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Test Rig for the In Situ Measurement of the Deformation Characteristics of Polymer Gears

This paper presents a new method for in situ deformation and wear measurement of polymer gears under operating loads. This new approach is based on the highly accurate measurement and tracking of time differences in the index pulses of encoders on the driving and driven side of a gear test rig. The change in these time differences over the test duration correlates with the tooth deformation. Through a specific combination of load cases, elastic, plastic and wear deformation components can be separated. This work also shows the potential of the method to investigate the influence of geometric and structural deviations on the resulting operating properties of polymer gears. The influence of process-related structural deviations or geometric deviations from the ideal shape on operating properties such as vibrations, wear or achievable service life can be investigated in much greater detail. Thus, on the basis of in situ measurements, a deeper understanding of influences on the operating properties of plastic gears can be gained.

Prüfstand zur In Situ-Messung der Verformungs-eigenschaften von Polymerzahnrädern

In diesem Beitrag wird eine neue Methode zur in situ-Verformungs- und Verschleißmessung von Polymerzahnrädern unter Betriebslast vorgestellt. Dieser neue Ansatz basiert auf der hochgenauen Messung und Verfolgung von Zeitdifferenzen in den Indeximpulsen von Encodern auf der An- und Abtriebsseite eines Zahnradprüfstandes. Die Veränderung dieser Zeitdifferenzen über die Prüfdauer korreliert mit der Zahnverformung. Durch eine spezifische Kombination von Lastfällen können elastische, plastische und Verschleißverformungsanteile getrennt werden. Diese Arbeit zeigt außerdem das Potenzial der Methode, den Einfluss von geometrischen und strukturellen Abweichungen auf die resultierenden Betriebseigenschaften von Polymerzahnrädern zu untersuchen. Der Einfluss von prozessbedingten Strukturabweichungen oder von geometrischen Abweichungen von der Idealform auf Betriebseigenschaften wie Schwingungen, Verschleiß oder erreichbare Lebensdauer kann wesentlich detaillierter untersucht werden. So kann auf der Grundlage von in situ-Messungen ein tieferes Verständnis von Einflüssen auf die Betriebseigenschaften von Kunststoffzahnrädern gewonnen werden.

In Situ Test Rig for the Measurement of the Operating Characteristics of Polymer Gears

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1. INTRODUCTION

Polymer parts are increasingly important for tribological applications because of their advantageous characteristics such as dry run capability and economic mass production by injection moulding [1]. However, parts made of plastic show significantly lower wear resistance in comparison to metal parts [1]. The wear behaviour of plastic and metal pairings typically consists of a run-in stage, characterised by higher non-stationary wear rates, and a following steady state at lower constant wear rates [2]. This is due to the alignment between the contact surfaces, the formation of wear reducing interlayers composed of wear particles and the transition from the edge area to the more resistant core area [3, 4]. A variety of geometric and microstructural factors can influence the run-in and the steady state; among others surface roughness [5] and structure [4, 6] of the metal pinion, degree of crystallinity and spherulite size [3], or tooth modifications [7]. Consequently, the in situ wear measurement is of interest as it permits the wear determination over time and therefore a distinction between the run-in and the steady stage. This knowledge allows for a better estimation of necessary wear addendums [8]. Therefore, a new approach for the in situ wear measurement of plastic gears paired with steel pinions is presented and applied to determine geometric and structural influences on the operating properties of a polyamide-steel gear set.

2. FUNDAMENTALS

2.1. Current methods and designs for in situ gear testing

Various in situ test methods for gears are described in the literature. There are two main approaches: test rigs for wear and lifetime measurement, and test stands for condition monitoring. The goal of condition monitoring is to decrease maintenance and repair cost by monitoring the health of a critical system [9], e.g. a gear drive. In condition monitoring test rigs sensor data, like acoustic or mechanical vibrations are recorded and the corresponding gears checked for signs of failure. Changes in these signals are correlated to certain types of failures, be it pitting, cracking of teeth or flanks [10], or excessive wear [11]. With corresponding models of the interaction of failure causes and vibrational profiles, inexpensive sensor systems to monitor a critical gear drives health can be built, using easily available vibrational data [12]. However, the underlying process for the gear defect, for example wear, cannot be quantified or analysed with these systems. For this reason, a variety of wear and lifetime test rigs has been designed. Almost all types of gear test rigs can determine gear lifetime trivially. The measurement of gear temperature by either thermographic camera or thermocouples is also widely established. However, there are only three main approaches to characterise the wear during operation: Gravimetric or tactile measurement or analysis of a system reaction.

A relatively easy way is to measure the weight loss of the system [13]. This method however, requires a substantial amount of wear leaving the system and does not allow the direct quantification of the reduction in tooth thickness. One tactile system uses a profile measurement to directly quantify gear wear [14]. This and similar systems using tactile measurements, however, require the gears to be big enough to be measured and the test rig to be stopped during the test run. Another approach is to measure a system reaction to the wear. For example, a test rig design be Hooke et al. [15] uses a pivoting gear block with a loading arm in contact with a fixed gear block. The rotation of the loading arm corresponds to the wear in the system. Due to its construction, this test rig might be highly susceptible to vibrations. To mitigate all these disadvantages, a new test rig for the in situ analysis of the deformation of gear sets under operating conditions was developed and validated at the Institute of Polymer Technology, LKT, which is now presented and employed in this work.

2.2. Influence of form and structural deviations

In general, geometric deviations from the ideal form lower product quality and function [16]. This is especially true for gears. Different tooth profiles [17], tooth geometries [18] and form deviations [19] can significantly influence the operating properties of polymer gears, like efficiency, transmission error, operating temperature, achievable lifetime or wear. Additionally, the structural properties, i.e. the makeup of the crystalline microstructure, can also significantly influence the operating properties of polymer machine elements [3]. With an increase in crystallinity and spherulite size the material strength [20], stiffness [21] and wear resistance [3] increase, whereas ductility and damping are decreased [21]. Using this knowledge to generate a favourable microstructure in polymer gears can lead to significantly improved lifetimes. However, little to no research has been done regarding the interaction of form and structural deviations. Most literature sources analyse either the one or the other. As of now, it is not known, whether the beneficial effects of a favourable morphology can outweigh the drawbacks of higher form deviations, and vice versa. For this reason, this work will present findings showing the influence of both, form and structural deviations, and their interaction on the operating properties of a polyamide gear set using a new in situ gear test rig.

3. MATERIALS AND METHODS

3.1. Materials and specimens

A wire cut steel pinion and polyamide 66 (PA 66) gears with a geometry according to VDI 2736, module 1 mm, were chosen for this research to represent a typical gear size in actuating drives, see Table 1. The chosen material, Ultramid A3K, a PA 66 by BASF SE, Ludwigshafen, Germany, is often used in gear applications due to its high wear resistance.

DIN 867	**Pinion**	**Gear**
Material	100Cr6	PA 66 Ultramid A3K
Module	1 mm	
Pressure angle	20°	
Number of teeth	17	39
Gear width	8 mm	6 mm
Profile shift	0.2045 mm	-0.3135 mm

Table 1: Technical specifications of the gear set

To analyse the influence of form deviations on the operating properties of the gear set, two different mould contours were manufactured, based on the same specifications. One contour was generated using the software KissSoft 2019 by KissSoft AG, Bubikon, Switzerland, the other was designed by Frenco GmbH, Altdorf, Germany, using their Spline Calculator.

The flank lines generated with the two programmes differ in the number of exactly calculated vertices of the contour and the interpolation between them. The KissSoft contour approximates the involute contour between the geometrically exactly calculated vertices with line segments, whereas Frenco uses splines. Due to these approximations, there will be small deviations from the geometrically exact involute curve in the low micrometre to nanometre range.

In addition to these approximation errors, inaccuracies in the generation of the overall gear contour from the individual flank lines can lead to variations for example in the tooth thickness over the circumference of the gear. Both mould inserts were manufactured on the same wire erosion machine with the same processing settings. As will be shown, these differently generated contours lead to differences in transmission error and vibration. The same wire cut pinion made out of 100Cr6 steel, flanks hardened to HRC 55 and ground to Rz 1 µm, was used for all tests.

The polymer gears were produced by injection moulding using an Arburg 370U-700-30-30 by Arburg GmbH & Co. KG, Loßburg, Germany. To determine the effect of structural deviations, two separate mould temperatures, T_{Mould} = 60 °C and 100 °C, were chosen to generate different morphological properties. Table 2 shows the main processing parameters.

Processing Parameter	**Parameter setting**
Screw diameter	18 mm
Mass temperature	290 °C
Mould temperature	60 °C and 100 °C
Injection / Packing / Cooling / Cycle time	1.5 / 6 / 20 / 35 s
Packing pressure	700 bars

Table 2: Processing parameters for the polyamide (PA66) gears

3.2. Functional principle of the new in situ gear test rig

The experimental setup used for the in situ deformation and wear measurement is shown in figure 1. Generally, the gear test rig consists of a DC motor, type DSM150N by Baumüller, Nuremberg, Germany, that drives the steel pinion on the input side. A hysteresis brake, type CHB-12 by Magtrol, Rossens, Switzerland, generates a braking torque on the output shaft, on which the plastic gear is mounted.

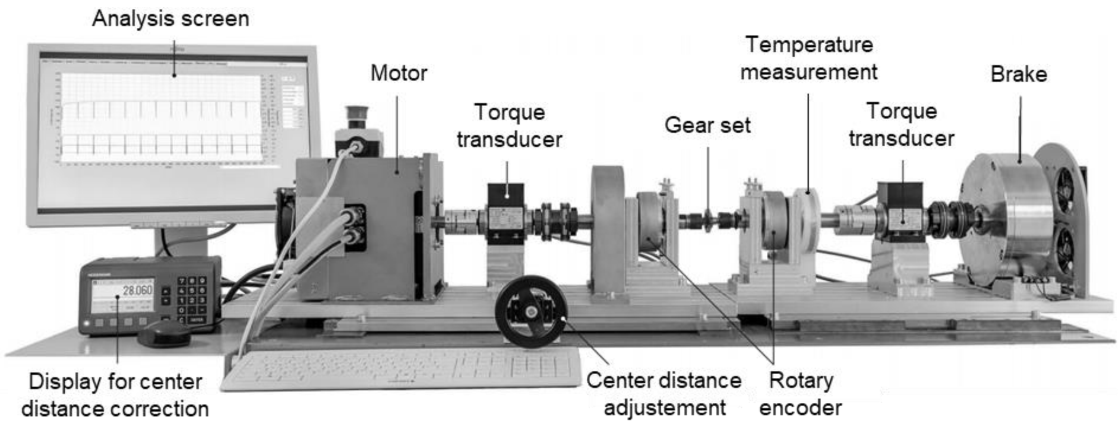

Figure 1: Main components of the in situ gear test rig

To analyse the gear temperature a thermocouple, Type K, can be used. Its signals are sent to the data logging PC via telemetry, type TEL1-PCM-IND by Kraus Messtechnik GmbH, Otterfing, Germany. The high resolution and accuracy of the torque transducers, type TMB307, by Magtrol, Rossens, Switzerland allow to closely examine the variations in operating torque of the gear set and their frequency spectrum.

The essential components of the measurement technology of the newly developed gear test rig are two rotary encoders, type A020 by Fritz Kübler GmbH, Villingen-Schwenningen, Germany, on the input and the output shaft. Deformation of the polymer gear's teeth, for example due to elasticity, thermal expansion, wear or creep, lead to a change in the relative angular displacement of the input and output shaft over time, see figure 2.

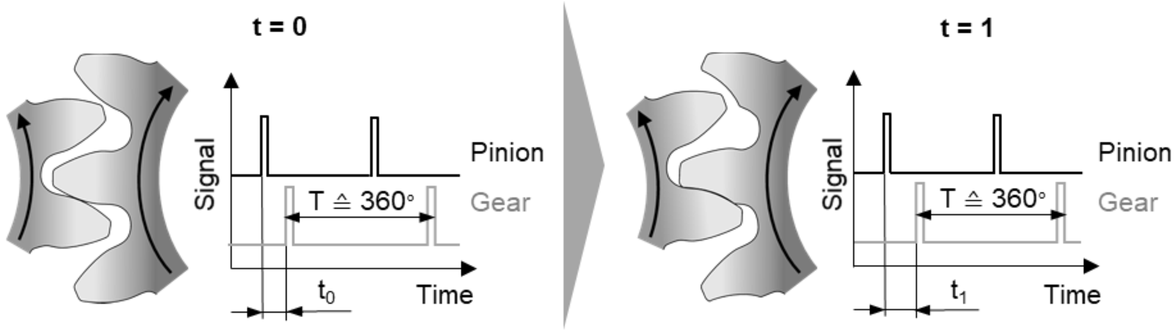

Figure 2: Tooth deformation induced angular displacement between input and output shaft

The encoders are used to calculate the angular displacement between the two shafts based on the increase in time differences between the index pulses of the encoders. Initially there is a time difference t_0 between the index pulses on the input and the output shaft. Due to deformation, this time difference increases during the test run to t_1. The change in time differences, $\Delta t = t_1 - t_0$, represents the increase in angular displacement. Using the fact that the time between two index pulses of the output shaft, T, is needed for one rotation, the increase in angular displacement can be calculated according to the following formula:

$$\Delta\varphi = \frac{\Delta t}{T} \cdot 360° \quad (1)$$

As a result of the measuring method, the calculated angular displacement is an integral measure of the tooth deformation. However, for the design of gears the local reduction in tooth thickness over the lifetime is the relevant design criterion to avoid premature failure due to wear. Thus, the angular displacement should be converted in an - albeit mean - change of local tooth thickness. Assuming a uniform and sufficiently small deformation over the tooth height, the change in tooth thickness Δs, at the diameter of interest, usually the pitch circle d, can be determined from the change in angular displacement Δφ, using equation (2):

$$\Delta s = \Delta\varphi \cdot \frac{\pi \cdot d}{360°} \quad (2)$$

In order to ensure comparability to existing measuring methods for tooth wear at the Institute of Polymer Technology (LKT), the measuring diameter dMWk = 38.146 mm according to DIN 3977 is used for calculating the tooth deformation. To achieve a sufficiently high accuracy, the encoders' signals are sampled at a frequency of 80 MHz using a data acquisition module, type NI 9401 by NI, Austin, Texas, USA. Thus, the experimental setup detects the time-dependent angular displacements with an accuracy of approximately ±0.0003° at 1 000 min-1. This corresponds to a measurement error in tooth deformation of ±0.1 µm given a plastic wheel with a 39 mm pitch diameter. Disturbing influences, like vibrations or speed fluctuations, could increase the measurement error; nevertheless, the system is still very accurate compared to other concepts and reaches the accuracy of common displacement transducers.

In order to distinguish between the wear and plastic tooth deformation on the one hand and the elastic deformation of the plastic tooth on the other hand, a load spectrum is applied. The test rig automatically switches between a high or “testing” torque and a low or “measuring” torque after a set amount of load cycles. During the “high” cycles, the gear is subjected to the testing conditions. The load in the “low” cycles is reduced as far as possible without risking smooth meshing. The reduction in torque leads to a decrease in elastic deformation. By setting the deformation measurement of the first “low” cycle to zero, the following measure-ments in the “low” state only include the lasting deformations, creep and wear, see figure 3. As a result of this method, the experimental setup does not have to be disassembled in order to determine the gear wear and creep. However, as of today, the test rig does not analyse thermal expansion, creep and wear separately. Further research into load spectrums and operating modes to separate these effects is currently being done at the LKT.

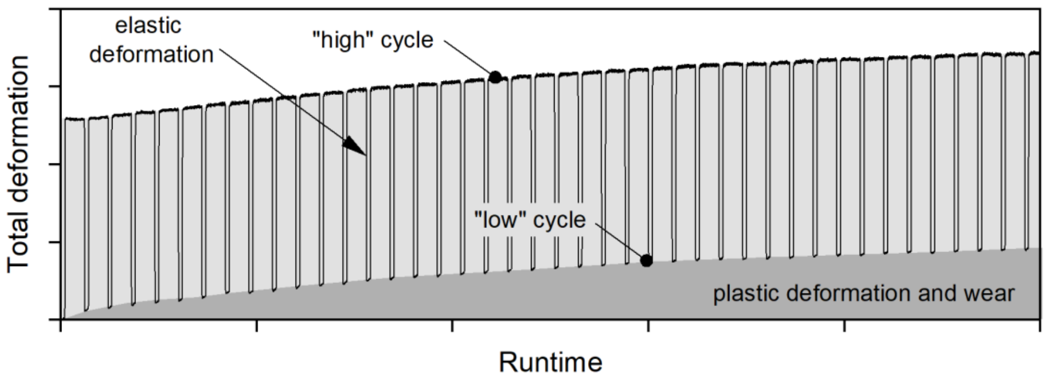

Figure 3: Distinction between elastic and plastic deformation and wear

3.3. Validation of the new wear testing method

The results of the new test rig were validated on the gear tests of the spline contour gears manufactured at 60 °C mould temperature. The gear test was performed on three specimens at an output speed of 1 000 min^{-1} for a duration of 10^6 load cycles, i.e. rotations. This results in a duration of 0.06 s per rotation on the output side, and about 1.0 s until the same teeth of pinion and gear mesh again, which is hereafter referred to as one meshing cycle. During the test the output torque was switched between the “high” setting of 1.0 Nm for 200 meshing cycles (= 3 333 load cycles), and 0.3 Nm for 40 “low” meshing cycles (= 666 load cycles). The deformation data was separated by load setting and compared to results of ex situ tests.

The design of this test collective is of particular importance. Ideally, the test rig is switched to the low cycles only for time it takes to collect sufficient data to evaluate the elastic deformation component. However, too fast changes between both states, e.g. by a very small number of low cycles, can lead to additional vibration whereas too slow changeover leads to the formation of two distinct deformation and temperature states. The combination of high and low cycles used in this work was chosen based on preliminary investigations to achieve a balanced measurement process. In further work, the influence of the test conditions on the measurement results will be investigated in more detail to develop an optimised testing strategy.

3.3.1 Thermal expansion

Compared to metals, polymers have a high thermal expansion. Due to frictional heating, the gear warms up during operation which reduces the measured deformation. The thermal expansion of the tooth Δs can be estimated with the following approximation.

$$\Delta s = \alpha \cdot s_0 \cdot \Delta T \qquad (3)$$

The coefficient of linear expansion for Ultramid A3K is given as $\alpha = 98 \cdot 10^{-6}$/K by BASF SE [22]. The tooth thickness in the pitch circle was measured as 1.438 mm. Assuming symmetrical expansion, half of the tooth thickness, 0.719 mm, can be used as the initial length s_0. A suitable reference temperature is needed to calculate the thermal expansion. Usually, an estimation of the spatial distribution of the temperatures, for example based on the approximation of the complex heat generation and transfer mechanisms using numerical and/or analytical methods, would enable a sound evaluation of the thermal expansion. However, this would have exceeded the scope of this work. Especially since the effects due to thermal expansion were expected to be rather small.

Previous research [6] on similar PA66-steel gear sets at the same load collective generated flank temperatures of 50 - 60 °C. Furthermore, literature, among others [23], has shown the tooth core temperature can be a good estimation for the average temperature of a tooth. Thus, the tooth core temperature was used to estimate thermal expansion. During the test, the tooth core temperature rose from 23°C to about 40 °C, giving a ΔT of about 17 K. Based on this data the thermal expansion Δs was estimated to be 1.2 µm.

A worst case calculation using $\alpha = 100 \cdot 10^{-6}$/K, the full tooth thickness of s_0 = 1.5 mm and ΔT = 20 K suggests an expansion of only 3 µm. Since the estimated thermal expansion is relatively small compared to the other measured deformations and the temperatures are relatively stable during the test runs, this factor was regarded as a negligible, reversible constant for all tests.

3.3.2 Elastic and creep deformation

A hydraulic pulsator test rig built by LUVRA Hydraulik und Regeltechnik GmbH, Nuremberg, Germany, was used to determine quasi-static tooth deformation in dependence of the applied force. Three representative teeth on three specimens of the spline contour manufactured at T_{Mould} = 60 °C were tested. The force was measured using a load cell, type U2B, the displacement was recorded with a displacement transducer, type W1T3, both by Hottinger Baldwin Messtechnik GmbH, Darmstadt, Germany. A single metal tooth was used to apply an increasing force on the polymer gear, which was generated by a hydraulic cylinder of the type HS5132 by LUVRA Hydraulik und Regeltechnik GmbH.

In the gear test the load on a single tooth is applied in about 1,54 milliseconds, equalling a rate of about 32 500 N/s. Limitations in the measuring equipment do not allow for such high loading speeds or tests at elevated temperatures. Therefore, the force was increased by a rate of 100 N/s until break at room temperature. The high load setting of 1.0 Nm corresponds to a force of 50 N, the low setting equals a force of 15 N. Furthermore, the creep characteristic of the specimens was determined in a creep test. A constant force of 50 N was applied to the specimens for about 24 000 seconds, which corresponds to 400 000 load cycles. Longer testing times were not possible due to technical limitations. One representative tooth of three specimens was tested. The resulting creep curve was extrapolated to 10^6 load cycles using a Power Law approximation to estimate the creep during the in situ tests.

3.3.3 Tooth wear deformation

The ex situ tooth wear deformation was evaluated using a coordinate measuring machine, type Leitz PMM 654, by Hexagon Metrology GmbH, Wetzlar, Germany. The measurements were performed by the Institute of Manufacturing Technology, FAU Erlangen-Nürnberg, Germany. The coordinates were measured in scanning mode with a scanning speed of 0.2 mm/s and 30 measurements per 1 mm. The tip radius was 1.0 mm. The measured contours of the three specimens and the 39 teeth were averaged and compared to the untested contour. The wear was evaluated as the biggest normal distance between the two mean contours.

3.4. Influence of form and structural deviations

After validation, the in situ deformation testing method is used in combination with the analysis of the gear temperature and the spectral analysis of the torque fluctuations to determine the influence of form and structural deviations on the operating properties of the gear set.

3.4.1. Characterization of form and structural deviations

Two mould contours (line and spline contour) and two mould temperatures (60 °C and 100 °C) were used to create four sets of specimens with different combinations of structural and form deviations to examine their interactions.

To characterise the form deviations, a two-flank rolling test according to VDI/VDE 2608 was performed on five gears using a ZWP06, manufactured by Frenco GmbH, Altdorf**,** Germany. Results are the radial composite deviation F_i”, the tooth-to-tooth radial composite deviation f_i”, and the runout by composite test F_r”. F_i” is a measure for the overall form deviations of the gear, whereas f_i” correlates with the form deviations of the teeth and F_r” is a gauge for the out-of-roundness of the gear.

The differences in contour design are so miniscule that there is no influence on the morphology of the parts. Thus, the microstructure across the gear flank was analysed using Fourier Transform Infrared Spectrometry (FTIR) with a Nicolet 6700 FT-IR, Thermo Electron Corporation, Madison, WI, USA, on 10 µm thin cuts of exemplary parts for the two mould temperatures.

The degree of crystallinity was calculated from the band ratios of the wave numbers 1 199 cm^{-1} (crystalline phase) and 1 180 cm^{-1} (amorphous phase) measured in extinction, as described in [24, 25]:

$$\text{Degree of Crystallinity } \mathrm{C_{FTIR}} = 1.867 \cdot \left(\frac{\mathrm{E}_{1199}}{E_{1180}}\right)^2 + 20.003 \cdot \left(\frac{\mathrm{E}_{1199}}{E_{1180}}\right) - 1.305 \quad (4)$$

3.4.2. Operating properties of the gear sets

In the future, the test rig will be fitted with additional sensor equipment and test modes to characterise a wider array of operating properties. Acoustic and structure-borne sound sensors are planned to measure the Noise-Vibration-Harshness (NVH) properties of polymer gear drives. A test chamber will be integrated to run tests at elevated temperatures and with lubricants. And lastly a test mode for temperature and frequency sweeps will allow the characterisation of geometric and material influences on the vibration properties over a wide spectrum of operating conditions.

In this work, the influence of the part deviations on selected operating properties (in situ deformation, temperature development and vibration) were analysed using the new gear test rig. The tests were run with the test scheme used for the validation, i.e. three test runs per specimen group at an output speed of 1 000 min^{-1}, a runtime of 10^6 load cycles, switching between 1.0 Nm for 200 meshing cycles and 0.3 Nm for 40 meshing cycles.

To exclude any effects due to pure shrinkage differences between the specimens, a compensation of the centre distance of the gears based on the process-induced shrinkage of the parts was applied. This compensation accounted for the difference between nominal and actual gear diameters.

For this work, three separate aspects of the operating properties were analysed. Firstly, the variations in torque due to vibrations caused by imperfections in the gears were analysed, since this is a measure of the influence of part deviations on additional dynamic loads which can lead to premature defects. To this end, the input torque signal sampled in one-second intervals with 5 kHz and then Fourier-transformed. There were no notable changes in the resulting frequency spectrum over time or between the specimens of the same group. Thus, a representative spectrum from the first “high” cycle is analysed.

Secondly, the influence of microstructure and form deviations on the gear temperature during operation was examined with a thermocouple embedded in the tooth root, since temperature greatly affects material strength and lifetime of polymer gears. Lastly, the influence of the deviations on the total plastic deformation (wear and creep) was investigated using the in situ method, since too high deformation and wear can cause the observed lifetime to be significantly shorter than expected.

3.5 Representation of the margin of error

The errors given in the results are the 90% confidence intervals (CI) for the mean and the difference of means, calculated based on a t-distribution, as follows:

CI_{mean} $$\bar{x}_1 \pm \mathrm{t}_{\mathrm{n}-1} \cdot \mathrm{s}/\sqrt{\mathrm{n}} \tag{5}$$

$CI_{Difference}$ $$\bar{x}_1 - \bar{x}_2 \pm \mathrm{t}_{\mathrm{n}_1+\mathrm{n}_2-2} \cdot \sqrt{\frac{(n_1-1)s_1^2 + (n_2-1)s_2^2}{n_1+n_2-2}} \cdot \sqrt{\frac{1}{n_1} + \frac{1}{n_2}} \tag{6}$$

4. RESULTS AND DISCUSSION

4.1. Validation of the testing method

Figure 4 shows the in situ deformation, calculated as described in section 3.2. Both, the “low” cycle and the “high” cycle curve of the in situ test exhibit a run-in phase, followed by a steady state of linear increase. This is consistent with the wear behaviour of polymer-metal systems [2]. The difference between the two curves is considered the additional elastic deformation as a result of the increase in load of 0.7 Nm between the “high” cycles and at the “low” cycles. Evaluating the in situ data results in an additional deformation of 58.6 ± 1.6 µm initially. This increases during the tests to 64.9 ± 11 µm after 10^6 load cycles, probably due to reduced stiffness caused by frictional heating of the material during the test.

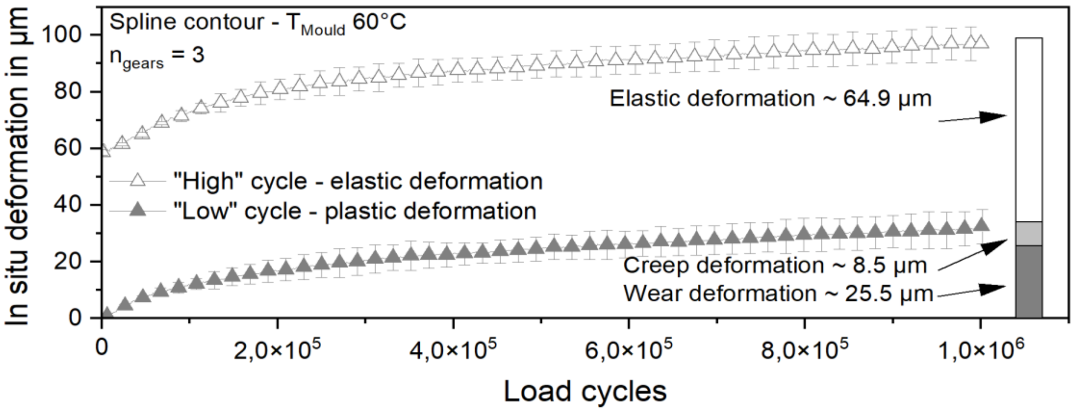

Figure 4: Correspondence of in situ and ex situ measurements

The ex situ pulser test for the elastic material properties shows a roughly linear increase of deformation with increasing force, see figure 5 on the left. To validate the measurement of the additional elastic deformation, the pulser test is evaluated at the forces which correspond to the load on the pitch circle at the respective torque level.

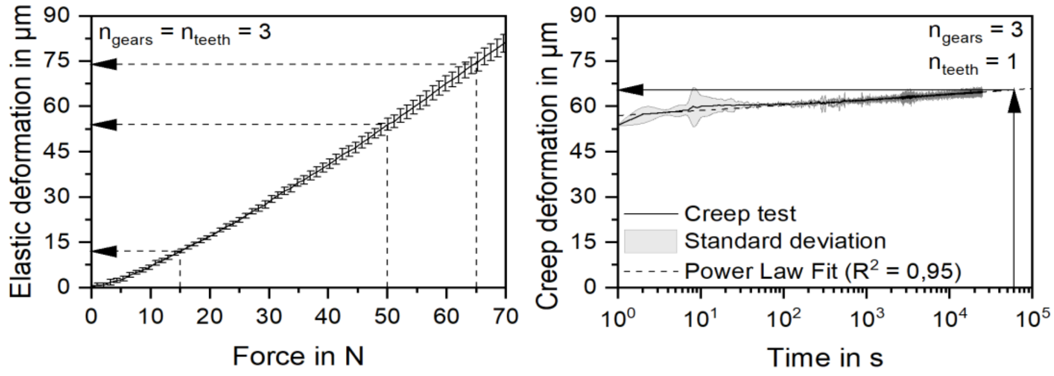

Figure 5: Elastic deformation (left) and creep (right) in the pulser test rig

In this setup 1.0 Nm torque corresponds to 50 N force in the pulser test and 15 N are equivalent to 0.3 Nm. This results in an elastic deformation of the tooth of from 12.2 ± 0.6 µm to 54.2 ± 2.0 µm for a difference of 42.0 ± 2.0 µm.

One explanation for this difference between in situ and ex situ measurements of the elastic deformation could be the torque differences in the meshing cycle due to dynamic loading of the teeth. The tested gear set has a nominal transverse contact ratio of 1.4. This means that on average 1.4 teeth are in mesh. At the beginning of the meshing cycle of the new tooth, the preceding tooth is still in contact. Both teeth share the applied load. This is followed by a phase in which only the current tooth transmits the load. Then the following tooth enters the engagement while the current tooth leaves the contact. A simplified schematic of this process is depicted in Figure 6.

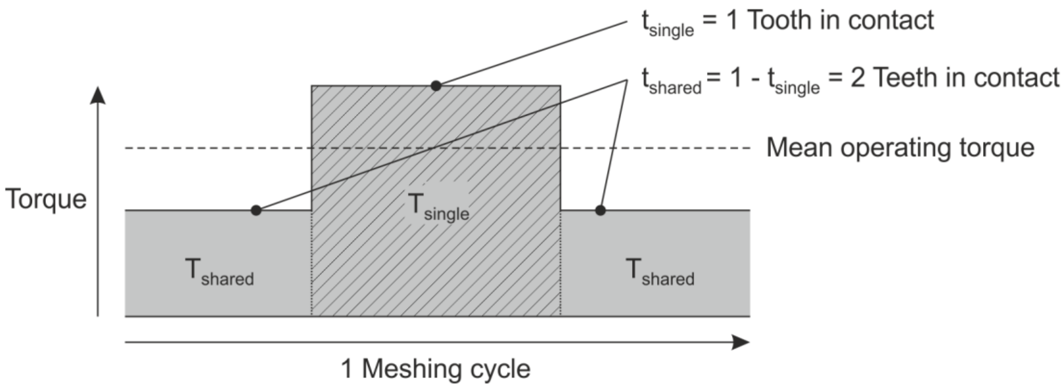

Figure 6: Simplified schematic of load sharing during the meshing cycle

Using the calculation given below, it can be determined for the given gear set that there is one single tooth in mesh (= t_{single}) for about 60 % of the meshing cycle. In the remaining 40 % two teeth (= t_{shared} = 1 - t_{single}) share the applied load.

$$1 \cdot t_{single} + 2 \cdot \left(1 - t_{single}\right) = 1.4$$
$$t_{single} = 0.6 \tag{5}$$

While the mean torque during the test was 1.0 Nm, the actual force on the teeth varies due to load sharing in the meshing cycle. It can reasonably assumed that the torque shared between the two teeth during the concurrent meshing phase is half as high as the torque a single tooth transmits (T_{single}). Based on this assumption the highest effective load on a single tooth in this gear set can be estimated as 1.25 Nm, with following calculation:

$$0.6 \cdot T_{single} + \ 0.4 \cdot \frac{T_{single}}{2} = 1\,Nm$$
$$T_{single} = 1.25\,Nm \tag{6}$$

If an effective single tooth load of 1.25 Nm (= 62.5 N) is assumed, the ex situ deformation is 74.1 ± 2.9 µm. Thus, the additional deformation in the ex situ test would be 61.9 ± 2.7 µm, agreeing with the in situ tests.

Another factor in the differences between the measurements of the elastic deformation could be the different testing conditions regarding temperature and strain rate, or loading speed respectively. Higher Temperatures result in a lower stiffness, increased loading speeds in increased stiffness. However, an exact analysis of these competing effects in the ex situ and in situ tests would require complex numerical modelling based on extensive thermo-mechanical material modelling, which is beyond the scope of this paper. Nevertheless, the effect of the increased temperature and loading speed on the material's modulus shall be estimated based on the thermo-mechanical characterisation of the material.

Supported by the literature [23], it is assumed that the core temperature of the tooth (23 °C for the ex situ and 40 °C for the in situ test) can be used as a suitable reference temperature for estimating the mechanical properties of a tooth. To estimate the relevant loading frequencies, the inverse of the loading speeds of the ex situ and in situ tests (0.5 s and 1.54 ms respectively) are used. The resulting loading frequencies in the in situ tests (650 Hz) are higher than those in the ex situ tests (2 Hz) by about three orders of magnitude. To determine the frequency-dependent modulus, Dynamic Mechanical Thermal Analysis (DMA) according to DIN EN ISO 6721 was performed using a rotational rheometer, type ARES-G2 by TA Instruments, New Castle, USA. Test specimen was an injection moulded tensile bar, size A1 made from Ultramid A3K according to DIN EN ISO 20753. In the DMA, temperature sweeps from 0 °C to 250 °C were carried out at 1, 10, 25 and 50 Hz. Based on the time-temperature superposition principle, frequency master curves for the storage modulus at 23 °C and 40 °C (and 50 °C for additional reference) were generated from these measurements.

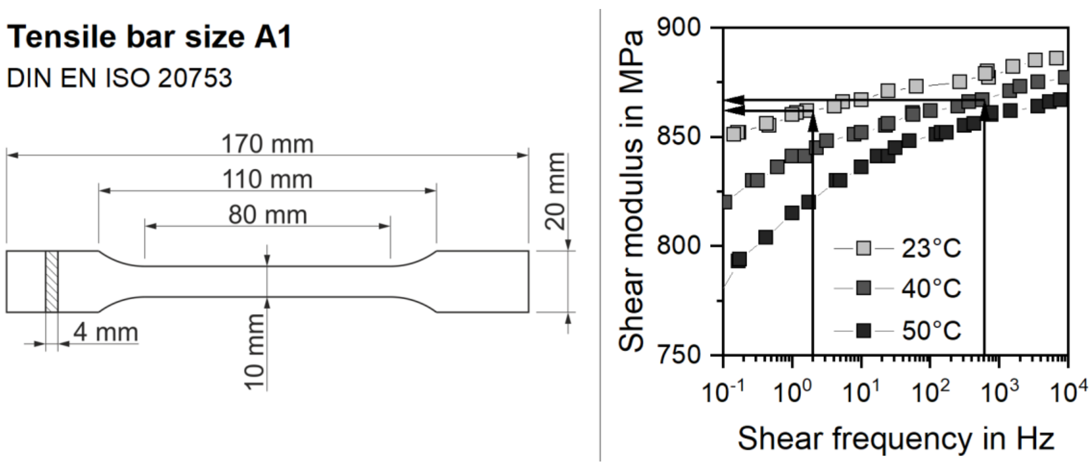

Figure 7: Tensile bar (left) and shear frequency dependent modulus (right)

Evaluated at the operating points of the ex situ and in situ tests, there is only an insignificant increase in the modulus from 862 MPa at 2 Hz and 23 °C to 867

MPa at 650 Hz and 40 °C, see figure 7. Based on these DMA measurements, only a negligible reduction of the deflection of might be expected as a result of the differences in loading speed and temperature for the chosen operating conditions. However, more research into the thermo-mechanical properties of polymer gears under typical operating conditions is certainly warranted.

In the validation tests the in situ plastic deformation after 10^6 load cycles was determined to be 32.4 ± 5.9 µm, see figure 4. This deformation consists of two main components, creep and wear. To determine the ex situ creep an extrapolated power law model was evaluated at one and at 60,000 seconds (= 10^6 load cycles). This gives a creep of 8.5 µm, see figure 5 on the right. The wear deformation was defined as the biggest difference between the mean tooth surfaces of the untested and the tested gears as determined by coordinate measurement. The wear was identified to be 25.5 ± 0.5 µm, see figure 8. Thus, the total deformation measured with ex situ methods was 34 ± 2.0 µm, matching the in situ measurement of 32.4 ± 5.9 µm.

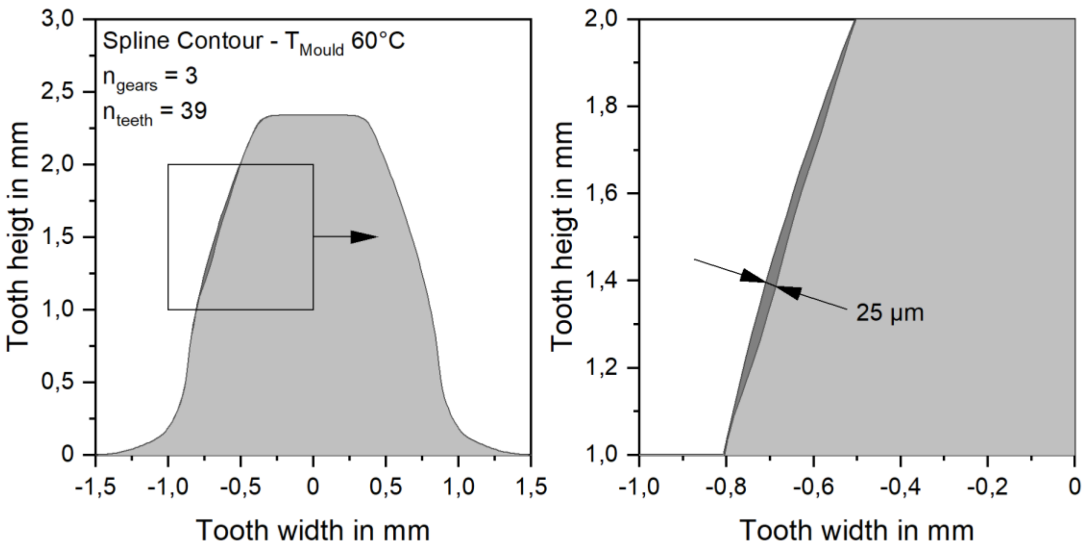

Figure 8: Mean tooth contours determined by coordinate measurement

Thus, both the elastic and the plastic deformation measured by the in situ test rig are validated using ex situ measurements, promising a very helpful tool to examine the behaviour of gear sets more closely. However, as of now, this method is only validated for this operating point. A more comprehensive validation of the system is currently being performed. Furthermore, the development of testing strategies for automatic separation of creep and wear in the plastic deformation data is subject of ongoing research at the LKT. Nevertheless, the presented method can be employed to investigate the influence of form and structural deviations on the operating properties of a polyamide-steel gear set.

4.2. Influence of form and structural deviations

There is a clear connection between the contour description and the form deviations of the test specimens in the two flank rolling test. Due to an unfavourable geometry generation the gears with the line contour show high tooth-to-tooth deviations, see figure 9 left side. Especially notable are the five peaks in the two flank rolling deviation of the gears with the line contour, figure 9 right side, which imply five slightly larger teeth on the gear. These five thicker teeth could generate additional vibrations during operation.

In contrast, the spline contour shows drastically lower tooth-to-tooth deviations, probably due to a better approximation of the geometrically ideal involute form and generation of more equally sized teeth. However, there is no statistically significant difference in the runout F_r" between the different contour designs or processing conditions. F_r" is mainly influenced by the position and the clearance of the shaft the gear is moulded onto in the injection-moulding tool.

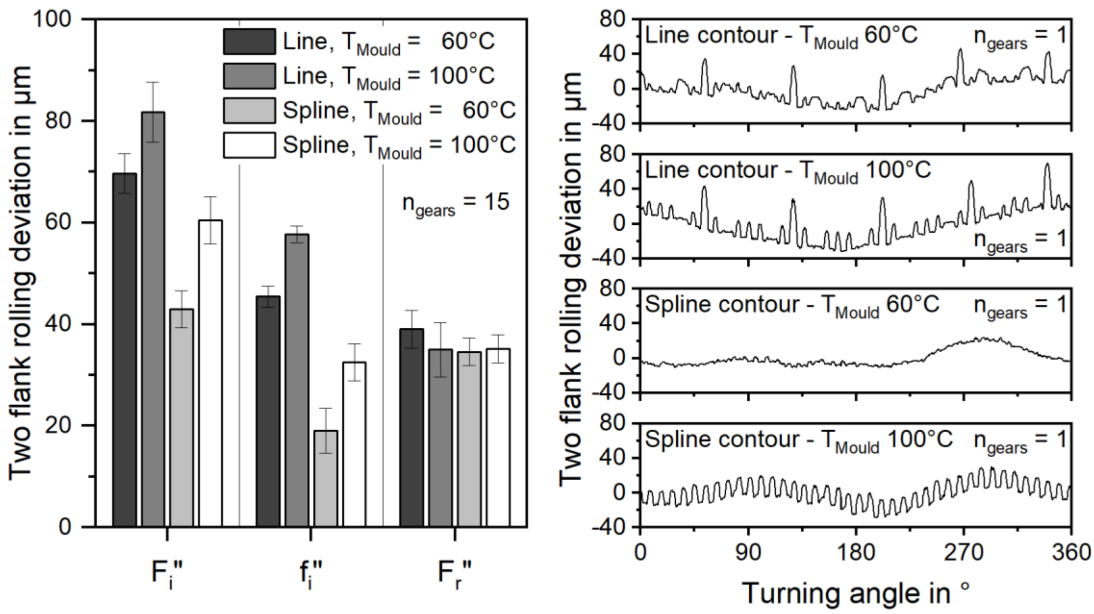

Figure 9: Two flank rolling deviations of the different specimens

There is also an influence of the mould temperature on the form deviations and the resulting flank rolling deviations. This is clearly seen in the characteristic peaks of the two flank rolling deviations, figure 9 on the right side. A small peak can be seen for every individual tooth of the gears manufactured at 100 °C. For the gears with the line contour, the five characteristic peaks are also slightly higher than those of gears produced at 60 °C mould temperature. Thus, it can be concluded that an increase in mould temperature leads to an increase in tooth-to-tooth deviation f_i" and radial composite deviation F_i".

There could be two main reasons. On the one hand, a higher mould temperature could lead to the melt staying warmer for longer, thus increasing viscosity and allowing a better moulding of possible contour errors. On the other hand, a higher mould temperature leads to a higher degree of crystallinity and formation of bigger sperulitic structures in the edge layer, see figure 10. This would increase the stiffness of the contact, thus leading to a higher impact of existing errors in the measurement, since there would be less elastic deformation of the contact area compensating for slight errors. A more comprehensive study of the geometry of the moulds and the gears itself is needed to determine and separate the exact cause, which is not in the scope of this but future work.

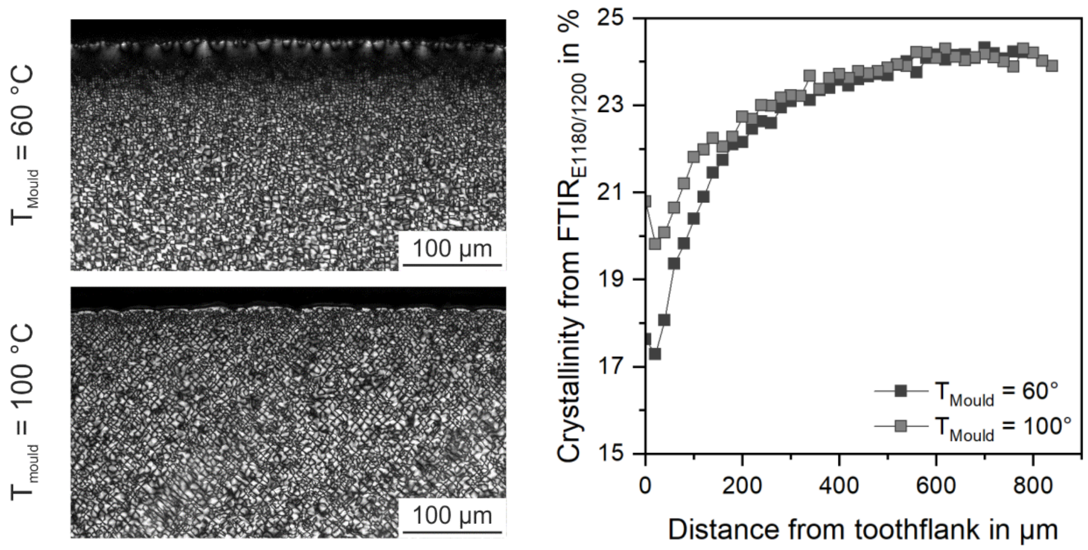

Figure 10: Morphological differences due to mould temperature

Combined these form and structural deviations have clear influences on the operating properties of the gear set. Figure 11 shows the spectral decomposition of the variations in the input torque signal. The peaks at 40 and 230 Hz represent natural frequencies of the test rig; the other smaller peaks are harmonics of the rotational frequency of 16.6 Hz.

An influence of the form deviations, especially the five slightly larger teeth of the line contour specimen, can be seen by the appearance of additional smaller peaks in the spectrum of the line contours, which are not present with the spline gears. Furthermore, the peak at 230 Hz is slightly higher and features a higher side band at 235 Hz for the line contour gears. This implies that the higher form deviations of the specimens, especially the five slightly thicker teeth, create additional vibrations. Since the amount of wear is roughly equal on all teeth, the slightly thicker teeth of the line contour specimens remain thicker compared to the other teeth over the test duration, explaining why the wear does not influence the variations in the input torque.

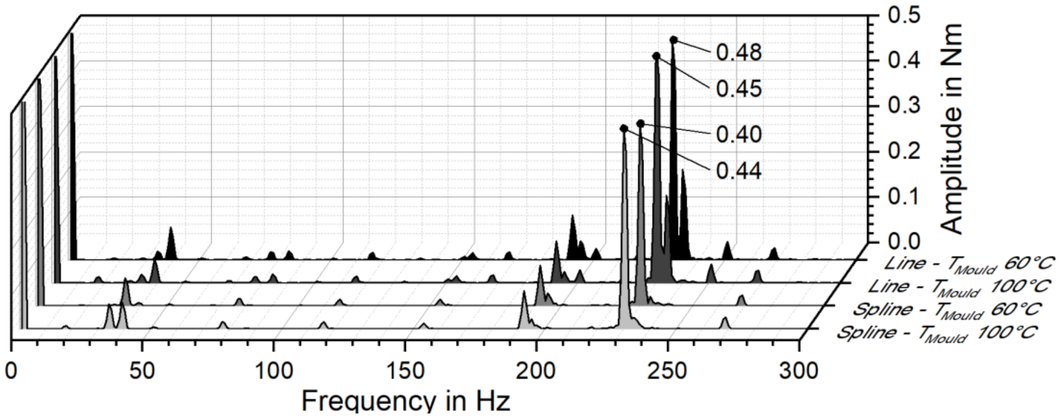

Figure 11: Influence of form and structural deviations on input torque variations

The peaks at the natural frequencies for the spline gears manufactured at 100 °C are slightly higher compared to the specimens produced at 60°C, possibly due to decreased damping as a result of the higher crystallinity. Otherwise, there seems to be no influence of the mould temperature on the torque variations, especially for the parts with higher form deviations.

The same relationship can be seen in the analysis of the tooth core temperature of the gears, see Figure 12 left side. There seems to be no influence of the form or morphological deviations on the core temperature. All gears reach and hold an operating temperature of 38 - 40 °C. The frictional heating seems to be independent of the form and structural deviations.

However, due to the measurement of the temperature in the tooth core via thermocouple being an integral measurement at a certain distance from the active flank, effects of local and temporal peaks in flank temperature might not have been registered. Thus, the investigation of the flank temperature by means of thermographic imaging might be of interest for future research.

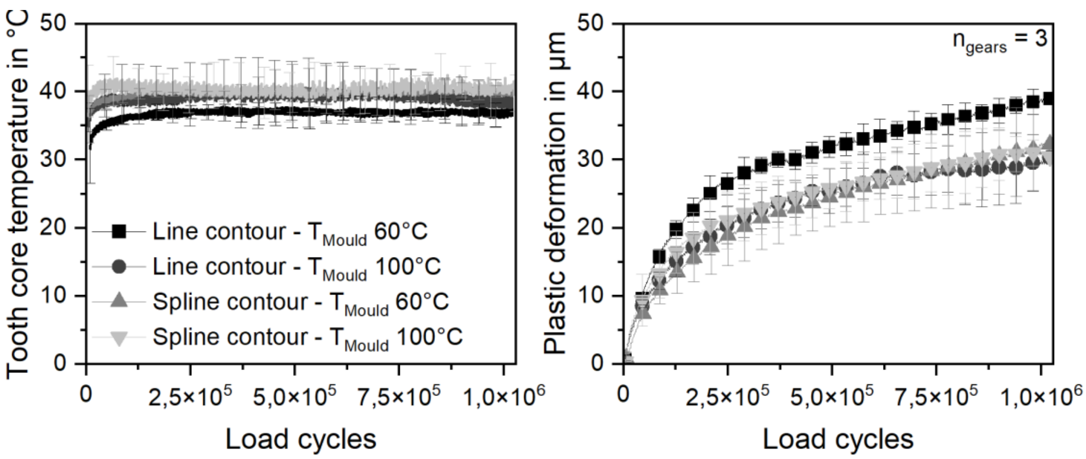

Figure 12: Influence of deviations on tooth temperature and plastic deformation

Regarding the plastic deformation, the specimens all show the characteristic behaviour of polymer-metal contacts. There is a run-in phase with an instationary change in deformation lasting for about 3.5×10^6 load cycles, followed by a phase of linear increase. There is no significant difference in the amount of run-in deformation for the specimens with the spline contour (60 °C specimen: 21.7 ± 4.6 µm after 3.5×10^6 load cycles, 100 °C specimen: 23.3 ± 3.5 µm) and the line contour specimens manufactured at 100°C mould temperature (22.9 ± 3.1 µm after 3.5×10^6 load cycles). However, for the line contour specimens manufactured at 60 °C, a slightly higher amount of run-in deformation (29.5 ± 0.9 µm after 3.5×10^6 load cycles) was detected when compared with the other gear sets.

One explanation could be that the interaction of unsuitable morphology and higher form deviations leads to increased wear. In turn, it seems sufficient to produce either better morphological properties or lower shape deviations to reduce wear. To verify that there is no advantage in improving both microstructure and form deviations, more extensive tests need to be carried out on samples produced at 100°C with low form deviations. Additionally, repetition tests to verify these preliminary findings at a higher level of significance should be carried out.

Finally, the rate of deformation increase in the stationary phase seems to be the same for all pairings. This suggests that there is no significant influence of the form or the morphological properties of the gears on the creep and wear behaviour after the gears have sufficiently adapted to each other. This implies that the potential for wear reduction mainly lies with reducing run-in wear by either improving geometric accuracy or morphological structure, or ideally both.

5. CONCLUSION

In this contribution, a new design for a test rig was presented and validated. By employing a specially designed load spectrum, the elastic and plastic deformation components under operating conditions could be separated. The validation with ex situ measurements showed a good agreement with the in situ measurements. The in situ plastic deformation of the validation pairing, 32 µm, could be related to creep deformation of about 9 µm in a gear creep test, wear deformation of 25 µm according to coordinate measurements, and a theoretically calculated thermal expansion of about 1.5 to 3 µm. The elastic deformation of 65 µm between “high” and “low” cycles could be reconciled with the deformation in the quasi-static pulser test, if the additional dynamic forces due to vibrations in the gear system of about 0.3 Nm are considered. However, further research into the testing methodology is needed to allow the automatic separation of thermal, elastic, visco-plastic (creep) and wear deformation.

Furthermore, the test rig was used to investigate the influence of form and structural deviations on the operational behaviour of a steel-polyamide gear set. The algorithms used to generate the gear contour significantly influenced the transmission error and the vibrations in the gear systems. The contour approximated with line segments showed significantly higher transmission error and more vibrations compared to the spline contour because of higher variations in the tooth thickness over the circumference of the gear.

An increase in mould temperature increased transmission error in both contours slightly, probably due to a stiffer and less error compensating contact behaviour. In general, there was no significant difference in wear rate between the specimens, once they reached the stationary wear phase. In the run-in phase, however, the specimens with the line contour and the low mould temperature showed higher run-in wear, whereas all other pairings performed roughly equal. This might be due to a detrimental interaction of transmission error and unfavourable morphology. Thus, it can be concluded that polymer gears can tolerate either a certain amount of form or structural deviations, but not both at the same time. Further research into the influence of these interactions of form and structural deviations should explore if this also true in the case of load carrying capacity and achievable system lifetime.

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