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MESHING BEHAVIOUR OF MINIATURISED PLASTIC GEAR PAIRS FOR POWER TRANSMISSION USING FEM

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ABSTRACT

In spite of the fact that miniaturised plastic gears for power transmission are widely used in many technical fields today, dimensioning for them has not been standardised, and there is little known literature on the subject. This paper shows the geometric features and specialised performance characteristics of these gears. Meshing behaviour, tooth deformation, flank pressure, and tooth root stress will be calculated as functions of tolerance, load, and possible profile modifications. Furthermore, points will be made regarding transmission and friction behaviour. From this analysis, recommendations about the design of such gear pairs will be made.

Index Terms – plastic gear, miniaturized, FEM, power transmission

1. INTRODUCTION

In many modern products, inexpensive, minaturised drive systems are used to move mechanical parts and assemblies. They are normally comprised of an electrical motor with high rotational speed, control electronics, and transmission gears. The motor and gears are often arranged one after another in a cylindrical form. For precision applications, the typical outer diameter is around 12 to 50 mm, which is only possible with gears whose module m is less than 1 mm. The gears are often made of plastic and can be cheaply produced in large quantities. These drive systems are used in many industries, and a broad range of motors is available for them. In contrast, there is a growing demand for miniaturised power transmission gears that are wear-resistant and quiet. To meet this need, it is necessary to know the influence of tooth geometry, load, and tolerances on gear performance.

2. MOTIVATION

Unlike for metal gears (e.g. DIN 3990, ISO 6336, AGMA 2001), there are no valid standards or guidelines available for plastic power transmission gears. Many of the calculations described in the literature (e.g. [2], [5], [7]) build on the VDI 2545 standard, which was retracted in 1996. According to [3] and [4], the VDI 2545 can also lead to significant under or over-dimensioning, depending on the application. In current publications, there is no agreement about which methods should be applied. Regardless of differing opinions in the literature, in practise, the VDI 2545 is most often used. Also, as compared to metal gears, there are only a few other known publications about plastic gears that contain insight into selected performance characteristics. These include:

- analytical calculation of transmission behaviour in [9], [10] and [15],
- experimental investigations of meshing behaviour in [13],
- finite element analysis of tooth deformation, load distribution, and tooth stress in [10], [12] and [14],
- analytical and experimental efficiency investigations in [16] und [17],
- analysis of the influence of temperature and humidity in [1], [8], [14] und [18].

The evaluation of these publications can be summarised by the idea that plastic gears show large tooth deformation under a load, compared to gears made from metallic materials. This leads to changes in meshing and transmission behaviour, as well as greater wear and noise. Furthermore, characteristics specific to plastics must be considered, such as their response to temperature, humidity, and friction. The analysis is made more difficult because the necessary material data is often incomplete or unavailable. In such cases, data from comparable materials, or material combinations, must be used. How much, and by what mechanism, the gear characteristics mentioned above are influenced has not yet been comprehensively investigated.
Almost all publications are the result of calculations and experiments with plastic gears whose module is $m = 1$ to 4 mm. The characteristic of precision toothing systems becomes clearer when the tolerances are more closely examined. The following investigation was prepared for toothing quality 8. Single pitch tolerance has a

significant influence on the meshing behaviour of a gear stage and is therefore more closely investigated. Fig. 1 shows the allowable tolerance relative to pitch for different moduli. Of note is the large increase for modules $m < 1$ mm. The relative tolerances here are 1 to 5 %, while gear teeth in mechanical engineering have much lower tolerances, around 0.1 to 0.4 %.

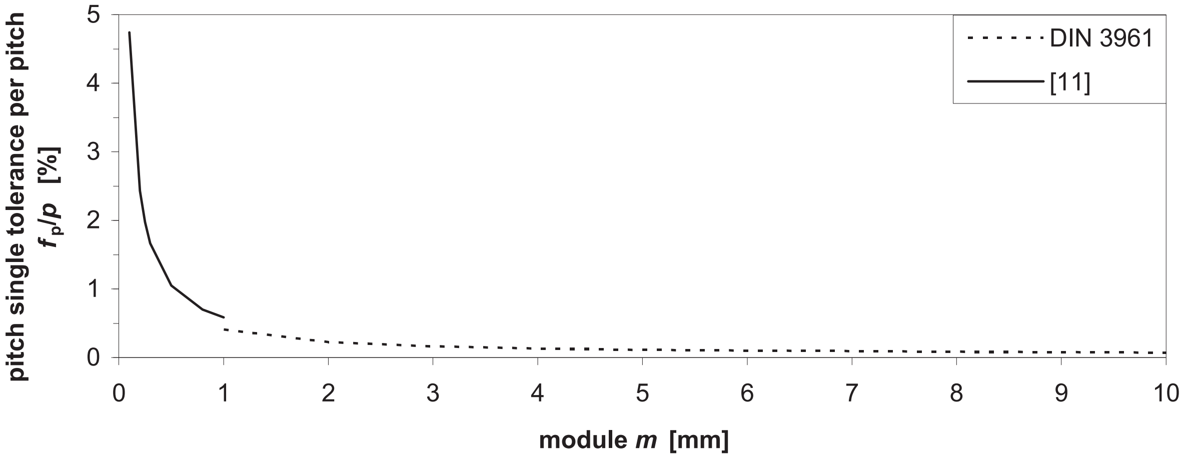

Fig. 1: Relative single pitch tolerances vs. module (example of quality 8, DIN 58405)

In plastic gears, the large deformation of the tooth tip mentioned above occurs in the circumferential direction under a load. Two calculations of the maximum deformation exist in the literature. For equal gears and identical operational parameters, [2] calculates the deformation to be 2.25 times higher than calculated in [5].

In the operating state, the single pitch tolerances and deformations are additive. In the worst case, for miniaturised gears, numbers as high as

- 2 ... 10 % according to [5] and
- 3 ... 15 % according to [2]

can be reached for real deviations in pitch under a load, in relation to the nominal value of the pitch.

This shows that the influence of tolerances on operation for gears in this module range cannot be disregarded, and that findings from mechanical engineering cannot be blindly adopted. The goal of this article is to investigate gear characteristics, such as tooth deformation, meshing behaviour, flank and root loads, as well as transmission and friction behaviour, as functions of toothing tolerances and operational parameters. To analyse these issues, essential computational methods must first be developed and validated. Next, parameters that have crucial influence over the characteristics mentioned must be chosen, e.g. torque and tolerances, which are subsequently subject to variance analysis. Furthermore, the influence of profile modifications is to be investigated. These modifications are to be technologically controllable for to make inexpensive precision gearing possible.

3. THE METHOD

Analytical methods for computing the gear characteristics mentioned above are very limited or are not applicable, and so a simulation method must be used. The Finite Element Method (FEM) is optimally suited to the problem at hand.

In simulations, it is generally desirable to minimise the amount of modeling and calculation. For that reason, legitimate simplifications are now described. The first step in the simplification process is to consider the system as a plain stress or plain strain state, hence the depiction of the three-dimensional gear in a two-dimensional plane (Fig. 2). All loads and deformations across the width can be considered constant, which means that the influence of gear width is assumed to be negligable. The gear wheels are not completely modeled. Only as many teeth as will mesh during the simulation need be modeled.

In addition to these simplifications, the mesh grid on the gears is also crucial for the quality of computational results. On the side of the tooth that meshes, the grid must be fine enough that the amount of contact produced by the Hertzian contact stress can be realistically computed. Preliminary investigations with different mesh grids showed that the edge length for surface elements of the gears studied here should be apporximately 2 µm. The mesh grids on the non-meshing side of the tooth can be significantly coarser.

The boundary conditions (see Fig. 2) do not require any simplifications or adjustments, as they reflect reality. Both gear centres are fixed to prevent translation. The driven wheel turns freely on its axle and is subject to an output torque M_2. The angular position φ_1 on the driving wheel is given. The tooth profiles have contact elements so that the teeth in the simulation can mesh with each other. These contacts are nonlinear boundary conditions in FE-analysis.

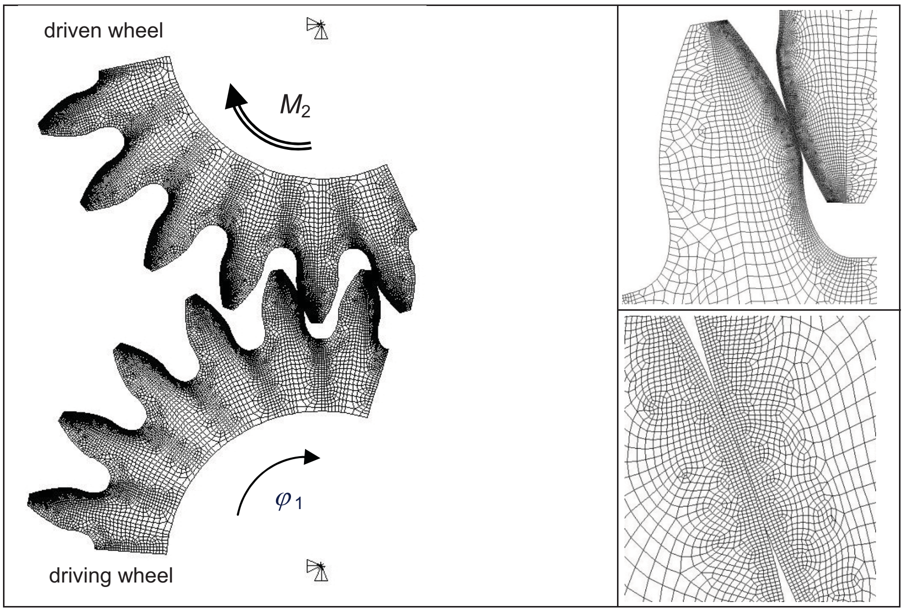

Fig. 2: FEA-model with boundery conditions and detail views of the FE-mesh grid

In the simulation, the driving wheel turns sequentially in small angular increments, $\Delta\varphi_1$. Preliminary investigations showed that a run-in process is necessary, which means that each pair of teeth must mesh during the initial simulation. Otherwise, large errors occur in the analysis. The minimum number of teeth per gear that must be modeled is determined by the run-in. Figure 3 shows important steps in the simulation. The pair of teeth that will be used in the analysis of the results is marked with dots.
The simulation described requires a large amount of time, since the run-in process must be computed for each change in both gear and operational parameters. Because of the contact area in the model and the material properties used, the simulation is nonlinear on multiple levels, further increasing the time it requires.

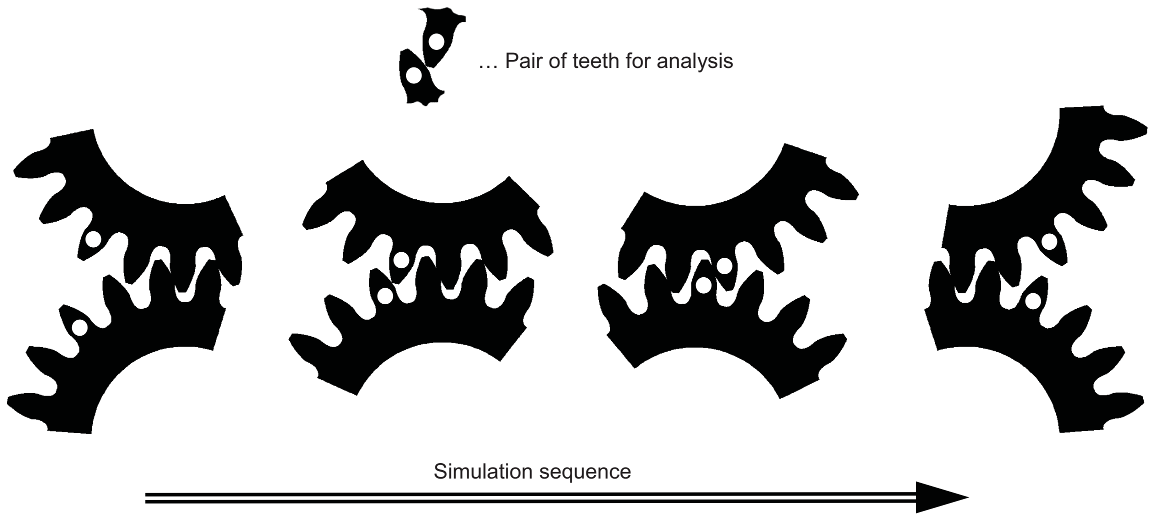

Fig. 3: FE-Analysis sequence

Before the simulation model can be used for analysis, it must be validated with reality. This validation shows whether the selected simplifications, the quality of the mesh grid, the boundary conditions, and the simulation sequence are legitimate. Appropriate validation parameters include all results of the simulation that can be measured on gears. Due to the very small dimensions of the gears considered here, all available measurement methods either cannot offer the necessary resolution or would significantly alter the gears' behaviour. For this reason, validation must be done indirectly, via metal gears, whose mechanical behaviour under various operating conditions is well-studied, and for which extensive publications and standards exist. Steel gears were simulated with the FE model created for plastic gears, and the results, such as flank pressure and tooth root stresses, are compared with accepted standards in order to calculate load capacity. The simulation agrees well with results from the standardised calculations, and so the model is estimated to be realistic. It must be noted, however, that the FEA run-in process described above must be be finished before evaluation can begin.

4. RESULTS

This section presents the results of the FE analysis for spur gear pairs. First, the gear variations that were analysed will be discussed in detail. The focus of the analysis is on meshing behaviour, tooth flank pressure, and tooth root stress. In addition, tooth deformation, as well as transmission and friction behaviour, are presented.

4.1. Parameters of the analysed gears

The gears chosen for this numerical analysis exemplify typical components of today's miniaturised stationary and planetary gears. All of the analysed gears had the following characteristics:

- reference profile DIN 58400,
- normal module m_n = 0,25 mm,
- normal pressure angle α_n = 20 °,
- number of teeth z = 21.

The parameters for the investigation are the tolerances on a single pitch f_p, the load, and the profile geometry of the teeth. Table 1 contains the variations in these parameters and the symbols used.

Table 1: Analysed gears (gv: spur gear; mgv: modified spur gear)

name	single pitch tolerance f_p driving wheel	single pitch tolerance f_p driven wheel	nominal load	material	additional specification
gv1	0 µm	0 µm	100 %	steel / steel	./.
gv2				POM / POM	
gv3	+16 µm	-16 µm			
gv10	0 µm	0 µm	50 %		
gv11	+16 µm	-16 µm			
gv13	0 µm	0 µm	10 %		
gv14	+16 µm	-16 µm			
mgv2	0 µm	0 µm	100 %		with tip relief on both gears
mgv3	+16 µm	-16 µm			
mgv10	0 µm	0 µm	50 %		
mgv11	+16 µm	-16 µm			
mgv13	0 µm	0 µm	10 %		
mgv14	+16 µm	-16 µm			

4.2. Meshing Behaviour

Two paired tooth profiles theoretically touch at one point that, for involute gears, moves along the path of contact from begin of the engagement until disengagement. This assumption corresponds well to the behaviour of metal gears. The FE analysis of plastic gears, however, shows a clear deformation of the contact area along the flank under a load, through which line contact occurs (Fig. 4). The length of this line is 15 to 20% of the tooth height under a nominal load. In this case, the concept of a point contact and a path of contact do not apply. Over one complete tooth mesh, a meshing area emerges as the sum of all line contacts.

The diagram in Fig. 5 shows the meshing area for variations without tolerances, and Fig. 6 shows the same for the gear pair with the maximum single pitch tolerances. Since the areas overlap, but are nevertheless to be differentiated, the closed curves in the diagram only represent the surrounding borders of the meshing areas.

For comparison, refer to the meshing behaviour of a steel gear (gv1). That process corresponds well to the theoretical path of contact, is limited by the two tip circles, and exhibits no meshing disturbances. In contrast, the POM gear pair (gv2) shows clear tooth tip interference at the points of engagement and disengagement, caused by the large tooth deformation that occurs. If a driving wheel with the maximum single pitch tolerance is paired with a driven wheel where that tolerance is at its minimum (gv3), the flanks of the teeth come into contact much later, and so the tip interference increases significantly as the gears disengage. For gears with the opposite tolerance situations, this massive mesh disturbance occurs as the gears first engage. The dashed curves represent the results of the gear pairing with tip relief. This profile modification avoids tip interference in both the variation without tolerances (mgv2), and the variation with the most unfavourable single pitch tolerance (mgv3). Gears are normally analysed for a nominal load that is equivalent to the maximum load. However, in practise, collective loads act together, each of which is much smaller. It is therefore instructive to analyse the mesh behaviour under different loads. As expected, gear pairs without tolerances (Fig. 5) have smaller meshing areas under smaller loads, and tip interference decreases. Nevertheless, at 10 % of the nominal load, this phenomenon does exist, though slightly. If single pitch tolerances are brought into the investigation (Fig. 6), tip edge interference diminishes only minimally. If gear pairs with such tolerance situations engage, significant mesh disturbances will occur, regardless of the current effective torque on the system. Both diagrams do show that tip relief can solve this problem.

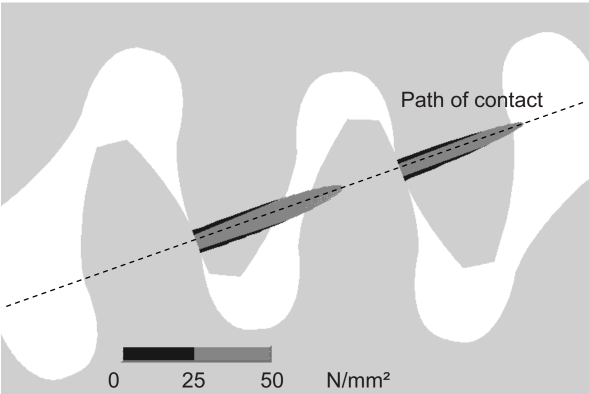

Fig. 4: Progression of the pressure on the tooth flank under a load

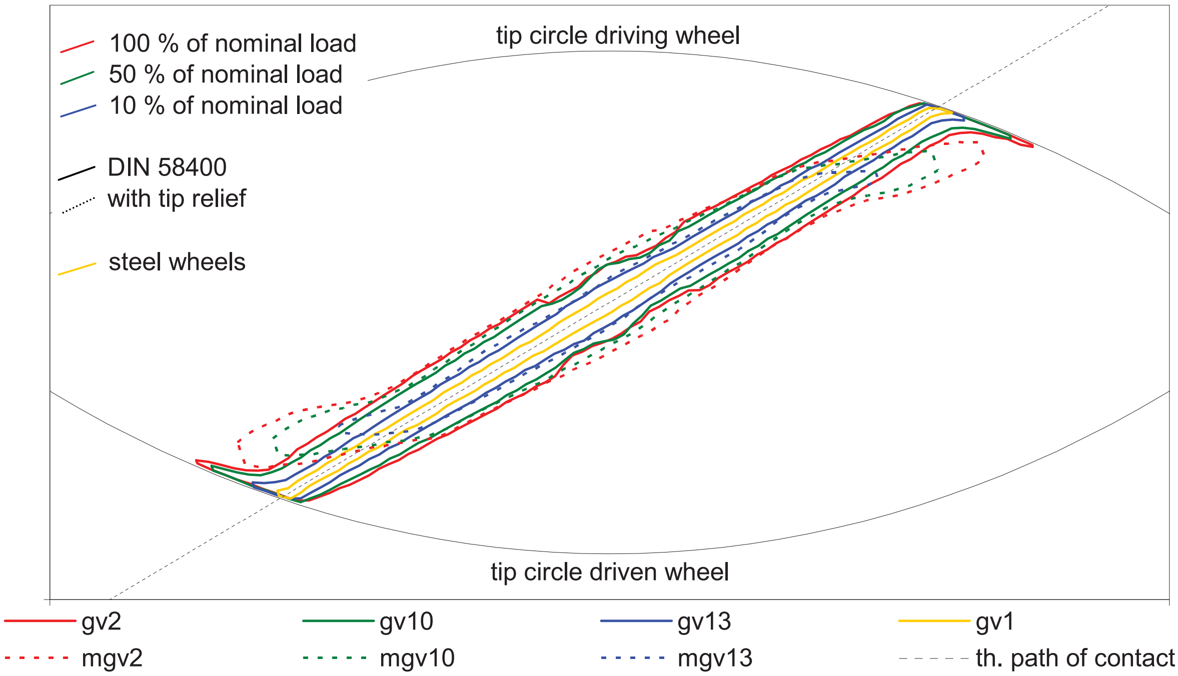

Fig. 5: Meshing behaviour for various loads and profile geometries with $f_p = 0$

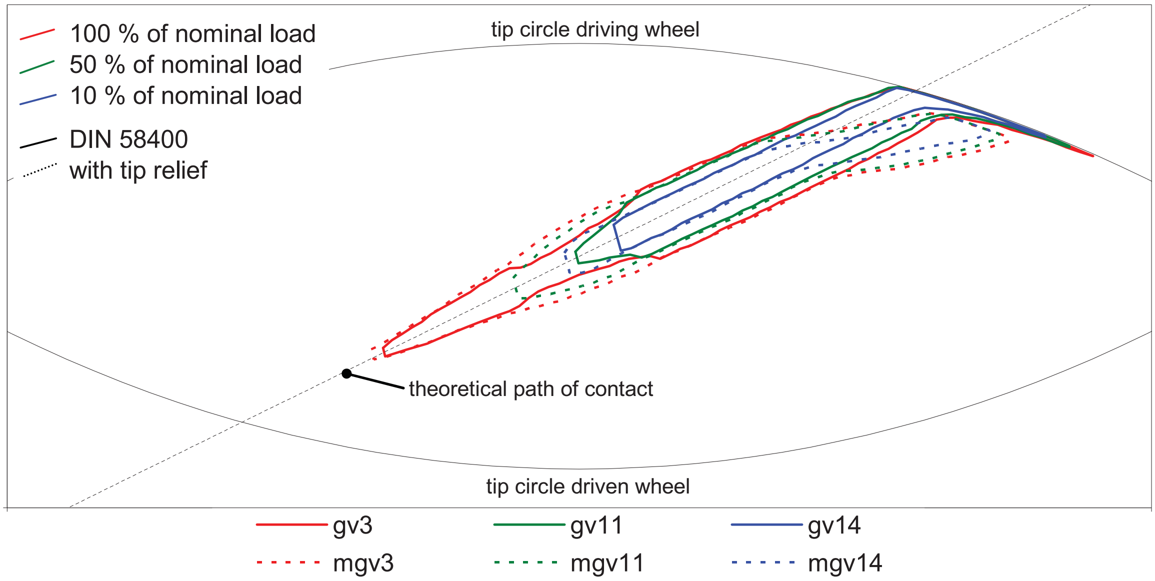

Fig. 6: Meshing behaviour for various loads and profile geometries with f_p = max

4.3. Tooth deformation

Due to their small elastic moduli, plastic gears undergo significantly larger deformations than metal gears. This has the same effect on mesh behaviour as pitch tolerances. In the literature, the value $\lambda = 0.1 \cdot m$ is given as a benchmark that is not to be exceeded. If larger values are achieved, meshing disturbances cause greater noise during operation, and the allowable tooth root stress is exceeded.

In Fig. 7, the progression of the tooth deformation λ is depicted over the driving wheel's angle of rotation. The teeth of the variation without pitch tolerance (gv2) reaches maximum deformation at $\lambda = 16$ µm. For the modified tooth profile (mgv2), the progression changes somewhat, and the maximum increases to $\lambda = 21$ µm. Both gears with maximum pitch tolerances (gv3, mgv3) do not reach their maximum until the end of the tooth mesh and are deformed twice as much as the gear without pitch tolerance.

The tolerance-free gear pairs (gv2, mgv2) keep within the guideline value, while the two pairs with tolerances do not. A comparison of meshing behaviours (Fig. 5, Fig. 6) shows, however, that only the two gears with normalised tooth profiles clearly showed tip interference. Single pitch tolerance in the modified pairs was corrected by tip relief, and the teeth meshed cleanly even at higher deformations.

In the graph (Fig. 7), the results of both analytical calculations are displayed. Since these are for tolerance-free gears, hence they correspond to the gv2 characteristic curve. According to [5], the maximum deformation correlates well with the simulation results, while, according to [2], values that are significantly too large are generated.

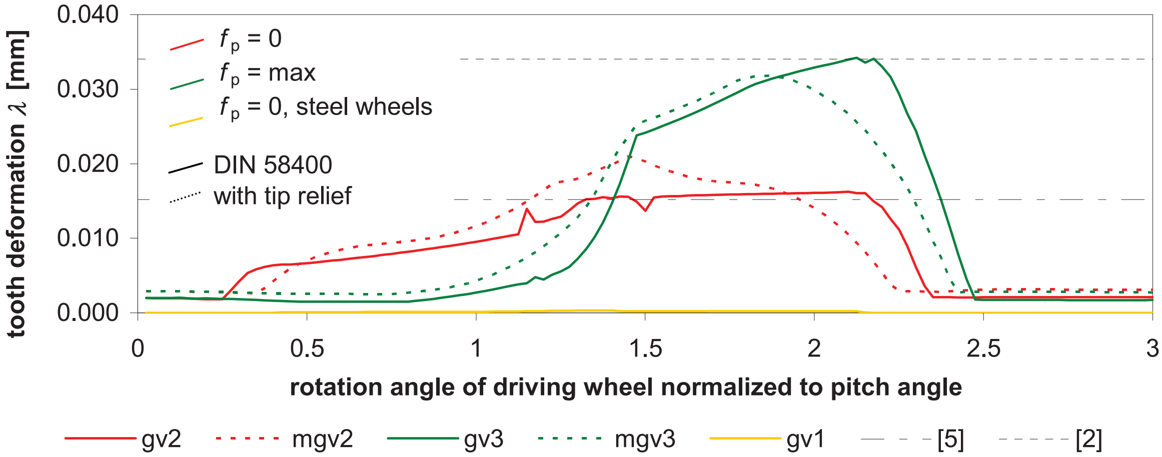

Fig. 7: Tooth deformation λ for different profile geometries and single pitch tolerances.

4.4. Tooth flank pressure

For comparison with plastic gears, the progression of tooth flank pressure over the angle of rotation for a steel gear pair (gv1) is depicted in Fig. 8. The characteristic curve shows the typical, known progression found in the literature. The plateau in the central section results from the single tooth meshing, while before and after, two pairs of teeth are meshing. Within each of these three regimes, the tooth flank pressure remains nearly constant. In contrast to this case, the plastic gear (gv2) shows distinctive maximum values at the beginning and end. This results from tip interference and exceeds the allowable tooth flank pressure for the material POM. In addition, because there is no single tooth mesh, no peak occurs in the middle of the characteristic curve. The gear with maximum single pitch tolerance (gv3) engages later and without great pressure, but does, when disengaging, exhibit values significantly outside the allowable range. If such a pattern of tolerances occurs, substantial wear will set in quickly, or operational disturbances will appear in the gear. Tip relief (mgv2, mgv3) prevents tip interference and thus significantly decreases tooth flank pressure. Therefore, even unfavourable tolerance situations can adhere to the allowable guideline for the material.

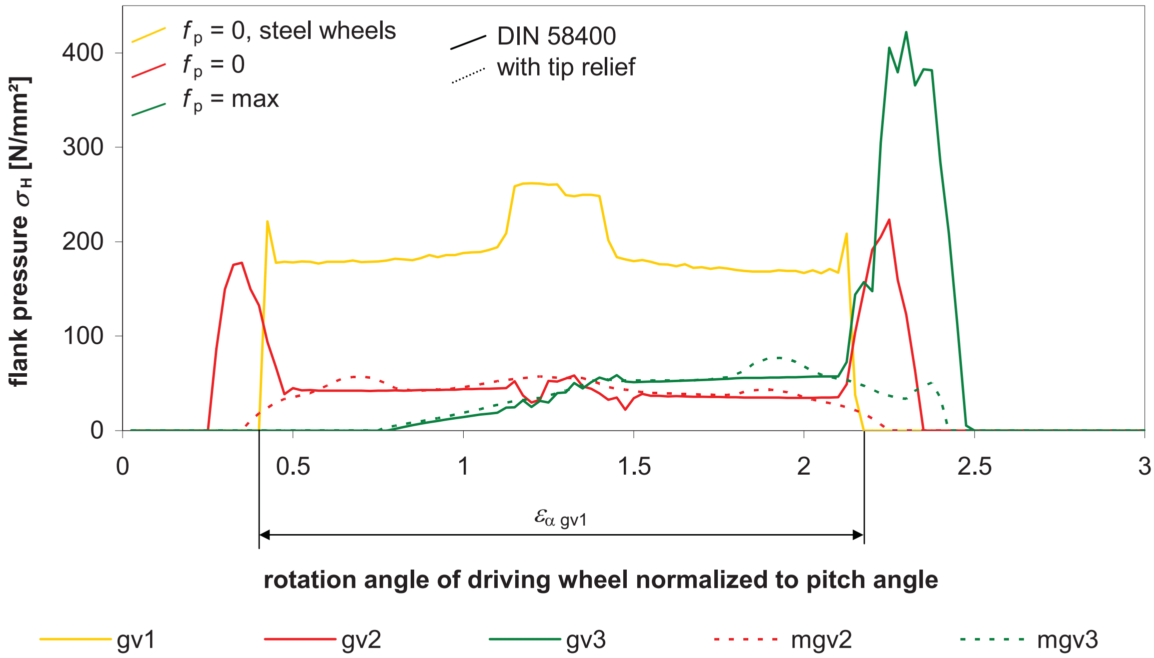

Fig. 8: Tooth flank pressure σ_H of a spur gear pair for various profile geometries and single pitch tolerances

By normalising the abscissa to the pitch angle, the actual transverse contact ration ε_α can be read from the graph. The values of ε_α are greater than the theoretical values in Gl. (1), due to tolerances and deformation. As an example, profile coverage is labeled for the steel gear pair in Fig. 8. Comparatively, deformation causes the plastic gears (gv2) to mesh for a much longer time. Tip relief on these gears (mgv2) reduces transverse contact ration somewhat. The teeth come into contact later, and lose contact sooner. Should two gears with maximum single pitch tolerances be paired, transverse contact ration is shortened substantially, even though the second law of gears still holds. It is evident by comparing the normalised (gv3) with the modified (mgv3) profile geometry that both variations begin to engage in the same way. When disengaging, however, differences are apparent, because the tip relief comes into contact.

$$\varepsilon_\alpha = \frac{g_\alpha}{p_e} \geq 1 . \qquad (1)$$

(ε_α - transverse contact ration; p_e - base pitch; g_α - length of path of contact)

Fig. 9 shows the flank pressure for gears with half the nominal load. A substantial reduction in the values is to be expected. The variation without tolerances and those with modified profiles (gv10, mgv10, mgv11) corresponds well with expectations. The progressions qualitatively follow those in Fig. 8, and are smaller by a factor of 2. One exception is the gear pair with large tolerances (gv11), for which the peak values, caused by enormous tip interference, are only minimally reduced at disengagement, despite a halved load.

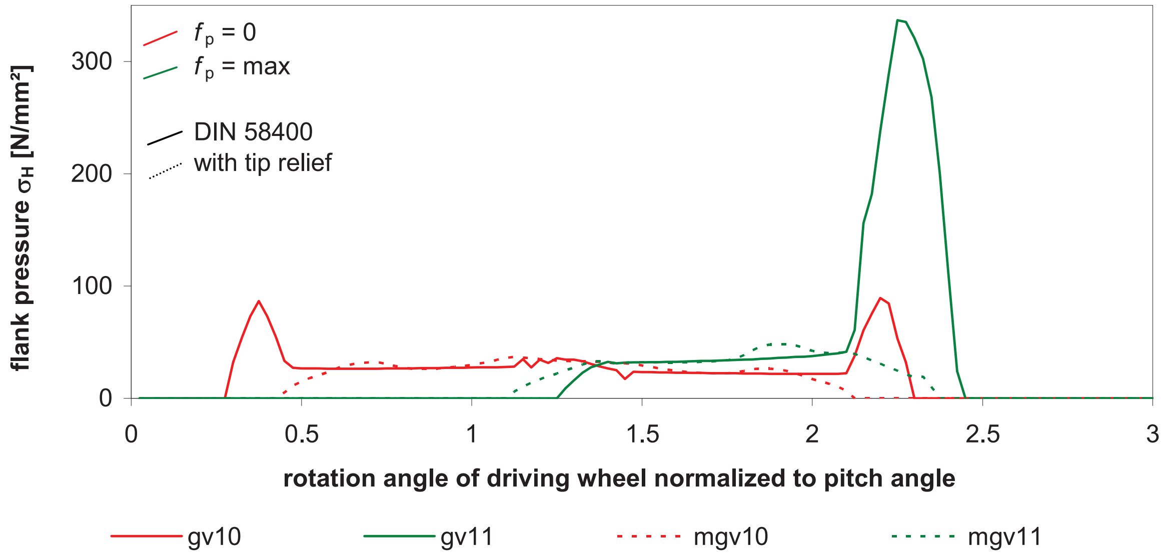

Fig. 9: Tooth flank pressure σ_H of a spur gear pairing for 50 % of the nominal load

4.5. Tooth root stress

In addition to flank pressure, tooth root stress is a second important parameter for dimensioning gears. The following graph shows the progression over the angle of rotation of the driving wheel. As a reference curve, the graph for a steel gear pair without tolerances is included in Fig. 10. As above, the salient features are the double tooth mesh at the beginning and end of the curve, as well as the single tooth mesh in the middle. The larger profile coverage for plastic gear pairs is apparent, as already mentioned. A comparison of the two gear pairs without tolerances (gv2, mgv2) shows that significantly higher stresses occur for the pairs with modified profiles, which is explained by greater deformation (see Fig. 7). The largest stress on the roots of the teeth occur in the variations with maximum single pitch tolerances. For these, tip relief does not alter the process much. In contrast to flank pressure, the tooth root stress values for all gear pairs are within the allowable range.

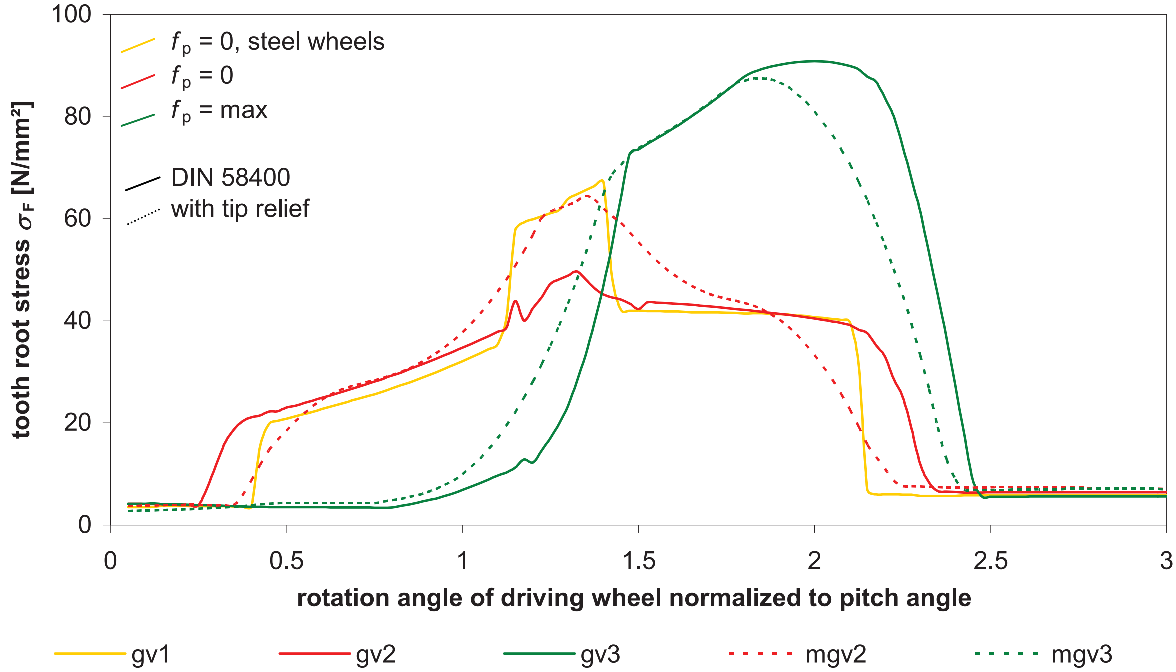

Fig. 10: Tooth root stress σ_F of a spur gear for various profile geometries and single pitch tolerances

4.6. Transmission Behaviour

The transmission error $\Delta\varphi_{ü}$ is calculated as the difference between the actual and nominal angles of rotation for the driven wheel:

$$\Delta\varphi_{ü} = \varphi_{2\_act} - \varphi_{2\_nom} = \varphi_{2\_act} - \frac{\varphi_1}{i} \, . \qquad (2)$$

($\Delta\varphi_{ü}$ - transmission error; φ_{2_act} - actual rotation angle driven wheel;
φ_{2_nom} - nominal rotation angle driven wheel; φ_1 - rotation angle driving wheel; i - ratio)

In Fig. 11, the transmission error $\Delta\varphi_{ü}$, is shown as a function of the driving wheel's angle of rotation. The steel gear pair (gv1) behaves almost ideally. As to be expected, the plastic gears show much larger deviations. Interestingly, the two gear pairs with normalised profiles (gv2, gv3) behave almost flawlessly over half the pitch angle as well. Also apparent is, however, the transmission error over the angle of rotation in the first part of the graph that can be attributed to the massive meshing disturbances. Tip relief only minimally worsens the behaviour of variations without tolerances (mgv2); however, there is clear improvement for the gear pair with maximum single pitch tolerance. It is thus expected that the transmission behaviour of a real gear pair with tolerances would not be worsened by tip relief.

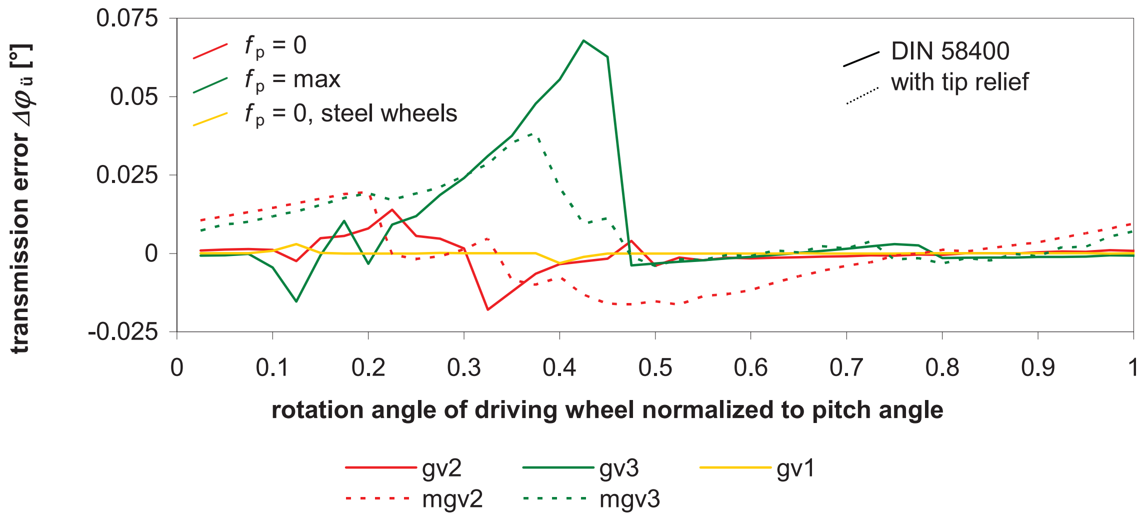

Fig. 11: Transmission error $\Delta\varphi_{ü}$ over the angle of rotation of a spur gear pair for various profile geometries and single pitch tolerances

4.7. Frictional work

The behaviour of the contact area on the meshed flanks influences the generation of noise and of heat on the tooth surface, and with that the efficiency and frictional wear. Not only is the effective flank pressure σ_H relevant, but also what distance s_G slides with that pressure. A physical quantity that takes both into consideration is the frictional work normalised to area, W_R/A. This is calculated according to Eq. (3):

$$\frac{W_R}{A} = \mu \int \sigma_H \, ds_G \, . \qquad (3)$$

(W_R - frictional work; A - area; μ - frictional coefficient; σ_H - flank pressure;
s_G - sliding distance)

The tooth flank pressure σ_H is discussed in detail in section 4.4. The sliding distance of these gear pairs can be seen in Fig. 12. For all variations without tolerances (gv1, gv2, mgv2), a symmetrical progression is apparent. At the point of engagement, there are large sliding distances that decrease continuously until the flanks make contact at the pitch point. At that point, there is no sliding. For the rest of the mesh, the sliding distance increases again, and large values are again reached at disengagement. If the case involves maximum single pitch tolerances, contact between the two flanks occurs short befor the pitch point. At disengagement, however, large distances are covered.

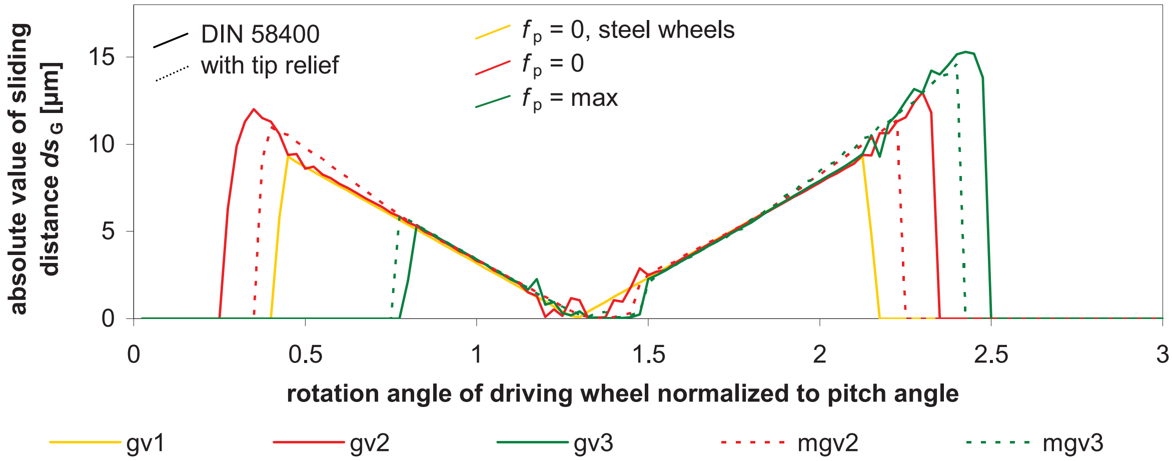

Fig. 12: Actual sliding distance ds_G of a spur gear pair for various profile geometries and single pitch tolerances

Fig. 13 depicts frictional work for one gear mesh. Of note are the high values for the plastic gears without tolerances (gv2) at the beginning and end of the characteristic curve. In these stages, large sliding distances are covered under high flank pressure. Because of this, local generation of heat and high frictional wear are to be expected on the tooth tips and on the corresponding tooth roots. These values are greatly exceeded by those for the variations with high tolerances (gv3). There, enormous wear is to be expected. Significant improvement in the frictional work is achieved by the gear pairs with tip relief. The characteristic curves do not have any peaks and are, despite higher coefficients of friction, on the same level as the steel gear pair.

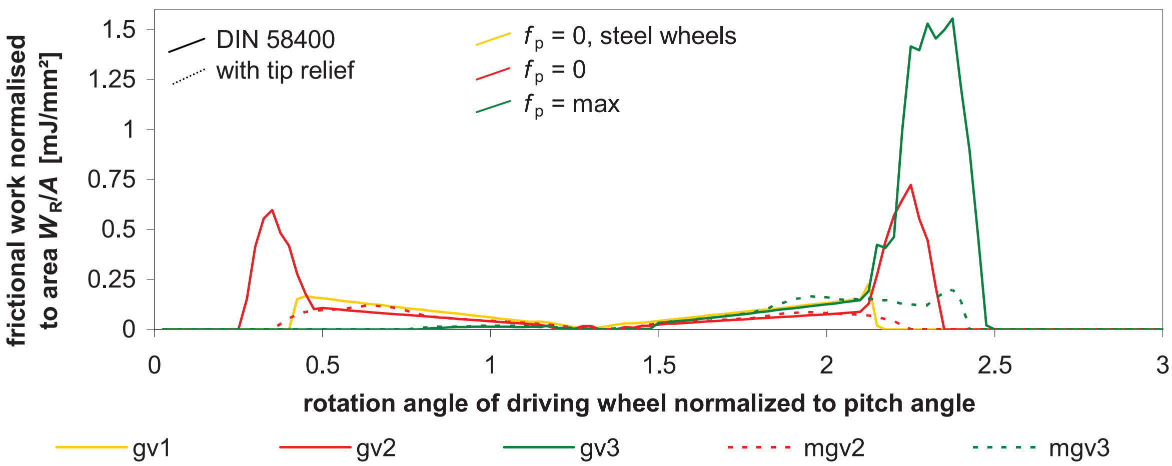

Fig. 13: Frictional work W_R/A of a spur gear pair for various profile geometries and single pitch tolerances

5. CONCLUSIONS

In summary, the main difficulty in the operating behaviour of miniaturised plastic spur gears used for power transmission is the large tooth deformation superposed with the single pitch tolerance. In the worst case, it results in a significant meshing interference at the beginning and at the end of the path of contact. Therefore, the tooth profile should generally be modified by tip relief for gears with modules $m < 1$ mm. This modification hardly influences transmission error, but the meshing behaviour is greatly improved. This causes less wear on the gears and reduces noise.

6. ACKNOWLEDGEMENTS

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