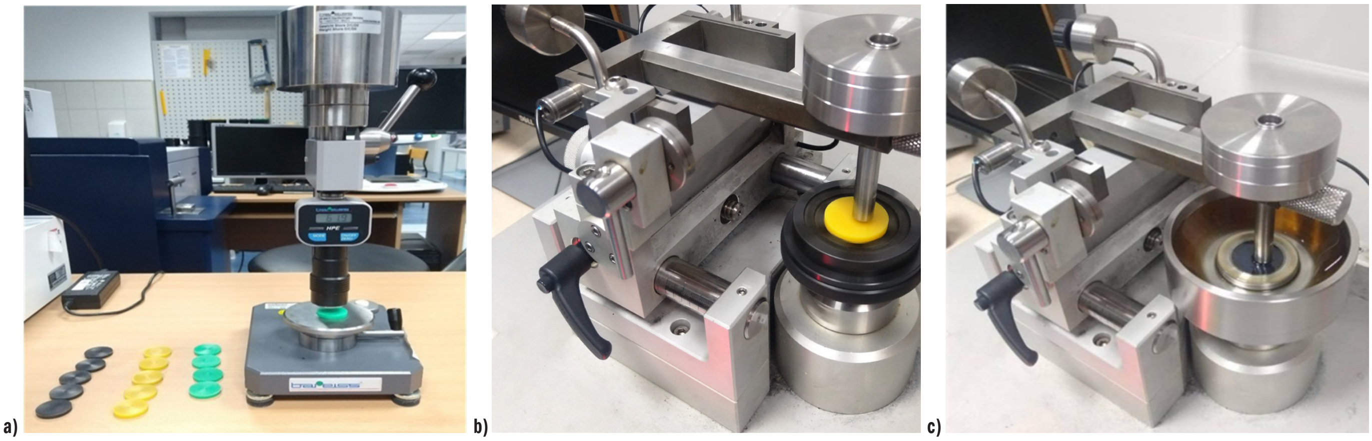

Fig. 2. a) Shore hardness, and friction and wear tests under: b) dry sliding friction conditions, and c) in oil

dry friction conditions. In the second stage, tests were conducted in oil with fully immersed disk (sample) and ball (counterbody), as shown in Fig. 2c.

After the friction tests, the samples were washed and dried. The cross-sectional areas of the friction surfaces were measured; this quantity was designated $S_{AR}$. Measurements were performed on ten consecutive sections along the circumference of the friction track (profilometer table rotated by approx. 35°). The resolution of the measurement was 0.1 µm, with measuring distance between 500 µm and 2000 µm (Fig. 3).

Based on the mean $S_{AR(Mean)}$ of the 10 measurements, volumetric wear was calculated according to Eq. (1):

$$V_{dics} = 2\pi R \cdot S_{AR(Mean)}, \tag{1}$$

where $V_{disc}$ is a volume wear [µm³], and $R$ radius of friction.

### 2.5 Oil Contaminants Test and Microscopic Analysis

Laboratory tests of oil contaminants were performed using the Parker Laser CM20 (Parker Hannifin, USA), a device designed to count solid particles in oil and classify them using of the optical scanning method (Fig. 4). The measurement procedure followed ISO 4406 [18].

It should be noted that the quality of measurements within 5 % can be achieved by using the ISO MTD (ISO Medium Test Dust) [18] and ISO ACFTD (ISO Air Cleaner Fine Test Dust [18]) procedures. As a result, it is possible to obtain results in accordance with the ISO standard [18] in the range of 7 µm to 22 µm, and the National Aerospace Standard (NAS) 1638 [19] in the range of 0 µm to 12 µm, and the Society of Automotive Engineers (SAE) AS4059 [20] code in the range of 0 µm to 12 µm. In the conducted research, it was assumed that contaminants would be classified into six categories of (4, 6, 14, 21, 38, and 70) µm. A Keyence VHX 7000 (Keyence International, Belgium) digital microscope was used for the microscopic evaluation of wear scars, with the focus on the analysis of wear mechanisms.

## 3 RESULTS

The values of the roughness parameters are given in Table 2. The samples exhibited differences in roughness and polishability. The *Ra* parameter varied slightly among the tested samples, while polyethylene showed the highest roughness.

Table 3 shows the statistical parameters of the Shore hardness test results, including the mean (*Mean*), standard deviation (*StdDv*), maximum (*Max*) and minimum (*Min*) values. The hardness of the

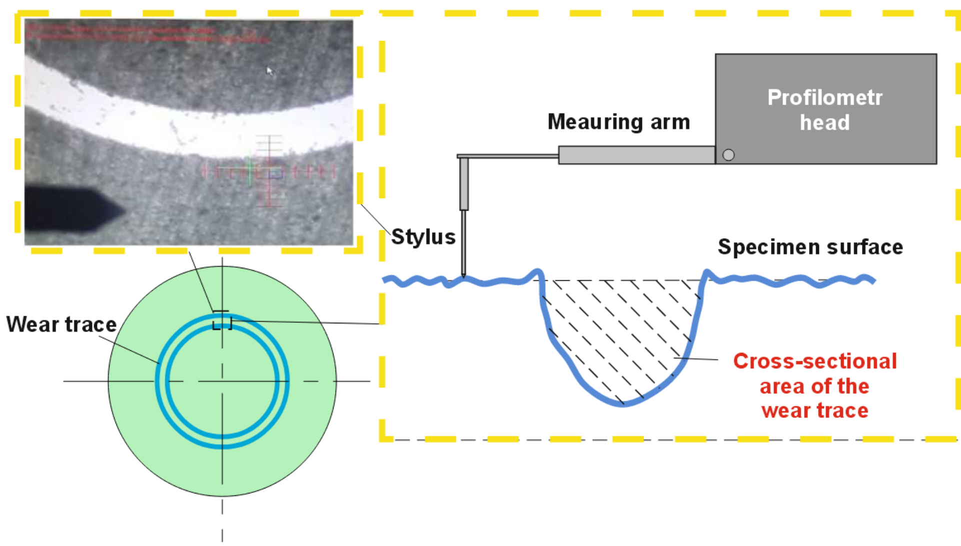


Fig. 3. Scheme for measuring the cross-sectional area of the SAR wear track on a contact (needle) profilometer

Table 2. Roughness parameters of the tested samples

| Material | Label | Sample no. | *Ra* [μm] | *Rq* [μm] | *Rv* [μm] | *Rp* [μm] | *Rt* [μm] | *Rz* [μm] |
|---|---|---|---|---|---|---|---|---|
| Polyamide | PA6G | 4 | 0.199 | 0.255 | −0.697 | 0.873 | 1.569 | 1.118 |
| | | 3 | 0.195 | 0.246 | −0.938 | 0.579 | 1.517 | 1.000 |
| Polyamide with molybdenum disulphide | PA6G+$MoS_2$ | 1 | 0.110 | 0.140 | −0.445 | 0.355 | 0.800 | 0.665 |
| | | 4 | 0.159 | 0.205 | −0.532 | 0.881 | 1.413 | 1.141 |
| Polyethylene | PE 1000 | 5 | 0.449 | 0.765 | −1.742 | 1.389 | 3.431 | 2.287 |
| | | 7 | 0.233 | 0.304 | −1.215 | 0.852 | 2.067 | 1.580 |

materials varied: polyamide with molybdenum disulfide had the highest hardness, while polyethylene the lowest. The statistical spread of the polyamide test results was the largest. The measured hardness of the polyamide samples did not match the manufacturer's data (Table 1), while the mean hardness of polyamide with molybdenum disulfide and polyethylene was consistent with the declared values. The tests were performed in accordance with ISO 868 [15]. A possible causes of hardness variations include surface processing on a lathe and grinder, changes in residual stresses in the surface layer, or even phase transformations that altered the crystalline phase content.

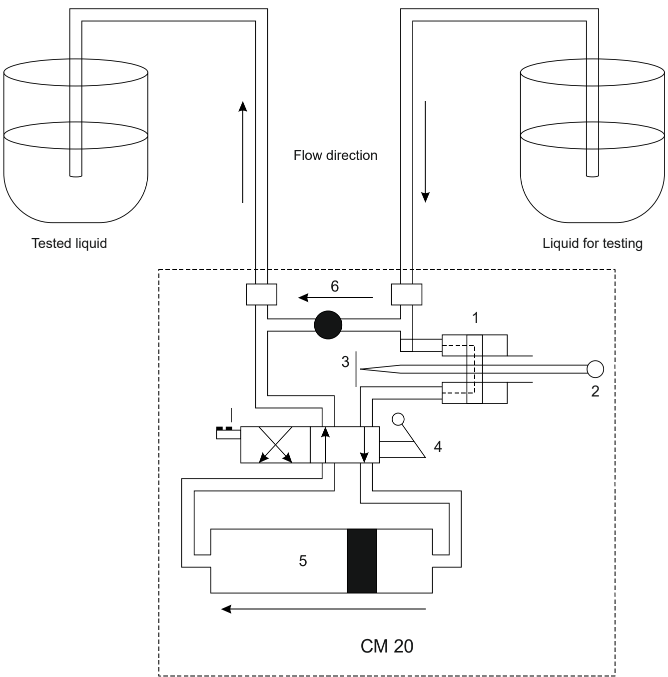


Fig. 4. Diagram of the measuring system for identifying the number of particulate contaminants in oil: 1 measuring chamber, 2 laser light source, 3 optical scanner, 4 switching valve (hydraulic), 5 dosing pump, and 6 flow sensor

In tribological tests, the coefficient of friction was determined. Two types of friction coefficients can be distinguished: one that representing the resistance to the initiation of relative motion (static coefficient of friction) and one representing the resistance during steady motion (kinetic coefficient of friction) [21]. Table 4 compares the value of the static coefficient of friction and the statistical parameters of the kinetic coefficient. The highest tangential coefficient of friction was observed for the PA6G material under dry conditions, while the lowest values were obtained for polyamide with molybdenum disulfide. In all cases of dry friction, the static coefficient was < 0.1, which can be considered low and characteristic of materials with good sliding properties [22]. As expected, lower kinetic friction values were obtained in tests with oil. The greatest variation between dry and lubricated conditions were observed for PE 1000. This material also had the lowest mean kinetic coefficient of friction in oil, while PA6G exhibited the lowest mean kinetic coefficient under dry conditions.

Table 3. Shore hardness ShD

| Material | Label | Min | Max | Mean | StdDv |
|---|---|---|---|---|---|
| Polyamide | PA6G | 64.8 | 75.9 | 69.8 | 3.9 |
| Polyamide with molybdenum disulphide | PA6G+$MoS_2$ | 73.0 | 79.9 | 77.9 | 1.5 |
| Polyethylene | PE 1000 | 56.1 | 62.3 | 60.2 | 1.4 |

The friction coefficient graphs are presented in Fig. 5. The values are higher in dry friction (solid lines). In addition, fluctuations at the beginning of the test without oil lubrication are clearly noticeable.

After approximately 6000 dry friction cycles, the curves stabilized without further fluctuations. For the two polyamide materials, the coefficient of friction decreased slightly. The material with the molybdenum disulfide additive (black line) showed a higher coefficient of friction than polyamide without the additive (yellow line). Polyethylene behaved differently, with friction coefficient values remaining nearly constant after approximately 4000 friction cycles. For samples tested in oil, the friction coefficient values initially increased until roughly 4000 friction cycles. Thereafter, the increase was minimal, and the friction coefficient values remain relatively stable.

Wear was determined by measuring the cross-sectional area of the wear scars. Figure 6 summarizes the profilograms for the PA6G samples tested under dry and oil-lubricated conditions. Table 5 shows the average values from the measured along the perimeter of the wear track. The variation in wear tests is evident. The highest wear was observed for the PE 1000 material and was significantly greater than

Table 4. Static and kinetic friction coefficient of selected samples

| Material | Label | Friction conditions | Start (static) | Min | Max | Mean | StdDv |
|---|---|---|---|---|---|---|---|
| Polyamide | PA6G | dry | 0.09 | 0.08 | 0.14 | 0.11 | 0.01 |
| | | oil | 0.06 | 0.05 | 0.097 | 0.084 | 0.008 |
| Polyamide with molybdenum disulphide | PA6G+$MoS_2$ | dry | 0.0002 | 0.002 | 0.18 | 0.15 | 0.01 |
| | | oil | 0.0003 | 0.007 | 0.097 | 0.086 | 0.007 |
| Polyethylene | PE 1000 | dry | 0.0003 | 0.0003 | 0.19 | 0.17 | 0.01 |
| | | oil | 0.047 | 0.028 | 0.05 | 0.045 | 0.005 |

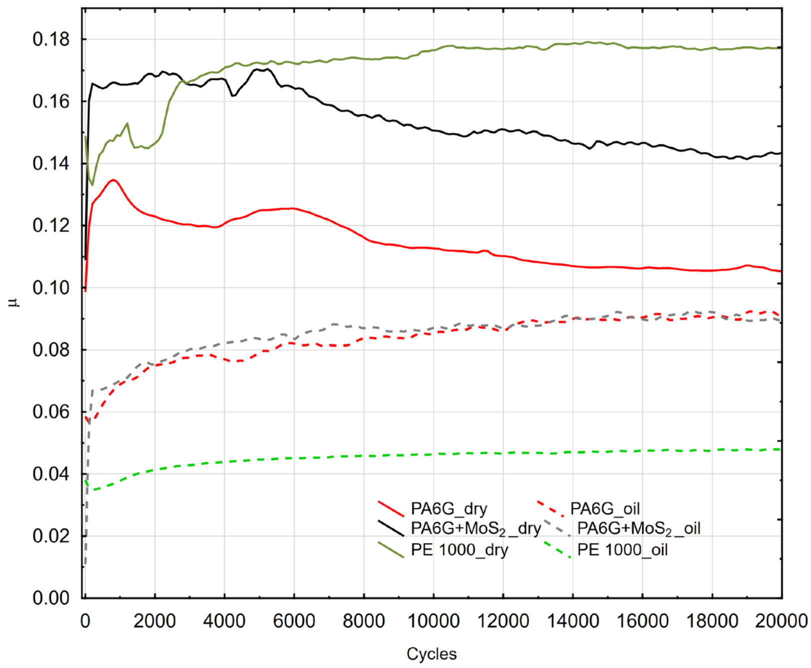


Fig. 5. Friction characteristics of the test kinematic pair; solid lines: dry friction , dashed lines: oil lubrication

Table 5. Tribological wear results (average values)

| | Material | Label | Cross-sectional area of the friction track [$\mu m^2$] | Max depth [$\mu m$] |
|---|---|---|---|---|
| Technical dry friction | Polyamide | PA6G | 160.806 | 1.02 |
| | Polyamide with molybdenum disulphide | PA6G+$MoS_2$ | 148.244 | 1.361 |
| | Polyethylene | PE 1000 | 1316.927 | 6.45 |
| Friction in oil | Polyamide | PA6G | 419.352 | 2.518 |
| | Polyamide with molybdenum disulphide | PA6G+$MoS_2$ | 232.344 | 2.215 |
| | Polyethylene | PE 1000 | 2136.330 | 6.315 |

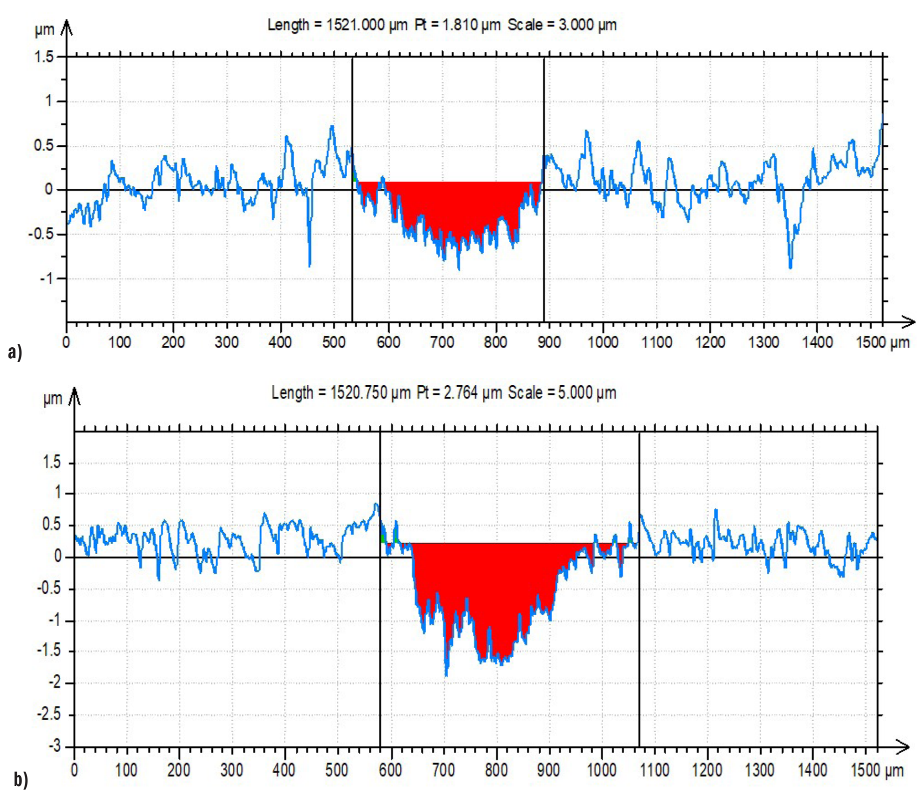


Fig. 6. PA6G friction path cross area: a) dry sliding, b) sliding in engine oil

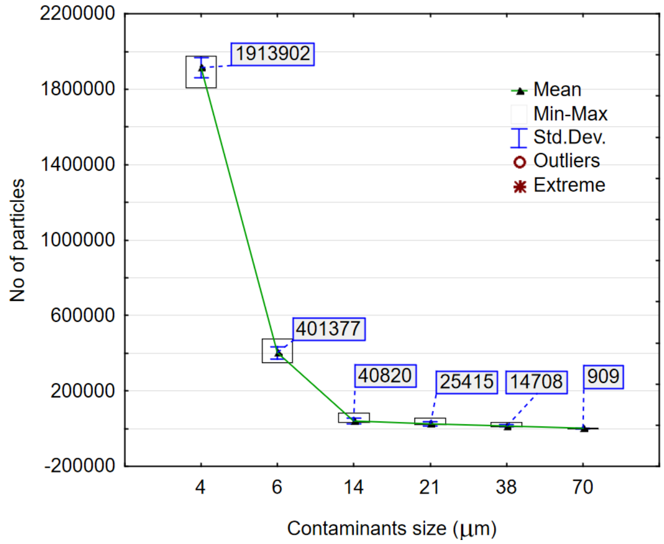


**Fig. 7.** Size distribution of particles suspended in test oil

that of both polyamides. In addition, wear was consistently higher under oil lubrication compared to dry conditions.

Figure 7 shows the particle size distribution suspended in the oil. The proportion of large particles in fresh oil (without any service history) was low, within the majority below 4 μm. Larger particles were likely associated with additives that improve oil properties. The aim was to evaluate the size of particles in oil, as additives often include microscopic soft metal particles ranging from 5 μm to 155 μm. In [22], a beneficial effect of such additives on the lubricating properties of gear oils has been shown. This may explain why the lubricating properties of the tested oils did not deteriorate despite the presence of larger particles. Other studies, including [4] and [21] emphasize that large particles with diameters from a dozen to several dozen micrometers correspond to the dynamic clearance that determines lubricating film thickness. Since dynamic clearance in some lubricated friction pairs should be only 0.1 μm to 1 μm [5], such larger particles are considered harmful. Their presence may affect the continuity of the lubricating film.

## 4 DISCUSSION

Measuring friction in chain drives is complex and therefore usually only possible through secondary variables, such as overall gearbox performance [22]. However, it is known that non-conformal contact and sliding friction occur between chain links (chain joints) and polymer sliders. In the modelling approach, the ball-on-disc method [22] with a 100Cr6 steel ball is typically used. Limited research has also been conducted using special tribometers [23]. As stated in [5] and [24], tribology is a tool for optimizing the durability and reliability of machines. Modelling studies based on tribology principles and methods therefore seem reasonable and were employed here.

The prerequisite for effective lubrication is the strong adhesion of the lubricant to the friction surfaces and the maintenance of an adequate lubricant film thickness, regardless of the conditions, such as friction speed, pressure and temperature [25]. This task is usually difficult to achieve [26]. The lubricant film is formed mainly through physical sorption of polar molecules and chemisorption of boundary layers on the friction surface [27]. In the machine junctions, however, load transfer occurs mainly through a hydrodynamic lubricant film [8]. In non-conformal contacts [28], high-pressure lubrication occurs [29]. The high contact pressure increases the lubricant viscosity and affects the elastic deformation of surfaces that interact indirectly through the lubricant film. For deformable materials, this may lead to macrodeformation [30].

Appropriate tribological properties can significantly improve engine power output and reduce energy losses, thereby lowering fuel consumption [31]. Some published works [32,33] confirm that, over time, a metal-polymer friction pair may transition into polymer-polymer interaction. Experimental results demonstrate that, lubrication with oil leads to different behavior. In the case of the plastics tested, the reduction in the friction coefficient with the oil film is significantly lower (Fig. 5), with the highest effect observed in PE 1000. This material also exhibited the highest measured wear depth and low abrasive damage under dry friction conditions (Fig. 8). Such behavior may be influenced by the so-called macrodeformation effect, defined as the uplift of a softer material (polymer) in front of the harder friction element (metal ball). This behavior affects the energy processes at the friction interface, where part of the mechanical energy is converted into thermal energy [30]. An increase in temperature increases the deformability of the polymer and, at the same time, leads to an intensification of macrodeformation and frictional resistance [31]. The high dry friction coefficient of PE 1000 thus indirectly indicate the relatively high surface deformability. However, under oil lubrication conditions, PE 1000 behaves differently. Among the materials tested, PE 1000 shows the greatest oil absorbtion and adsorbtion capacity. While its wear depth oil changed only slightly compared to dry friction, the cross-sectional area of the wear mark increased significantly, mainly influenced by the broader wear track (Fig. 9). The extent of friction damage may be influenced by the elastohydrodynamic effects of the oil film, determined, among other things, by the relatively high surface deformation of PE 1000. Unfortunately, no similar studies were found among the available publications for similar PE materials and oils tested under similar conditions. Therefore, in the future, research in this direction should be continued.

However, the higher wear observed in the oil test samples was unexpected. One of the mechanisms affecting the wear process under lubrication conditions may be the Rebinder effect, i.e. a reduction in surface energy density, caused by oil adsorbtion onto the friction surface [32]. The formation of defects on the surface of the polymer material and the decrease in surface tension can be explained by the existence of so-called chunking pressures [33]. This phenomenon can be particularly destructive on a friction surface with microcracks [34] initiating further damage at the surface. The disentangling action of liquids also leads to penetration of liquids in material micropores and micro-cavities, initiating further damage through adsorption-induced disentanglement effects [35]. Increased wear of polyamide in oil-lubricated polymer-polymer friction pairs compared to a dry-operation pair was demonstrated in [36], where oil diffusion into surface microcracks and micropores reduced mechanical strength. Figure 8 presents microphotographs of the wear marks. In the case of samples with the highest wear in oil, it displayed significant differences were observed compared to dry friction. Clear traces of abrasive wear were observed, most noticeable in PE 1000, with less severe effect in PA6G , and minimal furrowing in PA6G+$MoS_2$. The effect of deep furrowing to the dry friction of this material was limited. This may be attributed to the influence of the addition of molybdenum disulphide, improving sliding properties. The work [36,37] noted that, polymers containing a so-called 'slip agent', often migrate to the surface of the polymeric component during the friction process. Figure 5 (black line) shows a clear decrease in the friction coefficient of PA6G+$MoS_2$ after about 6000 dry friction cycles. In addition, this material showed the smallest wear difference under dry and oil friction conditions.

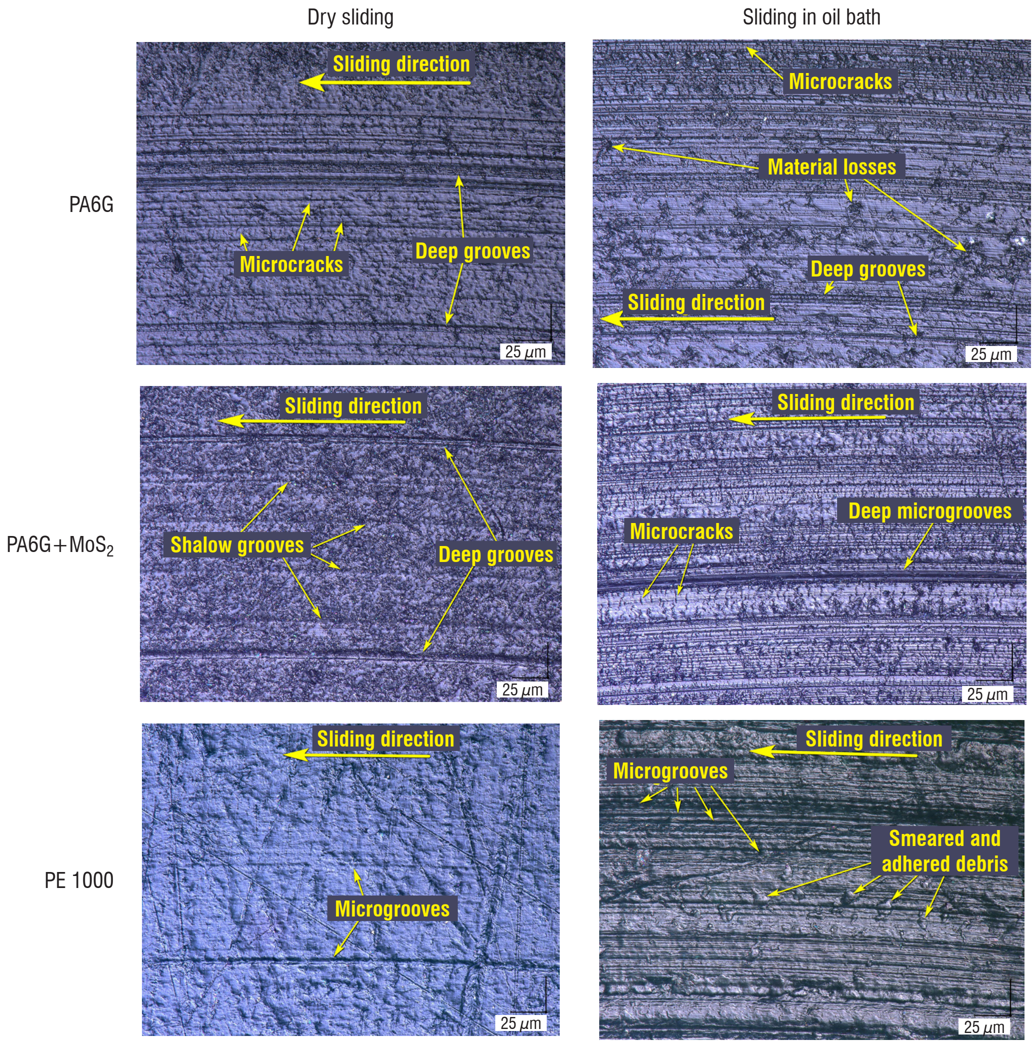


Fig. 8. Wear path of samples tested in dry friction and oil

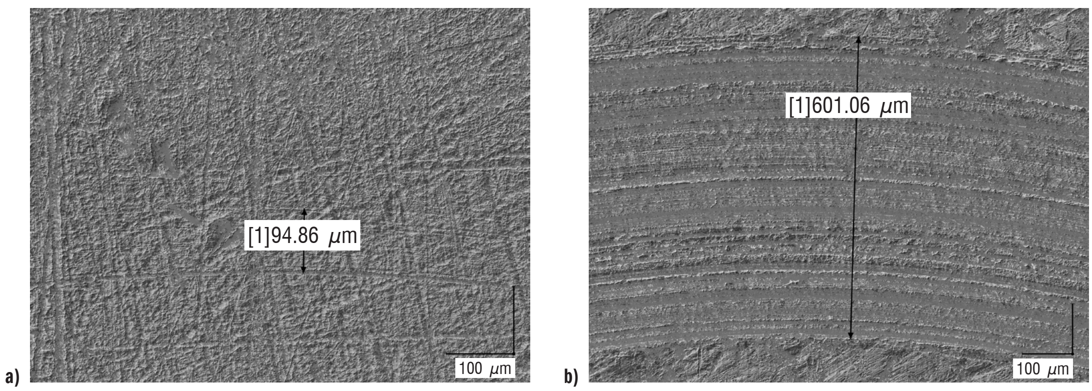


Fig. 9. Microphotographs of the wear path on PE 1000 material samples; a) dry sliding, and b) oil bath sliding

The work [35] concluded that fatigue wear can occur at lower contact stresses when oils with additives are used. The oil investigated in this work contains additives [37] designed, among other things, to stabilize viscosity under varying operating conditions. In addition, Rebinder and Shchukin [37] further noted that lubricant molecules absorbed on a surface can easily migrate along the walls of the microdamage, reaching critical areas that determine the strength properties of the material. It is also likely that the size of the particles in the oil may have an influence [38]. According to ISO 4406 [18], particles larger than 4 μm are considered as a reference value for contaminants. In new oils with no service history, such as the one used in this work, these particles may include additives to the oil base. The influence of these particles is limited to absorption into the friction surface. In the case of suspensions, solid particles impart non-Newtonian behavior to the fluid, with various types of viscosity anomalies [39]. Thus, the progression of tribological damage in the tested samples is most likely influenced by the synergistic effect of a multiple factors. In the literature [34], the following factors are mentioned, i.e. load, speed, temperature, working environment, equipment design, material quality, type of lubricant and many others.

Therefore, case-by-case analysis and comparative studies are key to determining the best materials for each application. In the study presented here, PA6G+$MoS_2$ showed the best anti-wear properties, on the other hand, PE 1000 showed the lowest friction resistance in the oil. Synthetic oils generally retain their lubricating properties longer than mineral ones. Their performance can be influenced by both the service life and the level of contamination resulting from exploitation **[39]**. Synthetic poly-alphaolefin oils (PAOs) are saturated oligomers produced by the catalytic oligomerisation of alphaolefins. PAOs are mainly used as base fluids in high-performance engine lubricants. Compared to mineral oils, PAOs exhibit superior physical and chemical properties **[16]**, including higher fluidity at low temperature, lower volatility and higher viscosity index at lower pour points **[32]**.

Finally, material cost is an important feature when selecting material. The price of PA6G, PA6G+$MoS_2$, PE 1000 depends on the type of product (shaft, plate, tube) and its size. Generally, PA6G and PA6 cost about the same, (approximately 20 €/kg). In contrast, PA6G+$MoS_2$ and PE 1000 are typically about 5 % to 10 % more expensive materials.

## 5 CONLUSION

The aim of this study was to experimentally verify the tribological properties of selected sliding thermoplastic materials with a defined Shore hardness in association with a metal component in a lubricated non-conformal contact. Model tests were conducted under technically dry sliding friction and engine oil lubrication for the metal-polymer pair. Based on the literature and experimental results, the final conclusions can be drawn:

1. The wear of polyamide is relatively low, partly due to the self-lubrication of this material. Under dry friction conditions, the addition of molybdenum disulfide (‘slip agent’) reduced wear in the form of molybdenum disulfide reduced wear by several percent compared to polyamide without this additive. The effect of the “slip agent” became even more pronounced during tests in engine oil, further reducing wear.
2. The PE 1000 material, with the lowest hardness, showed the highest wear. It was several times higher than the wear of both polyamide-based materials. Under oil lubrication, wear increased by about 61.6 % compared to dry friction conditions. While the depth of the abrasion trace was similar in both cases, surface abrasion damage was more severe in oil. It is possible that in both cases the depth of the friction trace was influenced by macrodeformation. Macrodeformation is mainly the uplift of the softer material (polymer) in front of the friction element (metal ball). This phenomenon alters the energy balance in the contact of the friction pair, where part of the mechanical energy is converted into thermal energy **[34]**. This leads to a local increase in deformation and friction resistance, which contributed to the highest dry friction coefficient among the tested materials. Most likely, large local deformation promotes the accumulation of lubricating liquid in the friction zone, explaining the low coefficient of friction observed under lubrication.
3. Despite the correlation between the friction coefficient and hardness of polymer materials postulated by **[38]**, the present study demonstrates the necessity of using direct measures related to tribological properties when selecting materials. Despite the fact that the so-called testing rush **[40]** should be avoided, model tests that refer to the real operating conditions as much as possible provide valuable insights for practical engineering applications.

## References


[1] Blau, P.J. The significance and use of the friction coefficient. *Tribol Int* 34 585-591 (2001) **DOI:10.1016/S0301-679X(01)00050-0**.

[2] Laber A. Modifying the operating conditions of friction pairs with solid lubricant based additives. *Tribologia* 5 137-145 (2011) **DOI:10.1108/ILT-04-2011-0028**.

[3] Bloch, H. (ed.) *Practical Lubrication for Industrial Facilities*. The Fairmont Press, Inc. (2009) Liburn.

[4] Becker, E.P. Trends in tribological materials and engine technology. *Tribol Int* 37 569-575 (2004) **DOI:10.1016/j.triboint.2003.12.006**.

[5] Łęciński, P. Tribology - what you need to know to ensure proper lubrication. *Inz & Utrzymanie Ruchu*, 5 22-30 (2019). (in Polish)

[6] Gil, L., Przystupa, K., Pieniak, D., Kozłowski, E., Antosz, K., Gauda, K., Izdebski, P. Influence of contamination of gear oils in relation to time of operation on their lubricity. *Appl Sci* 11 11835 (2021) **DOI:10.3390/app112411835**.

[7] Lawrowski, Z. Tribology: *Friction, Wear and Lubrication*, Wydawnictwo Naukowe PWN (1993) Warszawa. (in Polish)

[8] Skoć, A., Spałek, J., Markusik, S. *Fundamentals of Machine Design. Volume 2*, Wydawnictwa Naukowo-Techniczne, (2008) Warszawa. (in Polish)

[9] Gronowicz, J. *Environmental Protection in Land Transport.* Wyd. Instytutu Technologii Eksploatacji, (2004) Poznań-Radom. (in Polish)

[10] Luft, S. *Basics of Engine Construction*. Wyd. WKiŁ, (2009) Warszawa.

[11] Capanidis, D., Tański, A. Investigations of resistance to abrasive wear of selected engineering polymers. *Tribologia* 244 25-32 (2012). (in Polish)

[12] Wieleba, W. The mechanism of tribological wear of thermoplastic materials. *Arch Civ Mech Eng* 7 185-199 (2007) **DOI:10.1016/S1644-9665(12)60236-2**.

[13] Chernets, M., Opielak, M., Kornienko, A. Comparison of Load-Carrying Capacity, Wear, and Resource of Metal-Polymer Corrected Spur Gears with a Gear Made of Polyamides PA6 or PA6+30CF. Stroj vestn-J Mech E 68 713-721 (2022) **DOI:10.5545/sv-jme.2022.242**.

[14] Fraunhofer. Plastic camshaft module reduces $CO_2$ emissions. *https://www.fraunhofer.de/en/press/research-news/2020/may/plastic-camshaft-module-educes-co2-emissions.html*, accessed on 2024-02-20.

[15] ISO 868:2003. Plastics and ebonite - Determination of indentation hardness by means of a durometer (Shore hardness). (2003) International Organization for Standardization, Geneva.

[16] PMO. Premium oils. *https://www.pmo-lubricants.com*, accessed od 2024-10-05.

[17] Cortes, D.M., Szczerek, M.M. *Tribotesting. Reproducibility and Repeatability Problems*. ITeE Radom, Universidad de Monterrey, Mexico, PH of IST-NRI in Radom, Poland (2010) Radom.

[18] ISO 4406:2021. *Hydraulic Fluid Power - Fluids - Method for Coding the Level of Contamination by Solid Particles*. (2021) International Organization for Standardization, Geneva.

[19] NAS 1638, Standard for classification of particle contamination in oils and fluids. (1964) National Aerospace Standard, Arlington.

[20] SAE AS4059. The American standard for fuel cleanliness (1988) Society of Automotive Engineers, Warrendale.

[21] Pieniak, D., Jedut, R., Gil, L., Kupicz, W., Borucka, A., Selech, J., et al. Comparative evaluation of the tribological properties of polymer materials with similar shore hardness working in metal-polymer friction systems. *Materials* 16 573 (2023) **DOI:10.3390/ma16020573**.

[22] Laber, A. Modifying the operating conditions of friction pairs with solid lubricant based additives. *Tribologia* 5 137-145 (2011). (in Polish)

[23] Bernhard, A. How the obscure science of rubbing built the past and will shape the future. *Popul Mech* (2020) *https://www.popularmechanics.com/science/a31996381/what-is-tribology/*, accessed on 2023-10-10.

[24] Eftekhari, M., Fatemi, A. Creep-fatigue interaction and thermo-mechanical fatigue behaviors of thermoplastics and their composites. *Int J Fatigue* 91 136-148 (2016) **DOI:10.1016/j.ijfatigue.2016.05.031**.

[25] Pieniak, D., Michalczewski, R., Firlej, M., Krzysiak, Z., Przystupa, K., Kalbarczyk, M., Seykorova, M. Surface layer performance of low-cost 3D-printed sliding components in metal-polymer friction. *Prod Eng Arch* 30 361-376 (2024) **DOI:10.30657/pea.2024.30.36**.

[26] Azman, N., Samion, S. Improvement of the Lubrication Performance of RBD Palm Stearin as an Alternative Lubricant under Different Sliding Speeds. *Stroj vestn-J Mech E* 63 15-24 **DOI:10.5545/sv-jme.2021.7139**.

[27] Kałdoński, T.J., Ozimina, D., Cudziło, S., Durejko, T., Gocman, K., Zasada, D., Kałdoński, T. Comparative investigations of perfluoropolyether oils and selected ionic liquids used as lubricants. *Biuletyn Wojskowej Akademii Technicznej* 62 235-251 (2013). (in Polish)

[28] Chernets, M., Opielak, M., Kornienko, A., & Radko, O. Predictive estimation of sliding bearing load-carrying capacity and tribological durability. *Stroj vestn-J Mech E* 67 363-368 (2021) **DOI:10.5545/sv-jme.2021.7139**.

[29] Woropay, M. Basics of Rational Machine Operation. *ITeE* (1996) Bydgoszcz-Radom. (in Polish)

[30] Bobzin, K., Kalscheuer, C., Möbius, M. P., Rank, M., Oehler, M., Koch, O. Triboactive coatings for wear and friction reduction in chain drives. *Tribol Int* 185 108562 (2023) **DOI:10.1016/j.triboint.2023.108562**.

[31] Meffert, D., Oehler, M., Sauer, B. Precise friction measurement in drive chains using a chain joint tribometer. *Tribol Online* 16 151-158 (2021) **DOI:10.2474/trol.16.151**.

[32] Pytko, S., Pytko, P. Surface tension and the Rebinder effect. *Tribologia* 3-4 143-154 (2007). (in Polish)

[33] Lawrowski, Z. *Tribologia*. Wyd. OW PWR, (2008) Wrocław. (in Polish)

[34] Hebda, M. *Processes of Friction, Lubrication and Wear of Machines*. Wydawnictwo Instytutu Technologii Eksploatacji-PIB (2007). (in Polish)

[35] Edenbaum, J. *Plastics Additives and Modifiers Handbook*. Springer, (1992) New York.

[36] Walczak, M., Szala, M., Pieniak, D. Effect of water absorption on tribological properties of thermoplastics matrix composites reinforced with glass fibres. *Adv Sci Technol Res J* 16 232-239 (2022) **DOI:10.12913/22998624/147515**.

[37] Rebinder, P.A., Shchukin, E.D. Surface phenomena in solids during the course of their deformation and failure. *Sov Phys Uspekhi* 15 533-669 (1973) **DOI:10.1070/PU1973v015n05ABEH005002**.

[38] Jia, B.B., Li, T.S., Liu, X.J., Cong, P.H. Tribological behaviors of several polymer-polymer sliding combinations under dry friction and oil-lubricated conditions. *Wear* 262 1353-1359 (2007) **DOI:10.1016/j.wear.2007.01.011**.

[39] Rawls, A.S., Hirt, D.E., Havens, M.R., Roberts, W.P. Evaluation of surface concentration of erucamide in LLDPE films. *J Vinyl Addit Techn* 8 130-138 (2002) **DOI:10.1002/vnl.10353**.

[40] Benda, R., Bullen, J., Plomer, A. Synthetics basics: Polyalphaolefins-base fluids for high-performance lubricants. *J Syn Lubr* 13 41-57 (1996) **DOI:10.1002/jsl.3000130105**.



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**Data availability** *The data that underpin the results of this research are available for use by other researchers. Additionally, the corresponding author can provide additional raw data upon reasonable request.*

**Author contribution** *Daniel Pieniak: Conceptualization, Formal analysis, Investigation, Methodology, Resources, Writing – original draft; Mateusz Otrocki: Conceptualization, Investigation, Resources, Visualization, Writing – original draft; Leszek Gil: Conceptualization, Investigation, Methodology, Validation; Agata Walczak: Formal analysis, Funding acquisition, Project administration, Resources, Writing – review & editing; Krzysztof Przystupa: Data curation, Project administration, Resources, Visualization, Writing – review & editing; Artūras Kilikevičius: Data curation, Resources, Supervision, Writing – review & editing*

## Ocena funkcionalnih lastnosti površin polimernih drsnih materialov za uporabo v mazalnih kovinsko-polimernih parih


**Povzetek** *V članku je predstavljena primerjalna študija mehanskih lastnosti površin in odpornosti proti obrabi polimernih drsnih materialov. Avtorji so se osredotočili na uporabo polimernih materialov pri izdelavi nadomestnih delov za vozila, ki so še vedno v uporabi, vendar za njih originalni deli niso več v proizvodnji oziroma so nesorazmerno dragi glede na tržno vrednost vozila. Predmet raziskave so bile ˝tribološke interakcije med verigo krmiljenja in drsnim elementom. Preizkušeni so bili trije komercialni materiali, označeni kot: PA6G, PA6G+MoS$_2$ in PE 1000. Uporabljeno je bilo komercialno mazivo PMO 5W40 Extreme 100 % (namenjeno vozilom brez filtrov trdnih delcev). Analizirani so bili koeficient trenja in obraba pri suhem trenju ter pri delovanju v olju. Najnižji koeficient suhega trenja je bil dobljen za PA6G, medtem ko je PE 1000 pokazal najvišji koeficient. Pri uporabi olja pa je PE 1000 izkazal najnižji koeficient trenja. Izvedena je bila mikroskopska analiza obrabnih sledi. Obseg obrabe je odvisen od vrste uporabljenega materiala in od načina interakcije (suho trenje ali prisotnost oljnega filma). Obraba pri suhem trenju je bila manjša kot pri uporabi olja, kar je lahko posledica vpliva Rebinderjevega učinka.*



**Ključne besede** *odpornost proti obrabi, polimerni drsni materiali, stik ob trenju, koeficient trenja*