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TRANSMISSION ERROR OF THE SPIRAL BEVEL GEARS

Fraňová Z.[*], Řehák K.[**], Prokop A.[***]

Abstract: *This paper deals with the determination of the transmission error in spiral bevel gears and its minimization using tooth profile modification in order to reduce noise and vibration of transmission systems. Gears are the primary source of vibrations transmitted through the shaft and bearings to the gearbox housing and adjacent surfaces that emit noise to the surroundings. To increase the level of comfort and due to the legislative requirements, increasing emphasis is being placed on reducing the noise and vibrations of machine components, including transmission systems. This leads to the need to identify noise sources and evaluate them in terms of expected acoustic performance. The quality of the gear meshing can be evaluated by transmission error that is closely related to the noise and vibrations. To evaluate the quality of gears based on transmission error, experimental approaches are used that are costly and require quality equipment. Therefore, it is efficient to determine expected transmission error already at the design stage using numerical methods. In the presented paper, FEM software ANSYS is used for numerical simulation of the spiral bevel gear mesh and the obtained results were used to study the effect of tooth profile modifications on the transmission error.*

Keywords: Transmission error, Bevel gear, Noise, Vibration, Tip relief modification.

1. Introduction

Transmission error can be expressed as a linear displacement at a base circle radius defined by the difference of the output gear's position from where it would be if the gear teeth were perfect and infinitely stiff. Many references have attested to the fact that a major goal in reducing gear noise is to reduce the transmission error of a gear set (Smith, 1999 and Åkerblom, 2008). The basic formula for calculating the transmission error is given as

$$TE = r_2\theta_2 - r_1\theta_1 = \left(\theta_2 - \frac{r_2}{r_1}\theta_1\right) r_2, \tag{1}$$

where θ_1 and θ_2 are the rotation angles of pinion and wheel respectively and r_1 and r_2 are pitch radiuses of the pinion and wheel, respectively. The advantage of specifying the transmission error as a linear measurement is a possibility of comparing gear quality independently of their dimensions.

Predicting the static transmission error is a necessary condition for reducing noise radiated from the gearbox. Since transmission error and variations in gear mesh stiffness are considered to be the main excitation mechanisms of gear noise, minimization of the transmission error should result in gear noise reduction (Smith, 1999 and Åkerblom, 2008). A suitable method for evaluating and comparing the transmission error is to determine its peak-to-peak value, which is calculated as the difference between the maximum and minimum value of the transmission error. This parameter describes the effect of all deformations and inaccuracies in the gear and is therefore considered one of the most appropriate parameters to predict the expected noise and vibration generated by the gear.

Many authors deal with the prediction of the transmission error at the design stage using theoretical and numerical methods. Tharmakulasingam (2009) and Lin (2017) are using the finite element method for transmission error determination and design optimization. Other authors use experimental approaches

[*] Bc. Zuzana Fraňová: Institute of Automotive Engineering, Brno University of Technology, Technická 2896/2; 616 69, Brno, CZ, 171331@vutbr.cz

[**] Ing. Kamil Řehák, PhD.: Institute of Automotive Engineering, Brno University of Technology, Technická 2896/2; 616 69, Brno, CZ, rehak@fme.vutbr.cz

[***] Ing. Aleš Prokop, PhD.: Institute of Automotive Engineering, Brno University of Technology, Technicka 2896/2; 61669, Brno; CZ, ales.prokop@vutbr.cz

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to determine the transmission error, including Smith (1999), Henriksson (2009) and Tůma (2014). The above-mentioned works deal mainly with the transmission error of the spur gears. However, fewer authors deal with the transmission error of bevel gears. The few available sources include Mermoz (2013), where a detailed FEM model is used to determine the transmission error of bevel gears, and Astoul (2014), where the results of FEM analyzes are compared with experimental data.

2. Methods

In the presented paper, a numerical FEM model is used to determine the transmission error of spiral bevel gears and to compare individual tooth profile modifications. The bevel gear geometry was created parametrically in the CAD software Creo Parametric so that it is possible to automatically generate gears of new dimensions by changing input parameters such as the number of teeth, module, pressure angle and face width. To reduce the computational time, only the number of teeth required for the calculation was included. By removing the remaining teeth from both the wheel and the pinion, the number of elements used for generated mesh was reduced. Since the meshing of the gears leads to contact analysis, great emphasis was placed on the appropriate settings of the contacts, which were represented by the pairs of tooth flanks.

Since contact is defined on the tooth flank surfaces, the quality of mesh in this area has a significant effect on the accuracy of the results, and therefore it is necessary to generate a fine and uniform mesh. For this reason, the hexahedral swept mesh was generated on the outer layer of the teeth. The element sizes for the discretization of the geometry were determined based on mesh size sensitivity analysis to obtain sufficiently accurate results in relation to the computational time.

For the correct simulation of gear meshing, it is important to properly set the boundary conditions and applied loads. The wheel is allowed to rotate freely about the axis of the pitch cone and the pinion rotates by a constant rotation angle of 1 ° in each step of the simulation so that it is possible to calculate transmission error at successive positions of the pinion as it goes through the meshing cycle. In the first step the wheel is preloaded with a partial torque and in the next step full torque is applied on the wheel. The remaining degrees of freedom are restricted by preventing axial displacement of the wheel and pinion on their outer faces.

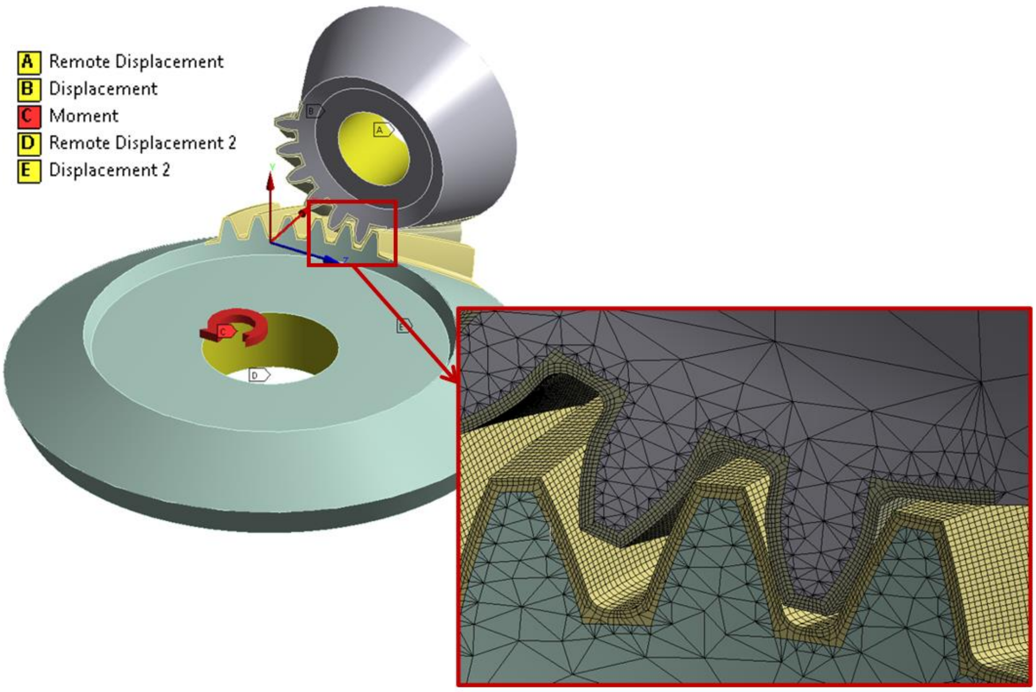

Fig. 1: Boundary conditions and generated mesh.

3. Results

Fig. 2 shows von Mises stress distribution of meshing teeth for load torque of 300 Nm. It can be observed that the highest stress is located at the teeth entering the contact where the edge contact occurs. Since the mating pair of teeth is deflected under load, it is not possible for the next tip to enter a contact in the pure involute position as there is interference corresponding to the elastic deflection resulting in edge contact. This effect leads to high stresses and decreased durability.

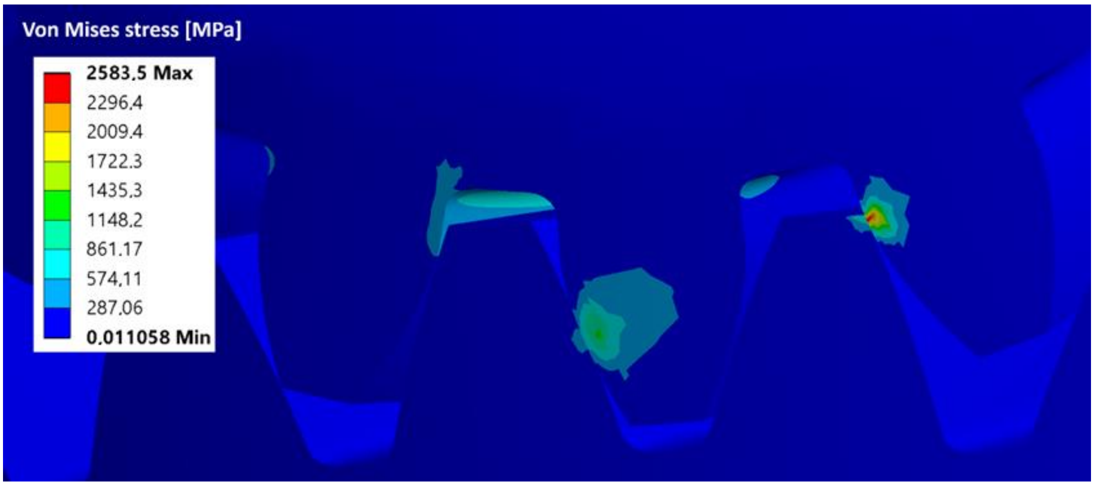

Fig. 2: Von Mises stress distribution.

To prevent stress concentrations on the tooth edges, caused by deflections under load, tip relief modifications can be implemented by removing material from the tip or the root of the teeth or both. Such modifications compensate teeth deflections under load so that the transmission error is minimized for a specific torque. Tip relief modification is defined by the amplitude C_a and the length of the tooth tip relief L_a.

In order to reduce the transmission error, two variants of the tip relief modification have been proposed in this paper. The first variant was a long modification starting just above the pitch diameter. The second variant was a short modification starting in the middle of the addendum. The amplitude of the modification C_a defined at the outer ends of the teeth was varied, while the length of the tip relief L_a was kept constant for each variant. Fig. 3 shows the transmission error as a function of the pinion rotation at a load torque of 300 Nm, for both short and long tip relief and the effect of the individual tip relief amplitudes. It is clear that the peak-to-peak values of the transmission error are considerably higher for all proposed amplitudes of long tip relief than for unmodified gears, and therefore long modification does not appear to be a suitable method for reducing the transmission error, considering a given type of gear at a given load.

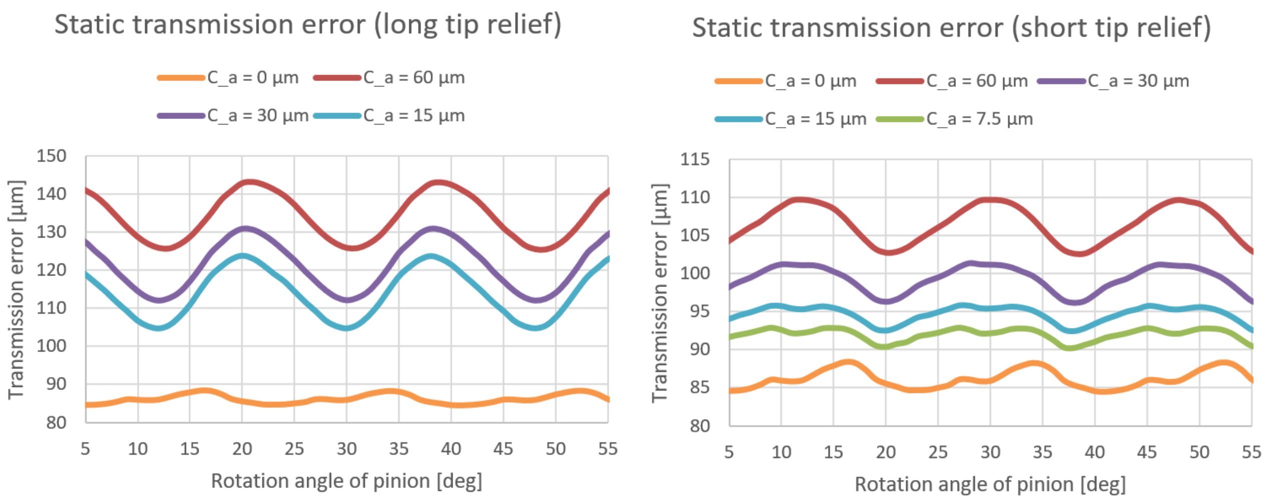

Fig. 3: The influence of short and long tip relief on the transmission error.

Since the long modification was not effective in reducing the transmission error of the gear set, a short modification was applied with the same amplitudes of tip relief C_a. In this case, the peak-to-peak values were significantly lower compared to the long modification. Modification with $C_a = 15\ \mu m$ even reduced the peak-to-peak value of unmodified gear by almost 12 %. Based on the results, an additional modification was applied with $C_a = 7.5\ \mu m$. The results showed that with this modification it was possible to further reduce the peak-to-peak value of the transmission error by up to 30 % compared to the gear without modification.

The effect of short tip relief modification with the amplitude of 7.5 μm on contact pressure distribution is shown in Fig. 4. By introducing this modification, the contact area was extended and the maximum contact pressure was reduced. These results support the theory (Astoul, 2014) that there is a correlation between minimizing the maximum contact pressure and reducing the transmission error.

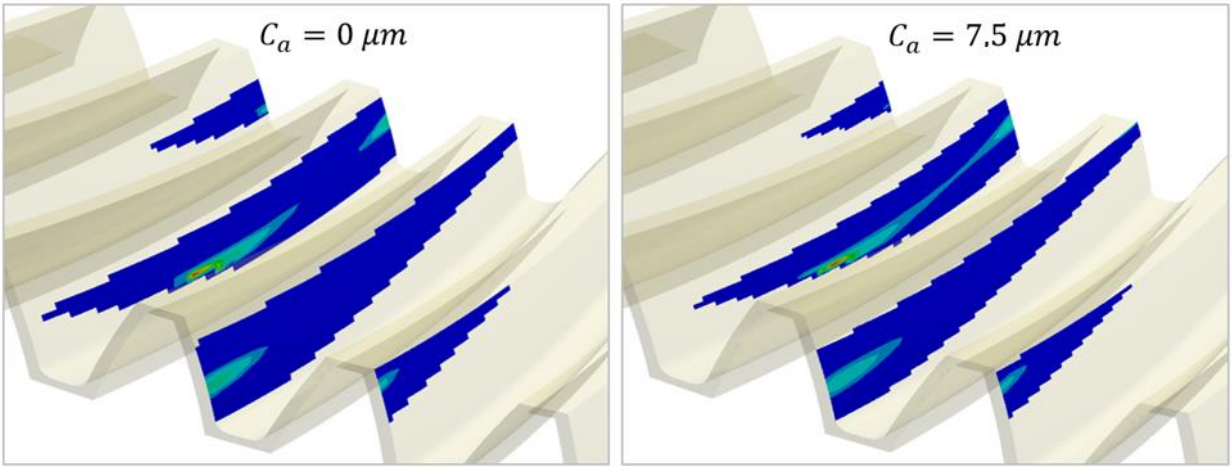

Fig. 4: Contact pressure distribution.

4. Conclusions

If the gear was to be operated primarily at a load torque of 300 N.m, a variant of a short modification with the amplitude of 7.5 μm would be the most suitable in terms of reducing the transmission error, and thus the expected noise generated by gear. In the case of a higher operating range, it would be necessary to carry out a more extensive sensitivity study and compare the achieved parameters. Based on the performed sensitivity analysis, the finite element method is an adequate way to determine the transmission error and to find a suitable modification for its reduction. However, creating a complete analysis is time-consuming, and therefore it would be effective to create a knowledge database that would help to eliminate unsuitable combinations. Last but not least, knowledge of the operating conditions is required for proper gear design.

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