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EXPERIMENTAL AND FEM ANALYSIS OF SINTERED STEEL WORM GEAR WEAR

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Summary

Crossed helical gears with steel worms and sintered steel wheels are widely used in motor vehicles and household appliances. Besides the experimental study of a prototype transmission, the simulation using the finite element method (FEM) is another possibility to test gears under load in the design phase and to optimize them. As part of this paper, a piece of software was developed that enables calculation of wear-related variables, such as the Hertzian stress σ_H, the minimum lubrication film thickness h_{min} and the glide path s_g, for every single point of the contact between worm and worm gear.

Key words: *sintered steel, transmission, gear, FEM*

1. Introduction

For a number of years the Chair of Mechanical Components and Power Transmissions (LMGK) at the Ruhr-University Bochum has been conducting research in the field of crossed helical gears with pinions or worms made of steel, and gear wheels made of plastic or sintered metal. Gearboxes of this type are widely used in motor vehicles and household appliances. For example, the trend towards increased comfort in motor vehicles has led to the utilization of more than a hundred servo drives in luxury automobiles (Fig 1). Servo drives are cheap, quiet, reliable and require minor or no maintenance [1].

In the design of crossed helical gears for a particular use, only in rare cases, for reasons of time and costs, a real prototype will be manufactured. The designer tries to design the gearbox with adequate security. There is a risk of oversizing the transmission, thus causing high costs [2].

Besides the experimental study of a prototype transmission, the simulation using the finite element method (FEM) is another possibility to test gears under load in the design phase and to optimize them [3]. Up to now, the FEM simulation of crossed helical gears has had only little application because of a complex model that should be created and the fact that it should have high mesh precision [4][5].

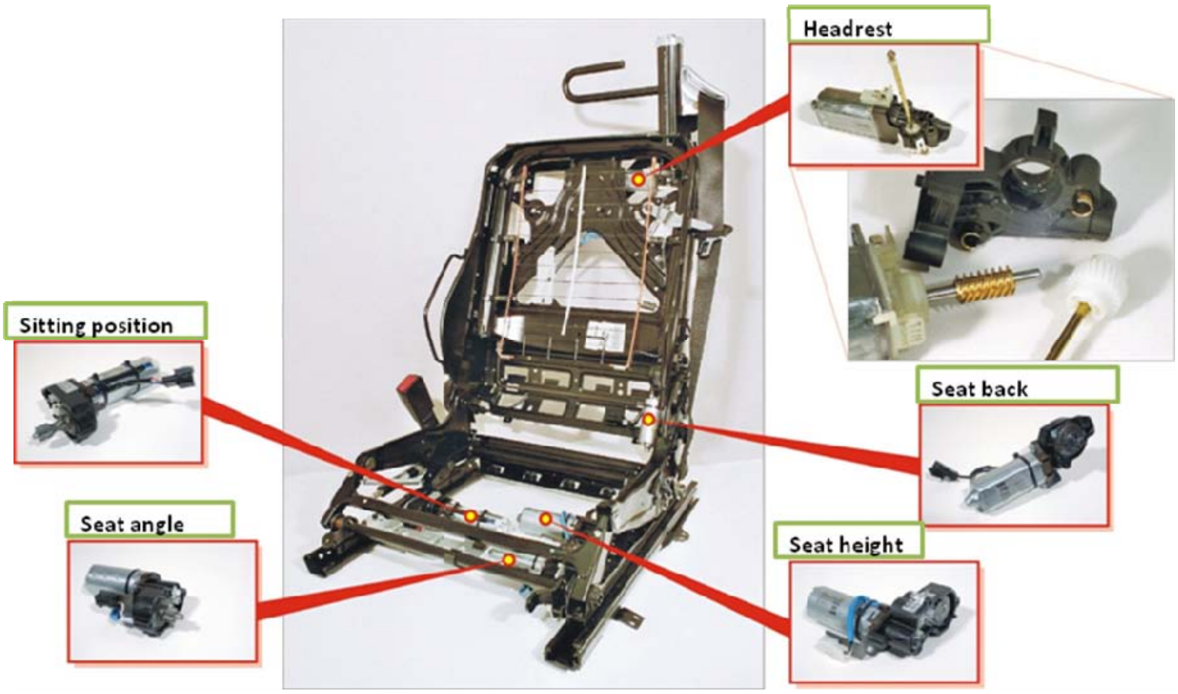


Fig. 1 Crossed helical gears in drive units for seat adjusting in BMW E46 C [6]

An increase in wear removal leads to a widening of the wear surface. Increased wear on the crossed helical gear can lead to early failure of the transmission. The determination of wear-related variables in the design phase is therefore very important [7].

2. Test Conditions

The practical tests were carried out by using five test benches with a center-to-center distance of 30 mm. A transmission of the asynchronous motor was mounted on the test bench and the output torque was applied via a magnetic particle brake. On each test bench, the engine and the gearbox, as well as the gearbox and the brake, were connected with a gear coupling. The measurement of the output torque was made on the transmission with a torque gauge bar via a slip ring transmitter. The speeds and output torques were controlled independently for each test bench. The test bench for the worm with helical gear and the position of the measuring points is shown in Figure 2. The parameters of the test gear pair are given in Table 1.

The test benches were made of aluminum. The test bench with measured points is shown in Figure 2 left and the test gear pair for crossed helical gears is shown on the right.

Test gears were measured before each test in the gearing measuring center Klingelnberg PNC 64. Figure 3 shows a case of wear on wheel teeth. A line along the tooth on the base circle of the wheel was measured in the gearing measured center. This line showed how great the wear width was.

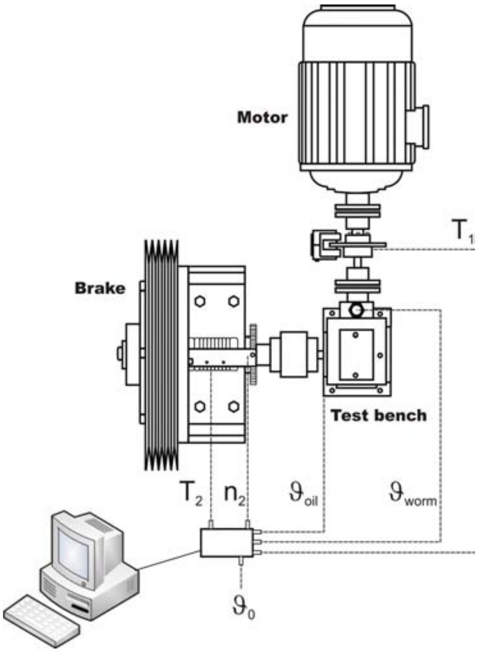


Fig. 2 Test bench and test gear pair

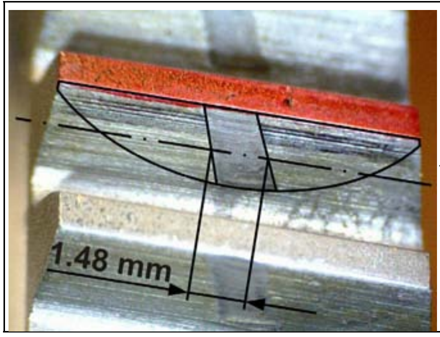
	Wear width l_B [mm]	Wear surface F_B [%]	Test duration L_h [h]	Output torque T_2 [Nm]	Input speed n_1 [min^{-1}]	Temp. ϑ_{oil} ($\Delta\vartheta_S$) [°C]
	1.48	23.6	120	20	5000	75 (55)

Fig. 3 Wear on tooth flank of the wheel

Table 1 Parameters of test gears

Parameter	Data
Center distance	30 mm
Module	1.252 mm
Ratio	40
Pressure angle	20°
Wheel material	Fe1.5Cr0.2Mo
Worm material	16MnCr5
Speed	1500 – 10000 min^{-1}
Torque	12-36 Nm
Lubrication	Klüber GH6 1500 (synthetic oil)

Lubrication of gearing was carried out by splash lubrication. The oil filling of the test drive was done is such a way so as to cover the worm completely with oil.

3. FEM investigation

3.1. SGGen

The Chair of Mechanical Components and Power Transmissions (LMGK) at the Ruhr-University Bochum has developed a net generator (Schraubradmodellgenerator - SGGen) which can calculate, using FEM, the Hertzian contact stress in the contact of the worm and the wheel. This net generator can generate a wheel flank in the new or wear state. It can also determine stresses for the gear flank in the new or wear state.

Software tool SGGen can automatically generate a gear pair for an analysis with the FEM. SGGen forms a net of crossed helical gears with any size as a FEM model for the MSC.Patran software with a MSC.Marc solver.

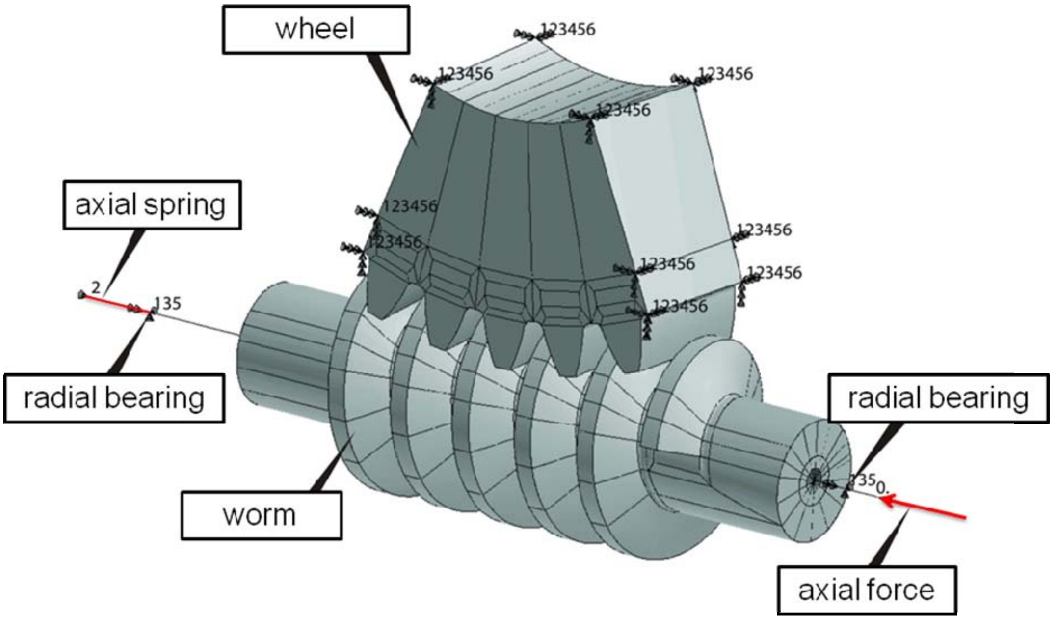


Fig. 4 Worm and wheel – constraints of the degrees of freedom

Figure 4 shows the model of the worm and the wheel with constraints and force. The wheel is presented as a segment of five identical teeth.

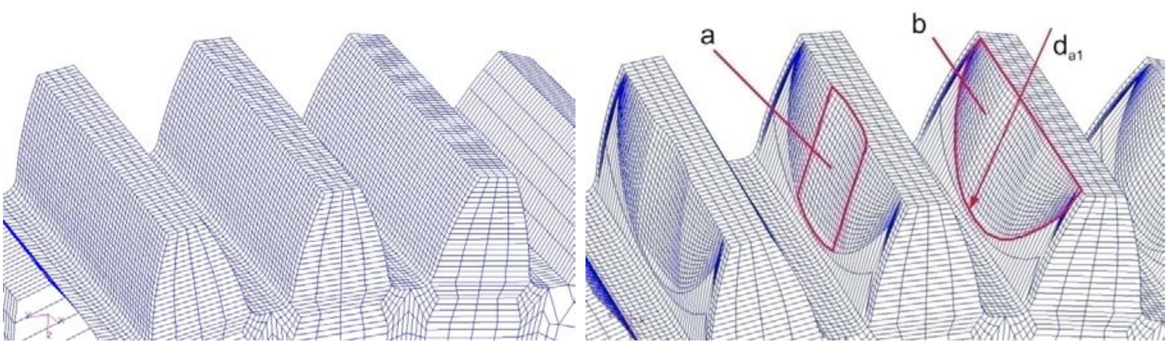


Fig. 5 FEM model of crossed helical gear in new and in wear state (a – wear surface, b – maximal contact surface, d_{a1} – addendum circle of worm)

Figure 5 shows the FEM model of the wheel in the new and the wear state. The wear state of the wheel flank is on the right side of the figure.

3.2. Contact lines of crossed helical gears

Crucial for the crossed helical gear are the location and the shape of the contact line in the contact pattern. The contact line of crossed helical gears is a curved line. The wheel has the shape of a helical gear. This means that there is a contact in point between the worm and the wheel in the beginning, and the contact point grows in the contact line due to wear on the tooth flank of the wheel. An important step in determining the physical properties of crossed helical gears is the calculation of contact lines for this gear type. Figure 6 shows contact lines in the radial section of a tooth flank with the wear of 21, 44, 78, 110, and 165 μm.

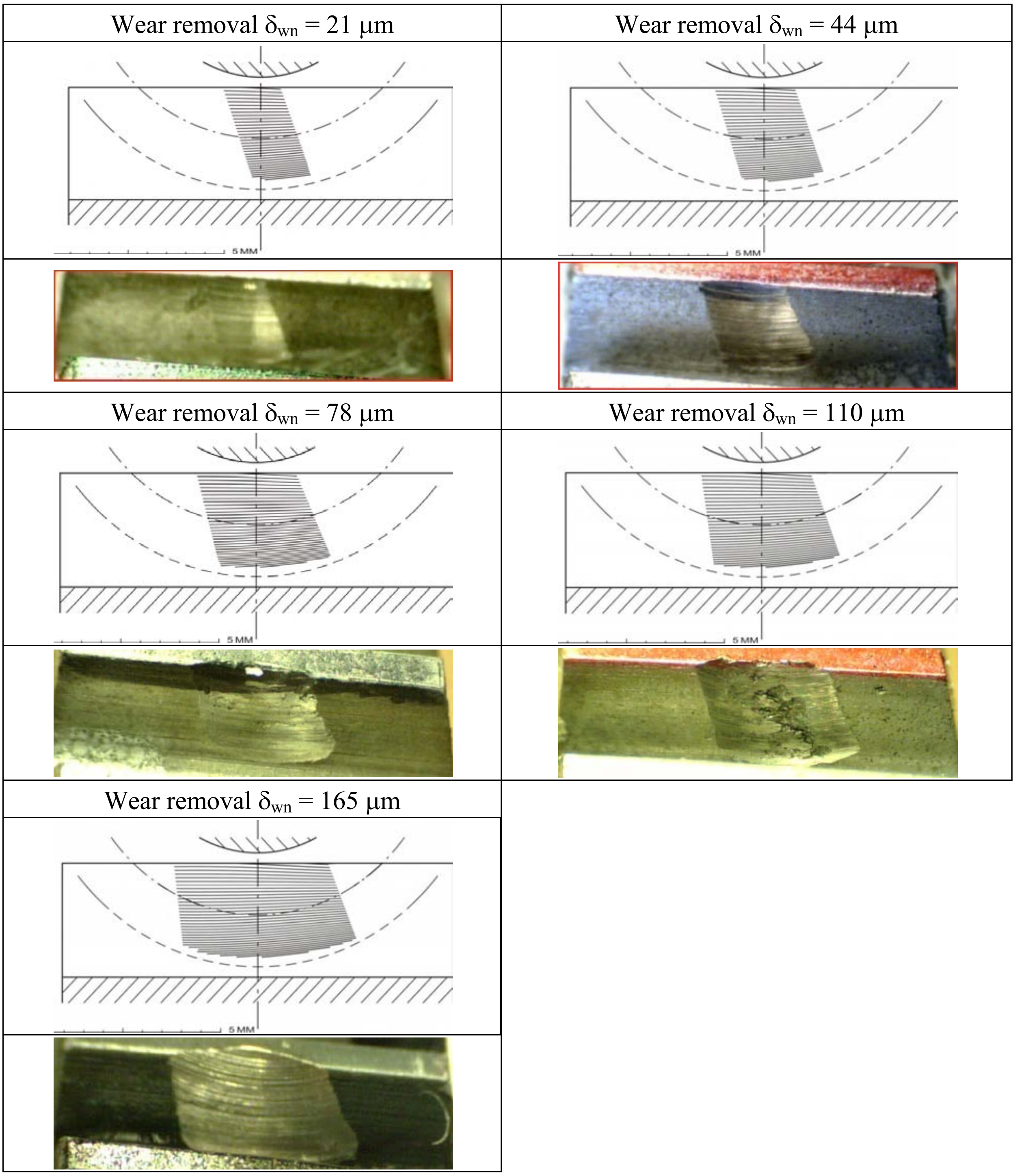


Fig. 6 Experimental and FEM results of wear in wheel flanks for different wear removals

The increase in wear removal leads to a widening of the wear surface, which results in lower Hertzian contact pressures [8].

3.3. Analysis of wear-relevant variables

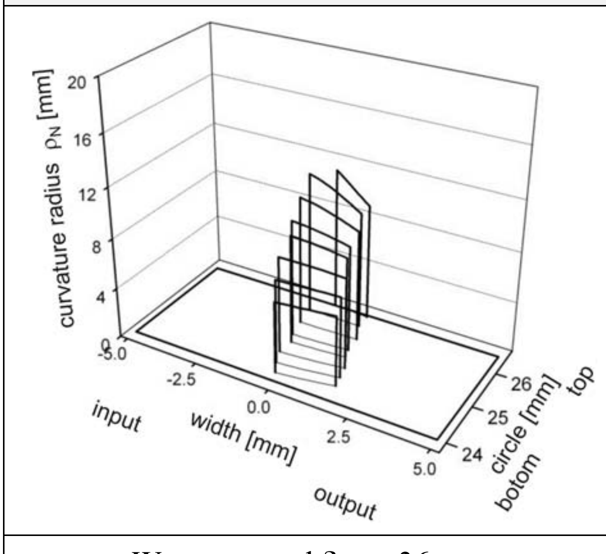
Modern FEM software applications calculate a variety of physical quantities. To use these quantities for the calculation of load capacity of crossed helical gears a software tool was created. Hertzian surface pressure σ_H in the tooth contact is of great importance for crossed helical gears. It is necessary to have at least 10 elements within the Hertzian width in order to take the actual curvature radius in the FEM model into account.

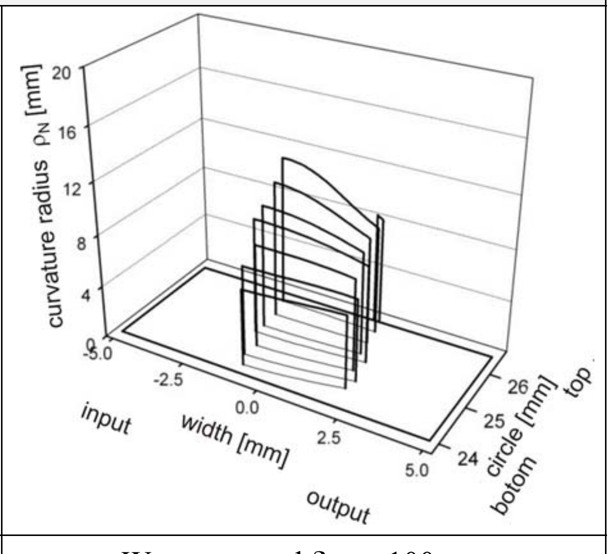
The developed software uses the ZSB program [9] and allows calculation of wear-related variables, such as the Hertzian stress σ_H, the minimum lubrication film thickness h_{min} and the sliding distance s_g for every single point of contact between the worm and the wheel. [10] defines the glide distance as the sliding path of the contact point of the worm in one of Hertz's areas of the worm gear. [11] develops a piece of software for the analysis of the contact pattern of any location and size of the worm gear. This allows the limitation of the contact area on the wear surface of the wheel and the worm and the calculation of normal tooth forces on the line load curve in the FEM model. This way the software can be used to determine the wear-relevant variables for a wheel with wear. Wear-related variables are the Hertzian stress σ_H, the minimum lubrication film thickness h_{min} and the glide path s_g of the wear wheel[12].

Wear intensity J_w of crossed helical gears can be calculated after determining the Hertzian stress σ_H, the minimum lubrication film thickness h_{min} and the sliding distance s_g for each contact point and for different material/lubrication combinations. However, currently there is no available calculation for wear intensities J_w in a single contact point. The wear intensity for single contact points can only be determined experimentally [13].

In the following section, the results of parametric studies are presented for the test gear data. Figure 7 presents the values of the equivalent curvature radius for different wear removals. It can be seen that the variation of wear rates has no influence on the equivalent curvature radius.

equivalent curvature radius ρ_n [mm]





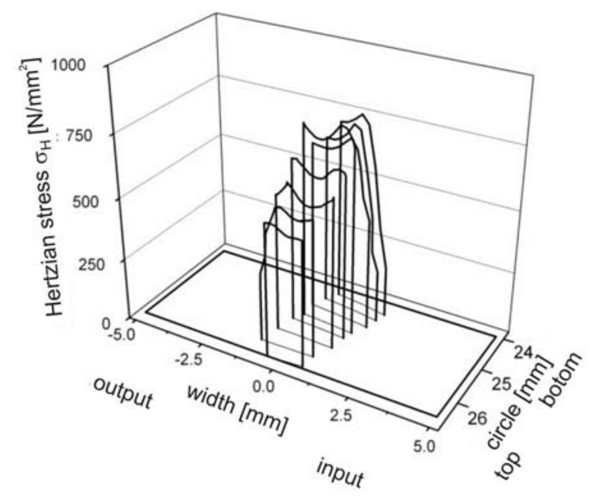
Wear removal δ_{wn} = 36 μm

Wear removal δ_{wn} = 100 μm

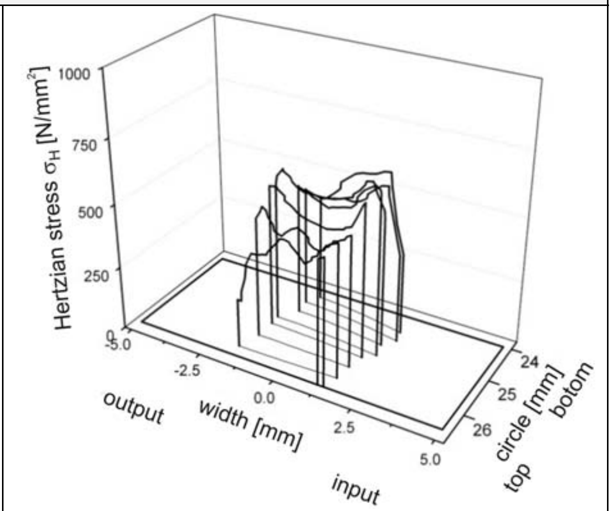
Fig. 7 Equivalent curvature radius for different wear removals

Figures 8, 9, 10 and 11 show the values of Hertzian stress σ_H, contact line load q, slide path s_g and minimum lubrication film thickness h_{min} for different wear removals. It can be seen that the increase in wear removals δ_{wn} leads to a broadening of the contact surface and to a reduction in wear-relevant variables such as Hertzian stress σ_H, contact line load q, slide path s_g and minimum lubrication film thickness h_{min}.

Hertzian contact stress σ_H [N/mm^2]; output torque T_2 = 12 Nm



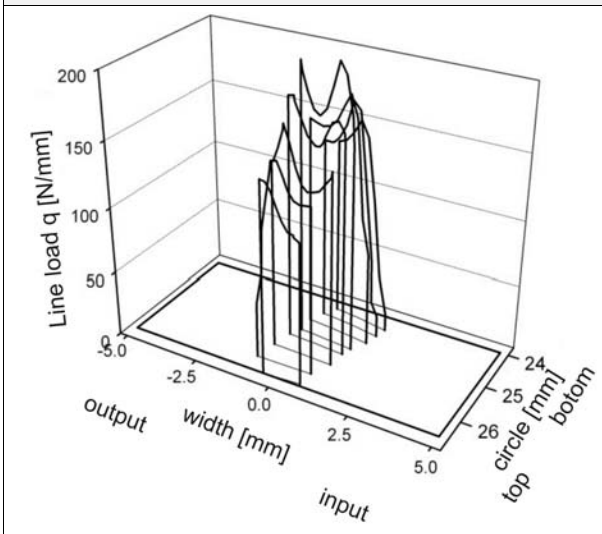
Wear removal δ_{wn} = 36 μm



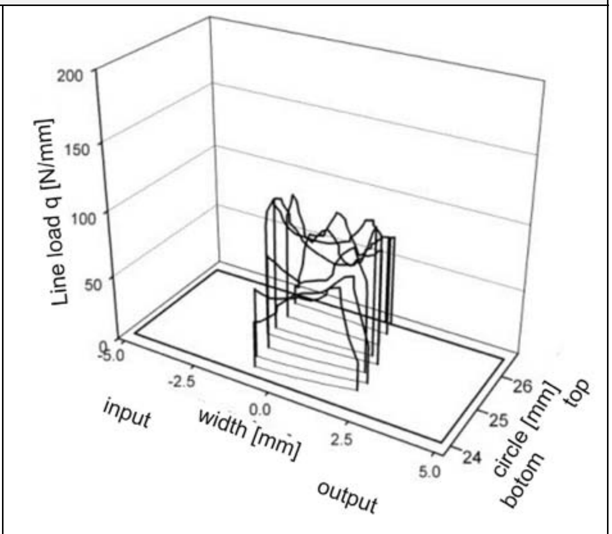
Wear removal δ_{wn} = 100 μm

Fig. 8 Hertzian contact stress for different wear removals

contact line load q [N/mm]; output torque T_2 = 12 Nm



Wear removal δ_{wn} = 36 μm



Wear removal δ_{wn} = 100 μm

Fig. 9 Contact line load for different wear removals

The wheel and the worm have an ideal geometry. The distribution of the Hertzian contact stress σ_H along the contact lines between the worm and the wheel with wear is not constant.

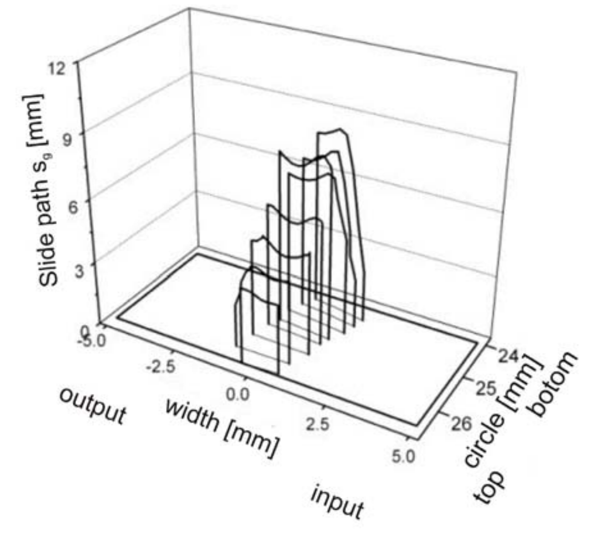
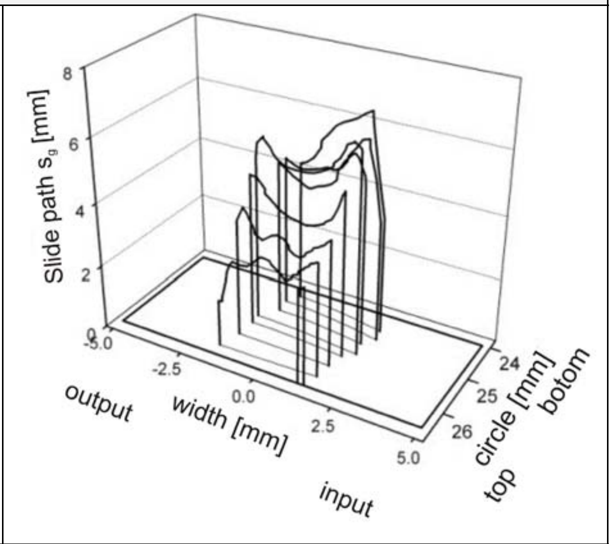
slide path s_g [mm]; output torque T_2 = 12 Nm	
	
Wear removal δ_{wn} = 36 µm	Wear removal δ_{wn} = 100 µm

Fig. 10 Slide path for different wear removals

The minimum pressure is always in the middle of the teeth. The smallest values of the minimum lubrication gap thickness h_{min} are also in this region of the flank.

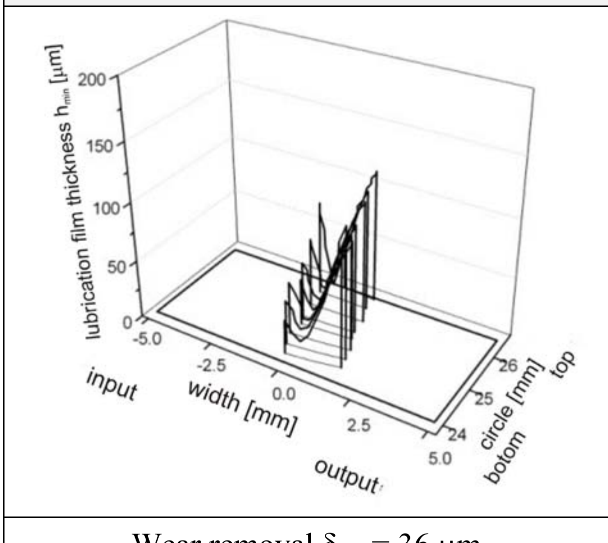
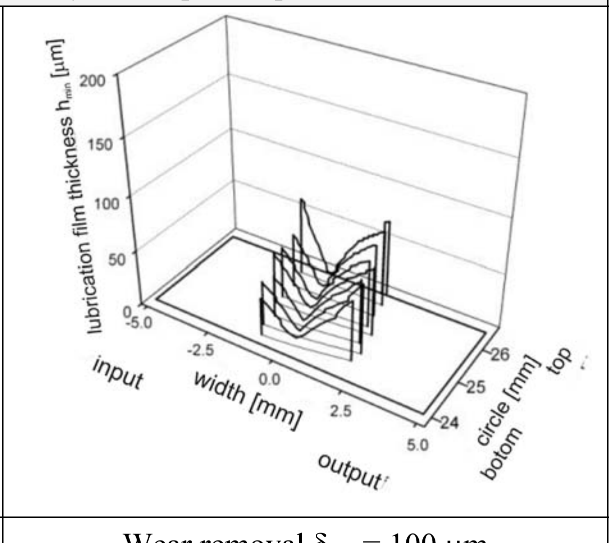
minimum lubrication film thickness h_{min} [µm]; output torque T_2 = 12 Nm	
	
Wear removal δ_{wn} = 36 µm	Wear removal δ_{wn} = 100 µm

Fig. 11 Minimum lubrication film thickness for different wear removals

4. Comparison of experimental and FEM results

To generate a wheel with wear, a method was developed which made it possible to generate a globoid form on a cylindrical flank. The globoid form was generated by superimposing a globoid flank in a cylindrical flank. The wear removal describes the highest loss of material due to wear on the base circle of the wheel. In order to generate wear, the globoid flank was rotated by the angle $\Delta\varphi$ in the wheel.

Figure 11 shows a wheel flank with and without wear in the transverse section with wear removal δ_{ws}. Wear removal δ_{wn} is the change of tooth thickness on the base circle in the normal section.

$$\delta_{wn} = \delta_{ws} \cdot \cos\beta_{s2} \tag{1}$$

Since the helix angle β_{s2} was relatively small ($\beta_{s2} = 7.507^{o}$; $\cos\beta_{s2} = 0.99$), $\delta_{wn} \approx \delta_{ws}$ could be adopted. Angle $\Delta\varphi$ could be calculated with the wear removal and the base circle of the wheel $d_{s2}/2$, which enabled the calculation of the coordinates of the globoid flank that were rotated compared to the coordinates of the cylindrical flank.

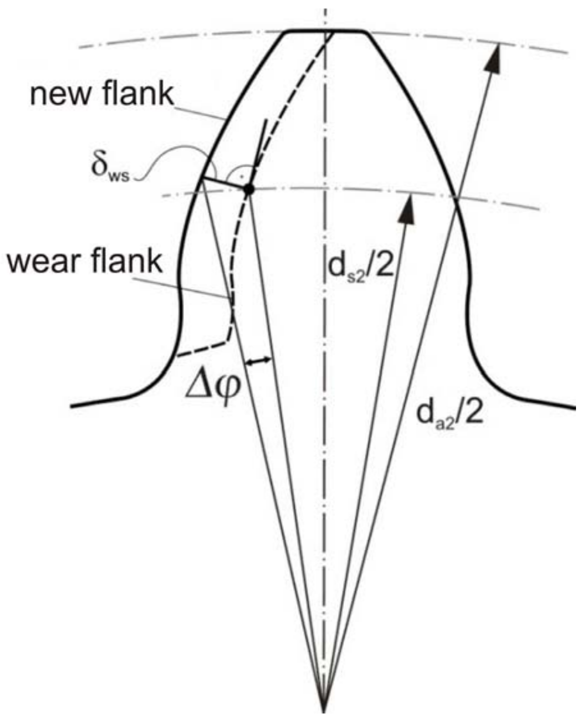


Fig. 12 Transverse section of a wheel with and without wear

Since the most wear on the tooth in the transverse section was unknown, the software calculated the angle that was formed on the base circle between the tooth flank with and without wear. Each point on the base circle of the cylindrical wheel had an assigned point on the globoid flank. The coordinates of the globoid flank were rotated until the distance between the two points corresponded to the given wear rate.

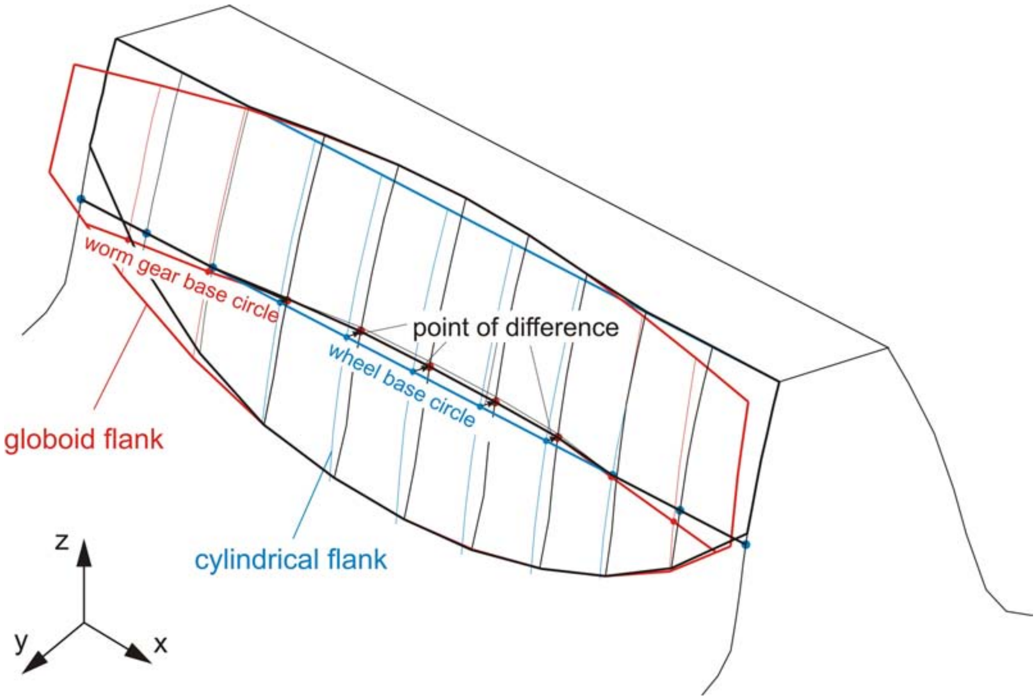


Fig. 13 Difference between the points of the cylindrical and globoid flank on base circle

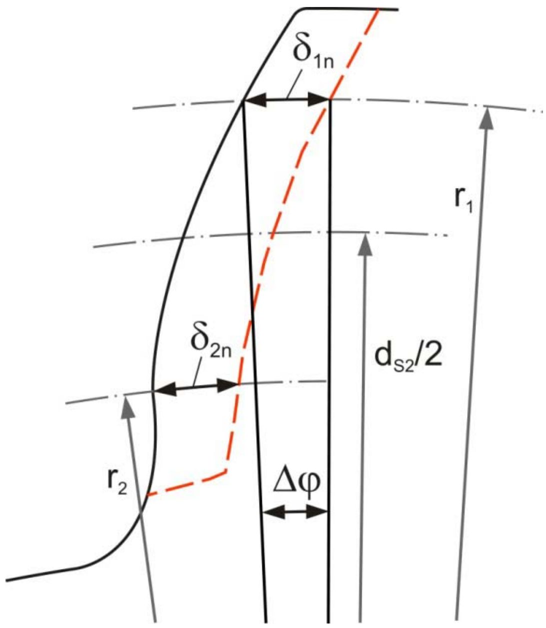


Fig. 14 Wear removal as a function of the tooth height

The wear part of the tooth flank had the form of a worm wheel, in which the worm had the function of the milling cutter. Figure 13 shows the superposition of the globoid flank of the wheel in the cylindrical flank Figure 14 shows the wear removal as a function of the tooth height.

The calculation of the wear removal in the normal direction is given in Equation 2.

$$\delta_{in} = \Delta\varphi \cdot r_i \tag{2}$$

Figure 15 compares the measured profile changes on the base circle of the wheel with the calculated profile changes.

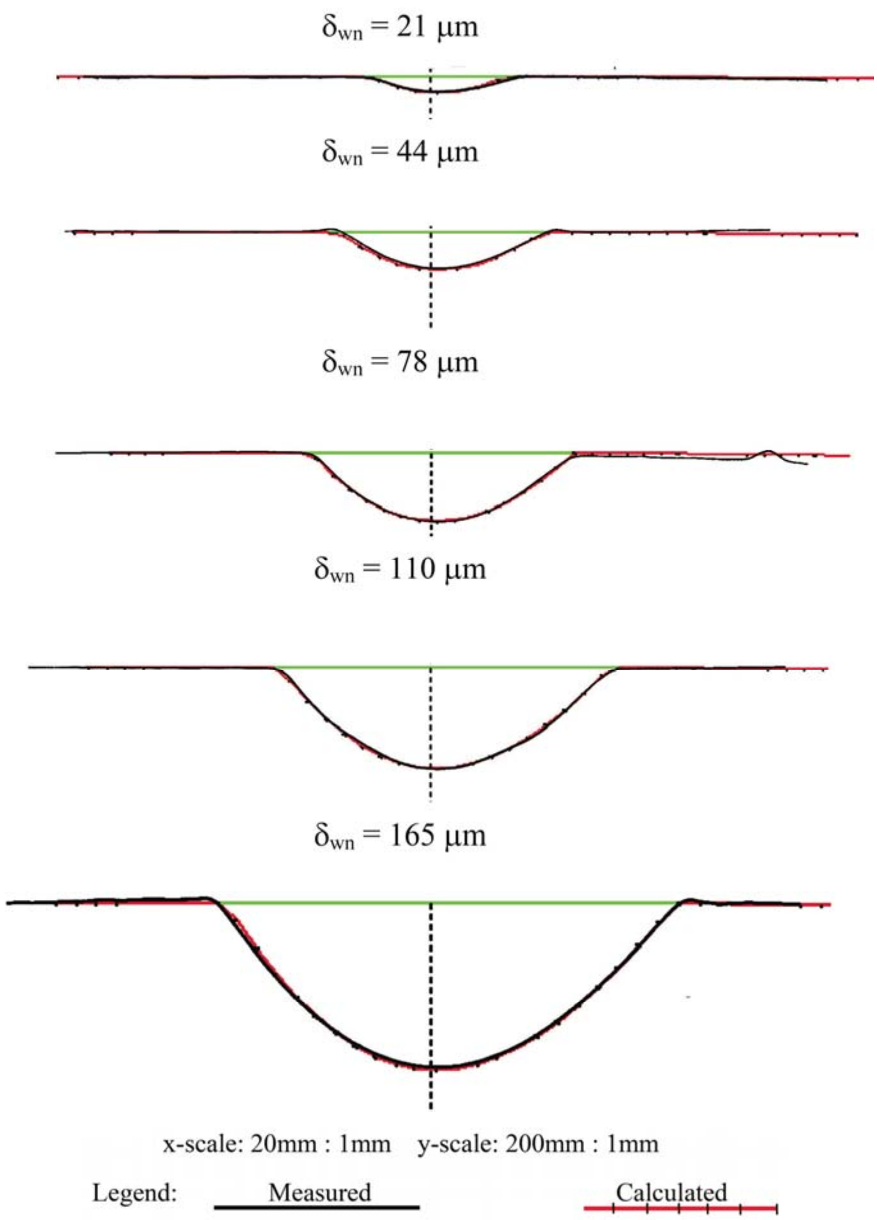


Fig. 15 Comparison of calculated and measured wear removals

5. Conclusion

FEM and the developed software tool can be used in the calculation of relevant parameters of crossed helical gears as well as for their design. However, before applying the method, it is necessary for this approach to be validated by a comparative analysis with experimental results. For crossed helical gears, the wear of wheel tooth is the most common cause of failure.

The conducted analysis shows that there is a high degree of agreement between the experimental and the software results for the wear-relevant values. Therefore, this type of calculation gives relevant information for the design of the optimal crossed helical gear drive.

The developed software allows the calculation of wear-related variables, such as the Hertzian contact stress σ_H, the minimum lubrication gap thickness h_{min} and the glide path s_g, for every single point of contact between the worm and the wheel.

Based on the results of the parameter study, the following can be stated:

- The calculated values of the Hertzian contact stress σ_H and the contact line load q show that the increase in wear removal δ_{wn} leads to a broadening of the wear surface and a reduction in the Hertzian contact stress σ_H and the contact line load q.
- The distribution of the Hertzian contact stress σ_H along the contact line between the worm and the wheel is not constant.
- There is a direct correlation between the distribution of the spare radius of curvature ρ_n and the Hertzian stress σ_H. The pressure distribution in the first approximation is obtained as a direct combination of the stiffness and the radius of curvature of the gearing.
- The minimum pressure is always in the middle of the teeth. The smallest values of the minimum lubrication gap thickness h_{min} are also in this region of the flank. In combination with the lower surface pressures, it can be assumed that similar wear occurs on the input and the output side. Here, the value of Hertzian stresses σ_H and the minimum lubrication gap thicknesses h_{min} are higher.

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