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INFLUENCE OF MANUFACTURING VARIATIONS OF PROTUBERANT TEETH SHAPES ON GEARING STRESSES OF HIGH-SPEED GEARBOXES

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Abstract: *The paper describes the solution of a current problem related to the technology of production of gear for high-speed gearboxes. The aim is to establish a reliable basis and procedure for evaluating the influence of manufacturing variations in the shape of the heel of the protuberant teeth on the stress and load capacity of the gearing. The capabilities of the software used so far bring rather uncertainty. A number of challenging FEA stress calculations for different variations of tooth heel shape deviations that occur in practice have been solved. The effect of changing the mechanical properties of the material with the depth of grinding is discussed. The results will be used both in the design of gearing and as a basis for deciding how to handle parts with defects, whether to continue using them or to scrap them. This will help to increase the efficiency of production and the reliability of gearboxes.*

Keywords: Gears, Protuberant gearing, FEA, Tension, Load capacity, Case hardening.

1. Introduction

Gears are manufactured at Wikov Gear, Ltd. mainly by roughly milling the gears by rolling. This is followed by carburizing and heat treatment. Finally, only the working surfaces of the gears are ground cleanly with precisely aligned shaped wheels.

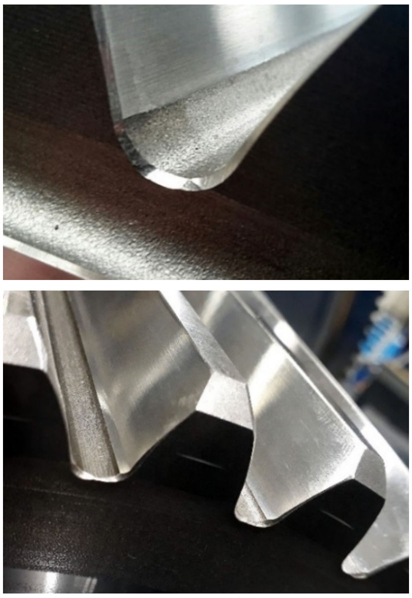

Fig. 1 Correct protuberant gearing (top) and notched gearing from the grinding wheel run-out (bottom).

In this procedure, so-called protuberant gearing is usually used. The shape of the tooth is modified so that a milling cutter with a modified profile is used to create a targeted undercutting of the tooth base during machining prior to case-hardenig. This has several advantages. The hardened surface layer in the base of the tooth is not removed, the involute runs out into the empty space and does not create a notch, the base of the tooth is not thermally affected by grinding and the radius of transition to the base circle is increased, reducing the notch coefficient.

The problem is that carburizing and heat treatment often results in some shape change. In the final grinding process, either material is missing and the wheel runs out into the void, or material is left over. In the latter case, the lower end of the involute ends up notched near the stressed tooth base, see Fig.1. The notch is also variable along the tooth length, with the greatest variation at the face of the gear, where unwanted stress concentration can occur.

The Wikov company uses the KISSsoft system for the design and verification of gearing. However, this only allows to deal with a condition where the notching and protuberanz undercutting is constant along the entire tooth length. Furthermore, KISSsoft exhibits extreme deviation and discontinuity of stress in the involute to tooth base transition, which is unexpected here. This is where the tension needs to be reliably known in order to decide how to deal with such a wheel further.

The main goal of the work is to calculate and assess the stresses in the tooth heel for different manufacturing

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deviations, for 2D cases for straight gearing and for 3D models of helical gearing in engagement. We use ANSYS T Grid 3D, ANSA, our own software GEN, ANSYS Mechanical APDL and COSMOS/M for model generation and complex computations with contacts.

2. Methods of calculation

Material. Defining the material properties of a cemented surface is difficult. We only have a hardness measurement protocol up to a depth of 3.2 mm, beyond that the progression is unknown (Härterei Reese , 2020). However, for the calculation of elastic problems we need to know the elastic modulus E and its distribution in the tooth profile. Unfortunately, elastic modulus is not directly related to hardness and cannot be calculated from it, e.g., (Belete, 2015; Aramidea, 2009; Elzanaty, 2014). Therefore, several calculations have been performed to appreciate the sensitivity of the stress distribution and magnitude on the variable E in the case-hardened layer. The first calculation was performed on a coarse 3D model of 203,818 elements, see Fig. 2. The modulus of elasticity varied linearly by 5% in the case-hardened layer.

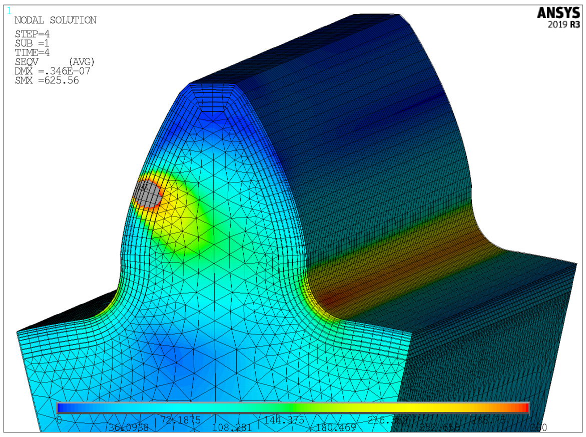

Fig. 2 Reduced HMH stress distribution, variable elastic modulus model.

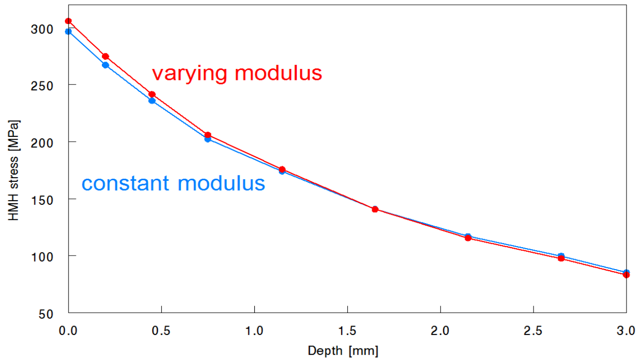

Fig. 3 Comparison of the stress distribution in the tooth base in depth for the constant and variable E.

We performed a detailed comparison of the maximum stresses on the compressive side of the tooth up to a depth of 3 mm with the constant-E model (see Fig. 3), and it turned out that the effect of the gradual change in surface E on the stresses is relatively small, and it is therefore correct to use constant-E models throughout the volume.

Discretization level and 2D vs. 3D model. To determine the optimal computational parameters, a series of test calculations were performed to evaluate the effect of the discretization level. The next step was to perform experimental planar (2D) and spatial (3D) calculations, see Fig. 4, where an almost perfect match between the 2D and 3D stress maxima in the tooth base can be seen. Thus, furthermore, 2D models can be correctly used for straight teeth. We can thus afford a much finer discretization.

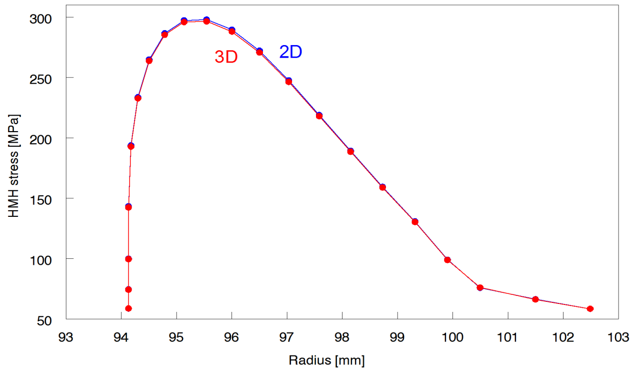

Fig. 4 Curve of reduced HMH stress over the tooth base surface without defect for 2D and 3D models.

The load during the contact of a pair of straight teeth was specified by Wikov, for 3D calculations it is modelled by the pressure in the area of the contact line or based on contact forces. The worst loading is when only one pair of teeth is currently in engagement and the contact force is applied at the largest radius.

3. Comparison of 2D calculations with KISSsoft results

The calculations were performed on a finer 2D model with 8,624 elements. In the surface layer, where there are large stress gradients, the layer thickness is 0.05 mm, see Fig. 5.

The comparison of the results calculated by ANSYS, COSMOS and KISSsoft can be seen in Fig. 6. The maxima in the tooth base are the same. Problematic are the results in the notch from the evolvent's run-out, where KISSsoft gives much higher values and, moreover, there is an unjustifiable discontinuity. KISSsoft seems to conservatively assume the norms and cannot correctly deal with unexpected changes in tooth surface shape.

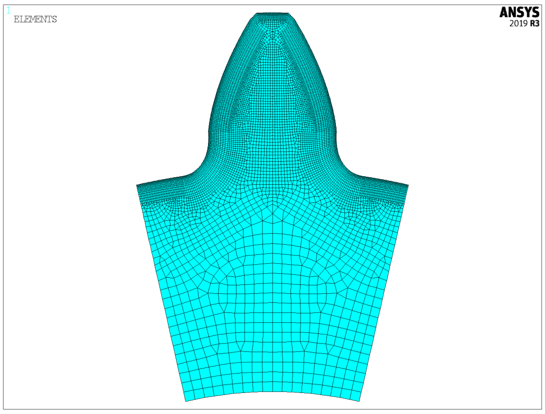

Fig.5 2D model of a tooth without defect, fine mesh.

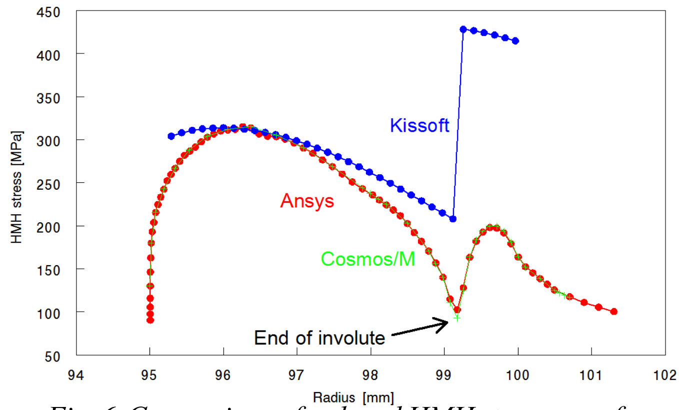

Fig. 6 Comparison of reduced HMH stress waveforms over the surface of a tooth with a defect.

4. A set of typical defects and a graded notch

An important part of the work was to perform several sets of 2D model calculations, with different types of defects. An example of the extensive results is shown in Figure 7, where the stress in the notch even exceeded the stress in the heel radius.

Next, to develop a methodology for qualifying the defects caused by the notching of the involute run-out, we solved a systematic set of cases in which the involute run-out gradually sinks into the radius of the base of the tooth due to radial expansion of the wheel after heat treatment, see Fig. 8.

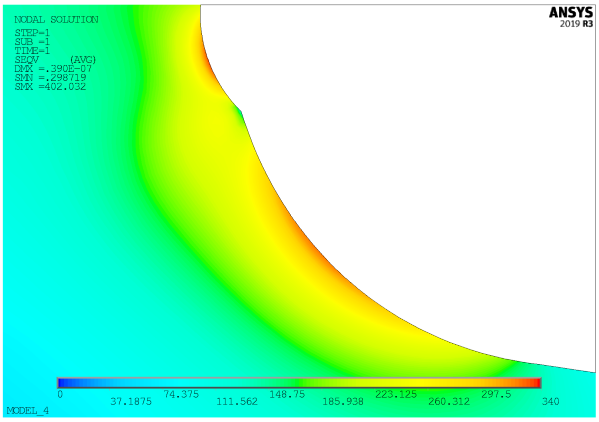

Fig. 7 Detail of the distribution of reduced HMH stress in the tooth base and the involute process.

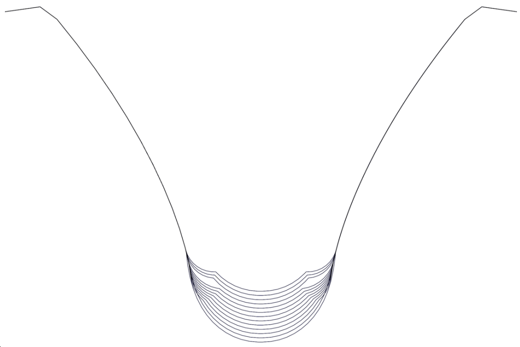

Fig. 8 Profiles of variants of the graded defect.

The results are shown in a condensed form of the reduced stress maxima in the heel radius and notch for all variants in Fig. 9.

The results show that even with increasing normal grinding depth, there is no extreme increase in the stress in the notch at the involute run-out. The results allow a qualitative and quantitative assessment of how to deal with not-perfect product.

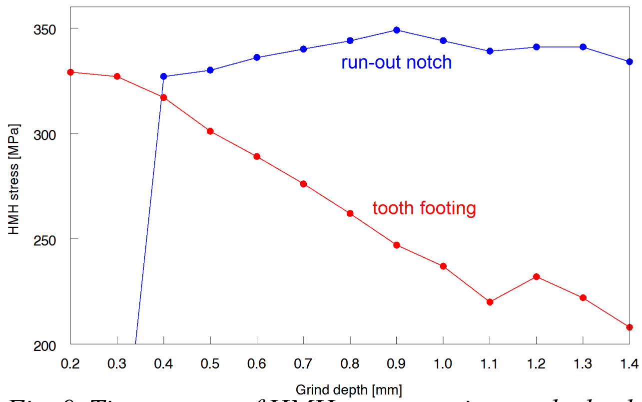

Fig. 9 Time course of HMH stress maxima at the heel radius and at the notch of the evolute process normal grinding-in of the involute.

5. Spatial models of helical gearing

Finally, numerically demanding 3D calculations of the helical gearing were performed, which cannot be correctly replaced by the 2D case. It was difficult to build a model of helical gearing with continuously varying defects between the gear faces. Therefore, we have developed our own software and can now model any varying shape of helical gear.

Entering the load is even more difficult. It is not possible to export the contact force at one time of the engagement from the Wikov program. So we performed experimental sets of calculations, first with loading the contact line with a normal force corresponding to the torque and in the second solution we modeled the whole gearing including the contact between the engaging teeth of the wheel and pinion. An order to model the contact accurately, finer discretization would be needed, but here the correct load transfer between the bodies is sufficient. The 671.454 finite elements model uses auxiliary beams through which, among other things, torque loads can be elegantly implemented.

After this verification, the calculation of the real corrected gearing given by Wikov with a modulus of 10 and a tooth inclination of 10° was performed, see Figure 10.

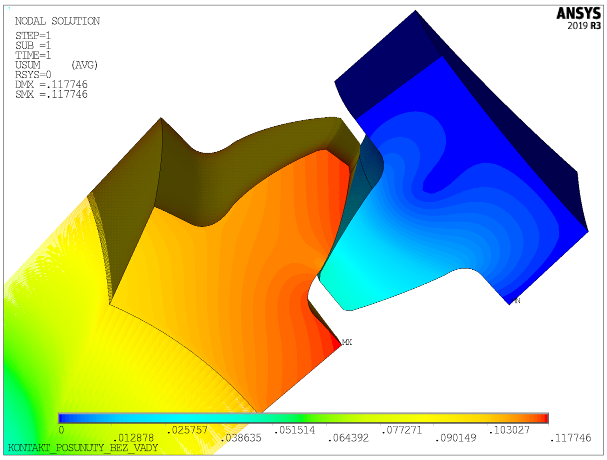

Fig. 10 Distribution of the total displacement on the engaging tooth pair.

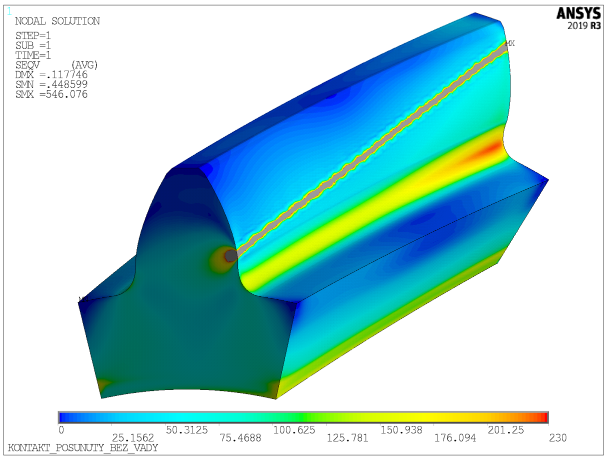

Fig. 11 HMH stress distribution on the side of the tooth.

Figure 11 shows the HMH stress distribution on the tension side of the pinion tooth. The straight contact area and the increasing tension at the tooth base and in the notch as the contact moves towards the tooth head can be seen very clearly. At this pinion face, the variable notch is the largest and there it can be observed a gradual increasing in tension towards the pinion face. We also see that in this case the stress in the involute notch is already higher than in the tooth base.

6. Conclusions

The paper describes the task of evaluating the manufacturing deviations of gear teeth in high-speed gearboxes. The need to solve this problem was caused in particular by uncertainties in the evaluation of not perfectly manufactured protuberant gears.

FEM calculations verified by two different software were used, the problem of the elastic modulus of the case-hardened material, the level of discontinuity, 2D and 3D analyses of differently loaded models were solved, and models of the whole wheelset with helical teeth and with axially variable defect size including contacts between teeth were also solved.

The results will be used in the design of the shape of the gearing and as a basis for deciding how to deal with parts that show defects in the grinding of the involute into the heel, when to release such a part for production and when it is necessary to throw it away.

Acknowledgement

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