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RESEARCH ARTICLE

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Damage mechanisms and tooth flank load capacity of oil-lubricated peek gears

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Abstract

Plastic gears are increasingly being utilized in higher performance ranges. Operating under oil lubrication can reduce friction and heat input. By running under oil-lubricated conditions, tooth damage such as abrasive wear recedes into a secondary role, and fatigue damage such as pitting becomes relevant for the durability of the gears. Polyether ether ketone (PEEK) is suitable for use in demanding operating conditions due to its very high strength properties, even at higher temperatures. In this work, the stress conditions affecting the tooth flank and the damage behavior of oil-lubricated PEEK gears are investigated and tooth flank strength parameters were determined for different temperature levels. Theoretical studies on the stress distribution on the tooth flank show the necessity to apply adapted profile modifications in order to reduce undesired stress conditions in the extended meshing area. In experimental investigations, the damage behavior of oil-lubricated PEEK gears is examined and the influence of temperature on the gear load capacity is discussed.

KEYWORDS

applications, friction, lubrication, theory and modeling, wear

1 | INTRODUCTION

Plastic gears are being produced in ever larger volumes and are used in new applications. In the past, their field of application was mostly limited to power transmission in actuators, but current developments show a trend towards applications with higher drive power. In the automotive sector, in addition to a large number of electric actuators driven by polymer gears, safety-critical applications such as braking and steering systems are increasingly equipped with plastic gears.[1] Today, plastic gears are typically applied in the drive unit of modern e-bikes. Even the drive train of small urban electric vehicles can be equipped with plastic gears.[2] High-performance polymer gears are there typically operated in an oil-lubricated system. The advantage of using plastic gears is the possibility of integral design and free shaping by injection molding. The production of high quantities in the injection molding process offers significant cost advantages. Due to the low material density, effective lightweight engineering can be performed. The advantage of PEEK in general is the possibility of operation in high temperature applications such as balancer gears in combustion engines. Furthermore, PEEK show a comparably high tooth root strength in comparison with other thermoplastic materials such as POM or PA46.[3] Under oil-lubricated operating conditions, PEEK gears are primarily subject to pitting damage.[4] The occurrence of pitting damage was already observed by Hachmann &

Strickle[5] in the 1960s with oil-lubricated polyamide gears. Van Melick[6] conducts research on PEEK gears and points out the formation of pitting. Lu et al.[7] investigate the flank load carrying capacity of oil-lubricated PEEK gears and generate pitting damages on the active gear flanks. Berer et al.[8] investigate the pitting performance of oil-lubricated PEEK disks and observe pitting like damages on the disk surfaces. However, until now there are no reliable tooth strength parameters available for PEEK materials in relevant gear design guidelines such as VDI 2736.[9]

2 | PROPERTIES OF POLYMER GEARS

Based on their thermos-mechanical properties resulting from the molecular structure, plastics can be classified into different subcategories: elastomers, thermosetting plastics as well as thermoplastics. In the environment of drive technology, semicrystalline thermoplastics are mainly used as gear materials.[10] The thermal and mechanical material properties of thermoplastics differ fundamentally from those of steel materials. In particular, tensile strength and Young's are significantly lower in comparison to steel grades. Further, strength and Young's modulus are highly temperature-dependent.[1] Not only temperature but also humidity and loading characteristics influence the mechanical properties. At high loading speeds, Young's modulus and tensile strength increase while elongation at break decreases. During static loading, plastics can be deformed by creep.[11] Increasing moisture contents also reduce the mechanical properties. Furthermore, the absorption of moisture leads to volume changes caused by expansion.[11] By increasing the material temperature, the fraction of crystalline areas is reduced resulting in a reduction of the mechanical strength properties such as tensile strength and Young's modulus. Exceeding the glass transition temperature leads to a significant reduction of the mechanical properties.

Viscoelastic material behavior is another parameter that influences the characteristics of thermoplastic polymers. Viscoelasticity results in the damping properties of thermoplastics which differ depending on the type of material and the occurring temperature.[1] The high damping coefficients of thermoplastics compared to steel materials are advantageous regarding the noise emissions in gear applications.

The resistance against liquid and gaseous chemicals is another beneficial characteristic of thermoplastics which makes these materials suitable for gear applications in challenging environments.

3 | DAMAGE TYPES OF POLYMER GEARS

Analogous to steel gears, typical damage characteristics are observed on plastic gears. As with steel gears, a bending stress with corresponding notch effect prevails in the tooth root area, which leads to tooth root fracture if the local bending fatigue strength is exceeded. In the area of the tooth flank, rolling stress can lead to the formation of pittings. While the underlying damage mechanisms are comparable, the mechanical-thermal material properties of plastics and steel differ fundamentally.

The occurring damage mechanisms are strongly dependent on the respective operating conditions. The most relevant damage types of thermoplastic gears are discussed in the following.

3.1 | Thermal damage

Friction in the tooth contact and viscoelastic losses inside the tooth cause plastic gears to heat up during operation. Particularly during dry running, the gears may rise noticeably in temperature.[12] If the melting temperature is exceeded, melting and thermal damages occur, which can result in gear failure. Particularly in dry or starve-lubricated systems, the maximum drive power is limited due to the low thermal conductivity of the plastic gears and the rise of the gear temperatures during operation. Grease lubrication is one possibility to reduce friction in the gear contact and thus minimize friction losses in order to increase the allowable drive power. Operation at oil-lubricated conditions reduce occurring friction losses in the tooth contact. The lubrication film separates the engaging tooth flanks. This way, heating caused by friction losses is reduced. Furthermore, the lubricant itself removes generated heat from the gear contact.

A common method for calculating the tooth temperature is included in VDI 2736.[9]

3.2 | Tooth root breakage

The service life of plastic gears is largely determined by their tooth root load carrying capacity. If local bending fatigue strength is exceeded, cracks occur in the area of the tooth root fillet, which eventually leads to tooth root fracture and, in most cases, to a total failure of the transmission system. The tooth root load carrying capacity of plastic gears is highly influenced by load induced deflections during operation.[3,13] The comparably low Young's modulus of thermoplastics leads to large tooth deflections under load in comparison to steel gears. The commonly used guideline for plastic gear design VDI 2736[9] is derived from DIN 3990[14] which was developed for the calculation and design of steel gears. The approaches used in DIN 3990[14] and VDI 2736[9] neglect the load induced deformations of the gear teeth which affect the tooth root stress. The effect of the stress reduction in the tooth root area due to deformation and increased actual contact ratio is not considered. Hasl[3] conducts comprehensive theoretical and experimental investigations on the tooth root load carrying capacity of plastic gears and derives a calculation approach to consider load-induced deformations.

3.3 | Frictional wear

Frictional wear occurs particularly in unlubricated dry running conditions in plastic gears. During tooth contact, the tooth surfaces are not separated by a lubricant film, resulting in direct contact of the tooth surfaces. Continuous material abrasion leads to a reduction in the tooth cross-section, which can subsequently result in fracture of the remaining profile.[15] Operation in oil-lubricated conditions can significantly reduce the abrasive wear occurring, as the tooth flanks can be separated from each other by the lubricating film. In VDI 2736[9] a calculation approach to determine the averaged linear wear of dry running plastic gears is included.[1]

3.4 | Pitting

When operating under oil-lubricated conditions, the transmitted power can be significantly increased compared to dry running.[12] Extensive experimental research[3,12] shows that thermal damages and abrasive wear can be prevented under oil-lubricated test conditions. However, fatigue damages such as pitting on the active tooth flank become more relevant.[1,4,6] Pitting results from material fatigue in the flank area subjected to contact stresses. Pitting damage typically occurs in areas of high flank pressure below the pitch diameter in areas of negative specific sliding. VDI 2736[9] contains approaches to calculate the flank load carrying capacity of plastic gears. The pitting formation mechanism strongly resembles the generation of pittings known from steel gears where pittings also occur in areas of high flank pressure and negative specific sliding. Furthermore, the optical appearance of pittings on plastic gears resemble those on steel gears.

3.5 | Influence of meshing interference

Due to their low stiffness, plastic gears tend to exhibit comparatively large tooth deformations under load. The tooth deformations result in an extension of the path of engagement and an increase in the contact ratio under load, which can have a positive effect on the tooth root load carrying capacity due to the load distribution over several tooth pairs. With regard to flank shear, however, Hasl[3] points out a possible negative influence on the flank load carrying capacity, especially for steel-plastic gear pairings. The load-induced deflections of plastic gears may cause meshing interferences, which lead to local pressure peaks in the area of premature and posterior meshing. These meshing errors can have a negative effect on the flank load carrying capacity of the respective plastic gears. Figure 1 shows an exemplary flank shear fracture caused by high flank pressures due to meshing interference in the area of posterior meshing at the dedendum utilization diameter d_{Nf}.

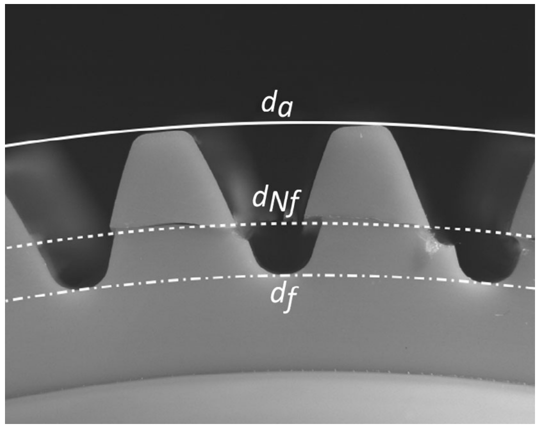

FIGURE 1 EXEMPLARIC flank shear fracture at a peek gear[3]

Adequate measures to reduce high flank pressures such as a tip relief on the driving (steel) pinion must be applied in order to ensure high flank load carrying capacity and to establish reliable strength data for new thermoplastic materials such as PEEK.[4]

4 | CALCULATION OF THE FLANK LOAD CARRYING CAPACITY OF POLYMER GEARS

The approaches to determine the tooth flank load capacity for plastic gears are derived from those developed for the calculation of steel gears according to DIN 3990.[14] In Equations (1) to (3) the calculation approach to determine the flank load carrying capacity of thermoplastic involute gears according to VDI 2736[9] is shown. The flank load carrying capacity is temperature and load cycle dependent. The rolling contact fatigue strength σ_{HlimN} thereby differs for the different grades of thermoplastics.[1]

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TABLE 1 Material properties of Vestakeep 4000G[8] and 5000G[10]

Property	Unit	Vestakeep 4000G	Vestakeep 5000G
Density (23 °C)	g/cm^3	1,30	1,30
Charpy-impact energy (23 °C)	kJ/m^2	7	9
Poisson's ratio (23 °C)	-	0.4	0.4
Thermal elongation coefficient	10^{-6} K^{-1}	60	60
Melting point	°C	ca. 340	ca. 340

$$\sigma_H = Z_E \cdot Z_H \cdot Z_\varepsilon \cdot Z_\beta \cdot \sqrt{\frac{F_t \cdot K_H}{b_w \cdot d_1} \cdot \frac{u+1}{u}} \leq \sigma_{HP}, \tag{1}$$

$$\sigma_{HP} = \sigma_{HlimN} \cdot Z_R / S_{Hmin}, \tag{2}$$

$$\sigma_{HlimN} = f(\vartheta_{Fla}, N_L). \tag{3}$$

The occurring flank pressure σ_H is compared to the permissible flank pressure σ_{HP}. According to VDI 2736[9] a minimum safety factor of $S_{Hmin} = 1.25$ for intermittent operation at N_L load cycles is required. For continuous operation at $N_L > 10^8$ load cycles a minimum safety factor of $S_{Hmin} = 1.4$ is required. The strength values σ_{HlimN} differ for the various grades of thermoplastics. Until now, VDI 2736[9] only contains strength values for a limited number of thermoplastics such as PA66 and PBT.[1] For PEEK grades no strength values are included in VDI 2736.[9]

5 | MATERIALS, TEST GEARS AND EXPERIMENTAL SETUP

5.1 | Gear materials

The plastic test gears for the experimental investigations are made of the nonreinforced high-performance plastics Vestakeep 4000G and Vestakeep 5000G. The two polyether ether ketones (PEEK) show very good mechanical properties even at elevated material temperatures. The glass transition temperature is in the range of approx. 150 °C. The melting temperature of more than 300 °C is significantly higher than those of typical thermoplastic gear materials such as POM and PA66. Table 1 shows further material properties of Vestakeep 4000G and Vestakeep 5000G. The properties resemble each other strongly. However, Vestakeep 5000G shows a slightly higher charpy-impact energy.

Figure 2 shows the temperature-dependent Young's modulus of the two investigated gear materials. Vestakeep 4000G and Vestakeep 5000G show comparable values of the Young's modulus. Material stiffness lowers significantly in the range of the glass transition temperature.

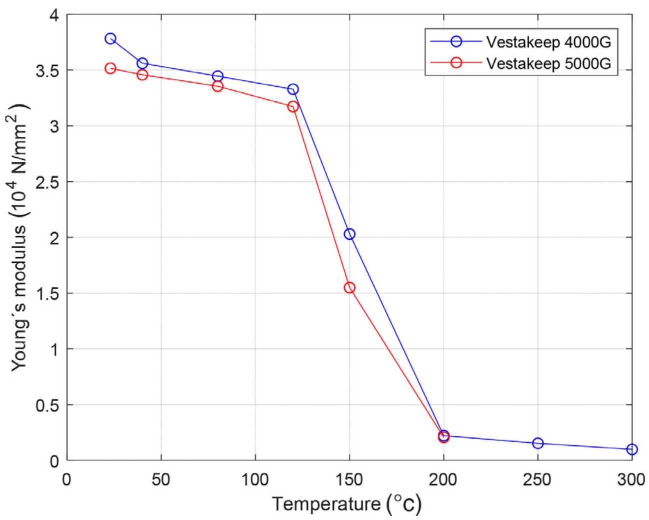


FIGURE 2 Temperature dependent Young's modulus of Vestakeep 4000G and Vestakeep 5000G according to ISO 527[5,16,17] [Color figure can be viewed at wileyonlinelibrary.com]

TABLE 2 KST-C test geometry main data

		Pinion	Gear
Normal module	(mm)	3	
Normal pressure angle	(°)	20	
Helix angle	(°)	0	
Face width	(mm)	22	20
Number of teeth	(–)	24	36
Tip diameter	(mm)	78.32	114.7
Usable tip diameter	(mm)	78.32	113.9
Tip edge radius	(mm)	-	0.75
Tip relief amount	(μm)	≈ 400	-
Modification length	(mm)	2	-
Transverse contact ratio	(–)	1.20	
Center distance	(mm)	91.5	
Material	(–)	Steel	Plastic

5.2 | Test gears

The herein performed experimental investigations are carried out in accordance with previous investigations[3,4,18] using test gears of a transmission ratio of

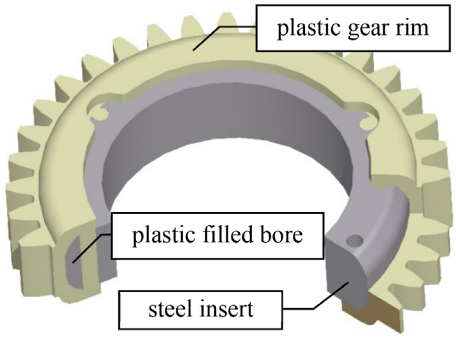


FIGURE 3 Design of plastic test gears[19] [Color figure can be viewed at wileyonlinelibrary.com]

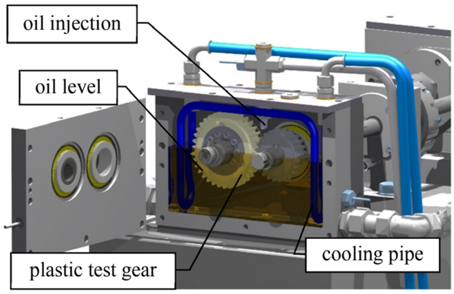


FIGURE 4 Test gear housing of FZG test rig according to References 18,21 [Color figure can be viewed at wileyonlinelibrary.com]

i = 1.5 at a design size of m_n = 3 mm (KST-C). The detailed specifications of the test gear geometry are documented in Table 2. Plastic gears are molded onto a steel insert. The steel insert is fixed to the test rig shaft by means of a conical adapter. This ensures a reliable shaft-hub connection. Holes distributed evenly around the circumference of the steel insert are filled with plastic during the injection molding process and ensure a sufficiently strong positive fit between plastic and steel insert. As a result of the injection molding process, the plastic gear features a tip edge radius of 0.75 mm. The detailed design of the plastic gear and the steel insert is shown in Figure 3.

All experimental investigations are performed using a case hardened and ground 16MnCr5 steel pinion. The steel pinion is designed with a circular tip relief to prevent premature damage on the plastic gear caused by meshing interference. The tip relief ensures that the sharp-edged tip edge of the steel pinion cannot engage with the dedendum flank area of the plastic gear, which would lead to significant pressure peaks in this area. The use of a tip relief significantly lowers the pressure in the extended area of contact and guarantees that the maximum Hertzian pressure is present in the area of single tooth contact. The results of a performed theoretical study of the pressure distribution along the path of contact for gears with and without a tip relief are presented in clause 5. The steel pinions reach surface roughness values of Ra ≤0.4 μm and gear qualities of $Q \leq 6$ according to DIN 3962.[20] The plastic gears achieve gear qualities of $Q \approx 12$. The polymer test gears exhibit roughness values of $Ra = 0.1$ μm. The roughness of the flanks is evaluated as the mean value of three teeth around the circumference measured in the direction of the involute in the middle of the face width.

5.3 | Experimental setup

All experimental investigations are performed at the FZG standard back-to-back test rig according to DIN ISO 14635.[21] The test rig with mounted test gears is shown in Figure 4. The test rig is run in back-to-back configuration at a fixed test torque. The electric engine only provides the occurring losses of the closed power loop.

TABLE 3 Main data FVA 3A[22]

Nomenclature	Value/type
Kin. viscosity at 40 °C	92 to 99 mm^2/s
Kin. viscosity at 100 °C	10.4 to 10.9 mm^2/s
Viscosity index VI	95
Density at 15 °C	0.879 to 0.886 kg/dm^3
Additive	4% Anglamol 99

The test rig can be run in different lubrication configurations. Beside dry running, grease and oil lubrication can also be configured and tested. Using oil lubrication, either oil sump or oil injection lubrication can be applied. Heating cartridges and cooling pipes in the gear housing as well as an external tempering unit for the injected oil allow lubricant temperature control in the oil sump. The oil temperature can be controlled within a range of ±3 K. The tests are run at a test gear speed of $n_2 = 2250$ min^{-1} with the steel pinion driving the plastic test gear.

All tests are performed under oil lubricated conditions at an oil-flow rate of 1.5 L/min. The applied lubricant is an ISO VG 100 reference mineral oil (FVA 3 + 4% Anglamol 99) with a sulfur-phosphorus-based additive package. The lubricant main data are shown in Table 3.

6 | THEORETICAL ANALYSIS OF THE FLANK STRESS

Due to the relatively low stiffness of thermoplastic materials, large deformations of the plastic teeth are to be expected under load. The deformation-induced increase in the path of contact results in pressure peaks during posterior engagement. Due to the relatively small radius

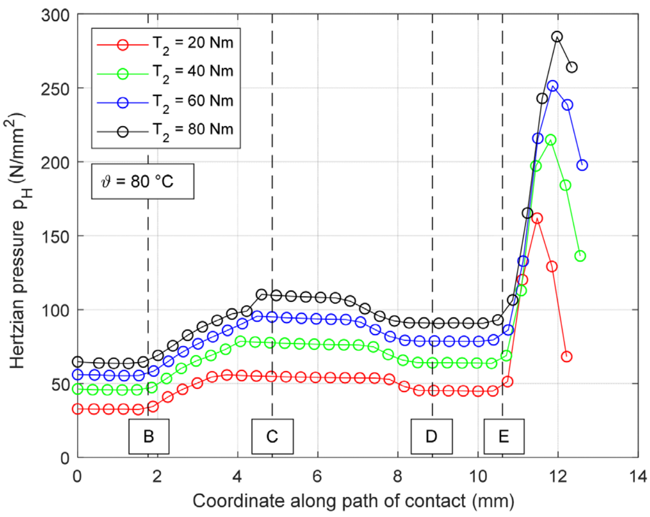


FIGURE 5 Influence of load on Hertzian pressure along path of contact, KST-C, unmodified, $E2 = 3300$ N/mm^2[24] [Color figure can be viewed at wileyonlinelibrary.com]

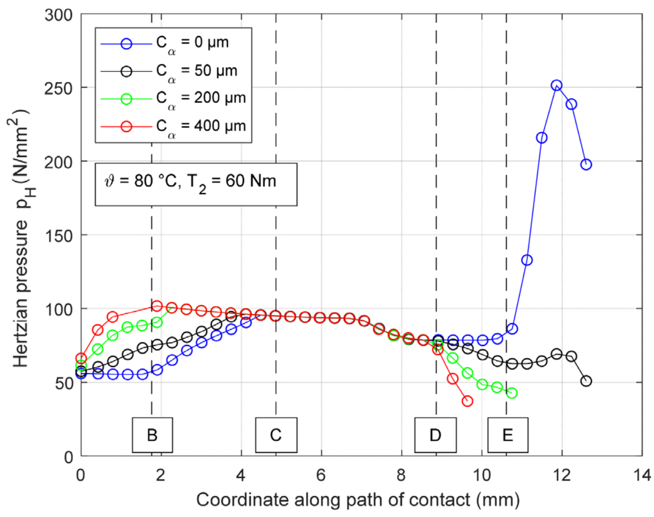


FIGURE 6 Influence of pinion profile tip-modification on Hertzian pressure along path of contact, KST-C, $E2 = 3300$ N/mm^2[24] [Color figure can be viewed at wileyonlinelibrary.com]

of curvature of the tip edge of the steel pinion, comparatively high flank pressures are obtained even at low torques. Load distribution and flank pressure is calculated using the state of the art gear calculation programme RIKOR.[23] RIKOR is able to calculate tooth deformation and load-distribution for different positions along the path of contact. Furthermore, it is possible to calculate flank pressure, considering the radius of curvature and profile modifications. Figure 5 shows the course of the Hertzian pressure along the path of contact calculated with RIKOR for an exemplary, unmodified steel-plastic (PEEK) gear pair at different torques. The main gear data of the gear pair can be taken from Table 2.

Even at low torques, the computed Hertzian pressure at the end of path of contact is significantly higher than the calculated flank pressure in the area of the pitch point. At even higher loads, the path of contact is further extended due to increasing tooth deformation, while the pressure values in the posterior meshing area continue to increase significantly. Without appropriate profile modifications on the driving steel pinion in order to reduce the stress peaks that occur, operation can lead to premature damage in the tooth root flank area of the plastic gear and ultimately to a failure of the entire gear unit.[24]

Figure 6 shows the influence of a long (2 mm at path of contact), circular tip relief of the driving steel pinion on the resulting tooth flank pressure for an exemplary steel-plastic (PEEK) gear pair also calculated using RIKOR. Whereas the Hertzian pressure in the posterior meshing area for an unmodified steel pinion is considerably higher than the flank pressure in the area of the pitch point, the contact pressure increase during posterior meshing can be significantly reduced by the use of a tip relief on the steel pinion.

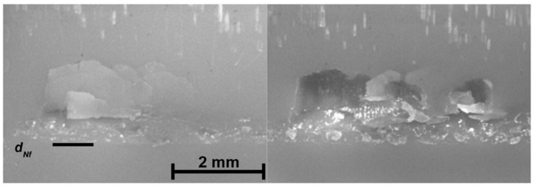
However, the amounts of the tip relief required for this are substantially higher than the usual correction amounts for conventional steel gears, which are usually in the range $C_{\alpha\alpha} \leq 50$ μm. For the calculation example shown in Figure 6, the flank pressure in the posterior meshing area can be reduced to the pressure level in the area of the pitch point by a reduction amount of $C_{\alpha\alpha} = 50$ μm. However, a further reduction of the pressure to a significantly lower pressure level is only achieved by applying considerably larger relief amounts in the range $C_{\alpha\alpha} \geq 200$ μm.[24]

7 | EXPERIMENTAL RESULTS

7.1 | Damage characteristics

When operating plastic gears in adequate oil-lubricated conditions, damage such as melting and abrasive wear become secondary. Instead, fatigue damage such as pitting is relevant with regard to the service life of the gears. Experimental investigations show formation of pitting after several million load cycles.[4] The material breakouts increase with progressive running time until the tooth flank is completely deteriorated.[4] Figure 7 shows the generation of a pitting damage of an exemplary plastic test gear at different load cycles. After exceeding the local fatigue strength, small debris breaks out of the tooth surface. The pitting area increases as the test run is continued.

Visually, pitting damage on the investigated plastic gears strongly resembles the formation of pitting on steel gears. In Figure 8, the specific sliding conditions of both pinion and wheel of the investigated test gear geometry



13·10⁶ Load cycles 15·10⁶ Load cycles

FIGURE 7 Exemplary generation of a pitting damage on a plastic (Vestakeep 5000G) gear, KST-C, injection molded[24]

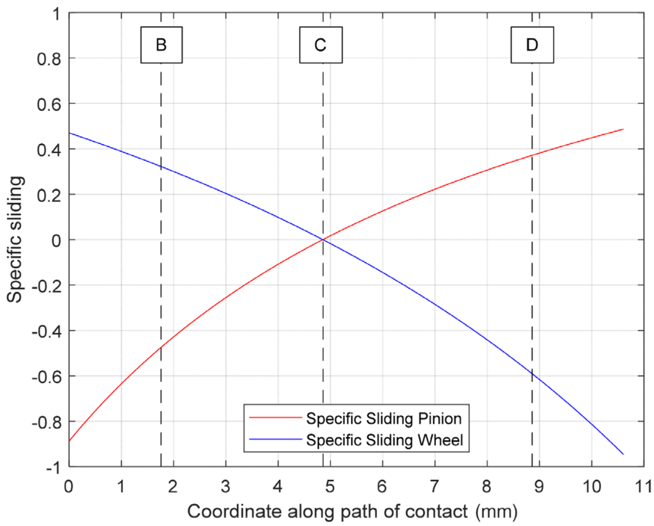


FIGURE 8 Specific sliding on pinion and wheel, KST-C[24] [Color figure can be viewed at wileyonlinelibrary.com]

KST-C are shown. Analogous to steel gears, pitting occurs preferentially in the dedendum flank area below the pitch point in areas of negative specific sliding.

Figure 9 shows the calculated Hertzian flank pressure along the path of contact for the experimentally investigated test gears at a torque of $T_2 = 43$ Nm. The flank pressure is calculated using RIKOR. Effects due to premature meshing are neglected since the plastic wheel utilizes a tip edge radius. The course of Hertzian pressure peaks in the area of single tooth contact where pitting damages are expected. Towards the end of the contact, Hertzian pressure decreases since the tip relief of the steel pinion operates in this area.[4]

In addition to investigating the damage behavior with test gears of test geometry KST-C manufactured from injection-molded Vestakeep 5000G and Vestakeep 4000G, spot tests are also carried out with test gears manufactured alternatively in the machining process (milled gears). Furthermore, test gears of alternative, helical geometries are tested. Objective of these investigations is to transfer the findings on damage behavior to alternative geometries and manufacturing processes and to verify the applicability of the conclusions regarding damage behavior.

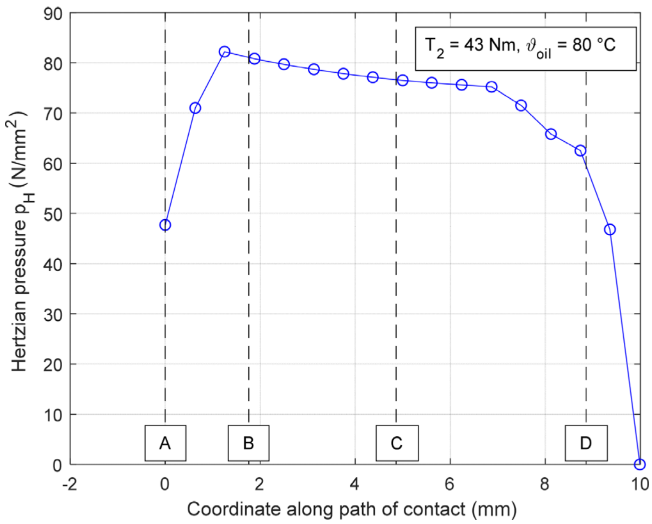
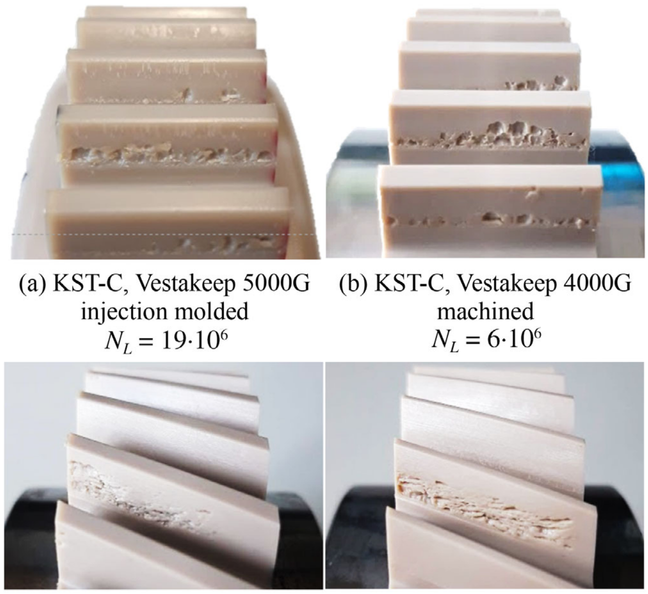


FIGURE 9 Hertzian pressure along path of contact, KST-C[4] [Color figure can be viewed at wileyonlinelibrary.com]



(a) KST-C, Vestakeep 5000G injection molded $N_L = 19 \cdot 10^6$

(b) KST-C, Vestakeep 4000G machined $N_L = 6 \cdot 10^6$

(c) KST-CS, Vestakeep 4000G machined $N_l = 12 \cdot 10^6$

(d) LL30, Vestakeep 4000G machined $N_l = 10 \cdot 10^6$

FIGURE 10 Comparable, exemplary pitting damages on different gear geometries, manufactured by injection molding & machining[24] [Color figure can be viewed at wileyonlinelibrary.com]

In addition to investigations on the damage behavior of a milled variant of geometry KST-C, investigations are also carried out on a conventional machined helical gear geometry (KST-CS) as well as on a loss-optimized low-loss gear geometry (LL30). A qualitative comparison of the observed damage characteristics is shown in Figure 10. Table 4 shows the corresponding basic data of gear geometries KST-CS and LL30.

Geometry variant		KST-CS		LL30	
		Pinion	Gear	Pinion	Gear
Normal module	(mm)	2.75		2.75	
Normal pressure angle	(°)	20		30	
Helix angle	(°)	25.6		26.0	
Face width	(mm)	22	20	22	20
Number of teeth	(–)	24	36	24	36
Tip diameter	(mm)	78.8	114.7	78.2	114.1
Center distance	(mm)	91.5		91.5	
Material	(–)	Steel	Plastic	Steel	Plastic

TABLE 4 KST-CS & LL30 geometry basic data

A comparison of the damage patterns presented shows that pitting damage occurs on all the test gears investigated. The damage occurs for both Vestakeep 4000G and Vestakeep 5000G materials (Figure 10a,b). Furthermore, the pitting damage is observed for both injection molded (a) and machined (b) test gears, regardless of the manufacturing process. The same applies to the conventionally helically geared variant KST-CS (c) and the loss-optimized variant LL30 (d). Damages are similar both visually in their appearance and in the location of their occurrence in the dedendum flank area below the pitch point in areas of high Hertzian pressure and negative specific sliding. Different geometry parameters such as helical design and low-loss design with reduced sliding speeds can affect lifetime. However, current guidelines such as VDI 2736[9] neglect the influence of helical geometry on the flank load carrying capacity. Further investigations on the influence of the helix angle on the gear lifetime are necessary.

7.2 | Load carrying capacity

Material strength of thermoplastic materials decreases with increasing temperature. In order to investigate the influence of the material temperature on the pitting load carrying capacity, comprehensive service life tests were carried out with injection-molded test gears with test geometry KST-C at two different temperature levels. Furthermore, the aim was to generate initial strength parameters with regard to the flank load carrying capacity of oil-lubricated PEEK materials. Test runs were performed as single-stage Wöhler tests at a load level of $\sigma_H = 77$ N/mm^2 ($T_2 = 41$ Nm). Failure criterion is met when 8% of the active flank surface area of a single tooth is damaged by pitting.

The results of the conducted fatigue tests with a material temperature of 80 and 120 °C, respectively, are shown in Figure 11. Results of the load carrying capacity investigations show a clear influence of the gear temperature on

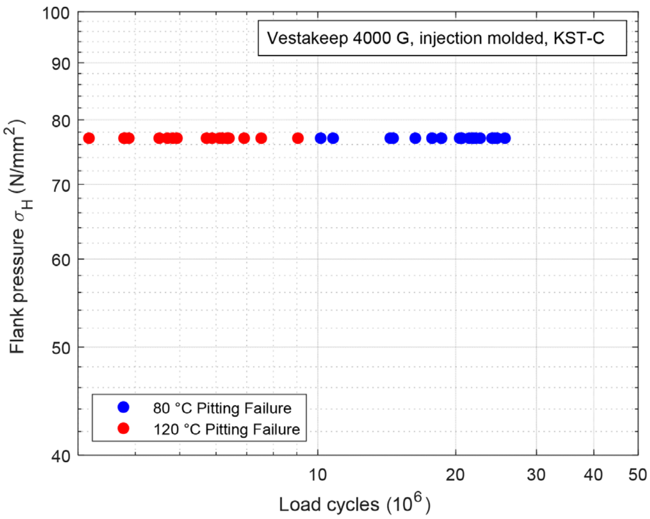


FIGURE 11 Influence of material temperature on flank load carrying capacity, Vestakeep 4000G, injection molded, KST-C [Color figure can be viewed at wileyonlinelibrary.com]

the achievable pitting lifetime. As expected, the achievable fatigue lifetime is reduced with an elevation of the material temperature. However, operation at temperatures of approx. 120 °C is possible with PEEK (e.g., as balancer gear in combustion engines) whereas other typical gear materials such as POM and PA66 can only be operated up to temperatures about 90 °C. The scatter of the test results for both temperature levels investigated is within the usual range for fatigue life investigations.[25]

8 | CONCLUSION

This article gives an overview on the relevant main failure modes of thermoplastic gears. While thermal damage and abrasive flank wear are primarily relevant in dry running, fatigue damage such as pitting dominates under adequate oil-lubricated conditions, provided that sufficient tooth root load carrying capacity can be guaranteed.

During operation, load-induced deformations can lead to pressure peaks in the extended meshing area, which can negatively affect the durability of the plastic gears. To ensure optimum flank load carrying capacity, adequate profile modifications have to be made in order to positively influence the course of flank pressure and to avoid stress peaks during posterior meshing. Due to the large deformation of the gear teeth resulting from low stiffness, comparably large tip reliefs have to be utilized in order to reduce stress peaks.

Experimental investigations with injection molded and machined test gears with different gear geometries in back-to-back tests show that comparable damage characteristics occur in oil-lubricated test conditions. The occurring pitting damages strongly resemble those known from steel gears both in optical appearance and in the local occurrence in areas of high flank pressure and negative specific sliding below the pitch circle.

Extensive pitting lifetime investigations at different temperature levels show a strong influence of the material temperature on the resulting pitting lifetime. With increasing material temperature, a significant decrease of the achievable lifetime can be obtained.

Future research will focus on further generation of strength properties at multiple load levels and temperatures in order to generate full S-N-curves. As a result, reliable strength parameters for the tooth flank carrying capacity of oil-lubricated PEEK gears will be available in the future, contributing to a safe and economical gear design.

NOMENCLATURE

b_w	common face width of the gear pair in mm
d_1	reference diameter in mm
d_{Nf}	dedendum utilization diameter in mm
F_t	tangential force in N
i	transmission ratio
K_H	factor for tooth flank loading
m_n	normal module in mm
n_2	test gear speed in 1/min
N_L	number of load cycles
S_{Hmin}	required minimum safety factor (flank)
T_2	wheel torque in Nm
u	transmission ratio
Z_E	elasticity factor in $\sqrt{N/mm^2}$
Z_H	zone factor
Z_R	surface roughness factor
Z_ε	contact ratio factor
Z_β	spiral angle factor
ϑ_{Fla}	tooth flank temperature in °C
σ_H	flank pressure in N/mm^2
σ_{HP}	permissible flank pressure in N/mm^2
σ_{HlimN}	rolling contact fatigue strength in N/mm^2

AUTHOR CONTRIBUTIONS

Christopher Martin Illenberger: Writing – review and editing (lead). **Thomas Tobie:** Supervision (supporting). **Karsten Stahl:** Supervision (supporting).

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DATA AVAILABILITY STATEMENT

The data that support the findings of this study are available from the corresponding author upon reasonable request.

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