High-Temperature Testing of Stanyl Plastic Gears: A Comparison with Tensile Fatigue Data

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Management Summary

This paper shows an experimental study on the fatigue lifetime of high-heat polyamide (Stanyl) gears running in oil at 140°C. Based on previous works (Refs. 1–2), an analysis is made correcting for tooth bending and calculating actual root stresses. A comparison with tensile bar fatigue data for the same materials at 140°C shows that a good correlation exists between gear fatigue data and tensile bar fatigue data. This insight provides a solid basis for gear designers to design plastic gears using actual material data.

Introduction

Plastic materials have been used in gearing for quite some time. Over the last decade the application field for plastic gears has extended from only low loads, positioning type of transmissions, to increasingly more demanding applications with high loads, high numbers of cycles and high temperatures. This implies that during the design process not only the quality of the geometry is important, but also the dimensioning with respect to stresses. The standards (ISO, DIN, AGMA) which are currently used by gear designers have a proven track record for metal gears; however, they certainly lack features that are of importance for plastic gears. In addition to this, the experimental data on gears available for today's gear materials are rather limited.

In a previous study, it was shown that the kinematics and stress distribution in a metal-plastic gear pair can be quite different from a metal-metal gear pair (Refs. 1–2). The main reason for this is the fact that the stiffness-strength ratio of plastics is lower, compared to steel. As a result, the deformation and tooth bending under loading are far more pronounced for plastic gears. Due to this load sharing over tooth pairs in contact (Fig. 1), the contact path and contact ratio are considerably different.

Figure 1b shows that the changes in load sharing influence the root stresses of the gear pair. It is not the modulus that directly affects the stresses, but the changes in contact ratio and tooth bending that—via load sharing—influence the root stresses.

From Figure 1a it is also observed that by going from a steel-steel pairing to a steel-glass-filled plastic pairing, the period of single tooth contact is halved. For the unfilled plastic case and the unfilled plastic at elevated temperature case, single tooth contact no longer occurs during meshing. With decreasing modulus the maximum load share decreases to a plateau value of approximately 2/3. This increase in contact ratio was shown to result in a substantial decrease in root stresses (Table 1).

The question now arises whether the root stresses can be related to material properties. For metallic materials, the correlation between material properties (tensile strength, fatigue strength, etc.) and the actual performance of a gear are quite well established. This is not the case for polymers. Along with

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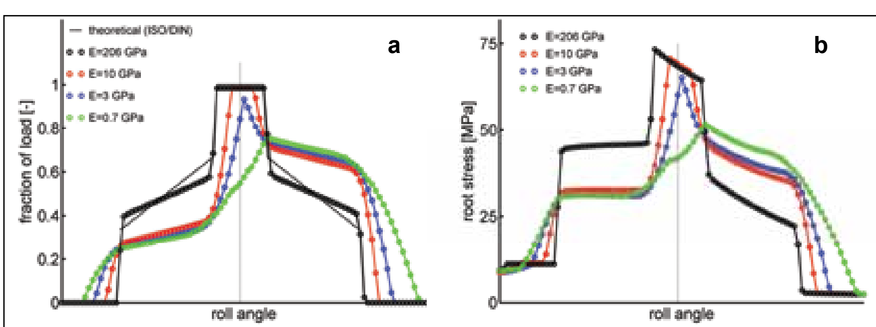

Figure 1—(a) load sharing and (b) root stresses as function of roll angle for a steel pinion and a steel gear (black circles), a glass-fiber (GF) reinforced-plastic gear (red circles), an unfilled plastic gear (blue circles) and an unfilled plastic gear at elevated temperature (green circles). The dashed black line is the load sharing according to (steel) theory (ISO 6336) and the dotted black line represents the pitch point.

Table 1—FEA Root Stresses versus ISO 6336 values upon varying the load share by changing the stiffness of the plastic gear in a metal pinion/plastic gear contact.			
Root Stress	Modulus	ISO 6336	FEA
Steel	E = 206 GPa	74.8 MPa	73.4 MPa
Plastic GF (30%)	E = 10 GPa	74.8 MPa	70.7 MPa
Plastic UF	E = 3 GPa	74.8 MPa	65.1 MPa
Plastic UF at high T	E = 0.7 GPa	74.8 MPa	51.7 MPa

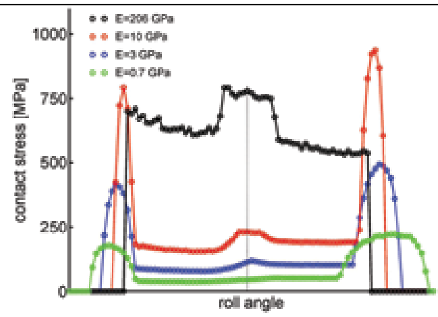

Figure 2—(Hertzian) contact stress as a function of the roll angle for a steel pinion meshing with a gear of various materials.

Table 2—Details of gear geometries		
	Pinion	Gear
Module	2 mm	2 mm
No. of Teeth	22	31
Pitch Circle Diameter	44 mm	62 mm
Base Circle Diameter	41.35 mm	58.26 mm
Tip Circle Diameter	48 mm	66 mm
Pressure Angle	20°	20°
Profile Shift	–	–
Tooth Width	13 mm	12 mm
Center Distance	53 mm	
Material	steel 16MnCr05	Stanyl
Root Radius Profile	0.38 mod	0.38 mod

differences in kinematics and stresses, the effect of testing conditions like temperature, strain rate, humidity, etc., play a huge role in their performance.

In addition, an increase in contact path length is observed in Figure 1 by going from the steel-steel mesh to the steel-plastic material at elevated T-mesh. The combination of increasing contact path length and tooth bending was also shown to have a big influence on the contact stresses in the same work (Refs. 1–2). The contact stress picture is shown in Figure 2.

As expected—based on contact mechanics and normalized calculations—the contact stresses near the pitch point where, according to classical theory, the maximum is found, decreases with modulus. However, due to the change in contact path, a preliminary contact of the tip of the plastic gear with the root of the pinion and a prolonged contact of the tip of the steel pinion with the root of the plastic gear are observed. This interference results in huge contact stress peaks at the beginning and end of the contact, due to the small contact radii and high forces, resulting in high contact stresses at high sliding velocities and, thus, in high pressure velocity values. Further study of the kinematics resulted in the expectation of substantial wear near the tip and the root of the plastic gear, for which observations in literature were shown to exist (Refs. 3–4).

Intent of this Study

Designing gears requires a degree of experience. Many new designs are based on proven concepts of the past. This is certainly true for plastic gears, where at the moment this comparative, best practice method is the safest way to operate. However, bottom line is that, in principle, a gear tooth is an odd-shaped bending beam, and the expected lifetime of this bending beam under fatigue loading should come close to the lifetime assessed in a lab scale test, provided that conditions are the same. For metals this is quite well established; however, for polymers this is certainly not the case. So if the aim is to assess the lifetime of actual gears under well defined testing conditions, this can be achieved by assessing the lifetime of tensile bars under the exact same conditions and trying to correlate the performance in terms of allowable stress for a certain number of cycles, via accurate numerical methods. There are three steps:

1. Generate high-temperature fatigue data for various Stanyl gears under oil lubrication. This approach was expected to result in fatigue failure of the gears by minimizing the amount of wear as much as possible while keeping the temperature as constant as possible.
2. Generate high-temperature fatigue data for various Stanyl tensile bars at the same temperature.
3. Incorporate the influence of tooth bending on the root stresses at various torque levels, and determine whether a correlation exists between fatigue lifetimes measured on gears and those measured on tensile test bars.

Materials and Methods

Stanyl is a high-heat polyamide PA 46 material made by DSM Engineering Plastics. The material is characterized by a high level of crystallinity (70%), which results in the retention of mechanical properties at temperatures above the glass transition temperature of all polyamide materials. Beyond this the material exhibits wear resistance and good fatigue properties at elevated temperatures. The following grades were incorporated in this research program:

- Stanyl TW341, an unfilled grade
- Stanyl TW200F6, a 30% glass-fiber reinforced grade
- Stanyl TW200B6, a 30% carbon-fiber reinforced grade

As a comparison material, PEEK Victrex 450G (unfilled grade) was tested.

Test temperature was 140°C, as lubricant Nuto H-68 oil was used (spray lubrication), a standard ESSO motor oil. All materials were subjected to a 140°C oil aging test, to ensure that no mechanical property deterioration occurred during the lifetime test runs.

Gear geometry. The gear geometries are listed in Table 2; injection molding of the gears was performed by IMS Gear, Donaueschingen, Germany; tool layout was designed so that at T = 140°C, the gears were of the required size (compensating for the shrinkage and thermal expansion).

The test rig (Figure 3). The gear testing was conducted the University of Berlin (Ref. 5) on two identical, in-house built four-square testing devices (Refs. 6–7). This device contains two gear pairs, i.e.—a metal driving gear pair and the testing gear pair—connected by two shafts. Via a torsional spring on one of the shafts, a preset moment is applied on both gear sets. Using such a closed-loop system is very beneficial in that the input power is only required to overcome the frictional and hydrodynamic losses of the lubricated gears and the frictional losses in the bearings. Via an electronic control system, the moment on the gear sets is monitored and, at a steep decrease of this signal (tooth breakage), the test is stopped.

All tests (four torque levels for each material grade, each torque level measured 7-fold) were run at 3,000 rpm running speed of the pinion. During the testing, the gears were lubricated via spray lubrication with oil from a thermostrated oil bath. The bulk temperature of the gear (measured via a thermocouple inserted in a gear tooth) was measured during various tests and proved to be very close to 140ºC. Also, the gear flank temperature, measured by infrared camera, proved to be close to the set temperature of the oil.

Tensile bar fatigue testing. The tensile bar fatigue tests were run on standard ISO 527-1A injection molded specimens. The specimens were subjected to a cyclic loading at a specified stress and a frequency of 8Hz. At this frequency the heating of the specimens, due to viscous dissipation, is negligible and the tests can be considered as isothermal, with the loading of the specimen cycles between a maximum value (max. stress level) and 10% thereof (min. value), implying R = 0.1 (ratio min/max value). The tests were performed on Zwich-Roell servo-hydraulic dedicated fatigue testing equipment, equipped with temperature chambers to control the environmental temperature.

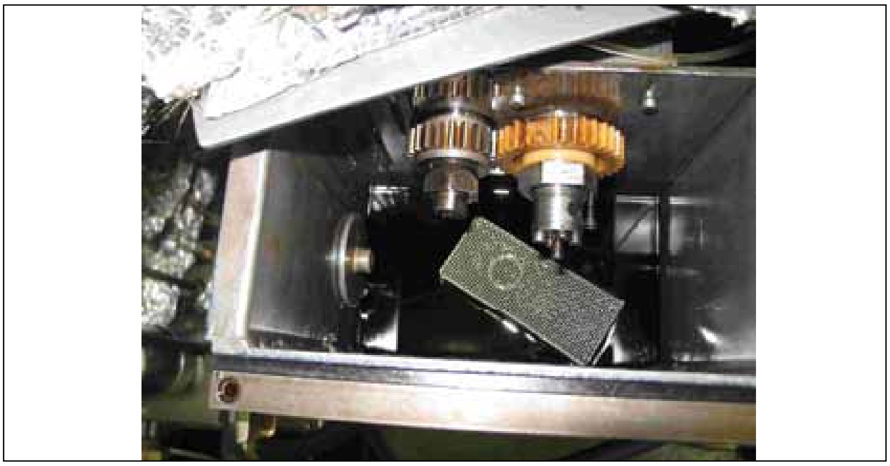

Figure 3—Detail of the gear test setup at the University of Berlin (Refs. 5–7).

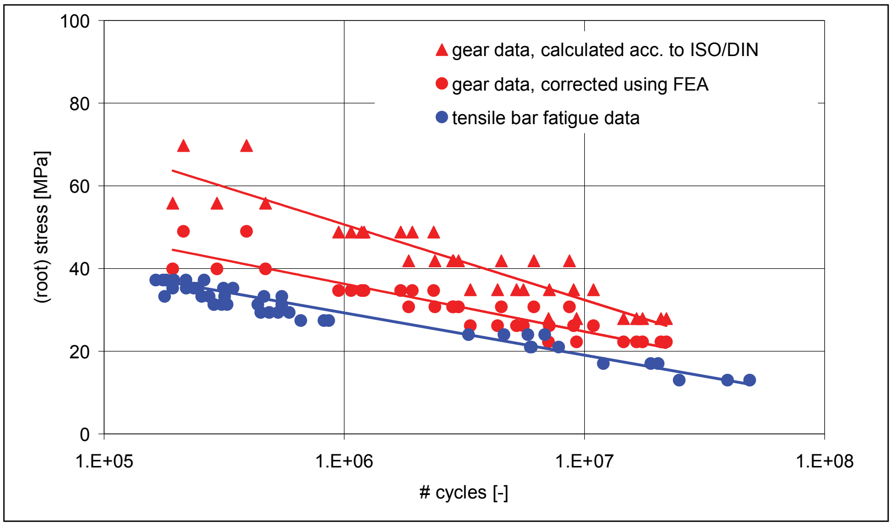

Figure 4—Experimental lifetime results for TW341 measured on gears and tensile test bars.

Experimental Results

Zooming in on Stanyl TW341. The tests of the unfilled material TW341 clearly showed a root failure, meaning that wear was negligible and that failure occurred in the region where the highest tensile stresses are expected during loading.

Via standardized calculations, the root stresses can be determined given the geometry and the applied torque. In Figure 4, the red triangles represent the data of ISO stresses versus the number of cycles until failure in a gear test. Clearly, the lifetime decreases linearly

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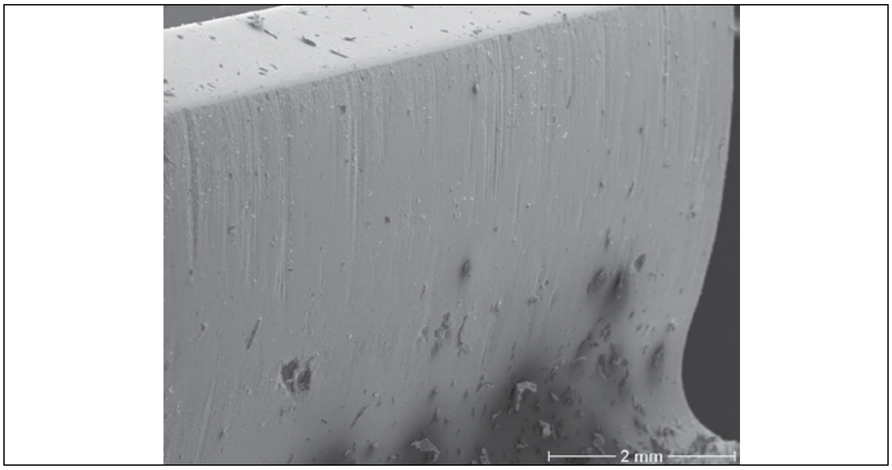

Figure 5—Detail of TW341 flank (SEM observation) after 3 million load cycles.

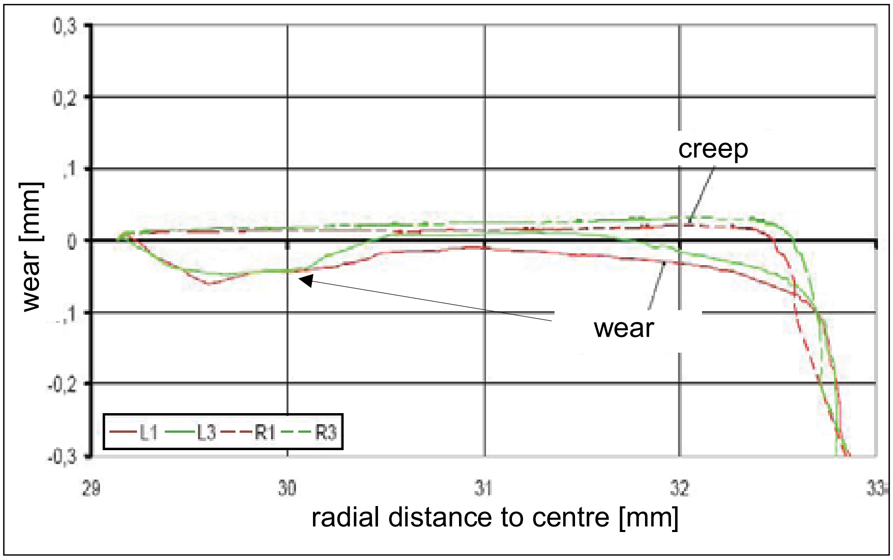

Figure 6—Wear of TW341 flanks after 1.1 x 10^7 load cycles.

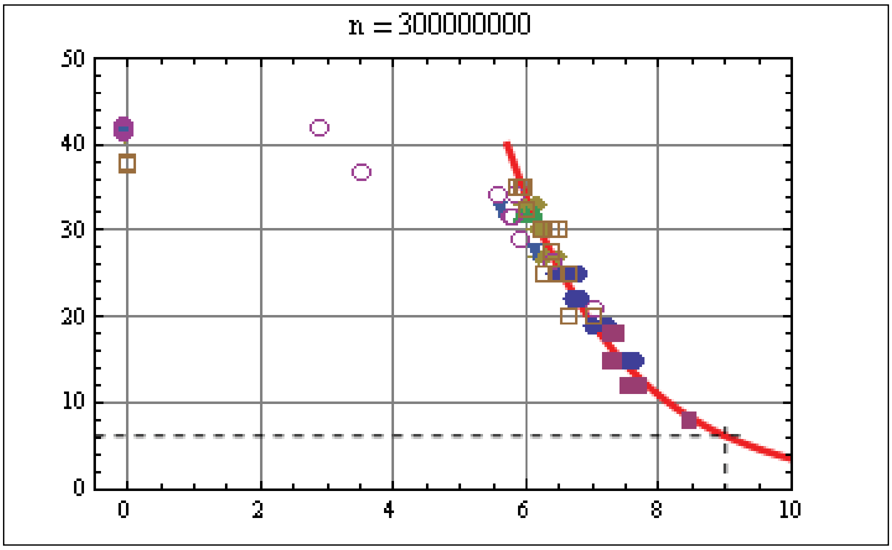

Figure 7—Fatigue results obtained for Stanyl TW341 tensile bar fatigue testing at 140°C.

on a logarithmic time scale.

The red circles represent the root stresses obtained by calculating these stresses by FEA. In this way a correction is made for the maximum stress due to tooth bending and load sharing. The blue circles, finally, represent fatigue lifetime measurements obtained upon tensile fatigue testing (ISO-527-1A test bars) at 140°C.

From Figure 4, the following can be concluded: Upon correcting the root stresses induced by the applied torque for tooth bending, a fairly good correlation between fatigue data measured on gears and tensile test bars respectively exists, the gear data being somewhat better. The reason for a better performance in gears compared to test bars is still subject to investigation. One of the reasons could be that the loading of a gear tooth is quite different from a test bar. A gear tooth experiences a peak load during a short period of the revolution, while during the remaining part of a revolution the load equals zero. In our case, this occurs at a frequency of 3,000 rpm or 50Hz, much higher than the applied testing frequency of the perfect sinusoidal loading at 8Hz in the test bars. The rate of deformation, to which these materials are quite sensitive, in a gear tooth is roughly 100 times higher than in the test bars.

As mentioned, the failure mode observed for the TW341 gear was a clear fatigue failure; hardly any signs of wear were observed in this case, as shown in Figure 5.

The low wear of TW341 at 140°C is further illustrated by one-flank profile measurements after over 10 million cycles (Fig. 6). For two separate gears (1 and 3) the deviation from the ideal involute profile is measured. Some minor wear scarring can be observed at the tip and root of the driven flank (L1 and L3) where, due to the tooth bending, interference occurs (Refs. 1–2). In addition, some deformation of the tooth due to creep can be observed by the positive wear on the idle flank (R1 and R3).

The good correlation observed

between tensile bar fatigue testing and gear testing was further exploited; tensile bar fatigue testing was further extended with a three-fold measurement at 8 MPa. This resulted in 300 million (300 x 10^6) load cycles without a fatigue failure occurring. The results obtained were subjected to a statistical analysis, and a sensitivity analysis was performed for the influence of the exact position of the failure point on the shape of the fatigue curve. A clear leveling off of the fatigue curve was observed, which was shown to be statistically significant.

The curve was further used to make an extrapolation to 10^9 number of load cycles, resulting in a stress level of 6 MPa.

In view of the good correlation with gear testing, it is now stated that Stanyl TW341 gears, when subjected to a stress level of 6 MPa, have a lifetime of at least one *billion* load cycles. These lifetimes are critical when considering oil pump-gear applications or balancer gear applications, to mention a couple of examples. With wear resistance at 140°C, Stanyl TW341 a good choice for these types of applications.

The resulting fatigue curve is shown in Figure 7. The red line through the data points has been fitted, followed by a statistical analysis, to establish whether the leveling off is significant or not. It was concluded that the leveling off indeed is significant, indicating the presence of a fatigue limit for Stanyl TW341 at 140°C. The factor n represents the number of cycles observed at 8 MPa stress.

Comparison of Stanyl TW341 and PEEK Victrex 450G. In Figure 8, a comparison is made between the lifetime of Stanyl TW341 and unfilled PEEK Victrex. The relatively short lifetime of the unfilled PEEK gears under these experimental conditions can be explained by the severe wear observed with PEEK 450G, as illustrated in Figure 9; the failure was clearly wear induced. Further, the relatively low level of crystallinity of PEEK results in a substantial drop in mechanical properties at/above the materials glass transition temperature of 143°C.

Zooming in on Stanyl TW200F6 (Stanyl 30% glass filled). Regarding the 30% glass filled grade, wear-induced fatigue failure was the observed failure mode. In Figure 10, gear testing results are compared with tensile bar fatigue testing results for both Stanyl TW200F6 and Stanyl TW200B6. The nature of the gear fatigue process seems to be somewhat different for the fiber-filled grades than the tensile bar fatigue process. For high loads, the lifetimes lie somewhat higher than the tensile bar fatigue data, similar to the unfilled material. For lower loads/higher lifetimes, the lines cross

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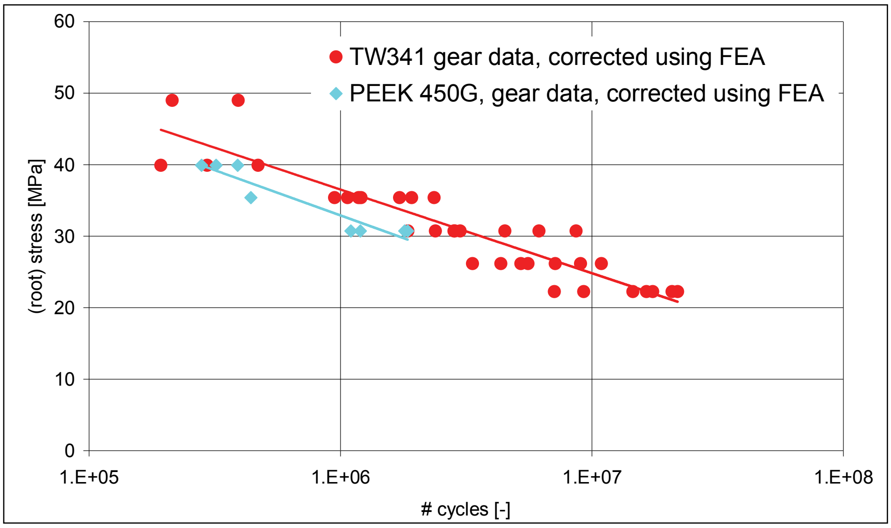

Figure 8—Root stresses (corrected for tooth bending) versus number of load cycles for gear testing of Stanyl TW341 (red) and PEEK 450G (cyan) at 140°C.

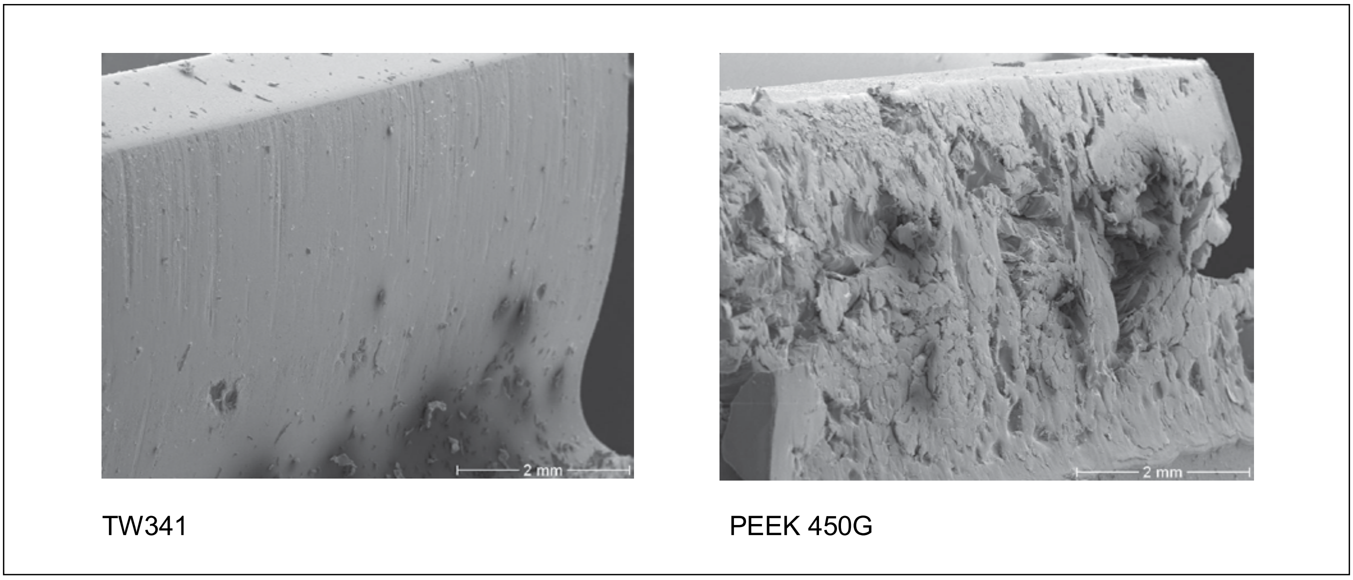

Figure 9—Wear comparison of Stanyl TW341 (left picture) and PEEK 450G after 3 million cycles at 140°C and a torque level of 10 Nm.

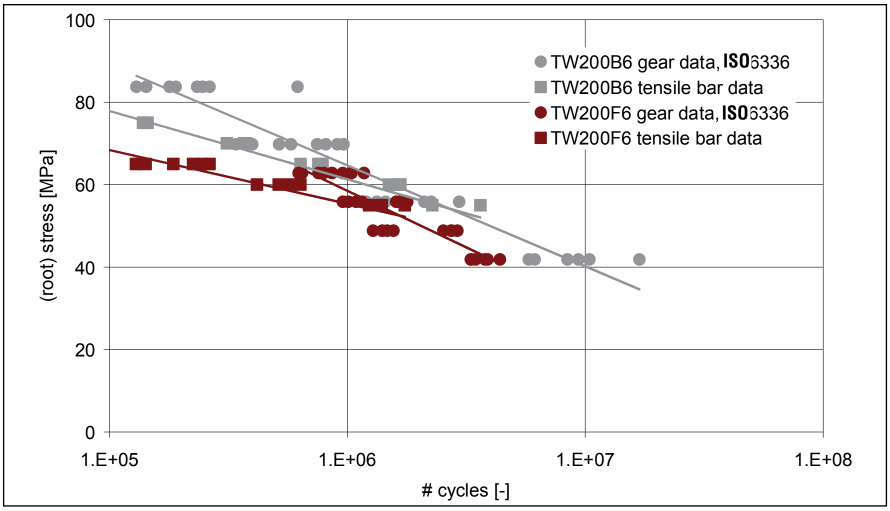

Figure 10—Tensile bar fatigue testing versus gear testing for Stanyl TW200F6 (top) and TW200B6.

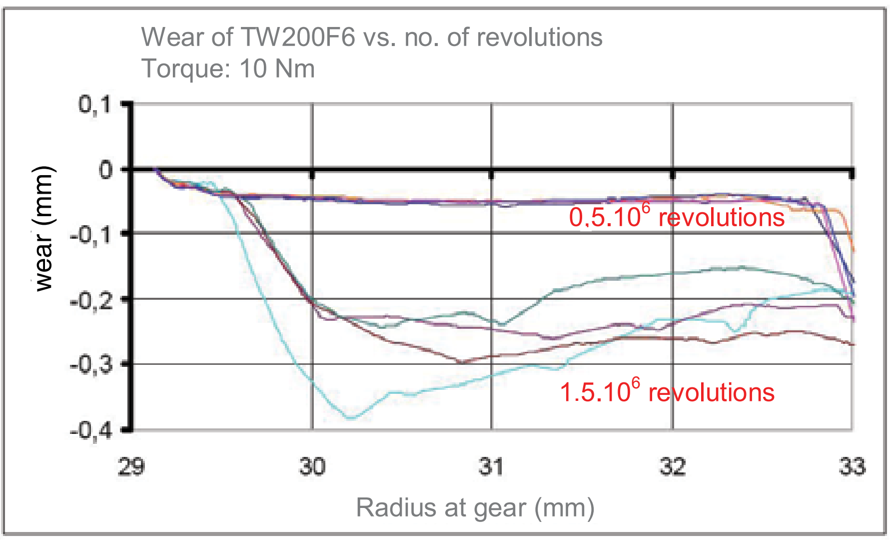

Figure 11—Illustration of wear observed during gear testing of Stanyl TW200F6 at 140°C under oil lubrication.

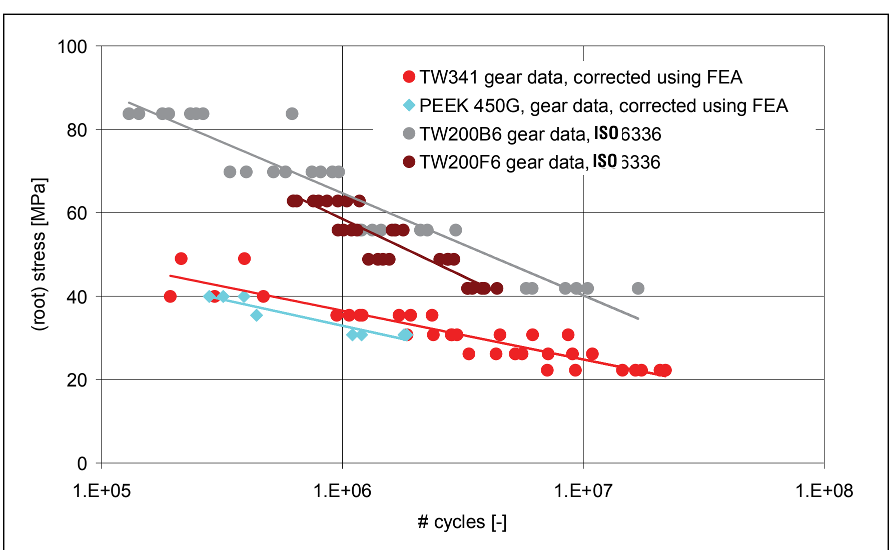

Figure 12—Gear fatigue results at 140°C under oil lubrication for various Stanyl grades and a PEEK material.

each other. This can be explained by the fact that, at these high numbers of cycles, the failure is wear-induced and not a true fatigue failure. The evolution of both failure processes over time can of course be very different.

For the fiber-filled materials, with Young's moduli of 5 to 10 times that of unfilled Stanyl, a correction for tooth bending is also made. However, the effect on the root stresses is only minor due to the relatively small bending of the teeth.

The final gear fatigue picture. Figure 12 presents the final gear fatigue picture after testing at 140°C with oil lubrication. In the instance of the unfilled materials, the root stresses have been corrected for tooth bending. A clear distinction can be seen between fiber-reinforced and unfilled grades. However, due to the steeper decay of the fiber-reinforced grades due to wear, the curves come closer to each other a very high number of cycles.

Conclusions

Various grades of DSM Engineering Plastics' high-temperature PA 46 have been tested as a gear at 140°C under oil lubrication. The data can be used for grade selection for high-temperature applications.

The concept of tooth bending of the plastic gear in steel-plastic gear combinations has been explored. Metal-based ISO-6336 results are an overestimation of root stresses calculated from applied torque levels, especially in (low modulus) unfilled plastic materials at elevated temperatures. Tooth bending is the underlying phenomenon for this, resulting in an increase of the contact ratio. This means that on average the applied loads are shared between more teeth, resulting in an overall lower root stress. The correct value of the root stress is therefore only obtained by using FEA, not by using ISO-6336.

When tooth bending is being taken into account and the correct root stresses are calculated, a fairly good agreement/correlation is obtained between fatigue data measured on gears and test bars respectively. This, however, is only the case if the failure mode is a true fatigue failure; for wear-induced tooth breakage, no correlation can be observed. For TW341 (true fatigue failure in the gear tests), a positive correlation with test bar fatigue was easily observed.

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