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Wear-coefficient analyses for polymer-gear life-time predictions: A critical appraisal of methodologies

S. Matkovič [a], A. Pogačnik [b], M. Kalin [a,*]

[a] Laboratory for Tribology and Interface Nanotechnology, Faculty of Mechanical Engineering, University of Ljubljana, Bogisiceva 8, 1000, Ljubljana, Slovenia

[b] Bauhar s.p, Seliska Cesta 6b, 4260, Bled, Slovenia

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ABSTRACT

Polymer gears are used in many devices due to their advantages, among other, dry-running possibilities. As part of their design, the life-time predictions for fatigue and wear are among the most critical and necessary for any application. In a wear life-time prediction, the wear coefficient is the key parameter. The VDI 2736 guideline considers the wear coefficients obtained from pin-on-disc tests. Due to the obvious possible differences in wear coefficients from pin-on-disc tests and actual gear tests, these data can vary considerably. Moreover, the methodology for determining wear coefficients in a real-scale gear test has several possible variations that can lead to significant differences. This study evaluates several methods of determining the wear coefficient, i.e., pin-on-disc, as well as seven methodologies based on real-scale gear tests. Two different polyacetal (POM) materials (Delrin 500P and Hostaform C9091), typical for polymer gears, were used for the pin-on-disc analyses and a re-evaluation from the literature. The Delrin 500P was additionally used in real-scale gear tests to provide a direct comparison of the wear coefficients with the selected methods. The overall conclusion is that the wear coefficient of polymers used in gear-design calculations should be obtained from real-scale gear tests. This can be concluded from empirical wear coefficient results, as well as from surface wear mechanisms analyses.

1. Introduction

Polymer gears are being used in ever-more-demanding applications, where accurate design calculations must be performed to predict their service lives. Two of the most popular polymer materials used for gears are polyamides (PA) and polyacetals (POM) [1]; however, other types, like PBT, PEEK, PPS, PAI [2], also find uses. The benefits of using polymer gears are their low weight, ease of manufacturing, design freedom, low cost, low noise and low vibrations. In addition, polymer gears can often operate without lubrication. These benefits are the result of the polymers having different material properties, compared with metals [3]. However, they also have some less-good properties. An example being the Young's modulus, which can be as much as one hundred times lower than the values obtained with steel [4]. Additionally, the Young's modulus of polymers can vary significantly with temperature. Another drawback is the thermal conductivity of polymers, which is also much lower than for steel (~100 ×) [4]. This results in increased operating temperatures, since the generated frictional heat is not removed from the teeth efficiently [5–7]. As a result, permanent deformation, and in some severe cases even melting, of the teeth can occur, which is not the case with metal gears. Other types of failure in polymer gears include flank fatigue [8] with pitting [9], root fatigue [10, 11], and wear [6,12]. Accordingly, to minimize the chance of thermal failure, a metal pinion is often selected in a pair with a polymer gear, which better dissipates the heat from the contact surfaces compared with a pair of meshing polymer gears.

One of the most significant advantages of polymer gears compared with their metal variants is their ability to operate in non-lubricated conditions, albeit with an increased risk of wear [13,14]. The wear on the tooth flanks leads to tooth-profile changes, e.g., tooth pressure angle changes, therefore changing the loading conditions on a tooth, which leads to different sliding velocities and contact pressures [13,15–17]. These changes can lead to an increase of the gear's contact temperature, which decreases the polymer's strength and can also result in tooth deformation and flank melting. Therefore, for dry-running gears in particular, the wear of a polymer-gear tooth is a key design parameter and must be verified in the design phase.

In contrast to metal gears, for which the calculation and design are standardized through DIN 3990 [18], ISO 6336 [19], AGMA 2001-D04 [20], etc, the design of polymer gears has no standard. In 2014, the VDI

* Corresponding author.
E-mail address: mitjan.kalin@tint.fs.uni-lj.si (M. Kalin).

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2736 guideline was proposed [21], and it is now the only generally available guideline that considers the wear of polymer gears.

Thus, although the VDI guideline is very important today for the design of polymer gears, it also has room for improvement. For example, the currently available databases required for a wear calculation only include POM/steel and PBT/steel contacts [21], which limits their applicability for other materials. However, a major obstacle is that the wear coefficients in these databases were obtained from pin-on-disc experiments. In other words, VDI 2736 gives the recommendation that the wear coefficient used in the prediction of a gear's wear life is obtained from pin-on-disc tests. The pin-on-disc wear coefficient is commonly calculated according to eq. (1) [22], where ΔV is the volume of the lost material, F is the normal load acting in the contact and s is the total sliding distance of the pin. The pin-on-disc wear-coefficient calculation is thus trivial. Nevertheless, it is well known that the wear of polymer materials, and so the wear coefficient, in pin-on-disc experiments, can vary since it strongly depends on many factors, such as sliding velocity, roughness, contact pressure, environment temperature and moisture, mechanical and thermal material properties, including any reinforcements and microstructure characteristics [23–28].

$$k_w = \frac{\Delta V}{F \cdot s} \tag{1}$$

However, the contact conditions in gears [29,30] are even more complex because they change during the gear's operation and are, in general, very different from those in pin-on-disc experiments. Namely, the motions of the surfaces in gear teeth are a combination of sliding and rolling in varying proportions as the roll angle changes [29], Fig. 1. This changes the load, deflections and contact conditions along the meshing line. Moreover, the inevitable wear causes the form of the tooth to change, which then modifies the conditions of the contacts along the meshing-line position and the gear's lifetime. For all these reasons, the values of the wear coefficient obtained from the pin-on-disc tribological tests are expected to vary from the wear recorded for a real gear in operation.

Similar to the contact conditions, the methods used to measure the wear of polymer gears when using gear test rigs [31] also have more variations compared to a pin-on-disc test. A gear-wear measurement in real time can be performed by measuring the movement of the bearing block, which rotates around the pivot with relatively large angles [14, 32]. In this way the measurement is made in terms of the reduction of the tooth's thickness, measured at the operating pitch point. However, in this method, the tooth's deformation due to the load cannot be separated from the wear, and thus the uncertainty in the measurements can be significant [33].

Other methods used for wear measurements are not in real time, but are performed on gears after a certain number of cycles, after which the test is stopped or continued. Typically, a weight-loss measurement on an accurate analytical balance would be made and the result compared to the weight of the gear before the test. A drawback of this method is that the humidity or a lubricant can affect the calculated values [31]. This could be, to some extent, eliminated by drying/cleaning the gears prior to the testing or examining the relationship between temperature, time, humidity and weight changes. Another indirect and very common method is to compare the form of the tooth's side-view profile before and after the test. This can be done using a microscope together with edge-detection software. In this way the changes from the original gear profile are determined and the extent of the weight loss is calculated. The drawback to this technique is that the deformations of the teeth due to a visco-elastic-plastic behaviour [3] can deviate from reality, and the in-situ deflections might not be encountered. This can, however, be compensated by measuring the un-worn tooth and then preforming a profile alignment. The common drawback of all indirect measurements is that when evaluating progressive wear, in multiple steps via intermitted tests, the gears need to be dismounted and removed from the test rig. Mounting the gears back to the test rig can introduce a mounting error that can cause misalignment and result in errors.

Accordingly, there are still numerous questions regarding how to identify a relevant wear coefficient for a polymer gear's life-time prediction, which is essential for the polymer gear's design and use in any application. Today, the question remains: what is the expected difference between the wear coefficient in gears and in a pin-on-disc test, and how the use of the latter will affect the calculated wear life time of the gears when compared to cases in practice? It is also not clear what are the uncertainties with all the possible methods to calculate the wear coefficient even in real-scale gear experiments, and how the different possible methodologies affect the predicted gear-wear coefficient. In this study we have attempted to answer these questions by measuring and comparing the wear coefficients used for polymer gears' design calculations, obtained using several methods. The wear coefficients obtained from standard, tribological pin-on-disc tests, an economical and rapid method in accordance with VDI 2736, were compared to the wear coefficients obtained using real-scale gear tests under the same contact conditions. The gear-wear coefficient was determined from gear tests

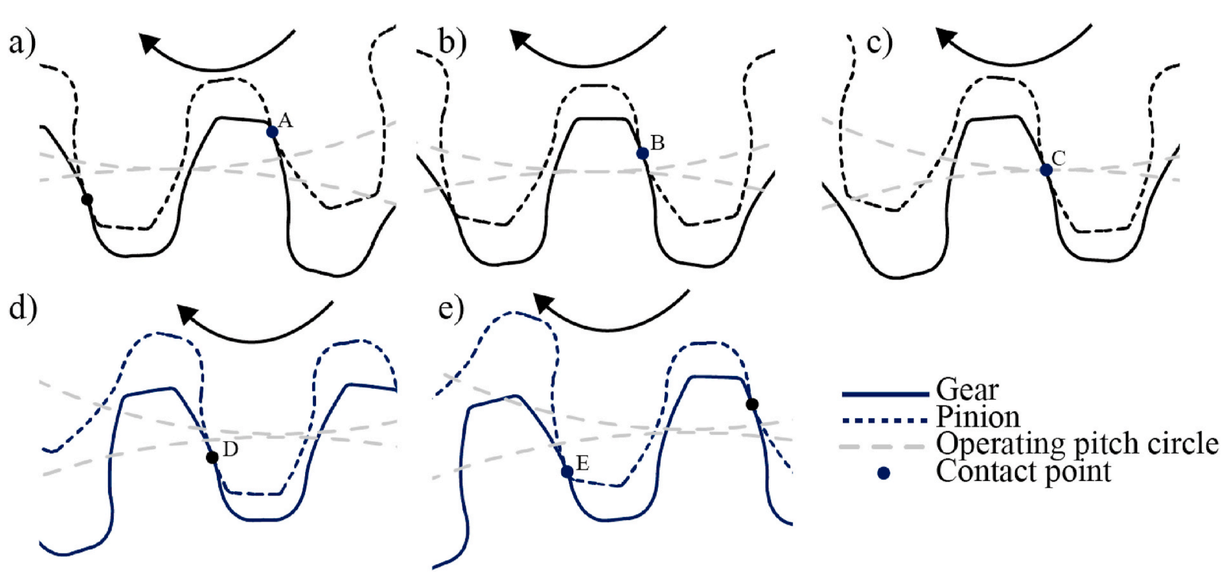

Fig. 1. Theoretical contact points for cylindrical gear meshing: a) point of first contact, b) highest point of single tooth contact, c) kinematic point, d) lowest point of single tooth contact, e) last point of contact.

with different methods: the gear weight-loss method, the tooth worn-out area, the tooth linear wear (thickness reduction) and by using a loaded tooth contact analysis (LTCA) and the commercial software KISSsoft [34]. The wear coefficients after different gear life times and intermittent tests were also analysed.

2. Experimental

2.1. Materials and specimens

The polymer samples were injection-moulded polymer pins (Φ2.93 × 15 mm) from two polyacetal materials that are typically used for gears: Delrin 500P (DuPont, Germany) and Hostaform C9021 (Ticona, Celanese, USA). The polymer pin's surface roughness before the tests was R_a = 0.5 μm. Due to the flat-on-flat contact geometry, special attention was devoted to achieving a parallel position between the pin and the disc prior to the test for the wear-coefficient measurements. An in-house methodology was applied to achieve parallel and equally rough polymer pin surfaces. This procedure also ensured that the running-in phase of the later pin-on-disc tests was short. The pin-on-disc test's counter-body rotating samples were 100Cr6 steel discs, measuring 24 mm in diameter and with a thickness of 8 mm. Their roughness was prepared to R_z = 1.5 μm ($R_a \approx$ 0.2 μm), measured using a stylus-tip profilometer (T8000, Hommelwerke GmbH, Schwenningen, Germany). The hardness was 62 HRC.

The second type of samples were gears for real-scale gear tests. A steel pinion versus the polymer gear combinations was studied, Table 1. Steel gears were produced by machining with a hardness of 62 HRC. The surface roughness of the pinon flank was the same as the surface roughness of the steel discs (R_z = 1.5 μm). Overall, gear quality 6, according to ISO 1328 [35], was achieved. The polymer gears were injection moulded from polyacetal Delrin 500P. They were produced using no-weld injection-moulding technology. Quality 10, according to ISO 1328 [35], was achieved.

2.2. Tribological tests

The tribological tests were performed using a pin-on-disc machine (CSM Instruments, Peseux, Switzerland) under dry-sliding conditions at room temperature (24 ± 2 °C) and humidity (50 ± 10%), Fig. 2. The testing machine consists of a turntable, which holds the steel disc and is driven by a servomotor. The polymer pins were fixed on the upper pivoting arm, where the normal load is applied. The tangential forces were measured through the deformation of the loading arm using an LVDT sensor. The contact conditions were set as close as possible to those in Ref. [28], which are used for the polymer-wear coefficient in the gear's life-time predictions in VDI 2736 [21]. The normal load was 27 N, resulting in a nominal contact pressure of 4 MPa. The sliding speed was kept constant throughout the tests at 0.5 m/s at the disc radius of 10 mm, which was used in all the experiments. The sliding distance was 5000 m. Considering the steel disc's revolutions, the corresponding number of cycles was 8.5×10^5. The coefficient of friction always reached a steady state, confirming the relevance of the testing distance. Each test was repeated six times and the results from all six tests are presented in the results section.

Table 1
Test gear specifications.

Parameter	Pinion	Gear
Material	100Cr6	Delrin 500P
Normal module, mm	0.8	
Number of teeth	17	22
Helix angle, °	0	0
Transmission ratio	1.29	
Facewidth, mm	8	6
Normal pressure angle, °	20	20
Profile shift	0	0
Gear quality	6	10
Reference profile	1.250/0.250/0.925	1.250/0.300/0.825

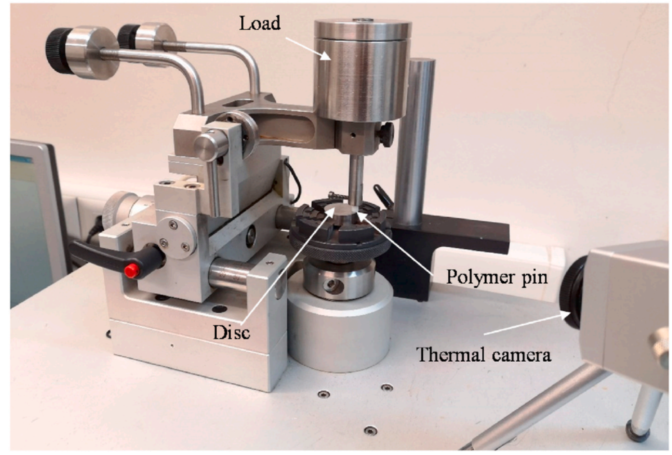

Fig. 2. Pin-on-disc apparatus with thermal camera on the side.

It is well documented that the temperature has a critical effect on the performance of polymers [36]. Thus, during all the tests, the pin temperature was monitored (Optris PI160, Optris GmbH, Germany). The emissivity was set at 0.92. The measurement area was set about 0.15 mm away from the contact surface. Both the maximum and mean temperatures were measured.

Before the tests, all the pins were weighed on an analytical balance (XA 210/X, Radwag, Poland) with a precision of 10^{-5} g. After the test, the measuring procedure was repeated to obtain the mass loss and subsequently the pin-on-disc wear coefficient (k_{Wpod}) in mm^3/(Nm) was calculated using eq. (2).

$$k_{\mathrm{Wpod}} = \frac{\Delta m}{\rho \cdot F \cdot s} \tag{2}$$

where Δm is the mass loss of the specimen, ρ is the density of the specimen, F is the normal force and s is the total sliding distance.

The worn surfaces were examined using a scanning electron microscope (SEM, JEOL JSM-IT100, Japan) at an accelerating voltage of 10 kV. Prior to the examination, the specimens were sputter coated (SCD005, Baltec AG, Liechtenstein) with a 20-nm-thick gold coating (sputtering at a 35-mm working distance using 30 mA for 100 s).

2.3. Gear-wear tests

The second set of experiments was performed using an in-house, open-loop test rig for polymer gears [36], Fig. 3. The test rig consists of the driving shaft and the driven shaft. The steel pinion is fixed on the driving shaft, which is controlled by a servomotor that adjusts the operating speed. The polymer specimen gear is mounted on the driven shaft, which is connected to a brake for adjusting the torque with high accuracy. The position of the driving shaft can be adjusted in the x and y directions. With two accurate torque sensors, the efficiency of the gear meshing can also be measured.

To monitor and control the gear's temperature on the flank and the root in real-time, an IR thermal camera (Optris PI400, Optris GmbH, Germany) together with an insulating chamber around the meshing gear pair was used. The camera is connected in a feedback loop to control the air flow into the chamber, which enables a constant controlled gear temperature with a ±1 °C variation on the desired area of the polymer gear, Fig. 4.

In order to compare the wear coefficient calculated from pin-on-disc measurements with the wear coefficient from the gear tests, the testing conditions were carefully chosen. The rotational speed of the pinion was

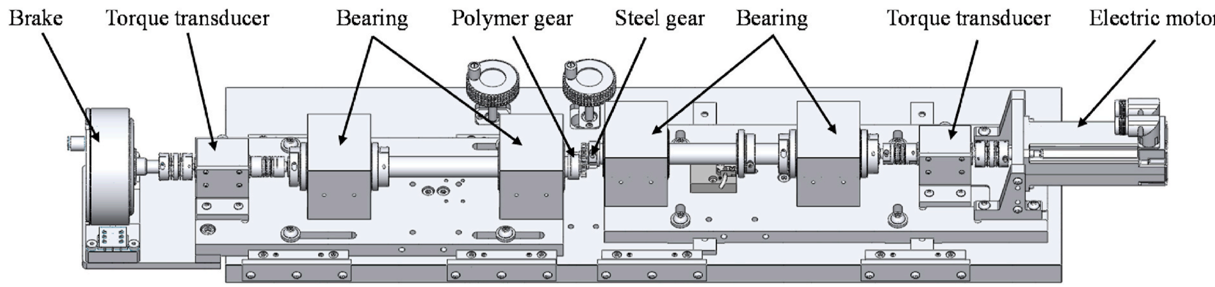

Fig. 3. Schematic of the polymer gear test rig.

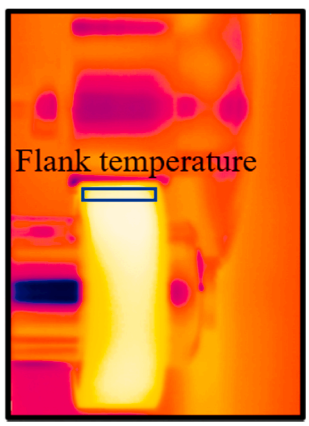

Fig. 4. Control of gear's flank temperature.

set to 1805 rpm, which corresponds to the 1395 rpm of the polymer gear. Considering the gear geometry from Table 1, this correlates with a sliding velocity of 0.5 m/s at the tip of the polymer gear. To prevent root failure before any measurable wear, the output torque was set to a low value of 0.35 Nm (root stress of 31.26 MPa according to the VDI 2736) and the temperature on the flank was set to 40 °C and kept constant throughout the test. The temperature was set to 40 °C as this is the polymer pin temperature that was obtained during the pin-on-disc tests.

To compare the wear coefficient in the pin-on-disc and the real gear tests, the first set of gear experiments was defined, which lasted for 10×10^6 cycles of gear meshing. The real-scale gear-wear coefficient was determined after this number of cycles, which corresponds to 92.3 h of operation for each test. After the test was completed, the polymer gear's weight loss was measured on an analytical balance, the same as used for the pin-on-disc tests, and an image of the worn tooth profile was taken using a Wild M3Z optical microscope (Wild – Leica, Switzerland) at 25 × magnification. Each test was repeated three times to determine the statistically repeatable and relevant average gear-wear coefficient of the Delrin 500P gears. The wear mechanisms of gear flank (Fig. 5) was observed under SEM using same conditions as for pins.

2.4. Progressive gear-wear test

Another set of gear tests was performed to investigate in detail how the flank's geometrical profile changes and how this compares to the wear coefficient obtained from different methods, which are explained in detail in the following sections. In this set of experiments, after several prior trial tests to establish the flank's wear-progression pattern, a single test gear pair was run up to 21×10^6 cycles and was stopped periodically at 15, 17, 19 and 21×10^6 cycles. After each intermittent test stop, the polymer gear's mass loss was measured on an analytical balance and an image of the worn tooth profile was taken using the optical microscope. Twenty-one million cycles were selected as the final stop because with a larger number of cycles tooth-root fatigue failure might occur.

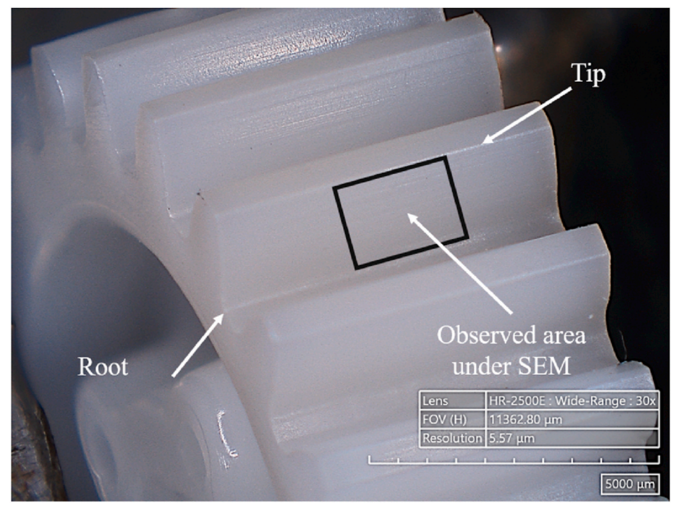

Fig. 5. Polymer gear observation area.

Moreover, at this stage the gears were still operating at a reasonably high efficiency of ~95%, and were thus fully functional.

2.5. Calculation of wear-coefficient-based gear-test results

The wear on the gears was evaluated using two different wear-loss measurement techniques: a mass-loss measurement using an analytical balance (the same as described for the pin-on-disc tests) and a gear-profile shape change (worn-out cross-section side view) using an optical microscope (same as described in the section on gear tests). By employing both of these techniques, the wear coefficient can be calculated with several different methodologies, which are described in detail as follows. The summary of all seven types of analysed gear-wear coefficients is presented in Table 2.

Table 2
Summary of the seven different types of gear-wear coefficients.

Symbol	Meaning	Calculation equation	Measurement technique for wear
$k_{Wweight}$	weight loss	eq. (7)	analytical balance
$k_{Wweight.pr}$	progressive weight loss	eq. (7) (M_W progressive)	analytical balance
k_{Warea}	worn-out area	eq. (8)	optical microscope
$k_{Warea.pr}$	progressive worn-out area	eq. (8) (A_W progressive)	optical microscope
k_{Wwm}	average linear wear on pitch line (reduction of tooth thickness)	eq. (9)	optical microscope
k_{Wltca}	Iterative LTCA	LTCA method in KISSsoft [34]	optical microscope
k_{Wvdi}	pin-on-disc wear data	VDI constant value [21]	defined in ref. [28]

According to VDI 2736, the wear of polymer gears is theoretically calculated with two different equations [21]: the local linear wear W_{local} (eq. (3)) and the averaged linear wear W_m (eq. (4)). The gear wear depends on the geometry of the meshing gears, such as the common face width (b_w), the profile line length of the active tooth flank (l_{Fl}) (eq. (5)) and the number of teeth (z). The equation for the averaged gear linear wear also considers the nominal torque (T_d), the number of load cycles (N_L), the local specific sliding (ζ), the degree of tooth loss (H_V) (eq. (6)) and the wear coefficient (k_w). The degree of tooth loss is calculated through the partial contact ratio of the pinion (ε_1) and the gear (ε_2), the helix angle at the base circle (β_b) and the gear ratio (u).

$$W_{lokal} = \frac{F_{n,local}}{b_w} \cdot N_L \cdot \zeta \cdot k_w \tag{3}$$

$$W_m = \frac{T_d \cdot 2 \cdot \pi \cdot N_L \cdot H_V \cdot k_w}{b_w \cdot z \cdot l_{Fl}} \tag{4}$$

$$l_{Fl} = \frac{1}{d_b}\left(\left(\frac{d_{Na}}{2}\right)^2 - \left(\frac{d_{Nf}}{2}\right)^2\right) \tag{5}$$

$$H_V = \frac{\pi\,(u+1)}{z_2 \cdot \cos\beta_b}\left(1 - \varepsilon_1 - \varepsilon_2 + \varepsilon_1^2 + \varepsilon_2^2\right) \tag{6}$$

2.5.1. Methods based on mass-loss measurements

Using mass-loss measurements of gears, two different wear coefficients can be calculated. The first one is the weight-loss wear coefficient $k_{Wweight}$, which is calculated using eq. (7) based on the VDI guidelines [21]. With this methodology, M_W is the mass difference between the unworn and the worn gear, T_d is the nominal torque, N_L is the number of load cycles, H_V is the degree of tooth loss and ρ is the density of the material. In addition, the gear-geometry values from Table 1, the degree of tooth loss H_V (according to eq. (6), H_V was 0.179) and the (previously described) material properties must be employed.

$$k_{Wweight} = \frac{M_W}{T_d \cdot 2 \cdot \pi \cdot N_L \cdot H_V \cdot \rho} \tag{7}$$

Another method to calculate the gear-wear coefficient is the progressive intermittent gear-wear test, were the gear's mass loss M_W was calculated by considering the progressive gear mass-loss between intermediate stops. Therefore, the $k_{Wweight,pr}$ annotation is used to distinguish the two wear coefficients. The stops were selected at 15, 17, 19 and 21×10^6 cycles, as mentioned earlier, and the wear coefficient between these periods was calculated.

2.5.2. Method based on optical worn-out cross-section tooth-profile analyses

A graphical evaluation of the wear was conducted by evaluating the wear by comparing the initial and the worn-out cross-section profile. In this way we can reliably measure the worn cross-sectional area (A_W) and also the linear wear, i.e., the tooth thinning (W_m) at the pitch line, as shown in Fig. 6.

Based on graphical observations of the worn profiles (Fig. 6), the wear loss was calculated from the side-view cross-sectional-area difference between the unworn original profile and the worn-out profile. A wear coefficient k_{Warea} based on the worn-out area is introduced. The profiles obtained at 10, 15, 17, 19 and 21×10^6 cycles were compared with the original tooth profile, the worn-out-area cross-section difference (A_W) in the profiles was multiplied by the gear width (b_w) and the number of teeth (z), divided by the normal load and the sliding distance, to obtain this wear coefficient, eq. (8). Like the wear coefficient $k_{Wweight,pr}$, the value of $k_{Warea,pr}$ can be calculated by considering the progressive worn-out-area difference.

$$k_{Warea} = \frac{A_W \cdot z \cdot b_w}{T_d \cdot 2 \cdot \pi \cdot N_L \cdot H_V} \tag{8}$$

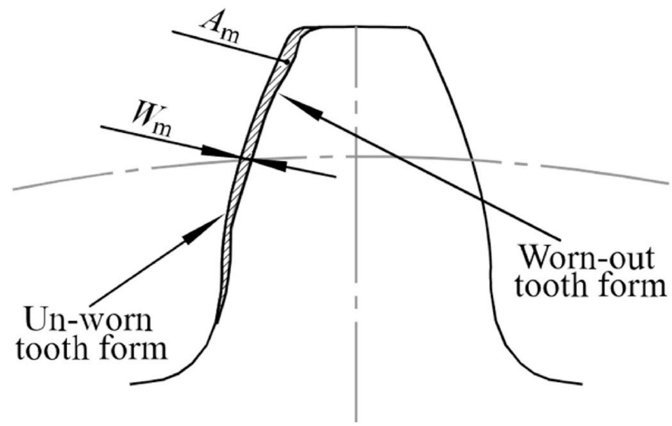

Fig. 6. Measurement of the tooth's worn-out cross-section area.

2.5.3. Wear coefficient based on the linear wear of a gear

By using an optically measured graphical evaluation of the worn-out tooth's profile, the wear coefficient k_{Wwm} can be calculated from eq. (9) based on the measured linear wear W_m representing the reduction of the tooth's thickness, i.e., gear thinning, on the pitch line. Linear wear W_m is an important parameter in polymer-gear design, recognised in VDI [21] as the limiting parameter for the polymer gear's admissible operation. It is suggested that W_m should not exceed 20% of the total tooth width on the pitch line. Accordingly, we used W_m and from its value calculated the wear coefficient k_{Wwm} as a benchmark, when comparing all the other wear-coefficient methodologies.

With this method the equation for gear-thickness reduction W_m (eq. (4)) is rewritten to obtain the value for he wear coefficient k_{Wwm} (eq. (9)). To perform the calculations, the length of the active tooth flank ($l_{Fl} = 1.09$ mm) was used (eq. (5)).

$$k_{Wwm} = \frac{W_m \cdot b_w \cdot z \cdot l_{Fl}}{T_d \cdot 2 \cdot \pi \cdot N_L \cdot H_V} \tag{9}$$

2.5.4. Wear coefficient according to LTCA from KISSsoft

According to the VDI guidelines and its suggested wear model [21], the worn tooth's profile along the meshing profile can be predicted. However, for this wear prediction, the gear-wear coefficient must be known. However, the wear coefficient is not usually known and has to be pre-set in a polymer gear's design and gear life-time predictions. By employing the generally used commercial software KISSsoft for the gear design and the loaded-tooth contact analysis (LTCA) [34], the wear coefficient can be inversely and iteratively obtained from the actual optically measured worn-out tooth's profile. This LTCA inverse iterative methodology was used in this study to determine the LTCA wear coefficient k_{Wltca}.

After the selected number of cycles, the worn-out tooth's profile side-view is measured and plotted along the unworn profile, Fig. 7. An initial wear-coefficient value k_{Wltca} is then assumed and introduced to the LTCA in KISSsoft. The software then returns the theoretically calculated worn-out profile. This profile can be plotted along the unworn and measured worn profile, Fig. 7. The best-possible fit is then achieved by iteratively correcting the LTCA wear coefficient k_{Wltca}, until the measured worn-out tooth profile and the one generated from the KISSsoft LTCA analyses are satisfactorily close. The wear coefficient k_{Wltca} that gives this best fit is then considered as the "correct" one. In our study, this analysis is shown after 17×10^6 cycles, where the results were found to be comparable for several other methods used in this work, as explained later.

2.5.5. Wear coefficient according to VDI 2736

The VDI 2736 guidelines [21] suggest two possible wear-coefficient (k_{Wvdi}) values that are the same for any POM material, but depend on

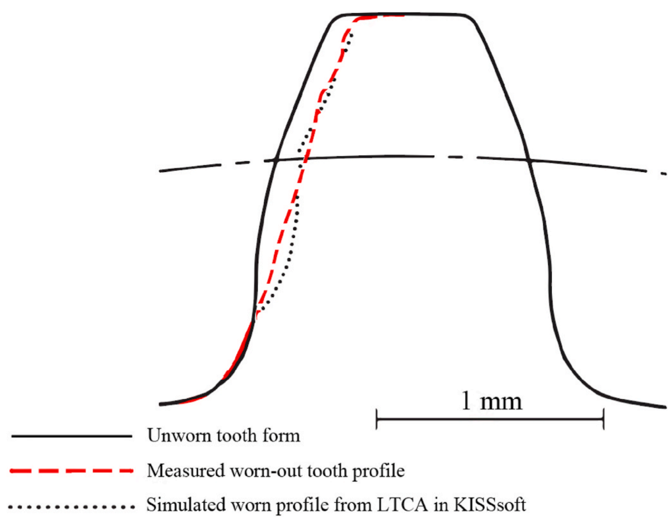

Fig. 7. Schematic presentation of the fitting of the measured, worn-out tooth profile and the simulated, worn profile from the LTCA in KISSsoft [34]. Note: the profiles are a realistic measurement and simulation of a POM gear after 17×10^6 cycles.

two roughness values of the counter steel gear. For steel gears with $R_z = 1.5$ µm the wear coefficient k_{Wvdi} for the POM gear is set as 3.40×10^{-6} mm^3/(Nm), while for the steel gear with $R_z = 0.45$ µm, the k_{Wvdi} for POM gear is set as 1.00×10^{-6} mm^3/(Nm) [21]. These values were obtained from Ref. [28], where a standard tribological pin-on-disc test was used with a stationary polymer pin sliding against a rotating steel disc. The same contact pressure, velocity, polymer material (Hostaform C9021), counter-body material (steel) and counter steel roughness were used as in this work. However, the pin's contact area was larger, i.e., 16 mm^2, compared to our 6.74 mm^2, and the sliding distance for the tests in Ref. [28] is not known.

3. Results

3.1. Wear coefficient obtained from tribological tests

During all the wear coefficient tribological experiments the steady-state polymer pin temperature was measured, and it was always about 40 °C (±2 °C). The wear coefficients obtained from all our pin-on-disc tribological tests (k_{Wpod}) using mass-loss measurements (eq. (2)) are presented in Fig. 8. The average value for the pins moulded from Delrin 500P was 1.58×10^{-4} mm^3/(Nm). For the Hostaform C0921 this value is

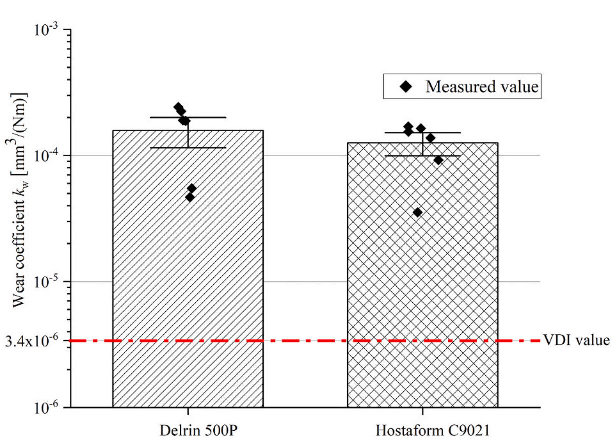

Fig. 8. Wear coefficient (k_{Wpod}) from pin-on-disc experiments with Delrin 500P and Hostaform C9021. Data points show measured raw data, column value represents the average, while scatter bars represent ± one standard deviation.

20% lower at 1.26×10^{-4} mm^3/(Nm). When the wear coefficient from the pin-on-disc tests in this study is compared with the wear coefficient in the VDI 2736 guidelines, which is suggested as 3.40×10^{-6} mm^3/(Nm) (a single value for any POM material), we see a huge discrepancy of two orders of magnitude, Fig. 8.

3.2. Gear-wear coefficient from continuous and progressive intermediate analyses

Fig. 9a presents the wear coefficients k_{Wweight} and k_{Warea} calculated for the gears running from zero up to a certain number of cycles, i.e., 10, 15, 17, 19, and 21×10^6 cycles. The wear coefficient (k_{Wweight}) calculated from the gear tests after 10×10^6 cycles using the weight-loss method (eq. (7)) was 2.00 ± 0.2×10^{-6} mm^3/(Nm) and from the worn-out-area method the wear coefficient k_{Warea} (eq. (8)) was 1.30 ± 0.3×10^{-6} mm^3/(Nm), which means a 35% difference, Fig. 9a. However, as the running cycles increase, the wear also increases (see Fig. 10) and the difference between the two methods becomes smaller. It can be seen from Fig. 9a that already after 17×10^6 cycles the difference is negligible and remains so small until the end of the test, i.e., after 21×10^6 cycles, where the difference was only 2%. Moreover, after an initial monotonic increase of the wear coefficient, for longer durations, i.e., from 17×10^6 to 21×10^6, exactly the same as when the difference between the two methods vanishes, the wear coefficient also does not change with the number of cycles.

Fig. 9b shows the wear coefficients obtained only on a certain section of the test, i.e., between two specific consecutive numbers of cycles, i.e., 0–10, 10–15, 15–17, 17–19 and 19–21 million cycles. The number of cycles between the different consecutive sections is here much smaller, and in accordance with the above finding, the differences between the mass loss and the worn-out profile area, i.e., $k_{\mathrm{Wweight,pr}}$ or the worn-out area $k_{\mathrm{Warea,pr}}$, are always significant (Fig. 9b), similar to those in the early stages of the continuous gear operation (see Fig. 9a). This shows that the measurement accuracy from the mass loss or the worn-out geometry indeed depends on the number of cycles considered and so the amount of wear in the particular section, and so is clearly improved with longer periods and large amounts of wear.

Moreover, from Fig. 9b it is also clear that there could be a huge difference in the calculated wear coefficient between consecutive periods. Namely, the mass-loss wear coefficient for the period of 10×10^6 to 15×10^6 cycles was 2.0×10^{-6} mm^3/(Nm), while it increased in the period of 15×10^6 to 17×10^6 cycles to a 7.1×10^{-6} mm^3/(Nm), which was the largest value measured in this study. This increase of 3.5 times is obviously due to a difference in the contact conditions at different lifetime periods, also clear from the gear-profile change in Fig. 10. Up to 15×10^6 cycles, there is wear mainly on the upper side of the pitch line (see Fig. 10c), where conditions were more severe with more relative sliding and a higher relative velocity. Between 15×10^6 and 17×10^6 cycles, the profile changes significantly, and there is no more involute shape on the flanks observed, Fig. 10d. Due to such a gear-profile change the meshing conditions are completely modified, which results in entirely different contacts in terms of pressures, slide-to-roll ratios and velocity. After this dramatic transition in the contact conditions, the running stabilises again and the wear coefficient becomes much lower again, comparable to earlier stages, Fig. 9b.

3.3. Wear coefficient obtained from linear wear of the gear

The wear coefficient k_{Wwm} was calculated for a worn polymer gear after 17×10^6 cycles. The number 17×10^6 cycles was selected to obtain information just after the most intensive wear phase (see Fig. 9b), and prior to any other potential non-wear changes on the gears occurring with further meshing, such as deformation and permanent tooth deflection. The measured average linear wear W_{m} (see Fig. 6) of the tooth after 17×10^6 cycles of meshing was 0.11 mm, which corresponds

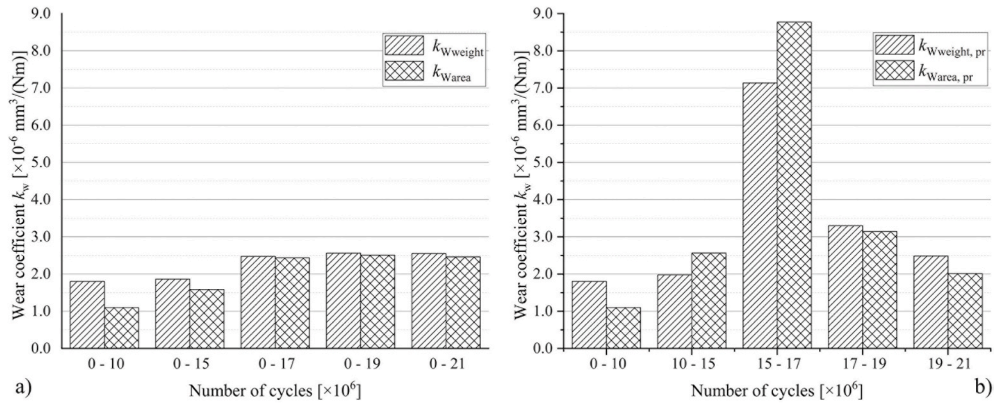

Fig. 9. Evolution of the wear coefficient from gear tests calculated with a) weight loss $k_{Wweight}$ and from worn-out area k_{Warea} and b) with progressive $k_{Wweight,pr}$ and $k_{Warea,pr}$.

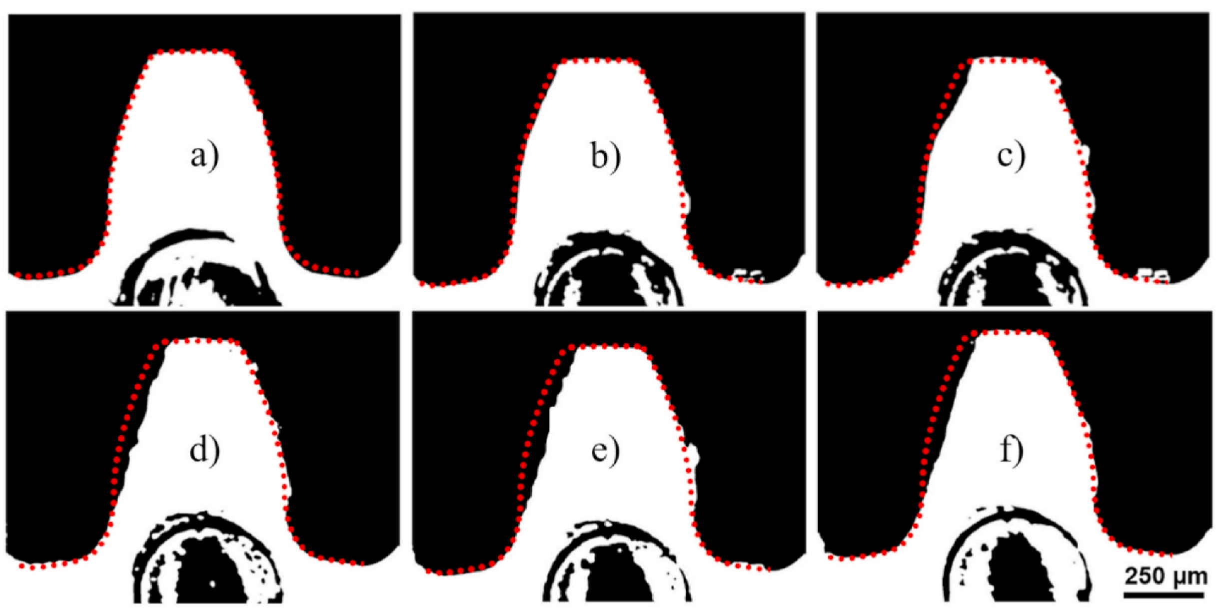

Fig. 10. A) New tooth profile and after b) 10×10^6, c) 15×10^6, d) 17×10^6, e) 19×10^6 and f) 21×10^6 cycles.

to a wear coefficient (k_{Wwm}) of 2.30×10^{-6} mm^3/(Nm) using eq. (9).

3.4. Wear coefficient obtained from gear tests using the LTCA method

The wear coefficient of k_{Wltca} from the LTCA method in the KISSsoft software [34] was calculated for worn polymer gears after 17×10^6 cycles, the same as the measurement from the gear's linear wear W_m. After some iterations, the LTCA profile correlates graphically very well with the real one, especially above the pitch line, as can be seen in Fig. 7. The wear coefficient k_{Wltca} calculated with the KISSsoft software was 2.60×10^{-6} mm^3/(Nm).

3.5. Overall comparison of the methods for wear-coefficient determination

Fig. 11 summarises all the wear coefficients calculated after different numbers of cycles and using various methods, i.e., $k_{Wweight}$ and k_{Warea} calculated by mass loss and worn-out area, as well as the iterative LCTA method k_{Wltca}, the VDI 2736 method k_{VDI}, and the pin-on-disc k_{Wpod}. In addition, it presents the "re-calculated" average linear wear W_m (see Fig. 6) that can be obtained theoretically-reversely from the wear coefficients by using eq. (4), but for a different number of cycles, as explained above and presented in Fig. 11. Moreover, a direct W_m

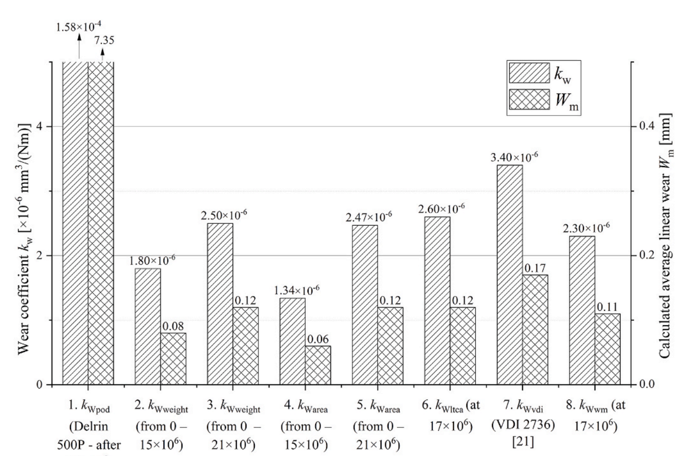

Fig. 11. Different wear coefficients k_W (primary axis) and linear wear W_m of the tooth (secondary axis) obtained from different techniques and methods used in this study.

measurement after 17×10^6 cycles is shown. Accordingly, Fig. 11 shows the variety of possible wear and wear-coefficient values we can obtain to predict the polymer gear's wear in a real application with common contact parameters and testing times in pin-on-disc and gear measurements.

As explained earlier, due to the most straightforward and accurate measurement of linear wear W_m, the wear coefficient that was calculated from this value can be considered as a benchmark for the other results. A comparison of the calculated wear coefficient (k_{Wwm}) from a direct linear wear measurement W_m (bar 8) with wear coefficients obtained using the weight-loss ($k_{Wweight}$) or worn-out-area (k_{Warea}) methods (see bars 2–5) matches very well only when the wear coefficient from the beginning of the test up to 21×10^6 cycles is compared (bars 3 and 5). The k_{Wwm} values are 7% and 8% lower than those obtained from the worn-out-area (k_{Warea}) and weight-loss methods ($k_{Wweight}$), which gave 2.47×10^{-6} mm^3/(Nm) and 2.50×10^{-6} mm^3/(Nm), respectively. Calculating the wear coefficient with data from the beginning of the test up to only the earlier stage, thus with a smaller number of cycles, i.e., up to 10×10^6 or 15×10^6, leads, remarkably, to 28–45% smaller wear coefficients (see bars 2 and 4). The importance of the long time-span considered for the calculation of the realistic gear life-time wear coefficient is therefore consistent with results in Fig. 9, which occur due to the different contact conditions, being too mild compared with the overall life time (e.g., in the early stages, see Fig. 10a–c), or rapid and sudden changes in the gear-flank wear profiles in the latter stage of operation (see Fig. 10c and d).

The wear coefficient k_{Wltca} (bar 6) calculation using the KISSsoft LTCA methodology [34], leads to a value of 2.60×10^{-6} mm^3/(Nm), which differs by only 11.5% from 2.30×10^{-6} mm^3/(Nm) of k_{Wwm} (bar 8) in a direct W_m measurement and an only 4–5% variation from long-term gear measurements using the weight loss ($k_{Wweight}$) or worn-out-area (k_{Warea}) methods. This suggests that the iterative methodology in the LTCA analysis using KISSsoft based on realistic gear testing is a very accurate route to wear prediction for use in applications. Moreover, it also has the advantage of a visual evaluation of the tooth profile's modification based on the incorporated wear model, which is additional information to the empirical wear coefficient and provides a qualitative insight into the tooth-form wear prediction, Fig. 7.

The k_{Wvdi} calculated using a fixed wear coefficient of 3.40×10^{-6} mm^3/(Nm) suggested in VDI 2736 [21] gives a 35% higher wear coefficient (bar 7) than was obtained based on actual direct, linear wear measurements (k_{Wwm}, bar 8) and more than 25% higher compared to long-term gear measurements based on the weight-loss ($k_{Wweight}$) or worn-out-area (k_{Warea}) methods, as seen in bars 3 and 5. From this it follows that the VDI method, which is based on the wear coefficient obtained in earlier pin-on-disc measurements [28], predicts a significantly higher gear wear than actually measured in the gear tests.

Similar to the VDI, the calculation of the wear coefficient obtained on the pin-on-disc apparatus k_{Wpod}, which equals 1.58×10^{-4} mm^3/(Nm) (bar 1), greatly overestimates the directly measured, linear wear coefficient k_{Wwm} of 2.30×10^{-6} mm^3/(Nm) (bar 8), by almost 70 times, and the long-term gear measurements based on the weight-loss ($k_{Wweight}$) or worn-out-area (k_{Warea}) methods, by about 63 times, which is an enormous difference. Accordingly, both methods based on the pin-on-disc method (k_{Wvdi} from VDI and k_{Wpod} directly from our measurements) provide wear coefficients that are too large compared to the real gear tests and they also vary between each other by about two orders of magnitude.

It is, however, interesting that the pin-on-disc wear coefficient, which is calculated after a much smaller number of cycles (8.5×10^5) compared to the gear tests, gives a higher wear coefficient than in the gear tests, which is just the contrary to the gear test methodology, where a small number of cycles led to lower wear coefficients. Such results from the pin-on-disc test might be related to many differences in the contact conditions between the pin-on-disc and the gears, and also to the "accelerated nature" of the wear in the pin-on-disc tests.

3.6. SEM analyses of the surfaces

SEM analyses of polymer pins after the POD testing are shown in Fig. 12. The wear damage on both POM materials is similar and is very severe. It shows the mechanically-based wear with evident scratching marks from sliding wear. Furthermore, small worn-out particles are found to adhere to the surface of polymer pins.

The gear-flank overview image of polymer gear surfaces after meshing for 10 million of cycles is presented in Fig. 13 with the characteristic meshing points marked from A to E. The meshing points are located based on the theoretical calculation of the meshing contact. In the area of a single tooth contact in the addendum region (B–C), the features are very different compared to the sliding marks present on polymer pins (Fig. 12). The close-up image of that region (Fig. 14 a) reveals characteristic features, namely the ridges, which are perpendicular to the direction of sliding and are smeared in the sliding direction. These features are approximately 120 μm and 3–5 μm thick, with an almost equal gap between all of them in the range of 20–25 μm. Similar features were also found in the area C–D. However, moving further away from the pitch line (near point A, shown in Fig. 14 b), the mechanism has changed and reveals sliding scratches mainly, oriented in the sliding direction. These scratches were also found in the opposite site of the gear area B-A, namely in the area D–E. These evidences confirm lower contact pressure in these regions, but pronounced relative sliding velocities. Moreover, in the D-E region, some wear debris were observed, which are pushed-out from the inner parts of the contact due to enhanced sliding velocities. In the pitch line – point C (Fig. 15), where polymer material is exposed to pure rolling motion, the wear features differ significantly compared to other flank areas. The pitch line lays between smearing features from addendum and dedendum region and has a width of approximately 70 μm. The worn surface is covered with a smooth and stressed surface layer that contain small holes with a diameter of 1–6 μm in this top surface layer.

With the increase of number of cycles, from 10 million to 17 million, the polymer gear-flank exhibits pronounced adhesion of wear particles (Fig. 16), where the wear particles cover almost entire single contact area, from point B to point D. However, this adhesive layer of wear particles (Fig. 17a) covers wear features in the layer below, which are very similar compared to gear-flank after 10 million cycles. In the double contact area region, from A–B and D–E, there is almost no adhered material, and the sliding wear marks are present (Fig. 17b), similarly to gear meshing after 10 million cycles.

4. Discussion

A life-time prediction is one of the most critical and necessary analyses in the design of polymer gears for use in applications. The life-time prediction relies on the wear coefficient that can be obtained in many different ways. In this study we have evaluated several possible methods to determine the wear coefficient, both in a conventional tribological pin-on-disc type device, as well as in real-scale gear tests obtained with various methodologies. The results are discussed below.

4.1. Gear life-time prediction based on pin-on-disc measurements

The wear coefficients k_{Wpod} obtained from the pin-on-disc apparatus for the two POM materials, namely Delrin 500P (1.58×10^{-4} mm^3/(Nm)) and Hostaform C9021 (1.26×10^{-4} mm^3/(Nm), do not correlate well with the values mentioned in the VDI 2736, k_{Wvdi} (3.4×10^{-6} mm^3/(Nm)) that are based on literature pin-on-disc data [28]. The difference is as large as two orders of magnitude. This is despite our pin-on-disc tests being performed under very closely controlled conditions, like those in Ref. [28], i.e., the same contact pressure, velocity, counter body material and roughness. However, the difference was the pin size and

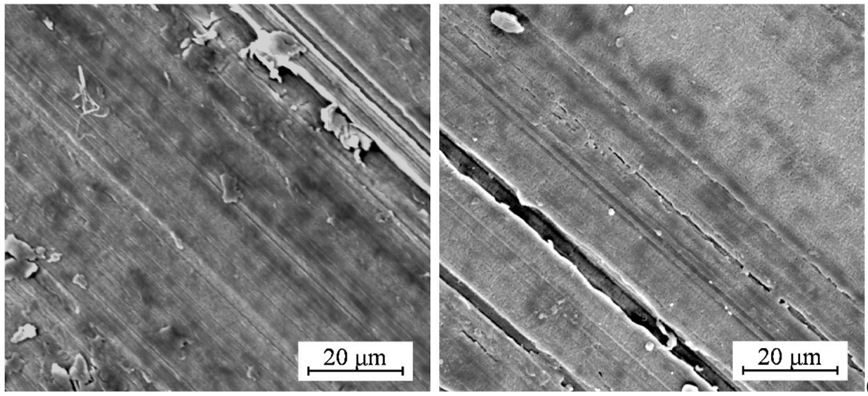

Fig. 12. SEM images of pin samples from pin-on-disk experiments after 8.5×10^5 sliding cycles, moulded from (a) Delrin 500P and (b) Hostaform C9021.

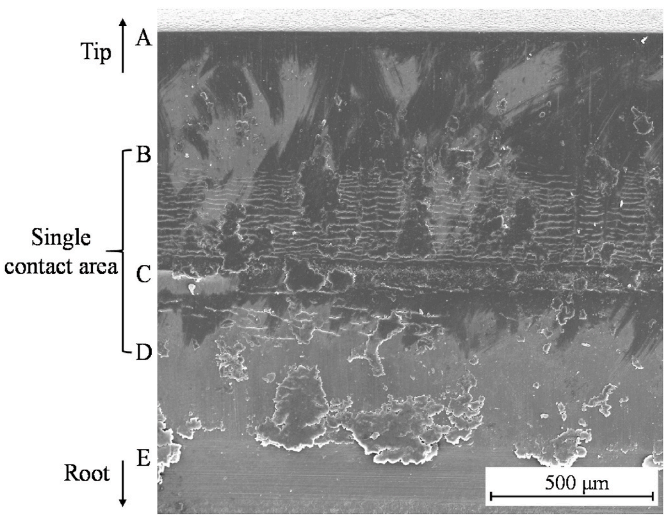

Fig. 13. SEM overview of polymer gear after 10 million of cycles.

the material that was in Ref. [28] the copolymer (Hostaform C9021) only, whereas we used two POM materials, one of those was Delrin 500P, which is a homopolymer. However, our tests with the Hostaform C9021 (the same material as in Ref. [28]) are also close to Delrin 500P data, i.e., about 20% of difference (see Fig. 8), but this is still almost as far as those in Ref. [28], with an about two orders of magnitude variation. Therefore, without commenting any further on the experimental details, it is clear that despite the similarity in the test conditions and the materials, the pin-on-disc results from various labs or testing devices can lead to significant differences.

However, since the wear coefficient is used for the prediction of the wear of polymer gears in applications, it is obviously an important value, so we have further investigated similar pin-on-disc studies. Several studies [23,27,28,37,38] evaluated the tribological properties of POM materials under conditions that always, at least partially, vary, also from our research or the details in Ref. [28]. Nonetheless, they were overviewed, and their wear-coefficient values are summarised in Fig. 18. From this it is clear that the pin-on-disc wear coefficients for the POM materials in contact with steel vary greatly, by more than two orders of magnitude, and are obviously influenced by many different contact conditions and operating parameters, as well as the test-rig designs and configurations.

When compared to actual gear performance, gear kinematics introduces a new level of complexity, since the load and the slide-to-roll ratio vary along the path of the contact, while the contact path is changing with the wear and the deformation. All these bring additional differences and uncertainties. It is also a common observation that any pin-on-disc wear coefficient greatly overestimates the wear of the actual gears. The VDI model (k_{Wvdi}) with a single POM value [21] results in a surprisingly small variation compared to the actual gears, with only a 35% difference from the direct linear wear coefficient k_{Wwm}. This is because the data used in the VDI are at the lower limits of all the available results, Fig. 18. The difference in our case, where our wear coefficient k_{Wpod} is located at the upper end of data (Fig. 18), is, however, 70-times too high compared to the direct linear wear coefficient k_{Wwm}. Unfortunately, it is impossible to claim that any of these two results are more comparable to the results for actual gears. Therefore, the wear coefficient obtained from the conventional pin-on-disc apparatus represents a too large variation and uncertainty to be used in predictions for real gear applications.

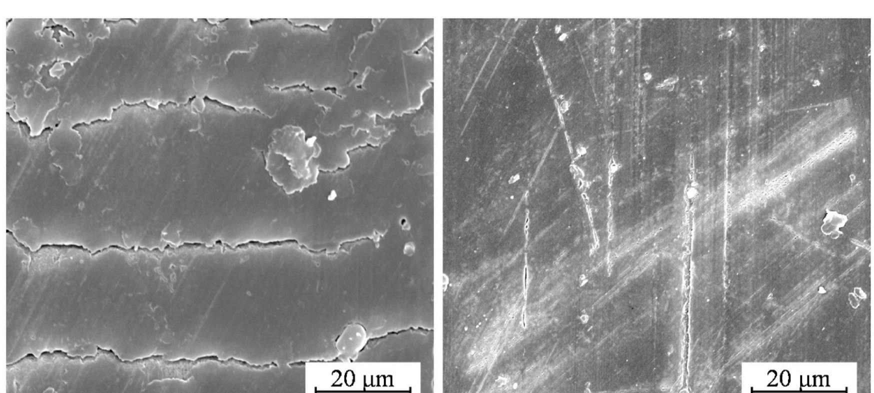

Fig. 14. Close up SEM images of polymer gear after 10 million of cycles at (a) single tooth contact area near point B and (b) double tooth contact area near point A.

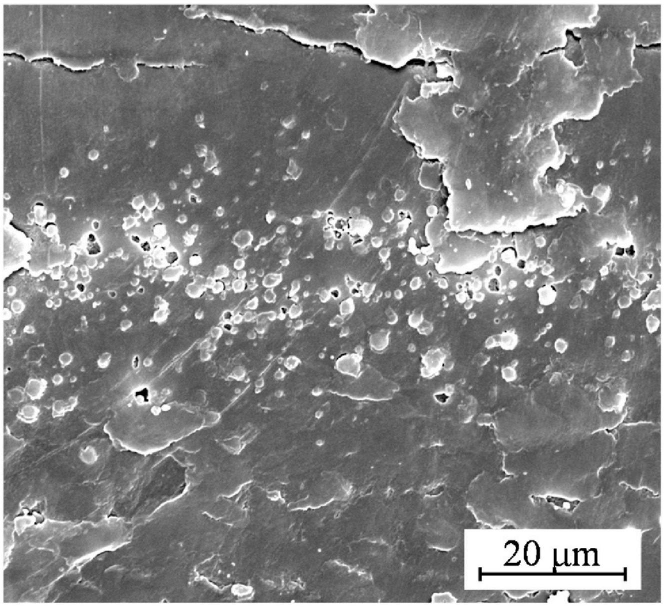

Fig. 15. SEM image of the pitch line after 10 million of cycles.

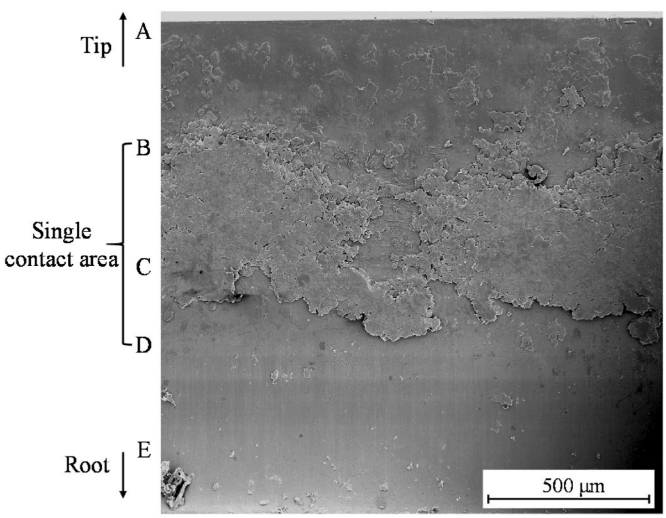

Fig. 16. SEM overview of polymer gear after 17 million of cycles.

4.2. Gear life-time prediction based on real gear tests

While the wear-coefficient calculation based on pin-on-disc measurements is trivial once the wear data are obtained, the wear-coefficient calculation from a real-scale gear is more complex and with more possible variations.

From our results it follows that both the weight-loss and tooth's worn-out-area cross-section evaluation are applicable for a determination of the gear's wear coefficient; however, it is obvious that the amount of wear, and thus the number of cycles, should be as long as possible, but, of course, under conditions of efficient operation. This is especially true for the tooth's worn-out-area cross-section method, which has an experimental accuracy limitation, as evidenced from Figs. 9 and 10. For the weight-loss method, a very accurate balance (e.g., 10^{-5} g) provides the appropriate accuracy even for small amounts of wear, Fig. 9a. In contrast, the optical method at 25 × magnification where the whole gear profile is seen, it is not accurate enough for small geometrical changes in the profile. This leads to significant errors in the wear coefficient for small amounts of wear. This is very pronounced in the earlier stages of gear running, i.e., up to 10 million cycles, where the difference between the two methods is approximately 40%, and even at 15 million cycles, the difference is still 20%, see Fig. 9a. However, at 17 million cycles and later, up to 21 million cycles, the differences between the two methods become negligible, at around 2%, Fig. 9a.

It turns out that when the wear is large enough so that the weight loss and tooth's worn-out-area cross-section methods give almost the same

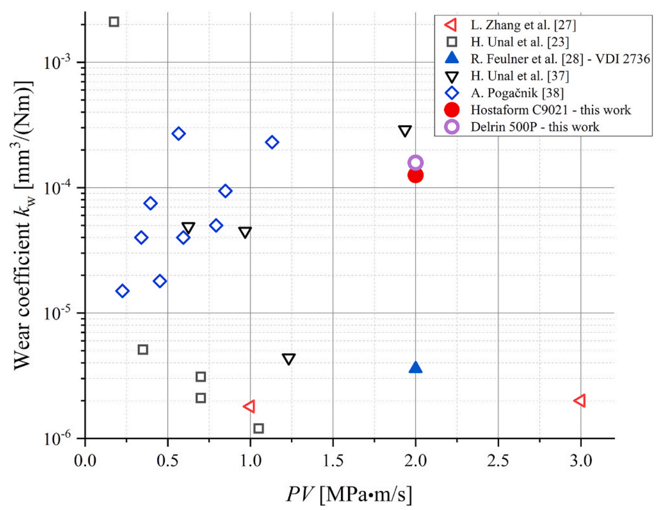

Fig. 18. Comparison of the wear coefficients for POM sliding against steel from various studies [23,27,28,37,38] and this work.

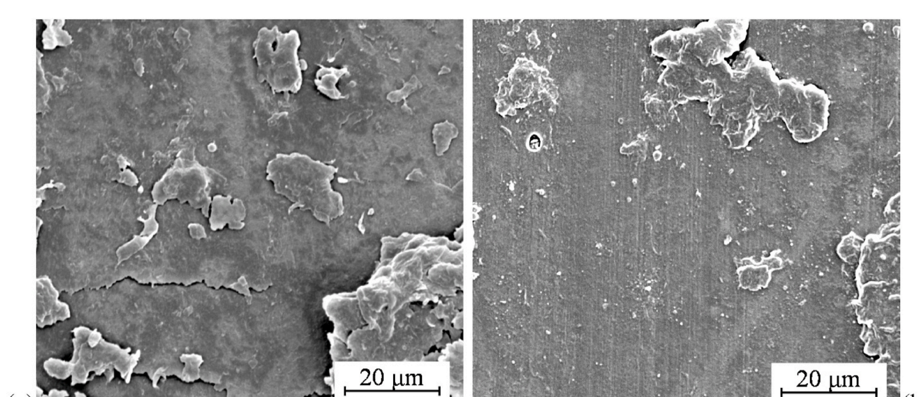

Fig. 17. Close up SEM images of polymer gear after 17 million of cycles at (a) single tooth contact area near point B and (b) double tooth contact area near point A.

results, also the calculated wear coefficients (k_{Wweight}, k_{Warea}) become almost the same as the wear coefficient calculated from a direct linear wear measurement k_{Wwm}, see Fig. 11, bars 3, 5 and 8. In our case this occurred after 17 million cycles (Figs. 9a and 11), where the difference between the three methods was only 8%. Accordingly, we propose that the relevant wear coefficient for a gear's life-time prediction can be calculated only after a large enough number of cycles, which can be assumed as a "rule of thumb" once the weighting, geometrical and direct linear tooth-thinning wear methods provide about the same results.

The progressive evaluation of the gear-wear coefficient after shorter periods is not appropriate for an evaluation because in the early stages of operation it leads to an under-estimation and lower wear coefficients, i.e., even 45% lower values were obtained after 10 million cycles, Fig. 9a. On the other hand, sudden and abrupt changes in the wear and tooth profile at later stages can lead to extremely high wear (Fig. 10c and d) and changes to the contact conditions where the wear coefficient can change dramatically; in our work by as much as 3.5 times, Fig. 9b. These are temporary changes, if the gears are not damaged to the level of malfunction. However, despite almost normal subsequent operation with high efficiency (95% or more), the wear predictions based on such a short running period could be totally inappropriate.

The VDI 2736 method [21] that suggest a fixed wear-coefficient value for all kinds of POM materials, which is an important drawback requiring improvements in methodology also due to novel materials' development. Moreover, this value that varies only due to the steel counter material's roughness, is obtained from a single pin-on-disc study [28], while Fig. 18 clearly show a large variation in the pin-on-disc data from many studies. While it is hard to establish which pin-on-disc results are closer to the gear application, the differences can be as high as two orders of magnitude (Fig. 11), i.e., far too much for a reliable gear life-time prediction. However, as mentioned earlier, the VDI wear coefficient k_{Wvdi} was only 35% higher than the direct linear gear-thinning wear coefficient k_{Wwm}. Considering the obvious differences in the contact conditions between the various pin-on-disc studies compared to the gear tests and their two-orders-of-magnitude variations (Fig. 18), this relatively small difference (but still too large for a realistic gear-wear evaluation) is more a coincidence of the pin-on-disc study selection, than a consequence of a validated methodology.

Among the two pre-defined methodologies, namely KISSsoft [34] and VDI [21], the KISSsoft gives a better and more realistic wear-coefficient estimation for a gear's life-time prediction. As seen from Fig. 11, it suggests a wear coefficient of 2.6×10^{-6} mm^3/(Nm), which is just 11.5% higher than that obtained from a direct W_m measurement, which will however depend on the number and the accuracy of the iterations. The reason for a good estimation is thus the iterative nature of the method, which considers actual gear data and an adaption to specific contact conditions and wear mechanisms while directly simulating the gear contacts. As mentioned, a visual observation of the deviations from real gear-tooth profile (Fig. 7) makes the method even more attractive and accurate.

4.3. Comparison of wear mechanisms

The conducted SEM analysis of both pin and gear surfaces clearly shows significant differences in the wear mechanisms. Polymer pins exhibit abrasive sliding wear across the whole surface, while gears have three distinctive regions. The first region is on double tooth contact area of gear flank (A–B and D–E), where abrasive sliding wear was observed. In this region, high slide-to-roll ratio is present, with predominant sliding. In the area where single tooth contact occurs (B–C and C–D), the deformation ridges were observed due to high contact pressures and lower slide-to-roll ratio. This was also observed in earlier findings [39]. The third region in the pitch-line (position C), where together with high pressures, only rolling motion was present, causing a well-defined deformation layer with many pit-shaped wear features. The observed wear mechanism on pin-on-disk pins was thus sliding-wear scratches, which significantly differs to those observed on polymer gear flanks, where three different regions were observed due to a difference in contact conditions and changeable combination of sliding velocity, contact pressure and slide-to-roll ratio. This differences in wear mechanisms between model pin-on-disc test and gear test leads to a very difficult comparison of the polymer wear behaviour in these two tests. We have discussed this phenomena in a greater detail in another recent publication [40].

5. Conclusions

1. The pin-on-disc wear coefficient from different studies varies greatly (up to two orders of magnitude) even under seemingly the same or similar tribological conditions. The pin-on-disc always overestimates the POM gear wear; from 25 to 35% in the VDI method, to even 63–70 times in our pin-on-disc study.
2. The wear coefficient obtained with conventional pin-on-disc devices represents too large a variation and uncertainty to be used in predictions of real gear applications.
3. The observed wear mechanism on pin-on-disk pins was sliding-wear scratches, which significantly differs to those observed on polymer gear flanks, where three different regions were observed due to a difference in contact conditions and changeable combination of sliding velocity, contact pressure and slide-to-roll ratio. This differences in wear mechanisms between model pin-on-disc test and gear test leads to very difficult comparison of the polymer wear behaviour in these two tests.
4. The weight-loss and the optical tooth's worn-out-area cross-section methods are both appropriate for determining the gear-wear coefficient when the worn-out profile is large enough, i.e., after a large number of cycles, so that the optical worn-out area can be measured with sufficient accuracy.
5. The wear coefficient for POM gears in this study could be established after 17×10^6 cycles.
6. As a "rule of thumb", the appropriate number of cycles for POM gears to obtain a valid wear coefficient can be the number of cycles when the direct linear wear measurement W_m gave "the same" wear coefficient as the weight-loss and the optical tooth worn-out-area cross-section methods.
7. The progressive evaluation of a polymer gear's wear coefficient for short running periods can lead to an underestimation of the wear at earlier stages of the operation and an overestimation due to abrupt wear with large changes to the tooth profile and the contact conditions.
8. The VDI 2736 method, which is based on a single pin-on-disc study with a single POM material, suggests much poorer applicability than the results from the actual gear tests.
9. The iterative methodology in the LTCA analysis with KISSsoft based on realistic gear testing gives a very accurate wear prediction, with about 4–10% of variation. However, the accuracy might improve with iterations and accurate fitting. Its advantage could also lie in a visual evaluation, and a qualitative insight into the tooth-form wear-model prediction.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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