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ORIGINALARBEITEN/ORIGINALS

Applicability of classic analytical calculation approaches for the design of plastic gears

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Abstract

Plastic gears are produced cost-effectively in high volume but mainly for applications under low load. Current trends are leading to the design and application of plastic gears for more demanding operating conditions. This requires finding the optimum macrogeometry for the application, which is determined in practice using iterative design processes. Numerical and analytical approaches are available for the design of gears. Due to their speed, only the latter are suitable for an iterative design process. However, the approaches currently used in the calculation programs are designed with a focus on steel gears. Their applicability to gears made of plastics, which material properties differ drastically from those made of steel, is therefore only partially possible.

The paper presents an investigation into the applicability of the classic analytical calculation approaches of Weber/Banaschek and Schmidt for contact analysis and the load-carrying capacity calculation of plastic gears. For this purpose, the results from analytical calculations are compared with those from numerical calculations. In addition, a comparison is made with typical standards (DIN3990, ISO6336) and guidelines (VDI2736).

As a result, the paper provides an initial statement on the transferability of results from steel-focused analytical approaches to gears made of plastic. In addition, calculation results are identified in which particularly large differences occur between the various calculation approaches and which must be considered in the design phase.

Availability of data and material The datasets generated and analyzed during the current study are available from the corresponding author on reasonable request.

Code availability Not applicable

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Anwendbarkeit klassischer analytischer Berechnungsansätze für die Auslegung von Kunststoffverzahnungen

Zusammenfassung

Kunststoffverzahnungen werden kostengünstig in großen Mengen hergestellt, allerdings hauptsächlich für Anwendungen mit geringen Leistungen. Durch aktuelle Trends werden Kunststoffverzahnungen zunehmend auch in Anwendungen höherer Leistungsbereiche eingesetzt. Dies erfordert eine optimierte Auslegung der Makrogeometrie, die in der Praxis durch iterative Auslegungsprozesse ermittelt wird. Zur Auslegung von Zahnrädern gibt es numerische und analytische Ansätze, wobei sich nur letztere aufgrund ihrer Schnelligkeit für einen iterativen Auslegungsprozess eignen. Die derzeit in Berechnungsprogrammen verwendeten Ansätze sind mit dem Fokus auf Stahlverzahnungen ausgelegt. Da sich jedoch die Materialeigenschaften von Kunststoffen drastisch von denen aus Stahl unterscheiden, ist eine Anwendbarkeit der analytischen Ansätze auf Kunststoffverzahnungen nur bedingt möglich.

Die Arbeit stellt eine Untersuchung zur Anwendbarkeit der klassischen analytischen Berechnungsansätze von Weber/Banaschek und Schmidt für die Kontaktanalyse und die Tragfähigkeitsberechnung von Kunststoffzahnrädern dar. Dazu werden die Ergebnisse aus analytischen Berechnungen, mit denen aus numerischen Berechnungen verglichen. Darüber hinaus wird ein Vergleich mit typischen Normen (DIN3990, ISO6336) und Richtlinien (VDI2736) vorgenommen.

Im Ergebnis wird eine erste Aussage zur Übertragbarkeit von Ergebnissen aus analytischen Ansätzen auf Zahnräder aus Kunststoff getroffen. Darüber hinaus werden Berechnungsergebnisse identifiziert, bei denen besonders große Unterschiede zwischen den verschiedenen Berechnungsansätzen auftreten und die in der Auslegungsphase berücksichtigt werden müssen.

1 Introduction

Modern plastic gears are used in a wide range of applications. Among the classic fields of application are household appliances, production plants, and actuators in the automotive sector, where plastic gears offer advantages over steel gears due to their low cost, low noise excitation, and the possibility of lubrication-free use. In addition, plastic gears can be produced fast and very easily in large quantities through the injection molding process. As a result, the number of plastic gears produced worldwide exceeds the number of steel gears. The number of published technical articles, on the other hand, shows an inverse relationship and deals predominantly with steel gears. This is also reflected in the availability of fast calculation methods for load capacity calculations, whether in the form of standards or analytical equations. While there are established standards for gearings made from steel, such as DIN3990 [1] or ISO6336 [2], there are no internationally recognized standards for plastic gears. Guidelines are also scarce on a national level. The Japanese standard JIS B 1759 [3] and the guideline VDI2736 [4, 5] in Germany are exceptions in this regard.

Due to the increasing trend in industry and economy to use polymer gears also for higher performance applications, a highly optimized design of the gears is necessary to utilize the maximum load capacity in the best possible way and thus open new fields of application. However, such a design can only be achieved through many iteration steps. Recent research has focused on the design of plastic gears using FE analysis [6–11]. Although these analyses can provide accurate results, they are only suitable to a limited extent for iterative calculation studies and the achievement of optima in macrogeometry design due to the long calculation times. Analytical calculation approaches, such as those of Weber/Banaschek [12] or Schmidt [13], can provide a solution to this problem. However, these analytical approaches were developed and validated for the calculation of gearings made of steel and do not consider the drastically different material properties of plastics. Because of this, the applicability of common analytical calculation approaches must be investigated for plastic gears.

2 State of the art

VDI2736 [4] provides guidelines for calculating the load-carrying capacity of thermoplastic gears. In four parts, this guideline covers the design and manufacturing of plastic gears, the calculation of the load capacity of spur and crossed helical gears and the execution of test rig trials. VDI2736 is largely based on DIN3990 [1] and experimental research by Hachmann/Strickle [14] from the 1960s on polyamide gears.

For the calculation of the load-carrying capacity of plastic gears, the temperature in the material, the tooth root stress, the contact pressure, and the deformation of the teeth are of particular importance. VDI2736 provides calculation approaches for this purpose. Equations 1, 2 and 3 show the basic equations for the calculation of temperature, tooth root stress, and contact stress according to VDI2736.

$$\vartheta_{\text{Fla}} \approx \vartheta_0 + P \cdot \mu \cdot H_V \cdot \left(\frac{k_{\vartheta,\text{Fla}}}{b \cdot z \cdot (v_t \cdot m_n)^{0.75}} + \frac{R_{\lambda,G}}{A_G} \right) \cdot ED^{0.64} \quad (1)$$

$$\sigma_F = K_F \cdot Y_{Fa} \cdot Y_{Sa} \cdot Y_\epsilon \cdot Y_\beta \cdot \frac{F_t}{b \cdot m_n} \leq \sigma_{FP} \quad (2)$$

$$\sigma_H = Z_E \cdot Z_H \cdot Z_\epsilon \cdot Z_\beta \cdot \sqrt{\frac{F_t \cdot K_H}{b \cdot m_n} \cdot \frac{u+1}{u}} \leq \sigma_{HP} \quad (3)$$

Numerous investigations have already shown that the equations in VDI2736 only represent the conditions in the mesh of plastic gears to a limited extent. For the tooth root stress, this can be attributed to the lack of consideration of the increasing total contact ratio of plastic gears under load [15, 16]. The flank load capacity is particularly dependent on the selected plastic but also on the lubrication, which influences the tooth contact temperature [17–19]. Because of this, higher load-carrying capacities are usually observed in practical tests compared to theory.

In addition to the calculation of the load carrying-capacity according to the standards and guidelines, analytical and numerical methods can also be used to estimate the load capacity. One representative of analytical calculation methods is the calculation software RIKOR [20]. RIKOR provides contact analysis for gears that can be used as a basis for determining root stresses and contact pressures. The calculation is based on a step-by-step analysis of the tooth stiffness. To calculate the tooth stiffness, a 2D analysis of the transverse section is first performed based on a substitute beam model for the tooth and a substitute half-plane model for the gear body. This approach was first introduced by Weber and Banaschek [12] in 1953. In the 1970s, Schmidt [13] extends the plane approach in the face width direction for spur and helical gears and calibrates the model via the tooth deflection by Weber/Banaschek. Schmidt's model represents a widely used approach in gear contact analysis and NVH calculation [21–24].

More universal than analytical models are numerical methods such as the finite element (FE) or boundary element (BE) method, which can account for a variety of influences, such as nonlinear material behavior or temperature dependencies. However, the results of FE calculations are highly dependent on the selected boundary conditions and the mesh. Therefore, the setup, as well as the validation and evaluation of FE models, takes more time than the setup of analytical models and requires a higher level of attention and expertise. In addition, the computation time of FE simulations is usually significantly longer than that of analytical methods.

When calculating and simulating gears made of plastic, the unique material properties should not be neglected. Due to the viscoelastic material behavior of thermoplastics, there are numerous dependencies that are not present in steel gears or are present at negligible scales. Among other effects, the Young's modulus depends on temperature and load speed or frequency. Because of this, the elasticity of thermoplastics decreases with increasing temperature. In addition, the stress-strain curve is nonlinear, even in the elastic range of the polymer. The strong dependencies lead to a more complex analytical design than for steel gears. This is why approximations are often used, such as temperature-dependent Young's moduli [25–27].

3 Models used for calculation

Analytical and numerical models are set up for the calculations. In order to obtain comparable results, the structure of the analytical and numerical model is kept as identical as possible. For this purpose, only the gear body and teeth are modeled (see Fig. 1), which has the advantage of eliminating the influence of the shaft deformation on the tooth contact and gear body tilting.

The FE simulation is modeled using GMSH [28] as the mesh generator and CalculiX [29] as the solver and is based on the work of Hasl [30]. The FE model is meshed with hexahedral elements, and the mesh is designed for fine meshing in the area of the tooth root to be able to represent the small radii of curvature in this area. The mesh density of the gear and tooth body is reduced since lower stresses occur there. In addition, the total number of elements and, thus, the calculation time can be reduced. Also, only a subset of teeth

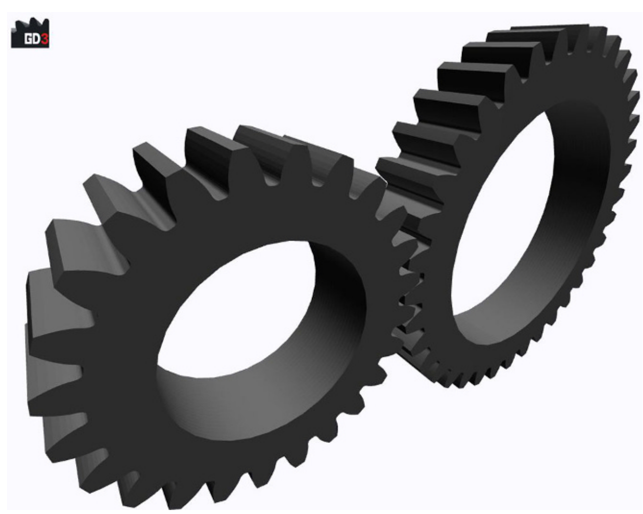

Fig. 1 Representation of the modeling of the wheel and pinion for the analytical model

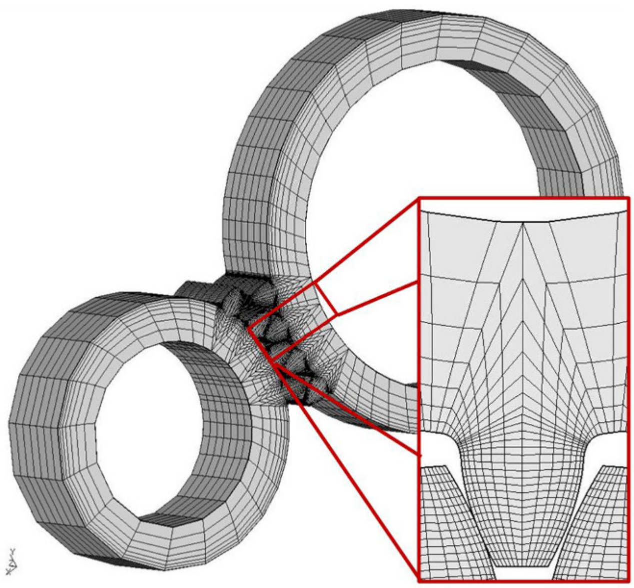

Fig. 2 Exemplary representation of the meshing in the FE model

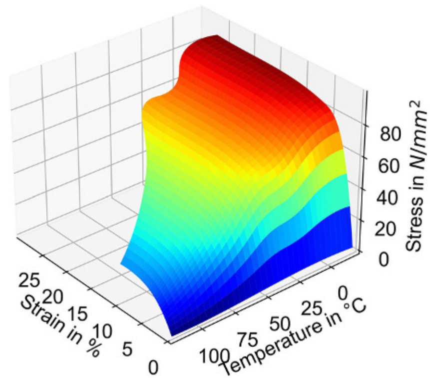

Fig. 3 Nonlinear material model for the polymer Delrin100NC100 [31] used in the FE model

is represented based on the theoretical total contact ratio to further reduce the calculation time. Figure 2 shows an exemplary representation of the FE model used. In the FE model, the wheel body is supported by fixing the wheel at the inner diameter so that the nodes of this surface cannot be distorted. The inner diameter of the pinion is fixed so that it can rotate to introduce a torque into the system via the pinion. In the analytical model, the gear bodies are supported by two infinitely stiff bearings for each gear. This best approximates the boundary conditions of the FE model. Furthermore, the torque is introduced into the system via the pinion as is in the FE model. The pinion is made from steel, and the wheel is made from plastic. The main material parameters used for the calculations are shown in Table 1. In the FE model, a linear and a nonlinear material model can be distinguished. For the former, the material data from Table 1 is used. In contrast, when using the nonlinear material model for calculation, the parameters are taken dynamically from stress-strain curves. Figure 3 shows the material model for a polyoxymethylene (POM, DELRIN100NC010 [31]) used in the nonlinear FE model. The common practice for combinations of steel and plastic gears is to select a steel gear that is wider than the plastic gear to avoid the harder pinion cutting into the wheel. This aspect is considered in both the analytical and numerical models.

Table 1 Basic material properties used for the calculation with the linear material model

	Steel	Plastic
Young's modulus at 20 °C in N/mm^2	210,000	3100
Poisson's ratio	0.3	0.4
Permissible bending stress σ_{FP} in N/mm^2	1650	35

The investigations are based on three illustrative gearings, for which measurement data is available from test bench trials. The gearings include two spur gears and one helical gear, and all have a center distance of 91.5 mm, which corresponds to the center distance of the FZG standard test rig [32]. Table 2 shows the main gearing parameters. The normal module varies between 1 and 3 mm, exceeding that of typical plastic gears found in many applications. The large moduli were chosen based on previous investigations conducted at the institute.

For the comparison between analytical and numerical calculations, we refer to the relevant parameters for the load capacity calculation, such as tooth root stress and flank pressure. These can also be found in the calculation guideline VDI2736 [5] for the load capacity calculation of plastic gears. In addition, the total contact ratio under load is also compared, which has proven to be particularly relevant for the load capacity of plastic gears.

All calculations are conducted for an operating point of a maximum of 100 Nm input torque and 20 °C material/oil

Table 2 Geometry parameters of the example gearings used for calculations

	kst-A		kst-C		kst-CS	
	Pinion	Wheel	Pinion	Wheel	Pinion	Wheel
m_n in mm	1.00		3.00		2.75	
z	72	108	24	36	24	36
b in mm	22	20	22	20	22	20
a in mm	91.5					
α in °	20.0					
β in °	0		0		–25.6	25.6
d_f in mm	70.4	106.9	65.3	101.8	66.3	101.4
d_a in mm	74.7	111.1	78.3	114.7	78.8	114.7
ε_α	1.209		1.198		1.369	
ε_β	0		0		1.000	
h^*_{aP0}	1.25					
ρ^*_{aP0}	0.30	0.46	0.25	0.33	0.25	0.25
Material	Steel	Plastic	Steel	Plastic	Steel	Plastic

temperature. The torque is chosen high enough to allow calculations not only in the range of elastic deformation but also in the range of plastic deformation.

4 Comparison of calculation results

4.1 Total contact ratio under load

First, we examine the total contact ratio under load. Due to the lower stiffness of thermoplastics compared to steel, the teeth of plastic gears deflect more under load. As a result, the total contact ratio of plastic gears under load increases more than that of classic steel gears. Figures 4 and 5 represent the calculated total contact ratio of the analytical and numerical models for the spur gears of Table 2. All the lines show a similar trend. However, the analytical model underestimates the total contact ratio compared to the numerical model by up to 15% for the kst-C gearing and even 20% for the kst-A gearing. Furthermore, the deviations depend on the chosen material model. Under consideration of the nonlinear material properties of polymers, the loaded total contact ratio shows a steeper gradient, which is to be expected since the loads at high torques are above the yield stress and, thus, in the range of plastic deformation. It is noticeable that the total contact ratio increases more for the kst-A gearing with module 1 mm than for the gearing with module 3 mm. This is consistent with research that has found correlations between the increase in total contact ratio and the number of teeth [33, 34].

Helical gears typically have larger total overlaps than spur gears. Figure 6 shows the results for the total contact ratio of the helical gear kst-CS (Table 2). The graphs show a similar pattern to the spur gear results described above, with relative deviations of up to 10%. Again, the overlap of the nonlinear material model increases more than that of the numerical and analytical simulations with linear material properties.

4.2 Tooth root bending stress

The load-carrying capacity of the tooth root is mainly influenced by the tooth root bending stress resulting from the force acting along the tooth profile. According to VDI2736 [5], the safety against tooth root fracture results from the ratio of the occurring tooth root stress and the tolerable tooth root stress. For this reason, we examine the maximum tooth root bending stresses that occur.

Especially at low loads, the root stresses calculated by the analytical and numerical calculations show a good correlation (see Figs. 7 and 8). With increasing load, the deviations between the models increase. As before, larger deviations can be determined for the gearing with a smaller

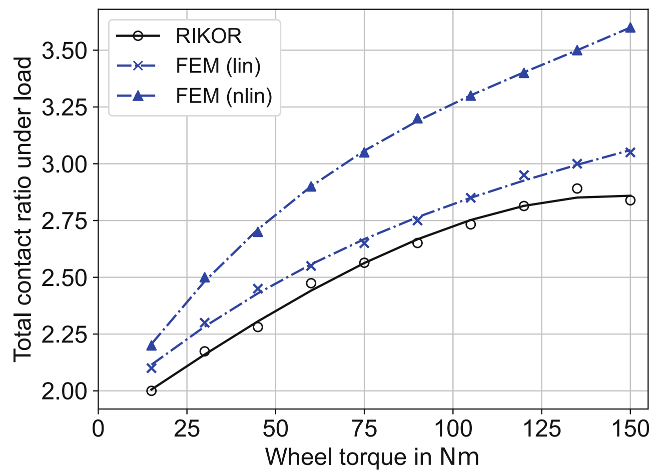

Fig. 4 Total contact ratio under load for kst-A gear as a function of the output torque

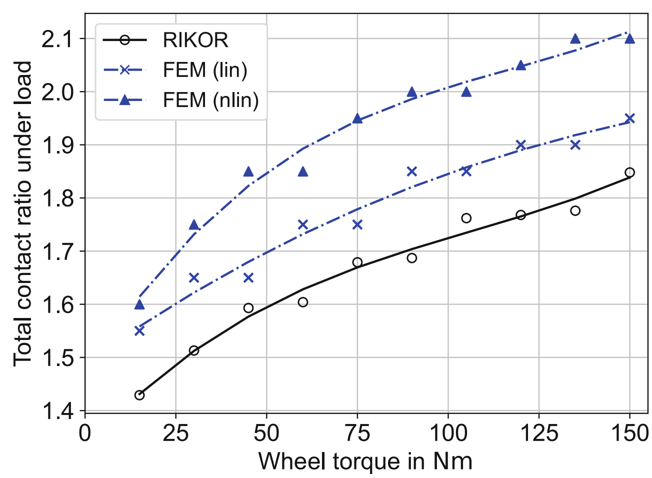

Fig. 5 Total contact ratio under load for kst-C gear as a function of the output torque

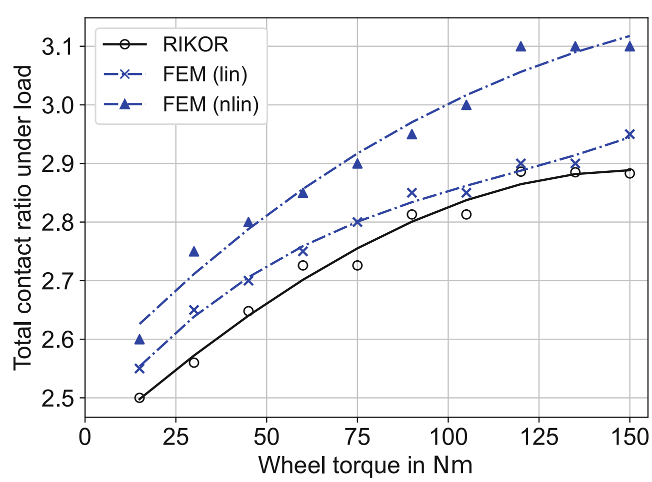

Fig. 6 Total contact ratio under load for kst-CS gear as a function of the output torque

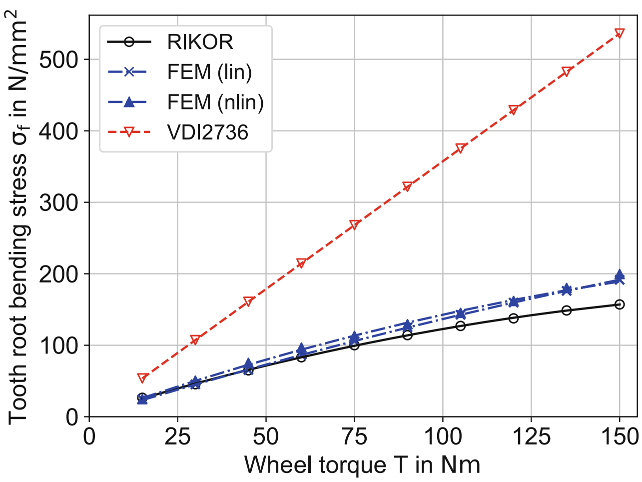

Fig. 7 Tooth root bending stress for kst-A gear as a function of the output torque

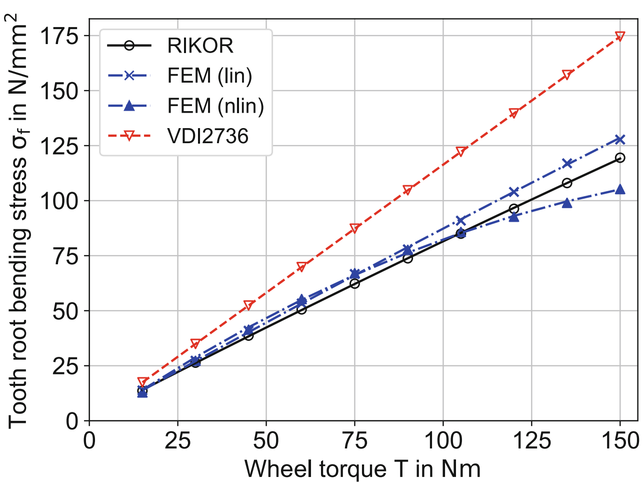

Fig. 8 Tooth root bending stress for kst-C gear as a function of the output torque

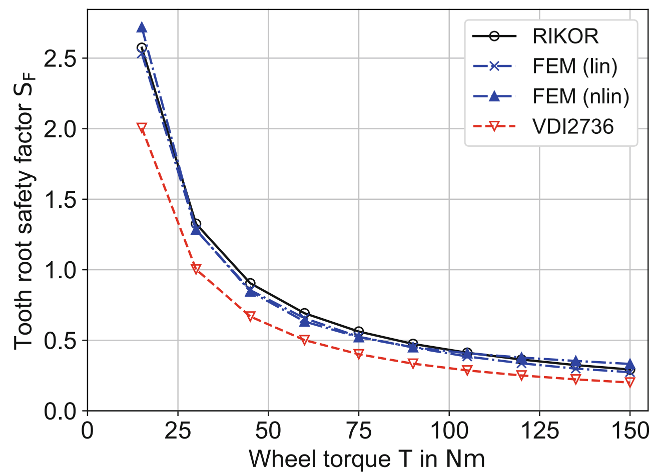

Fig. 9 Safety factors for kst-C gear for continuous operation ($N_L = 10^8$, $S_{Fmin} = 2$) at 20 °C at different wheel torques

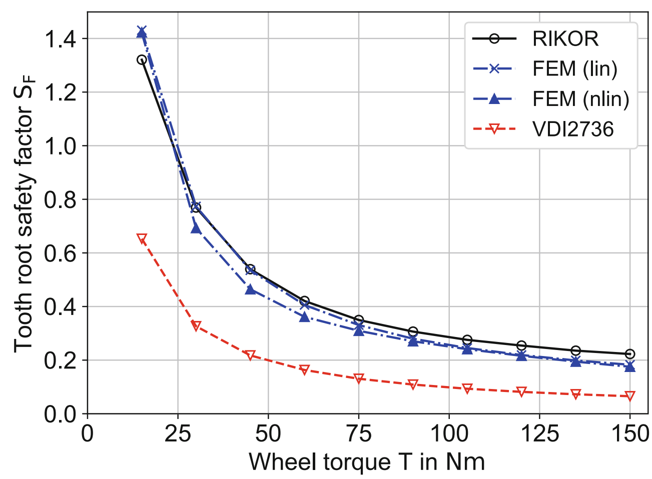

Fig. 10 Safety factors for kst-A gear for continuous operation ($N_L = 10^8$, $S_{Fmin} = 2$) at 20 °C at different wheel torques

module (kst-A max. 20%, kst-C max. 10%). An influence of the material model can be observed for the gear with module 3 mm (Fig. 8), where the slope of the curve of the nonlinear material model flattens with increasing load. The beginning of the flattening correlates with the transition of the total contact ratio from 60 Nm output torque to values greater than two, which causes a reduction in the average load per tooth. This effect can also be seen for module 1 mm (Fig. 7), although it is much smaller due to the already high initial total contact ratios.

For both gearings, tooth root stresses according to VDI2736 are significantly higher than those according to the analytical and numerical calculations. This is caused by the increase in profile contact ratio not being fully considered in the calculation guideline. As a result, VDI2736 yields significantly lower safety factors, meaning a safety of 1 for the kst-C gearing is already reached at a 25% lower input torque compared to the numerical and analytical calculations (Fig. 9). For kst-A (Fig. 10), the safety factor according to VDI2736 will not reach safety factors above 1. A comparison with DIN3990 and ISO6336 shows that these standards do provide results that are within the range of VDI2736. However, the values are slightly lower resulting in a safety factor of 0.58 according to DIN and 0.65 according to ISO for the kst-C gearing at an output torque of 45 Nm.

The highest stresses at the root of the tooth are also determined for helical gears, according to VDI2736. However, the results between the different methods are more similar than for spur gears (Fig. 11). The helix angle factor Y_β has a positive effect here, as it considers the more favorable meshing conditions found on helical gears. Furthermore, there is a greater deviation between the calculation results of RIKOR and the numerical simulations than for spur gears, which requires further investigation.

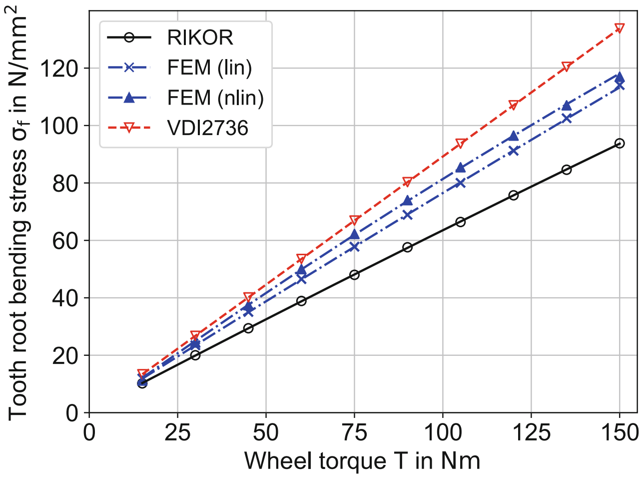

Fig. 11 Tooth root bending stress for kst-CS gear as a function of the output torque

4.3 Contact stress

The Hertzian contact pressure is used to calculate the flank load capacity. For this purpose, the maximum flank pressure occurring over all mesh positions is determined. The curve of the analytical model shows an almost constant increase, which is in good agreement with the results of VDI2736. Different curves are found between the analytical and the numerical calculations (Figs. 12 and 13). For both the linear and nonlinear material models, the flank pressure initially increases before it drops abruptly at a threshold torque and rises again thereafter. The limiting torque of the drop correlates for kst-C (Fig. 13) with the transition of the total contact ratio to values greater than two. For the kst-A gearing (Fig. 12), there is also a drop in contact stress that corresponds to the increased total contact ratio, although the effect is less pronounced. However, the results of the numerical model show intensive growth at higher input torque values. This drastic increase cannot be fully explained with the material model. The not quite plausible results of the numerical models show the limited abilities of FEA for contact stress calculation which is in part due to the contact algorithm used.

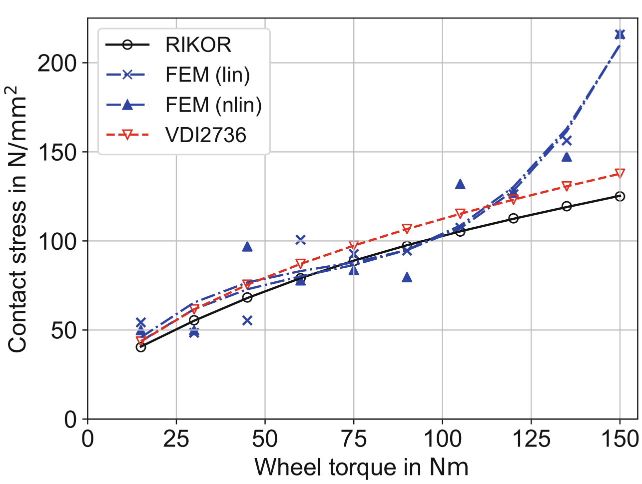

Fig. 12 Flank pressure for kst-A gear as a function of the output torque

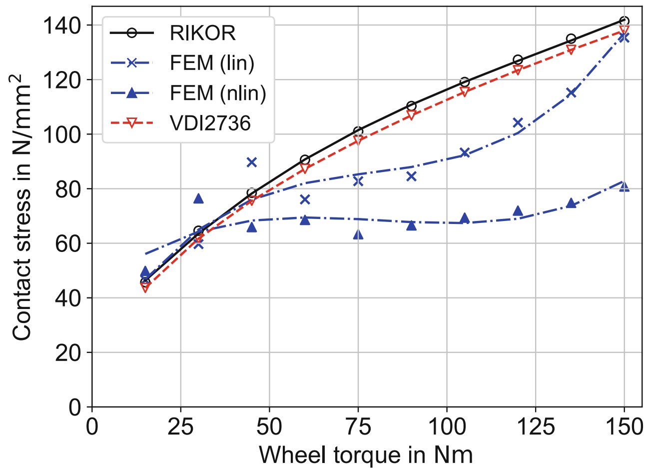

Fig. 13 Flank pressure for kst-C gear as a function of the output torque

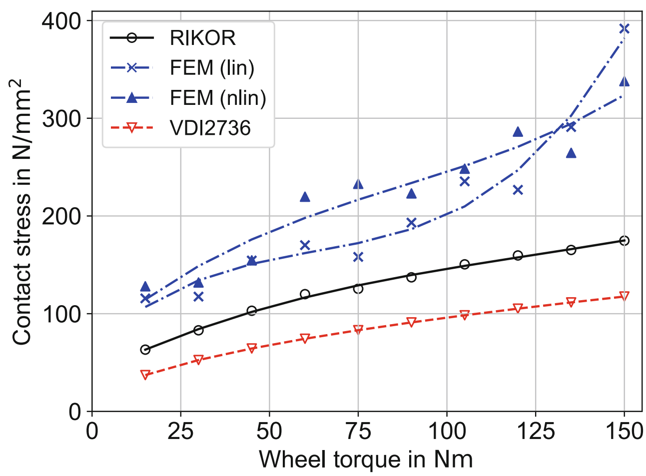

Fig. 14 Flank pressure for kst-CS gear as a function of the output torque

The wave-like progression is also evident in the numerical results for the helical gearing kst-CS, although it is hardly noticeable for the nonlinear material model (Fig. 14). Again, the numerical results are only partially plausible. It is noticeable that the results of VDI2736 show a lower pressure than the higher-quality calculation methods. This can be attributed to the contact ratio factor Z_ε, which considers the influence of the effective length of the contact line based on the radial and overlap contact ratio, especially for overlap contact ratios ε_β greater than one.

5 Conclusion

The design of plastic gears for transmissions for higher power levels can only be achieved by optimizing macrogeometry through iterative studies of numerous variants. FE methods are not suitable for such a design process because of their long calculation times and complex setup. In contrast, analytical methods are characterized by their significantly shorter calculation times while still achieving sufficient accuracy. Current analytical calculation approaches are only designed and validated for steel gears. The paper compares the calculation results for analytical and numerical simulations for tooth root stress and flank pressure that are relevant for determining the tooth root and flank load capacity. In addition, the total contact ratio increases under load, which is highly relevant for plastic gears.

The comparison shows a good agreement of the tooth root stress between the numerical and analytical calculation results when using the material parameters associated with the operating point. The material model (linear, nonlinear) is found to play a minor role at the selected operating point, especially for the range of sufficient operating safety. For the flank pressure, however, significantly larger deviations were found between the numerical and the analytical models. Since the analytical results are in good agreement with the results of VDI2736, this may be because of an inadequate contact model in the numerical calculation, which requires further investigation. The guideline VDI2736 produces significantly overestimated tooth root bending stresses, resulting in low tooth root safety factors. On the other hand, the flank stresses calculated according to VDI2736 are in the range of the numerical and analytical results. Larger deviations can be seen throughout the results of all methods for the helical gearing, which must be investigated further in future research.

Overall, the current analytical calculation methods are a good starting point for an optimized design of plastic gears. However, further investigations should be carried out on the influence of the helix angle on the calculation results as well as on the geometric peculiarities of plastic gears, such as the pronounced rounding at the tip edge that influences the contact pressure or the more severe influence of temperature for flank damage.

6 Nomenclature

The nomenclature is shown in Table 3.

Table 3 Nomenclature

Symbol	Entity	Description
a	mm	Center distance
b	mm	Tooth width
d_a	mm	Tip diameter
d_f	mm	Root diameter
h^*_{aP0}	–	Addendum factor rack
$k_{\vartheta,Fla}$	$K \cdot (m/s)^{0.75} \cdot mm^{1.75}/W$	Heat transfer coefficient of the plastic gear
m_n	mm	Normal module
u	–	Gear ratio
v_t	m/s	Tangential velocity
z	–	Number of teeth
A_G	m^2	Heat-dissipation surface of the mechanism housing
ED	–	Relative tooth-engagement time with respect to ten minutes
F_t	N	Nominal tangential load
H_V	–	Degree of tooth loss
K_F	–	Tooth root load factor
K_H	–	Tooth flank load factor
$R_{\lambda,G}$	$K \cdot m^2/W$	Heat transfer resistance of the mechanism housing
P	W	Nominal output
Y_{Fa}	–	Form factor
Y_{Sa}	–	Stress correction factor
Y_β	–	Helix angle factor
Y_ε	–	Contact ratio factor
Z_E	–	Elasticity factor
Z_H	–	Zone factor
Z_ε	–	Contact ratio factor
Z_β	–	Spiral angle factor
α	°	Pressure angle
β	°	Helix angle
ε_α	–	Contact ratio
ε_β	–	Overlap ratio
ϑ_0	°C	Ambient temperature
ϑ_{Fla}	°C	Flank temperature characteristic
μ	–	Coefficient of friction
ρ^*_{aP0}	–	Tip radius coefficient

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