STRENGTH ANALYSIS OF ASYMMETRIC INVOLUTE GEARING

Ondřej Miláček[1], Michal Vašíček[2]
[1]ČVUT v Praze, Fakulta strojní, ondrej.milacek@fs.cvut.cz
[2] ČVUT v Praze, Fakulta strojní, michal.vasicek@fs.cvut.cz

PEVNOSTNÍ VÝPOČET ASYMETRICKÉHO EVOLVENTNÍHO OZUBENÍ

Abstract: *In automotive drivetrain there is standardly used symmetric involute gearing. It is given mainly by the production line equipment of manufacturers. But especially in field of automotive where engines are usually turbocharged there are torque demands at both tooth sides very different. From this reason it is not meaningful to have symmetric tooth profile. The strength analysis of asymmetric involute profile will be introduced in this contribution.*

Key words: *involute, gearing, asymmetric, automotive, FEM analysis*

INTRODUCTION

Involute gearing has been produced for more than hundred years, therefore there are many sophisticated methods for its designing and dimensioning including loading conditions during its lifetime. The description of its geometry is quite demanding and the easiest type of it is the symmetrical one which is also standardized and commonly used. This designing software is able to predict the lifetime and safety coefficients against failures. Of course all necessary parameters of gearwheels material and its heat and technological treatment have to be known. Furthermore this project of testing gearwheels is interesting also from the point of view of material because they are made of PM (powder metal). In this contribution will be described the FEM analysis of asymmetrical variant of the 3rd speed from the gearbox MQ200.

STRESS ANALYSIS OF SYMMETRIC INVOLUTE GEARING

As it was already mentioned in the introduction, stress calculation at gearwheels has been provided for many years. For correct gearing dimensioning it is necessary to precisely describe loading conditions (loading spectrum) and lifetime demands. It is done by using of many coefficients for each phenomena which are used in final relation for the stress analysis. Basically there are two kinds of stress calculation, i.e. contact stress between tooth flanks and bending stress in the root fillet of teeth. If the S-N curve of the material is known, safety coefficients against pitting and tooth breakage can be then calculated as final result. During this time many programs were created for gearing stress calculation. In this case there was used program "KissSoft". This program is worldwide known. Its appearance can be seen in figure 1.

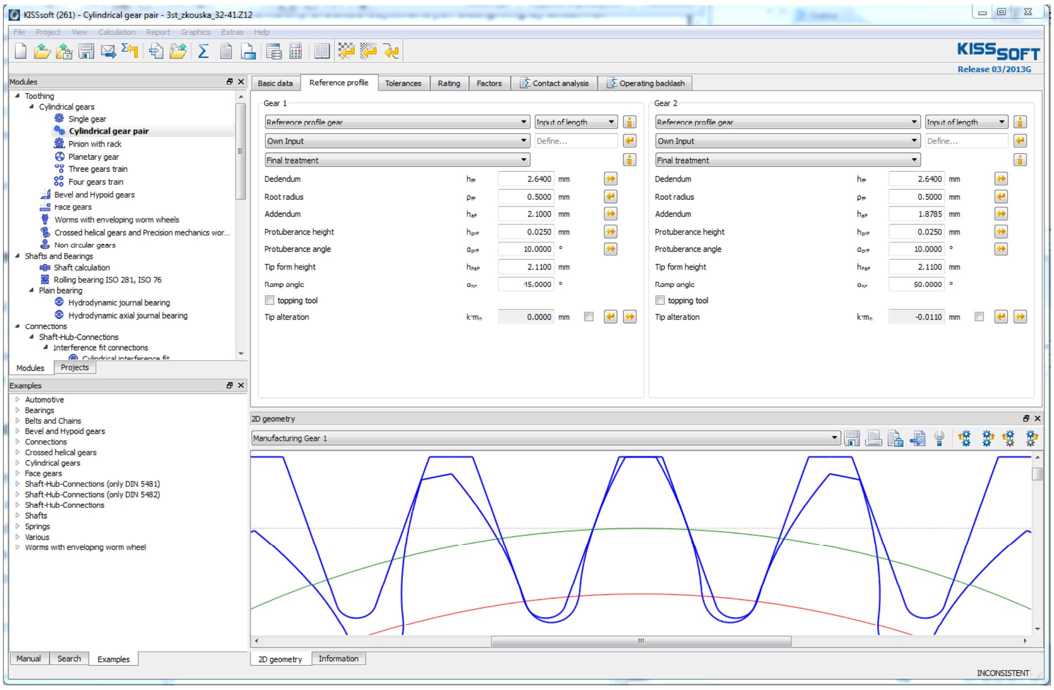

Fig. 1 *Appearance of the program KissSoft. In this case manufacturing of a gearwheel by cutting tool.*

CREATION OF ASYMMETRIC TOOTH PROFILE

First step in this work was the creation of the software for designing of asymmetric gearset. This was done in the program Python. It is possible to set here all necessary geometrical parameters at both tooth flanks separately. Some parameters are common for both tooth flanks, e.g. shift profile coefficient. Its appearance is depicted in figure 2. The target was to create similar

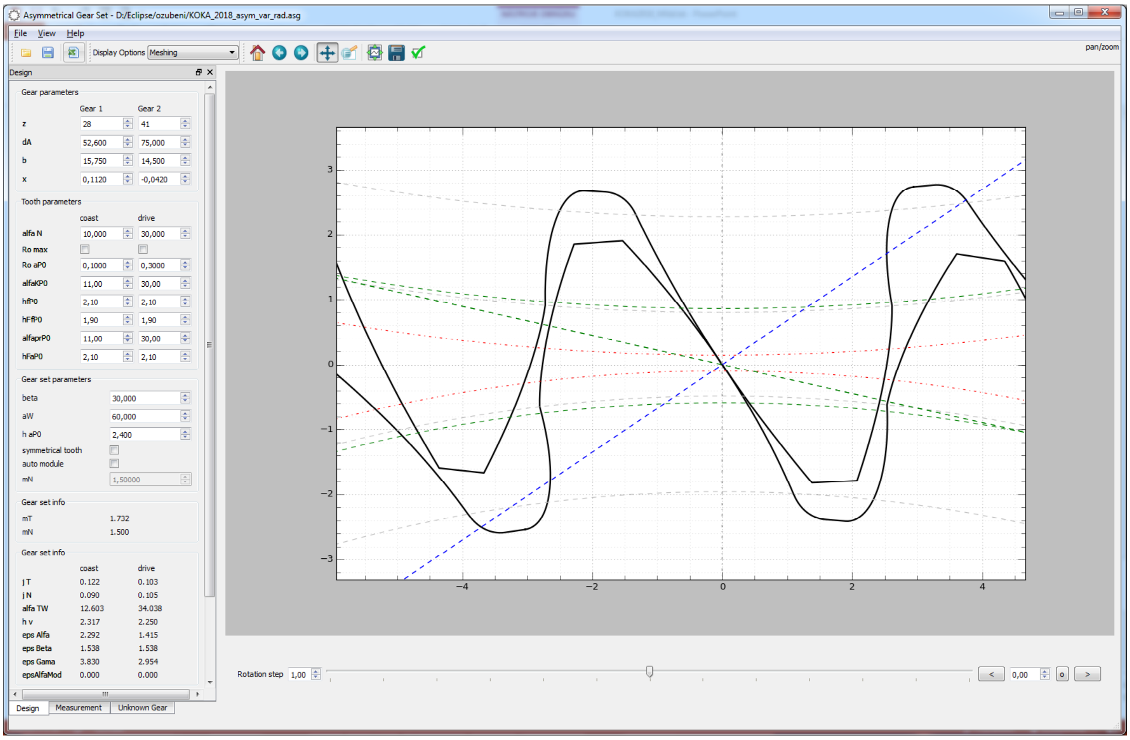

Fig. 2 *Appearance of the program for asymmetric gearing. In this case meshing of a two gearwheels.*

program as KissSoft, also with animation of trochoidal manufacturing by asymmetric cutting tool. This program shows the gearset including clearances between teeth. The output of this program are coordinates of tooth profile points. Of course it is possible to create symmetrical profile too.

These coordinates were then used as an input for the program Catia for creation of real tooth profile using interpolating spline curves. By pattern function was then created whole gear profile and 3D model of the gearwheel.

The biggest advantage or the reason why asymmetric gears are interesting is the possibility of reducing contact stress at tooth flanks which leads to higher resistance against pitting. This contact stress can be also calculated analytically using formula 1 for single tooth contact,

$$\sigma_H = \sqrt{\left(\frac{F_n}{\pi b}\right)\left(\frac{E}{2(1-\nu^2)}\right)\left(\frac{1}{\rho_1}+\frac{1}{\rho_2}\right)} \tag{1}$$

where ρ_1 and ρ_2 are osculating radii of flanks in touching point. At the opposite side there is a disadvantage in higher radial forces from meshing leading to higher loading of bearings and shafts.

METODOLOGY OF THE STRESS CALCULATION FOR ASYMMETRIC GEARING

For stress calculation on asymmetric gears there are no standards. From this reason the only possibility is the usage of FEM analysis. Originally we wanted to apply 3D touch of both tooth flanks. This would be meaningful only in case that all flank profile modifications (microgeometry – e.g. crowning) would be included in the 3D model. Unfortunately it would be very demanding for both the computation and model creation. From this reason we have decided to provide stress analysis only in 2D.

Tested gearing is helical one, original symmetric gearing and newly designed asymmetric gearing have same helix angle β. So its influence at recalculation should be identical for both variants.

First step was to calculate contact and root bending stresses for symmetric gearset under given loading conditions in the program KissSoft.

Second step was the creation of identical symmetric profile in our python program and its subsequent usage for tooth profiles and gearwheel Catia model creation. At this gearset was then provided FEM analysis under identical loading conditions as the symmetric gearing resulting again into Hertzian and bending stresses.

Third step was the computing of comparing coefficient between stresses obtained from KissSoft and FEM analysis. Hertzian stress can be calculated either analytically using formula 1 or from FEM analysis. In our case we have decided to use FEM results which can be seen in formula 2.

$$k_{H\,FEM} = \frac{\sigma_{H\,KissSoft}}{\sigma_{H\,FEM}} \quad \text{... for contact stress} \tag{2}$$

In case of root bending stress we obtain its value for each tooth side – see figure 3. It is interesting that higher von Mises stress is always at the coast side. But this stress is the compressive one. If we are thinking about root stresses, we have in mind the resistance against tooth breakage. This kind of failure starts always at the drive side where tensile stress appears. From this reason is for the recalculation coefficient for root stresses used its value from the drive side, although it is not the maximal one. This coefficient is calculated for each gearwheel separately - see formula 3.

$$k_{F\ FEM_{1,2}} = \frac{\sigma_{F\ KissSoft_{1,2}}}{\sigma_{F\ FEM_{drive\ side_{1,2}}}} \qquad \text{... for root bending stress at each gearwheel} \qquad (3)$$

If we know these recalculation coefficients, we can apply same methodology for other gear geometry, in our case the asymmetric one.

Fourth step was the loading (FEM analyzing) of our asymmetric gearing. Resulting stresses were then multiplied by appropriate recalculation coefficients to get stress values which include all aspects (loading conditions) from KissSoft.

FEM ANALYSIS DESCRIPTION

As an input we had 3D model of the whole gearset created in Catia. For the FEM analysis

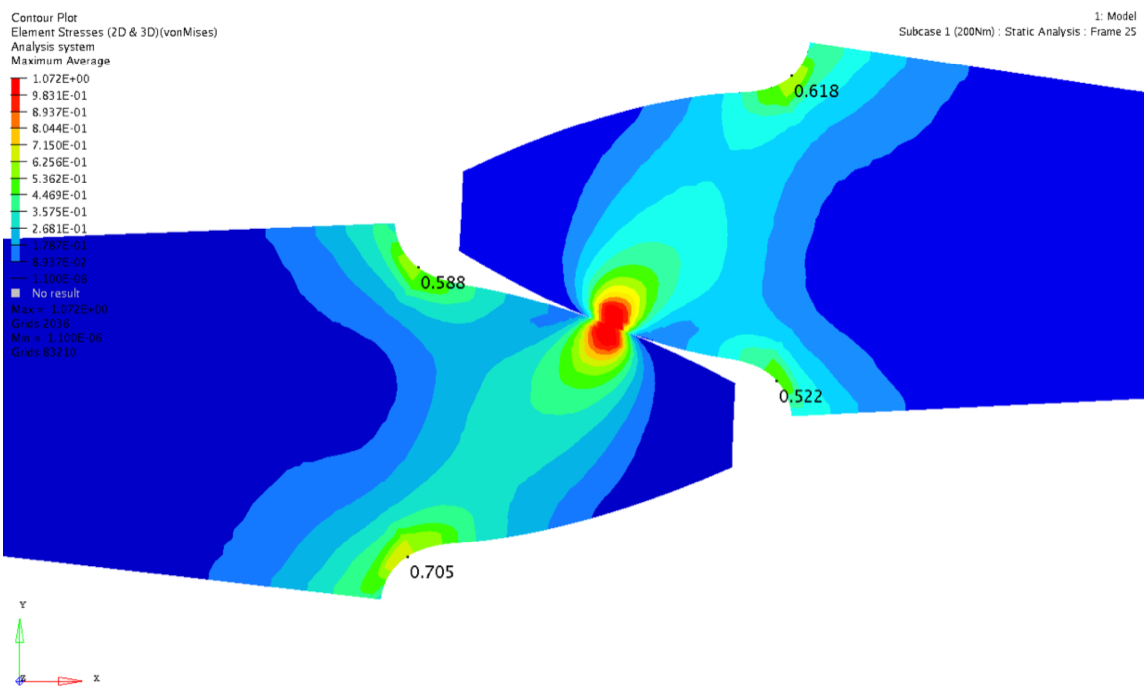

Fig. 3 *FEM results for angular cutout belonging to one pair of teeth od original symmetric gearing.*

whole gearwheel is not necessary. Originally we used instead of it only circular sector belonging to one tooth. This was sufficient only for contact stress. In case of root bending stress the area with local root stress maximum was quite close to models border which is created by elements with restricted displacement – see figure 3, which can influence stress distribution in the tooth root. From this reason we have decided to use a model with an angular cutout belonging to 3 gear pairs. Comparison between models with one and three teeth pairs is depicted in figure 4. Gears were in the position when the contact appeared between drive flanks. Drive side is this one with higher mesh angle. From this model was then created a cross section which determined our 2D geometry. This geometry was then simply extruded for 1 mm therefore loaded assembly was in final spur gearset with the width of 1 mm. Applied loading torque was then the nominal one divided by the common facewidth of the gearset.

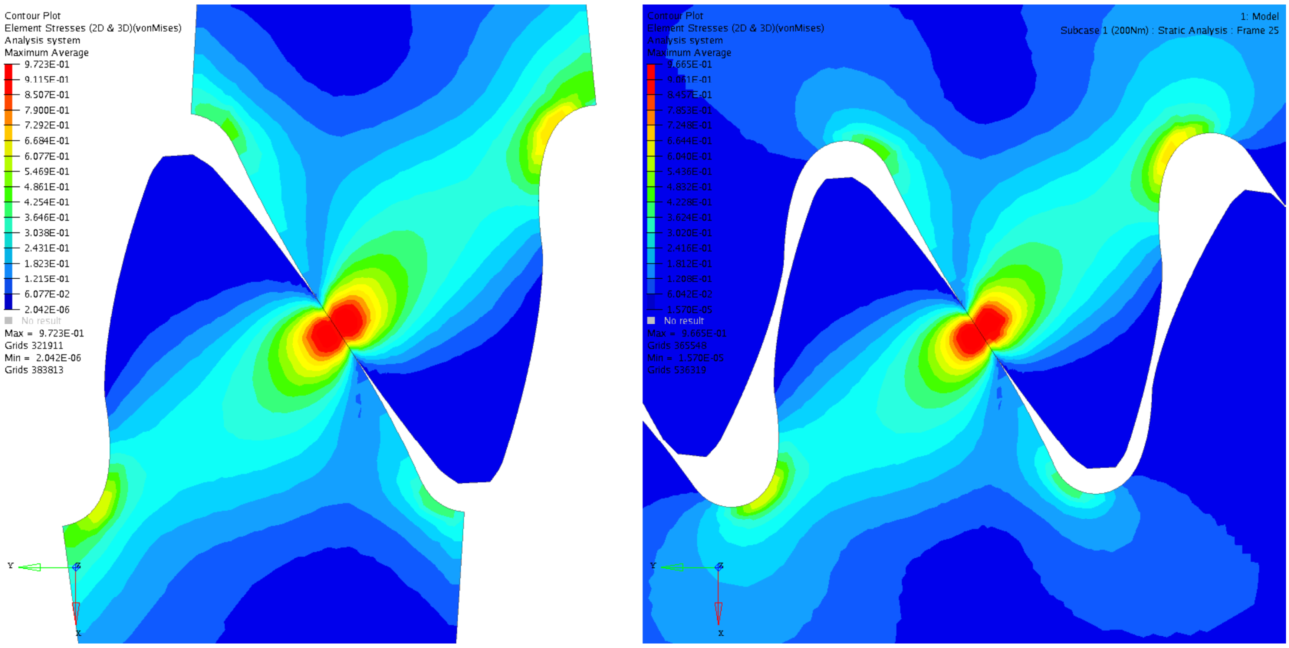

Fig. 4 *Comparison between FEM results of asymmetric gearing for angular cutout belonging to one (left) and three (right) pairs of teeth.*

3D FEM model was created and solved using Altair's Optistruct solver. Average mesh size is 0,1 mm and utilizes hexa- and pentahedral elements of first order. Material model is linear, defined by parameters obtained by measurement. Contact between teeth is treated by symmetrical node to surface small sliding penalty contact.

FEM ANALYSIS RESULTS

It is clear that usage of model with three teeth radial cutout of the gearwheel is more realistic than the one with one tooth radial cutout, especially for root stresses. These values have slightly changed. This change cannot be determined in general because it is strongly dependent at gearing geometry (amount of teeth, etc.). In our case have this stress at drive side (tensile) at one gear increased by 6% and at the second one by 12%. In case of contact pressure there was again an increase by 5%. These changes are between one and three teeth model.

For the comparison between symmetric and asymmetric gearing were used meshing models with three teeth. From the FEM analysis it is clear that at asymmetric gearing contact stress decreased by 9 %. In case of root bending stress at drive sides the decrease is for the first gear 17 % and for the second one 22 %.

CONCLUSION

The advantage of asymmetric gearing is in decreasing of contact pressure at drive flanks. According to FEM model root stresses at drive sides have decreased too. At the opposite side there is due to higher mesh angle significant increase of radial forces from gearing meshing. These forces are loading shafts and bearings, which leads to bigger shaft deflections and angular deviations. From this reason stress analysis of shafts and new microgeometry of the gearing (e.g. crowning) has to be newly designed. In case of PM (powder metal) material usage which has always some porosity, the gearing geometry change into asymmetric shape is the way how to compensate this characteristic feature of PM parts.

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