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TRANSMISSION ERROR OF THE SPIRAL BEVEL GEARS

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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

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

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. 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 prediction of the trasnmisison error by numerical methods, Tharmakulasingam (2009) and Lin (2017), but only few of them are focused on the bevel gears, Mermoz (2013) and Astoul (2014).

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. The modification of teeth was considered, see Fig. 1.

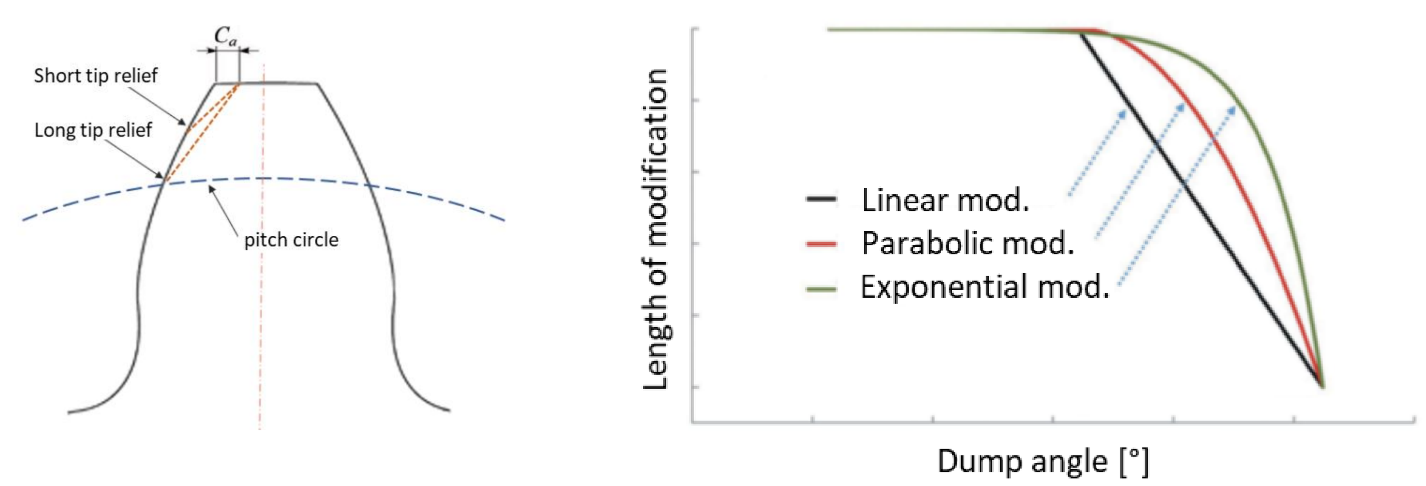


Fig. 1 Modification of teeth

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, see Fig. 2.

RESULTS

The initial study was performed at constant value of transmitted torque 300 Nm. It can be observed, Fig. 3, 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. To prevent this contact the tip relief modification can be used.

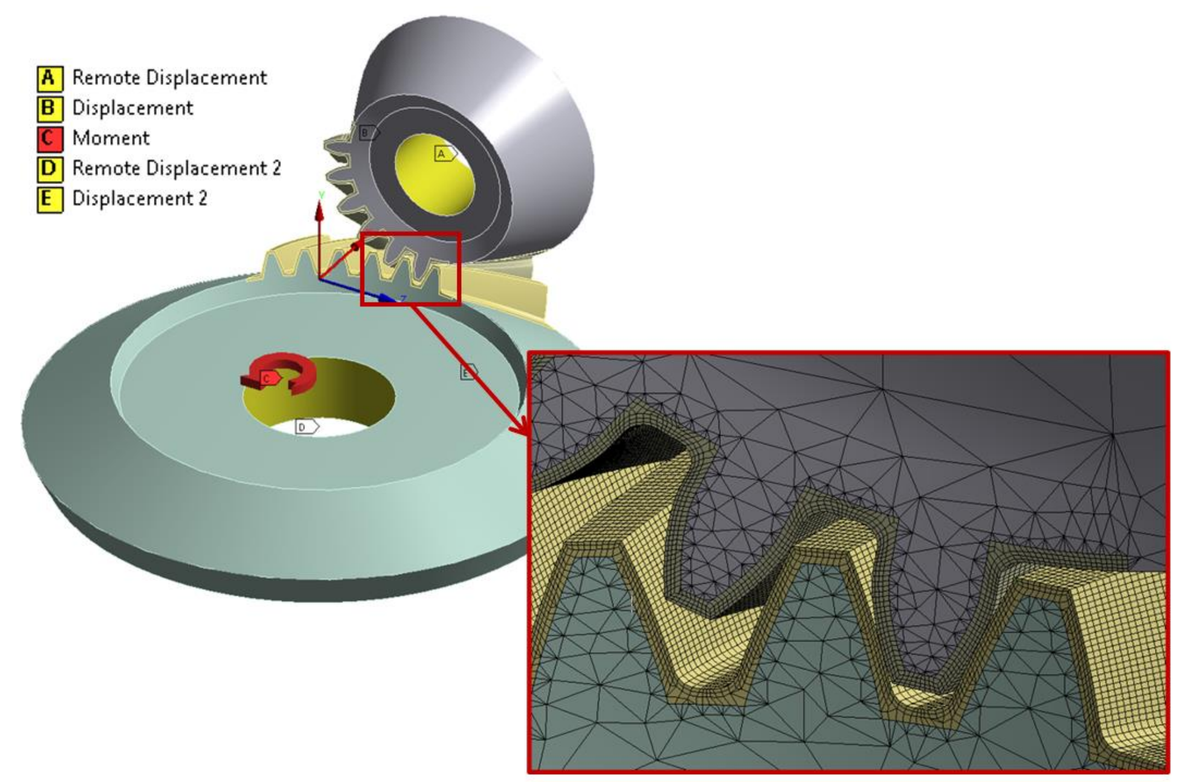


Fig. 2 Boundary conditions and generated mesh

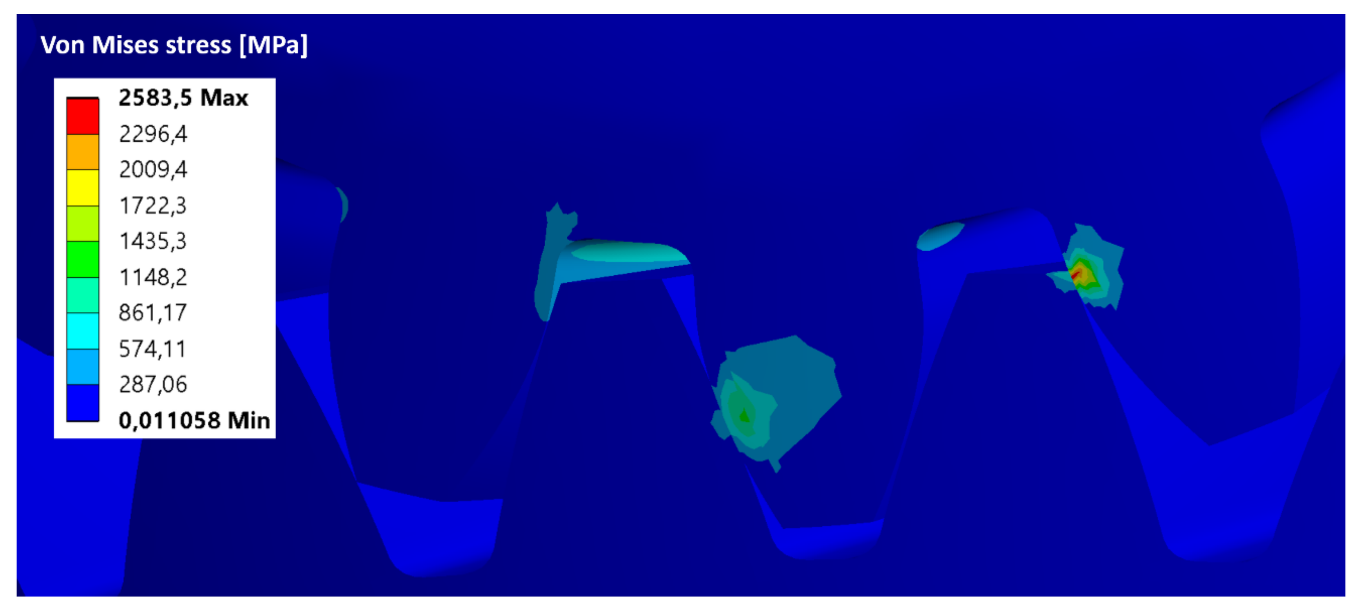
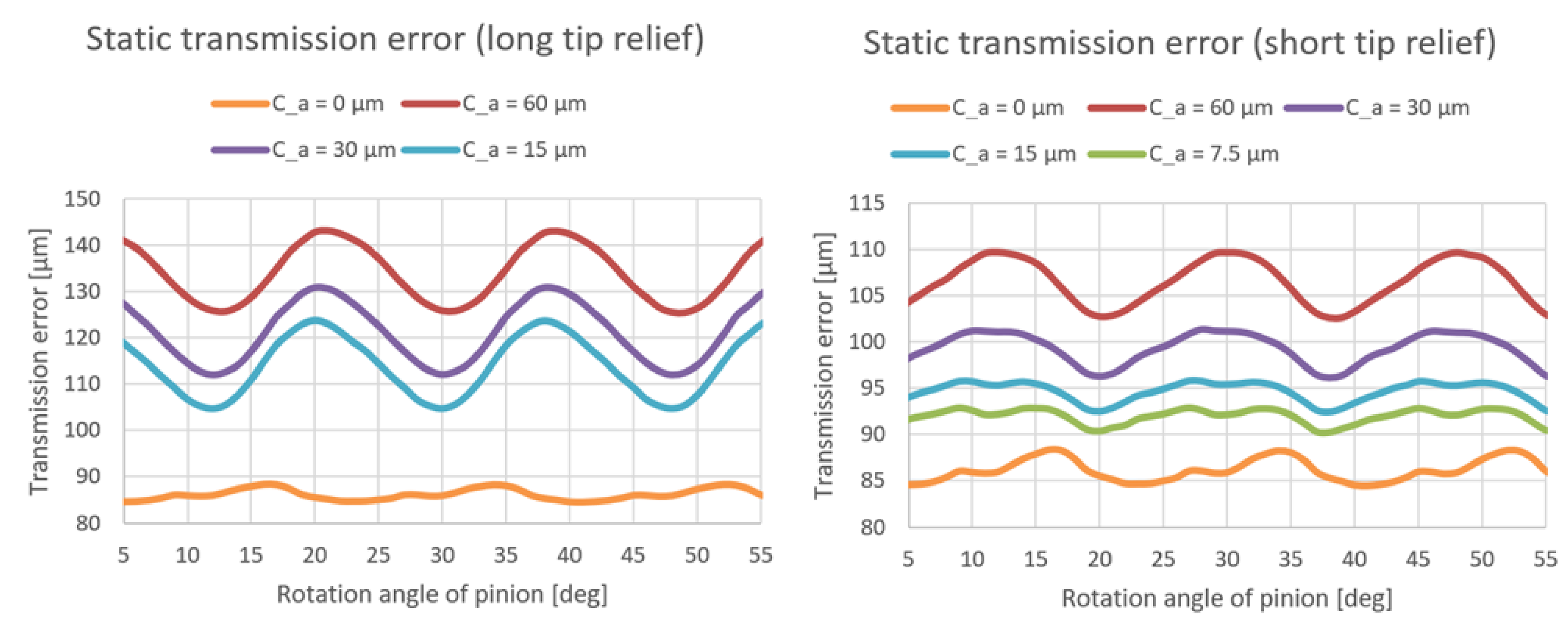


Fig. 3 Von Mises stress distribution.



Modification	C_a = 0 µm	C_a = 7,5 µm	C_a = 15 µm	C_a = 30 µm	C_a = 60 µm
Long tip relief	3,8384	-	19,1882	18,6707	17,8435
Short tip relief		2,6751	3,3889	5,1804	7,1063

Fig. 4 The influence of short and long tip relief on the transmission error.

The transmission error for long and short tip relief is shown in Figure 4 including the peak to peak value. The effect of short tip relief modification with the amplitude of 7.5 µm on contact pressure distribution is shown in Figure 5. At the same operation conditions the parabolic modification was applied and its shown in figure 6 including peak to peak value.

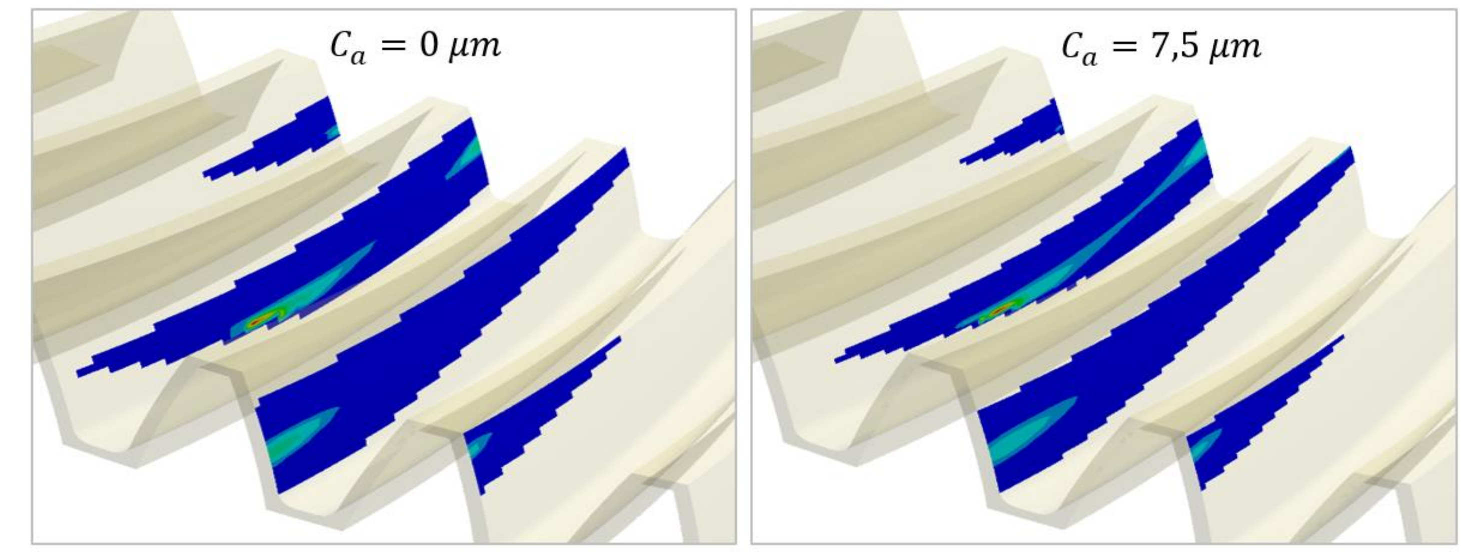
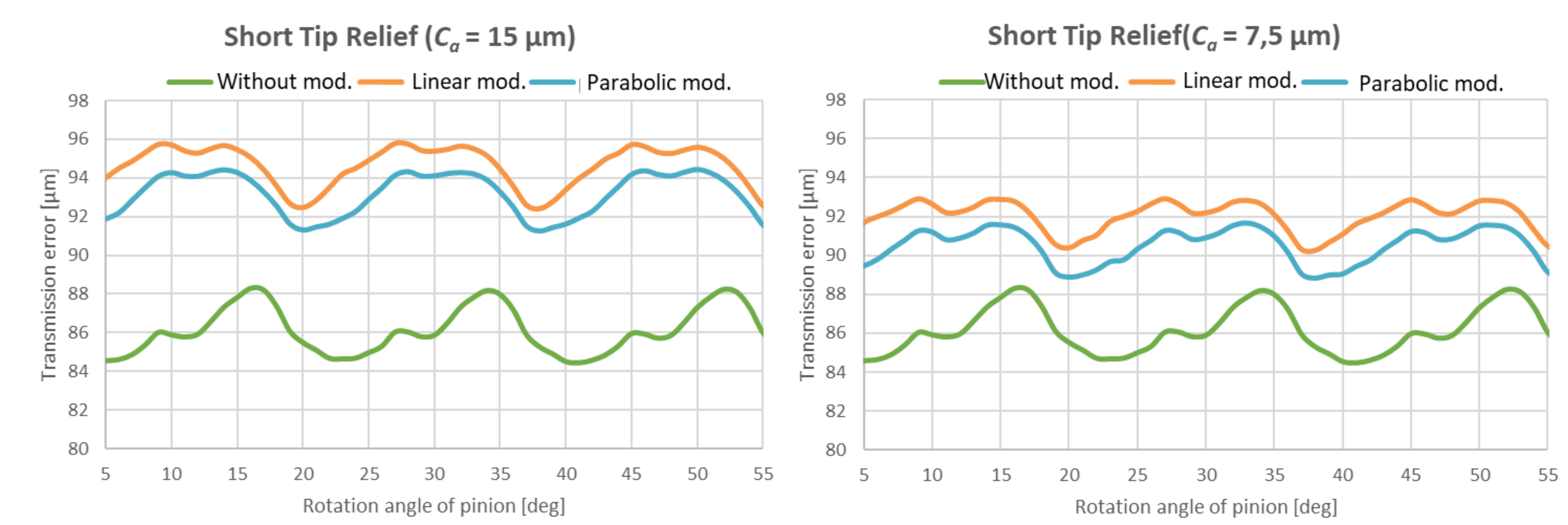


Fig. 5 Contact pressure distribution



	w/o mod.	Linear mod.		Parabolic mod.	
		C_a = 7,5 µm	C_a = 15 µm	C_a = 7,5 µm	C_a = 15 µm
Peak-to-peak [µm]	3,8384	2,6751	3,3889	2,8291	2,6751

Fig. 6 The influence of modification type

SUMMARY

The methodology for determination of TE, which is based on the FEM, is developed on spiral bevel gears. One of the most important advantages of this methodology is the parametric model, which enables easy modification of input parameters. The whole approach is presented on gear pairs with gear ratio 2. Based on the result the operation conditions play important role. If the presented gear is operated primarily at a load torque of 300 Nm, a variant of a short linear modification with the amplitude of 7.5 µm or parabolic modification with the amplitude of 15 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.

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