



Application of high performance polymer gears in light urban electric vehicle powertrains

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Abstract

The application of high-performance polymer gears in power transmissions is increasingly gaining focus in current research and industrial practice. The well-known advantages of thermoplastics such as cost-efficient production and low density compete with moderate strength and limited operating temperatures. However, the properties of thermoplastics can be used effectively to address the challenges of more efficient powertrains with higher power density, reduced fuel consumption, and reduced manufacturing costs. This work focuses on the implementation of high-performance polymer gears in the powertrain of a light electric vehicle. For demonstration purposes, a series transmission of a prototype of a light electric vehicle is modified and investigated considering the specific properties of polymers and steel materials. Due to its material properties, the high-performance polymer VESTAKEEP® 5000 G was used for the investigations. Different constructional modifications to the serial transmission as well as modifications of the micro geometry of the gears were applied to improve the load distribution and provide sufficient load carrying capacity to the polymer gears. Assuming usual driving cycles and lifetime, the calculations according to VDI 2736 [1] and RIKOR [2] indicate a sufficient load-carrying capacity for the polymer gears used inside. Under the given assumptions, the use of polymer gears in a powertrain of light electric vehicles appears to be possible.

Einsatz von Hochleistungskunststoffen im Antriebsstrang von urbanen Elektrokraftfahrzeugen

Zusammenfassung

Der Einsatz von Zahnrädern aus Hochleistungskunststoffen in Anwendungen mit höherer Leistungsübertragung rückt zunehmend in den Fokus der aktuellen Forschung und industriellen Praxis. Den bekannten Vorteilen thermoplastischer Kunststoffe wie kostengünstiger Herstellung und niedriger Dichte stehen niedrige Festigkeiten und begrenzte Betriebstemperaturen gegenüber. Die Eigenschaften thermoplastischer Kunststoffe können jedoch genutzt werden, um die Herausforderungen effizienterer Antriebsstränge mit höherer Leistungsdichte, geringerem Kraftstoffverbrauch und reduzierten Herstellungskosten zu begegnen. Diese Arbeit konzentriert sich auf die Implementierung von Zahnrädern aus Hochleistungskunststoffen in den Antriebsstrang eines urbanen Elektrofahrzeugs. Zu Demonstrationszwecken wird das Seriengeräte eines Prototyps eines Elektrokraftfahrzeugs modifiziert und unter Berücksichtigung der spezifischen Materialeigenschaften von Kunststoff- und Stahlwerkstoffen untersucht. Aufgrund seiner Materialeigenschaften wurde hierfür der Hochleistungskunststoff VESTAKEEP® 5000 G für die Untersuchungen verwendet. Verschiedene konstruktive Modifikationen am Seriengeräte sowie Modifikationen an der Mikrogeometrie der Zahnräder wurden vorgenommen, um die Lastverteilung zu verbessern und den Kunststoffzahnrädern damit eine ausreichende Tragfähigkeit zu verleihen. Die Berechnungen nach VDI 2736 und RIKOR zeigen unter der Annahme üblicher Fahrzyklen und Lebensdauer eine ausreichende Tragfähigkeit für die darin eingesetzten Kunststoffzahnräder auf. Unter den gegebenen Annahmen erscheint der Einsatz von Kunststoffzahnrädern in einem Antriebsstrang von Elektrokraftfahrzeugen deshalb grundsätzlich möglich.

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1 State of knowledge

Polymer gears are constantly expanding their areas of application in the field of drive technology due to various advantages such as the absence of corrosion, good damping properties, and the possibility of cost-efficient mass production of parts using the injection molding process. Moreover, against the background of current challenges of climate change, the low material density of polymers offers great lightweight construction potential, which can be used in particular to save damaging greenhouse gases with regard to the complete process chain. However, the significantly lower mechanical strength and low stiffness compared to steel pose major challenges. Furthermore, the high temperature dependence of the material properties leads to limited operating temperatures in applications. Exceptions are high-performance polymers such as PEEK, which can also be operated at higher temperature regimes. Wear and low thermal conductivity, especially in dry running, limit the service life of polymer gears [3, 4]. Therefore, in applications with higher power transmission, a simple substitution of steel gears by polymers is usually not possible [5, 6].

Thermoplastics are predominantly used in transmission systems designed to transmit motion so far, like [7]:

- Control of electric brake systems (parking brake)
- Electric steering systems
- Gear pumps
- Powertrain of electric two-wheelers

The wide application of polymer gears as power-transmitting machine elements in the automotive sector is not known yet. However, polymer gears are already being used increasingly in the powertrain of electrified bicycles. There, gears made of thermoplastic materials can be found as driving machine elements that transmit the power of the electric motor to the wheels via a transmission supporting the driver. By using modern polymers and under oil lubrication, the power density can be increased many times compared to dry running [4]. According to [8], this results in a “soft” (thermo-) elastohydrodynamically lubricated (TEHL) contact under load characterized in particular by the high compliance of the plastic material. Numerical calculations according to [9] show a comparatively low hydrodynamic pressure in the rolling-sliding contact, which prevents a significant pressure-induced increase in the lubricant viscosity and can influence the friction behavior positively. Thus, experimental studies in [10, 11] have shown that very low coefficients of friction in the range of superlubricity are technically possible in the lubricated polymer contact.

Due to various advantages such as insensitivity to changes in center distance and broad basic knowledge available from the steel sector, polymer gears are usually

designed as involute cylindrical gears. Depending on the requirements and application, spur, helical or internal gears are used. Currently, the design, as well as the calculation of the load-carrying capacity of polymer gears, is carried out according to VDI 2736 [1], the actual valid guideline for the load-carrying capacity calculation of thermoplastic cylindrical gears. VDI 2736 [1] provides calculation approaches for the tooth root stress (see Eq. 1), which is based on the theory of a fixed bending bar. To calculate the tooth root load carrying capacity, the single teeth of a plastic gear are represented by the model of a fixed cantilever beam. The area of maximum bending stress appears in the area of the tooth root at the contact point of 30°-tangent and the tooth root radius. According to VDI 2736 [1] the flank pressure can be calculated by Eq. 2, following the approach of the Hertzian pressure theory. This guideline is largely based on DIN 3990 [12, 13] developed for steel gears and tries to consider the different material properties of polymers compared to steel. However, the influences due to the high compliance of polymer gears, such as the increasing contact ratio under load, are not adequately represented in the current guideline as shown in [14, 15].

$$\sigma_F = K_F \cdot Y_{Fa} \cdot Y_{Sa} \cdot Y_\varepsilon \cdot Y_\beta \cdot \frac{F_t}{b \cdot m_n} \quad (1)$$

$$\sigma_H = Z_E \cdot Z_H \cdot Z_\varepsilon \cdot Z_\beta \cdot \sqrt{\frac{F_t \cdot K_H}{b_w \cdot d_1} \cdot \frac{u+1}{u}} \quad (2)$$

Calculating the service life of polymer gears regarding the failure mode tooth root breakage or pitting, the damage accumulation hypothesis by Palmgren und Miner [16] can be used. This assumes that every dynamic load contributes to the damage process. According to the theory, failure of the component occurs as soon as the damage sum reaches $s > 1$. For calculation of the damage sum s , for each load level i the number of occurring load cycles (n_i) is determined and compared with the number of bearable load cycles (N_i) from the S/N-curve. The quotient of the occurring number of load cycles to the bearable number of load cycles is the partial damage d_i of each load level. According to the damage accumulation hypothesis by Miner [16], the sum of all partial damages leads to the damage sum s of the total amplitude collective.

$$s = \sum d_i = \frac{n_i}{N_i} \quad (3)$$

2 Aim of this work

Polymer gears are gaining importance, particularly due to the described advantages. As shown in [4] the use of modern polymers and operation with oil lubrication can increase the power density considerably in comparison with dry

running. This extends the range of applications to power-transmitting powertrains in small electric vehicles, for instance. Against this background, the objective of this study is to develop and evaluate a transmission demonstrator for a small electric vehicle using high-performance thermoplastics gears next to the conventional steel gears.

3 Serial test vehicle

The transmission demonstrator developed in this work is intended to be applied in a small urban electric vehicle the ILO1 developed by Emm! solutions GmbH, which is classified in the L7e vehicle class according to § 20 Absatz 3a Satz 4 Anlage XXIX StVZO. The prototype of this concept car is based on the Renault Twizy and offers a development and measurement platform for a new mobility concept with focus on autonomous driving vehicles (see Fig. 1). Vehicles in this class are restricted to a maximum empty mass of $m=450\text{ kg}$ and to a maximum drive power of $P=15\text{ kW}$. The serial gear unit used in ILO 1 is a fixed-speed two-stage steel helical gearbox made out of steel components.



Fig. 1 ILO 1 serial test vehicle

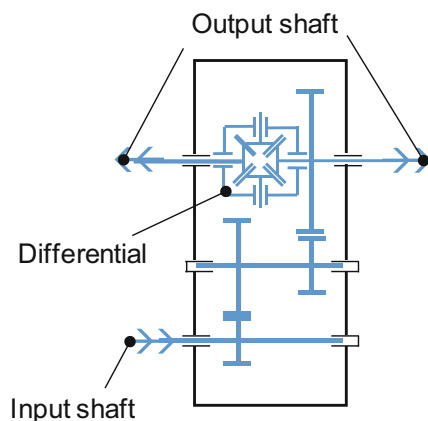


Fig. 2 Schematic illustration of the series gearbox

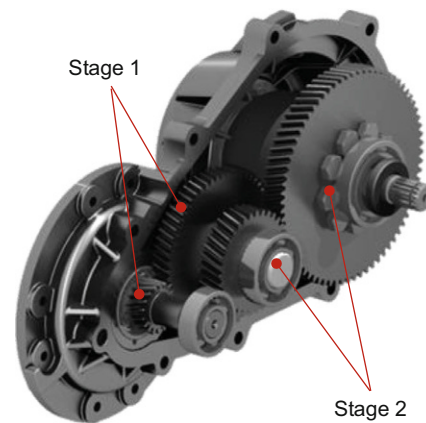


Fig. 3 Serial steel gearbox

The estimated electric range is about 50–80 km depending on the driving style.

Fig. 2 and 3 shows the schematic illustration and the components of the two-stage series cylindrical gear transmission in the opened condition. The electric machine delivers its torque directly to the pinion of stage 1. Stage 1 consists of two cylindrical gears out of steel, with teeth numbers of $z_{11}=14$ and $z_{12}=61$ ($i_1 \approx 4.36$). Torque is transmitted to the rear wheels via the output shaft of stage 2 consisting also of two steel cylindrical gears with $z_{21}=34$ and $z_{22}=72$ ($i_2 \approx 2.12$). In overrun mode or during braking, power is transmitted backwards from the wheels to the electric machine, which then acts as a generator. The total transmission ratio of the gear unit is $i_{\text{total}}=9.23$. The expected service life, which was taken into account in the design, is 60,000 km according to the manufacturer's specifications. The recommended lubricant is a gear oil of type 75W-90 API GL-4+.

4 Definition of performance parameters

For the basic dimensioning of the powertrain presented, it is initially necessary to determine the load spectra to which the powertrain is subjected. For this purpose, the Worldwide Hardened Light Vehicles Test Procedure (WLTP) [17] cycle is used. By specifying a route, speed, and acceleration profile, load spectra can be derived from which torque requirements and the expected service life can be estimated. Depending on the power-to-weight ratio of the vehicle, the assumed route profiles differ slightly. Due to an empty mass power-to-weight ratio of $\leq 34\text{ W/kg}$, the ILO 1 is classified in class 2 of the WLTP cycle. This results in the speed profile shown in Fig. 4 as a function of time. A total of three stages are passed through, which differ in terms of the maximum velocity that is attained [17]. This test cycle was chosen to represent the vehicle's driving perfor-

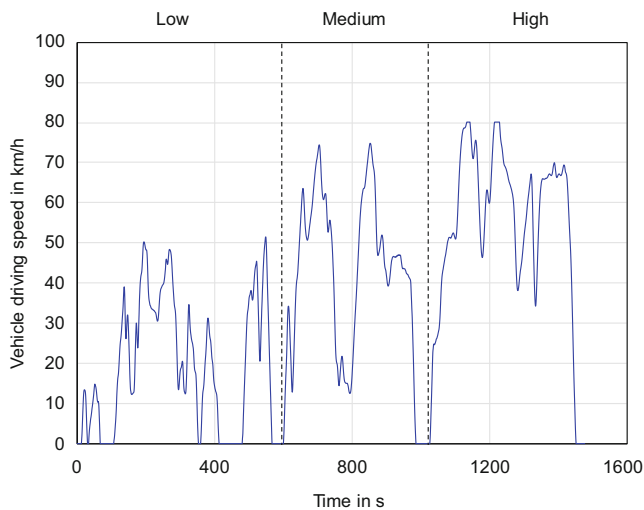


Fig. 4 WLTP test cycle of class 2 vehicles

mance as realistically as possible. With the profile “high”, which is adapted to the driving performance of the ILO 1 ($v_{\max} \approx 80 \text{ km/h}$), generally more critical conditions are assumed for a more conservative gear design.

The associated torque and speed characteristics at the gearbox output for the conditions assumed in this study can be derived from the WLTP. To determine the appearing torque on the gear teeth, the total vehicle resistance F_{Vehicle} has to be calculated consisting in total of four components and can be calculated by means of the following equations: the air resistance F_{Air} , the rolling resistance F_{Roll} , the acceleration resistance $F_{\text{Acceleration}}$ and the gradient resistance F_{Gradient} [18, 19]. For further explanations, please refer to [19].

$$F_{\text{Vehicle}} = F_{\text{Air}} + F_{\text{Roll}} + F_{\text{Acceleration}} + F_{\text{Gradient}} \quad (4)$$

$$F_{\text{Air}} = 0.5 \cdot \rho_{\text{Air}} \cdot c_W \cdot A \cdot v^2 \quad (5)$$

$$F_{\text{Roll}} = f_R \cdot m_F \cdot g \cdot \cos \alpha_{\text{Gradient}} \quad (6)$$

$$F_{\text{Acceleration}} = \lambda \cdot m_F \cdot \frac{\Delta v}{\Delta t} \quad (7)$$

$$F_{\text{Gradient}} = m_F \cdot g \cdot \sin \alpha_{\text{Gradient}} \quad (8)$$

In this case, the total driving resistance F_{Vehicle} is calculated in the horizontal. With knowledge of the total driving resistance F_{Vehicle} at the different conditions, the required torques T , can be calculated at the wheels with the following equation [9] by using the dynamic wheel radius r_{dyn} .

$$T = F_{\text{Vehicle}} \cdot r_{\text{dyn}} \quad (9)$$

Applied to the ILO 1, this results in the speed-torque distribution shown in Fig. 5. Each point according to the classification in “Low”; “Medium” and “High” (see Fig. 4) represents an operating condition of the discretized assumed WLTP-test cycle. Finally, the operating characteristic can

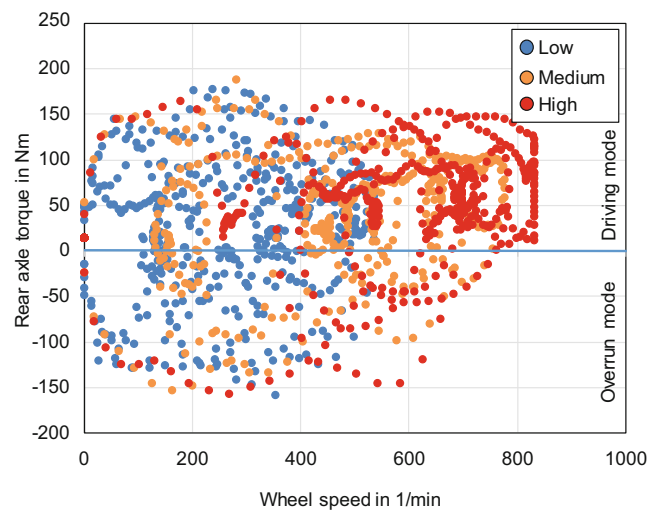


Fig. 5 Torque and speed characteristics at the driven rear axle

be used to determine the requirements for dimensioning the hybrid-polymer-steel transmission.

To calculate the required torque speed and acceleration values are derived from the WLTP cycle while the vehicle data are taken from the manufacturer’s specification.

5 Dimensioning of a polymer-steel hybrid-gearbox

The design of the macro geometry of both gear-stages is performed with the aim of achieving a sufficient load carrying capacity of all cylindrical gears involved. A major challenge in this context is the compliance with further requirements, which are briefly summarized below:

- Application of polymers as gear material for at least one gear-stage
- Comparable driving performance to the serial gear-box
- Further use of the serial gear-housing while retaining the center distances and dimensions of the original serial gear-housing
- Further use of the serial electric machine and application of a two-stage transmission with comparable gear-ratio ($i_{\text{total}} \approx 9$)
- Operation of the newly developed gear-box both in the ILO 1 and on a gear test-rig

The simple substitution of a steel gear by a polymer gear of the same size is generally not possible. Calculations by VDI 2736 [1] show, that the stresses occurring in the tooth root as well as the pressure on the tooth flank result in values that exceed the strength limits of polymers likely leading to problems and premature failure of the polymer gears. For example, the calculation of the flank pressure according to VDI 2736 [1] for an mean drive torque of

Table 1 Properties of polymer materials [22] for different temperatures

Property	VESTAKEEP® 5000 G (at $\vartheta = 23\text{ }^\circ\text{C}$)	VESTAKEEP® 5000 G (at $\vartheta = 80\text{ }^\circ\text{C}$)
Young’s modulus at $23\text{ }^\circ\text{C}$ in N/mm^2	3515	3355
Poisson’s ratio	0.41	0.41
Density in g/cm^3	1.30	1.30

$T = 30\text{ Nm}$ ($\approx 50\%$ of maximum) results already in a flank pressure of $\sigma_H > 200\text{ N}/\text{mm}^2$ in the first stage and is therefore many times higher than the bearable flank pressures of the used polymer.

To enable the use of polymers as gear material, various approaches were pursued to reduce the occurring stresses to a tolerable level. Due to the torque and forces required to guarantee sufficient driving performance, the second gear-stage (output to the wheels) cannot be designed using polymer gears. For this reason, polymer gears are possible only at the first gear-stage.

Based on former experimental investigations [20] the unreinforced PEEK material VESTAKEEP® 5000 G has emerged as the most promising candidate material for this project. It is shown, that under oil-lubricated conditions the load-carrying capacity of this material is almost completely limited by the flank load-carrying capacity. Premature failures due to wear or tooth root fracture or wear could not be observed. Besides lower production costs fiber-reinforced polymers are not considered in this study as they can influence the wear behavior negative, as described in [21]. The main material properties of VESTAKEEP® 5000 G are shown in Table 1 for a ambient temperature of $\vartheta = 23\text{ }^\circ\text{C}$ just as for a relevant operating temperature of $\vartheta = 80\text{ }^\circ\text{C}$.

Based on the results in [20, 21] the occurring flank pressures at the polymer gears in the first gear-stage are the crucial point in this study when designing the powertrain.

Table 2 Main gear geometry parameters

		First stage		Second stage	
		Polymer pinion	Polymer wheel	Steel pinion	Steel wheel
Normal pressure angle	$^\circ$	25		20	
Helix angle	$^\circ$	-8	8	15	-15
Number of teeth	-	19	50	32	109
Normal module	mm	2.0		1.5	
Profile shift coefficient	-	0.20	-0.04	0.31	0.04
Tooth width	mm	52	50	19	18
Addendum diameter	mm	43.2	104.8	53.6	172.4

The use of a steel-polymer pairing with the higher loaded pinion made out of steel results in comparably high flank pressures that prevent the use of polymer in this application. For this reason, the first gear-stage is designed as a pure polymer-polymer pairing in order to reduce the occurring flank pressure to a tolerable level. In addition, the transmission ratios are shifted from the first stage to the steel-steel pairing of the second gear-stage, taking into account the available packaging space. This measure enables the reduction of the tooth forces occurring at the first stage, which has a positive effect on the tooth root and the tooth flank load carrying capacity.

Despite the applied measures, it is necessary to increase the joint tooth width to 50 mm, in particular to reduce flank pressure on the polymer pinion. The resulting widening of the original gearbox was solved by means of an intermediate aluminum spacer which is bolted between the two gear-housing halves. Analogous to the series gearbox, the sealing of the two gearbox halves and the spacer is done with flat paper seals. Despite the additional spacer made of aluminum (weight $\approx 1\text{ kg}$), the total weight remains almost constant compared to the series gearbox.

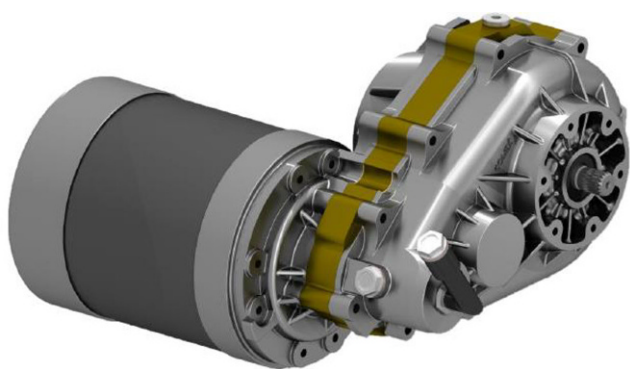


Fig. 6 Adapted gear-housing with intermediate spacer (yellow) and electric machine

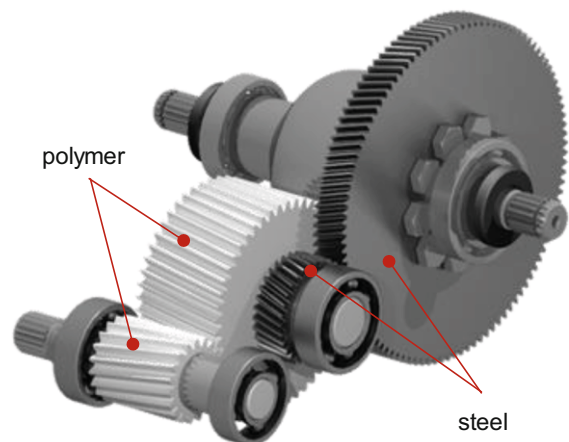


Fig. 7 Polymer steel hybrid gearbox

Fig. 8 Load spectrum and S/N curve of VESTAKEEP® 5000G—linear damage accumulation

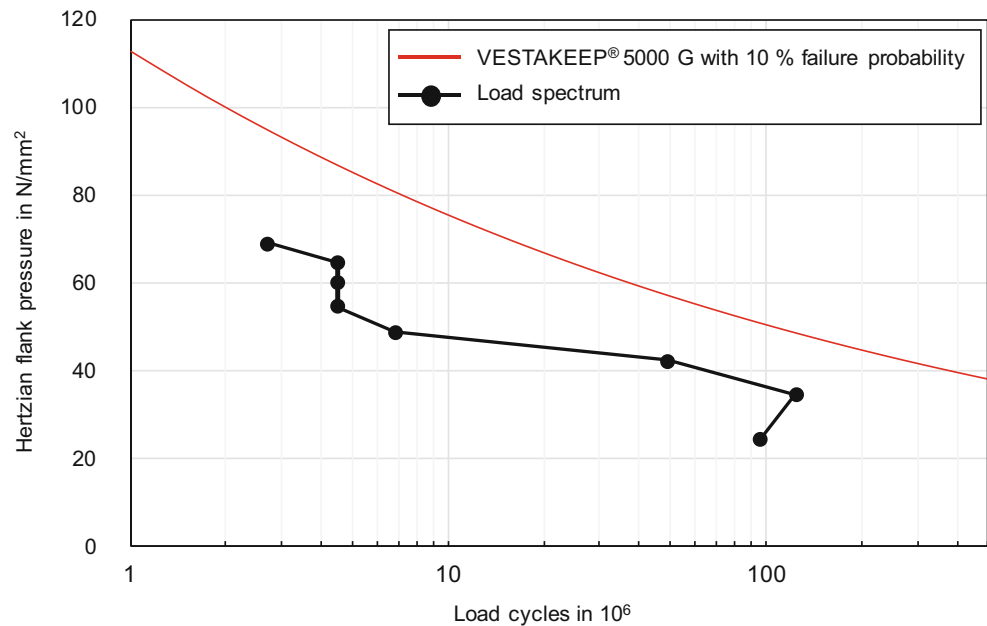


Fig. 6 shows the adapted gear-housing including intermediate spacer. The serial gear-housing halves can still be applied. The use of polymer materials on the second gear-stage and the differential-gear is not possible due to the high load. Both second gear-stage and differential will therefore continue to be machined from case hardened 18CrNiMo7-6 steel material, which ensures sufficient load-carrying capacity. The resulting macro geometries of the two gear stages are shown in Table 2.

Fig. 7 shows the components of hybrid polymer-steel cylindrical gear transmission in the opened condition. Experimental investigations [20] show that the load-carrying capacity of polymer gears made of VESTAKEEP® 5000G under oil-lubricated conditions is almost completely limited by the flank load-carrying capacity. Premature failures due to tooth root fracture or wear could not be observed. Based on these test results, the proof of sufficient tooth flank load capacity is considered crucial in this work and is described in the following. In order to obtain a reliable prediction of the tooth flank load-carrying capacity, the relevant characteristic of the tooth flank load-carrying capacity, the Hertzian pressure, is calculated for all determined load levels using VDI 2736 [1].

Since only limited strength values are available so far for the considered material, results obtained in [20] on the flank load-carrying capacity for the material VESTAKEEP® 5000G are used. The corresponding S/N-curve at a material temperature of $\vartheta_{\text{Material}} \approx 80^\circ\text{C}$ (see Table 1) for a failure probability of 10% is shown in Fig. 8. For higher load cycles the S/N-curve was extended by linear extrapolation. In order to estimate the service life of the polymer gear, the experimental results are compared with the load spectrum determined according to the WLTP-cycle. The correspond-

ing flank pressure σ_H of the polymer gear are calculated according VDI 2736 [1] (see Eq. 2) for the occurring torques for a temperature of $\vartheta = 80^\circ\text{C}$. The corresponding number of load cycles for each flank pressure is also derived from the load spectrum. Analogous to the reference vehicle, the target service life is also 60,000km. Since the WLTP-cycle is mainly designed to determine consumption values, extra loads were assumed to simulate additional acceleration at full load and thus an additional tightening of the collective. The individual operation points are also shown in Fig. 8. Based on the life expectancy of the polymer gear considered here, the stress leads to a resulting damage sum $s \approx 93\%$. Under the conditions described, damages to the tooth flank within the required service life are therefore not to be assumed. The use of the polymer-polymer gear stage seems possible.

Due to the low material stiffness compared to steel materials, the application of polymer gears results in comparably high deformations under load. As a result, the operation of unmodified gears may lead to unexpected transmission errors during premature and posterior meshing as well as to significant pressure peaks which can lead to undesired flank damages. To prevent these effects, profile modifications can be applied to optimize the pressure distribution. The occurring pressure distribution during operation of the polymer-polymer stage is calculated using the load distribution program RIKOR [2]. Fig. 9 shows the calculated pressure distribution of the polymer gear-stage at a maximum input torque of $T = 57\text{Nm}$ before and after applied profile modifications. The wave-shaped pressure distribution on the tooth flank at the beginning and the end of the mesh is attributed to numerical inaccuracies in the software program.

Fig. 9 Hertzian pressure at polymer gear stage for maximum input torque $T = 57 \text{ Nm}$ without (a) and with applied profile modification of $50 \mu\text{m}$ (b)

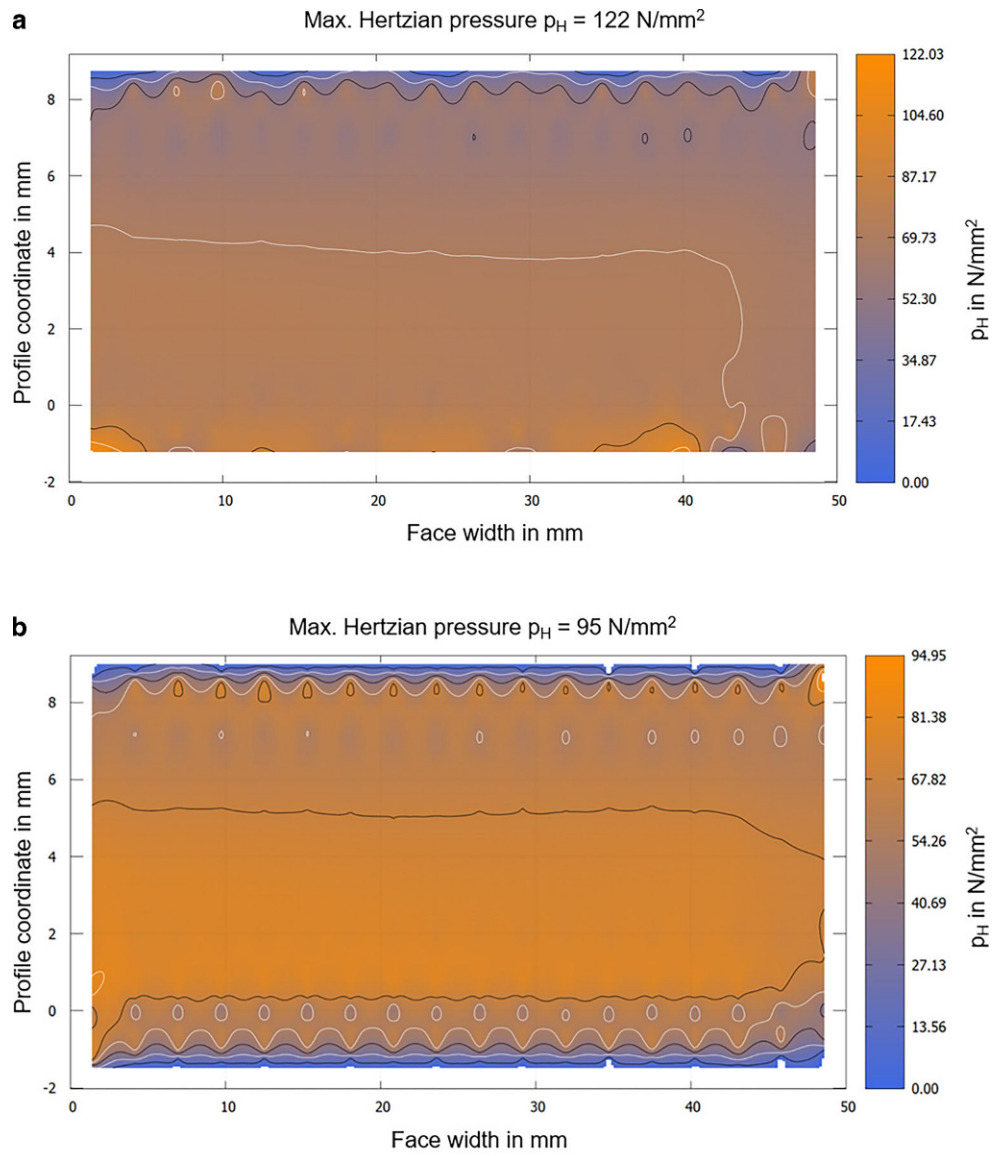
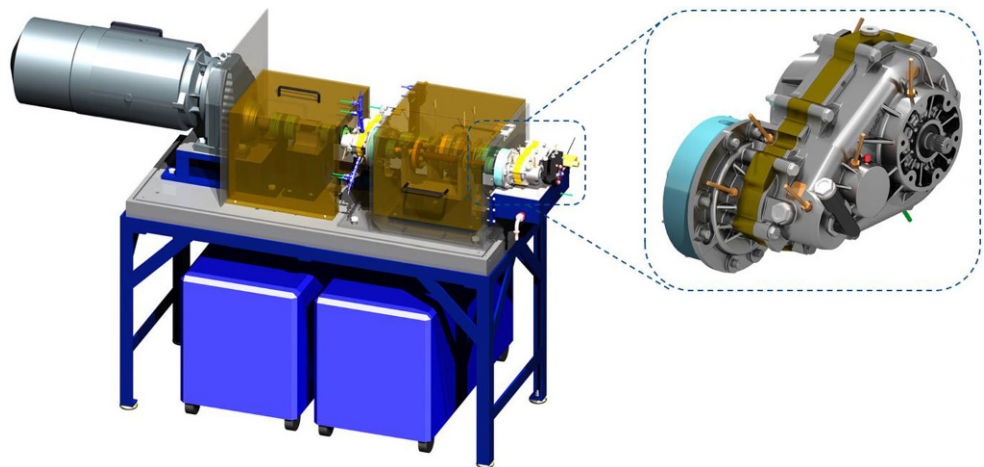


Fig. 10 Universal test rig for electric drives with mounted test gearbox



Adequate profile modifications on polymer pinion (root relief of 50 μm) and wheel (tip relief of 50 μm) effectively reduce pressure peaks during premature and posterior meshing. As a result, the risk of preliminary flank damage is drastically reduced. At maximum possible torque, the calculated Hertzian stress stays below $p_H < 100 \text{ N/mm}^2$ and supports the assurance of sufficient load-carrying capacity. The positive effect of profile corrections is also shown in experimental investigations in [20].

6 Outlook—Investigation of performance and efficiency

Additionally, a test rig is developed and set up for the experimental investigation of the presented gear concept. Fig. 10 shows the schematic structure of the test rig. Next to the functional verification by load-carrying capacity investigations under operating conditions close to the application, further tests with focus on efficiency are conducted. As shown in [10, 11], the use of polymers as gear material can also offer potential in terms of increased efficiency due to very low coefficient of friction in the area of superlubricity under specific lubrication conditions. Parallel to that, the hybrid polymer-steel transmission is tested in practice, installed inside the powertrain of the ILO 1 vehicle.

7 Conclusion

Due to various advantages, polymers are firmly established in technology as construction materials suitable for a wide range of applications. However, the use of polymer gears in applications with higher power transmission is not common but becoming increasingly the focus of research and development projects. Therefore in line with this trend, this work presents a transmission concept for a small urban electric vehicle using polymer gears. To meet the high demands on the material, the high-performance polymer VESTAKEEP® 5000G was used. Due to the limited strength of polymers compared to steel, the original series transmission was modified with focus on a reduction of the detrimental flank pressure. In addition to a plain polymer-polymer stage at the gearbox input, the carrying tooth width was also increased to ensure adequate load-carrying capacity. A subsequent estimation of the service life through load capacity calculations according to VDI 2736 [1] shows that the use of polymer gears in this application is technically feasible. Furthermore, it is shown that with the applied profile modifications, the occurring pressure distribution can be positively influenced. The calculated damage sum with a design for the aimed service life of 60,000 km results in $s \approx 93\%$ equivalent that failure of the polymer gears within the ser-

vice life is not assumed. Under the assumptions made and the given framework, the use of polymer gears in this type of powertrain appears possible. The exact verification of the developed hybrid polymer-steel transmission will be carried out in experimental investigations on a new designed test rig as well as in a practical field inside the vehicle. The aim of this initial investigations was to verify the general applicability of polymer gears as a power-transmitting machine element using the example of a transmission demonstrator in a small electric vehicle powertrain. Weight and cost savings were not in the focus of this preliminary study and are not expected due to the required framework conditions (use of the same housings, low quantities,...). These topics will be addressed in further optimization studies.

Table 3 Nomenclature

Symbol	Entity	Description	Source
b	mm	Face width	[1]
b_W	mm	Common face width	[1]
d	mm	Reference circle diameter	[1]
d_i	–	Partial damage sum	[1]
F_t	N	Nominal tangential force	[1]
$F_{\text{Acceleration}}$	N	Acceleration resistance	[19]
F_{Air}	N	Air resistance	[19]
F_{Gradient}	N	Gradient resistance	[19]
F_{Roll}	N	Roll resistance	[19]
F_{Vehicle}	N	Vehicle resistance	[19]
i	–	Gear ratio	–
K_F	–	Factor for tooth root load	[1]
K_H	–	Factor for tooth flank loading	[1]
m	kg	Mass	–
m_n	mm	Normal module	[1]
n_i	–	Occurring load cycles	[16]
N_i	–	Bearable load cycles	[16]
P	kW	Power	[1]
$r_{\text{dyn.}}$	mm	Dynamic wheel radius	[19]
s	–	Damage sum	[16]
T	Nm	Torque	[1]
u	–	Gear ratio	[1]
Y_{Fa}	–	Form factor	[1]
Y_{Sa}	–	Stress correction factor (notch effect)	[1]
Y_β	–	Helix angle factor	[1]
Y_ε	–	Contact ratio factor	[1]
Z_E	$\sqrt{\text{N/mm}^2}$	Elasticity factor	[1]
Z_R	–	Surface roughness factor	[1]
Z_β	–	Spiral angle factor	[1]
Z_ε	–	Contact ratio factor	[1]
σ_F	N/mm^2	Root stress	[1]
σ_H	N/mm^2	Flank pressure at the pitch cylinder	[1]

8 Nomenclature

The nomenclature is shown in Table 3.

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