KISSsoft AG - +41 55 254 20 50
Uetzikon 4 - +41 55 254 20 51
8634 Hombrechtikon - info@KISSsoft.AG
Switzerland - www.KISSsoft.AG

Path of contact calculation KISSsoft 04-2010

1 Introduction

The tooth contact under load is an important verification of the real contact conditions of a gear pair and an important add-on to the strength calculation according to standards as ISO, AGMA or DIN. The contact analysis simulates the meshing of the two flanks over the complete meshing cycle and is therefore able to consider individual modifications on the flank at each meshing position.

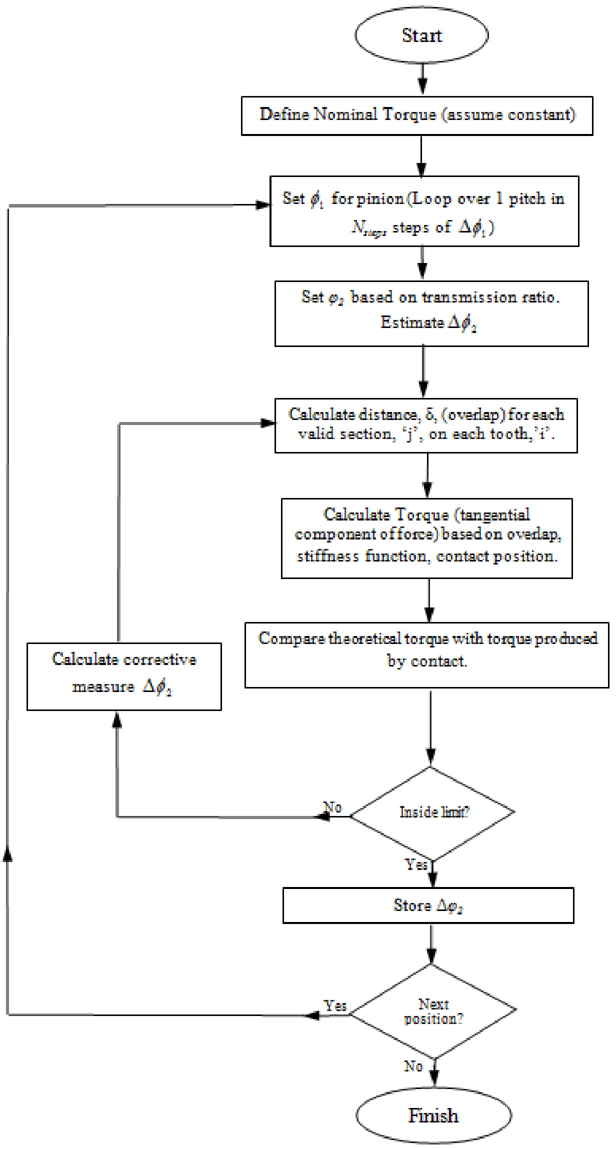

Fig. 1: tooth contact analysis according to Peterson

The tooth contact analysis is therefore mainly used to reduce noise which is caused by the effect of shock load at meshing entry due to elastic bending of the loaded teeth. It is further used to optimize load distribution by analyzing the effectiveness of gear profile modifications considering the misalignment of the gear axis due to shaft and bearing deformation under load.

1.1 Basic calculation method

The tooth contact analysis simulates the meshing contact assuming a constant nominal torque. The calculation procedure has been defined by Peterson: For a given pinion rolling position (rotation angle $\phi 1$) the corresponding gear rolling position $\phi 2$ is determined with an iterative calculation (see figure 1).

The calculation considers the local elastic deformation due to several effects and the corresponding stiffness's which appear under load: stiffness from bending and shear deformation c_Z, stiffness from Hertzian flattening c_H and bending stiffness of the tooth in gear body rim c_{RK}.

This calculation procedure is repeated for all the meshing cycle. Comparisons with FE calculations showed a very good correlation.

The final stresses include the load increasing factors calculated by the standard, such as application factor K_A, dynamic factor K_V and load distribution factor $K\gamma$ in planetary gears or gear pairs. For the tooth root stress, also the gear rim factor Y_B according to ISO6336 is considered.

Before KISSsoft Release 04-2010, also the load distribution factors $K_{H\alpha}$, $K_{H\beta}$ for Hertzian pressure resp. $K_{F\alpha}$, $K_{F\beta}$ for root stress was considered. This has been changed for the enhanced tooth contact analysis, please find the details below.

2 What is new in Version 04-2010 ?

With the KISSsoft Release 04-2010, the tooth contact analysis for cylindrical gears has improved significantly. In addition to the preceding releases, the stiffness model was extended to better take the load distribution in the width direction into account, which is a significant characteristic of helical gears, but also other effects are now considered, finalizing in the 3D display of results.

2.1 Coupling between the slices

For a tooth contact of helical gears, the meshing field is different to a spur gear. The contact lines for a spur gear are parallel to the root line, and herewith also the load distribution in length direction is uniform. The contact lines for a helical gear are diagonal over the tooth, which means the load is not uniformly distributed over the length of the tooth. Still the unloaded part of the tooth has a supporting effect and influences the deformation of the tooth as well.

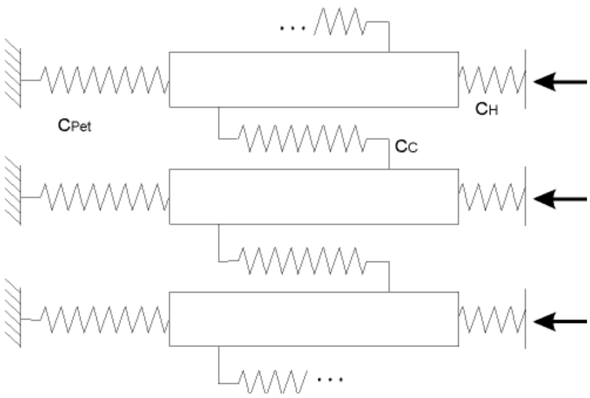

Fig. 2: stiffness model according to Peterson and KISSsoft Release 04-2010

This supporting effect of the unloaded areas has to be considered for the contact analysis of helical gears.

For this purpose, in KISSsoft 04-2010 the gear is in lengthwise direction divided in slices. The single slices are then connected between each other with the coupling stiffness c_c so that a supporting effect between the slices can be considered (see figure 2).

$$c_{Pet} = f(c_Z, c_{RK})$$

c_{Pet}	stiffness tooth root following Peterson
c_Z	stiffness from bending and shear deformation
c_{RK}	stiffness from deformation through rotation in the gear blank
c_H	stiffness from Herztian flattening following Peterson

The coupling stiffness c_C is defined as follows:

$$c_C = 0.04 \cdot A_{\sec}^2 \cdot c_{Pet}$$

c_C coupling stiffness
A_{sec} Number of slices

The coupling stiffness is related to the contact stiffness and hence individual for each gear pair, and is verified for different gear types with FE calculations and other established software. The number of slices A_{sec} depends on the accuracy setting which is defined from the user. However, the single coupling stiffness c_C is defined in a way that the system coupling stiffness is independent of the number of slices and therewith also independent of the user settings.

In figure 4, the same gear calculation is compared between the KISSsoft release 04-2010 and the previous release. It is a spur gear (helix angle β= 0°) with a larger face width for the pinion (b_1=50mm) than for the gear (b_2=44mm). The supporting effect of the unloaded face area outside the meshing contact causes an increased edge pressure within of the meshing contact. This effect can now be considered with the coupling stiffness between the slices.

In the previous KISSsoft releases the forces remain constant (figure 4a), where as in the KISSsoft release 04-2010, the normal force at outer ends of meshing contact is increased (figure 4b). Note that figure 4a shows the pinion face width b_2=50mm, whereas in figure 4b only the common face width b=44mm is displayed.

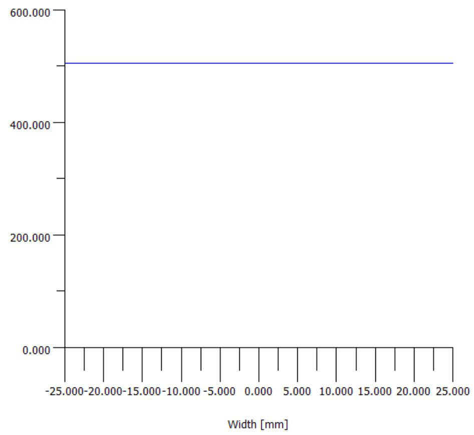

Fig. 4a: previous KISSsoft releases shows constant normal force (line load) over face width

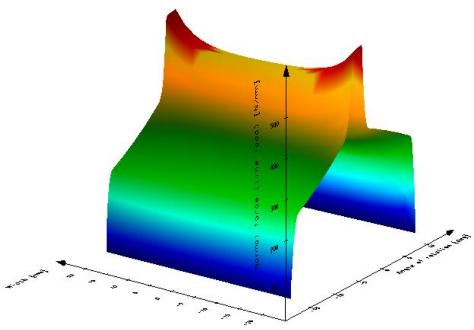

Fig 4b: KISSsoft 04-2010 shows the increased normal force (line load) at edges of contact area

2.2 Decreased stiffness on the side borders of helical gears

For helical gears, the tooth may be cut by the cylindrical bodies (see figure 3), which results in reduced tooth thickness s_{red} compared to a tooth that is not cut having a tooth thickness s_n. Whenever force is applied to the tooth with reduced tooth thickness, it will result in higher deformation due to lower stiffness. This effect is considered with the reduced coupling stiffness c_{Pet_border} for the slices of teeth.

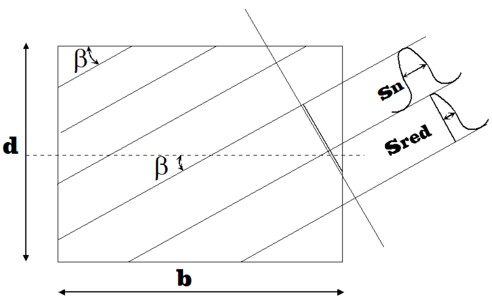

Fig. 3: decreased rigidity on the side borders

In KISSsoft 04-2010, the following formula is applied, which is also verified with FE calculation and other established software.

$$c_{Pet\_border} = c_{Pet} \cdot \sqrt[2]{\frac{s_{red}}{s_n}}$$

c_{Pet_border}	coupling stiffness for slices with reduced tooth thickness
c_{Pet}	standard coupling stiffness
s_{red}	reduced tooth thickness at border
s_n	standard tooth thickness

In figure 5, the same gear calculation is compared between KISSsoft release 04-2010 and the previous release. It is a helical gear (helix angle β=15°) with the equal face width b=44mm. In the previous Releases the effect of reduced coupling stiffness at border wasn't considered, therefore the normal force (line load) at border isn't increased. In KISSsoft release 04-2010, the normal force at the start as well as end of contact is increased.

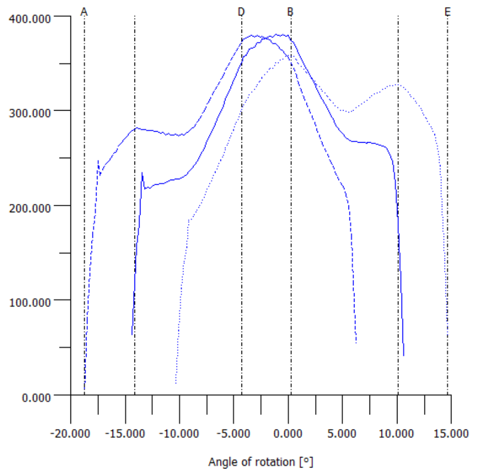

Fig. 5a: previous Releases don't show higher normal forces (line load) at ends

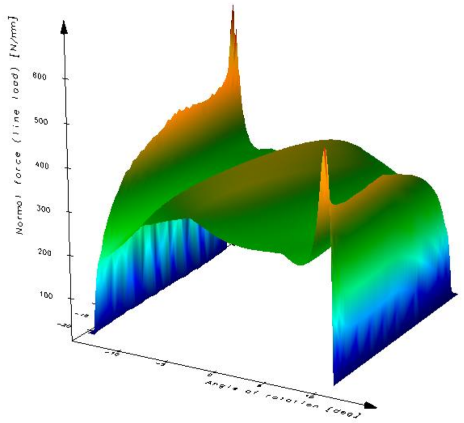

Fig 5b: KISSsoft 04-2010 shows higher normal forces (line load) at start and end of contact

2.3 Revised calculation of tooth stiffness of helical gears

For helical gears, the contact stiffness c_{Pet} following Peterson is calculated based on the effective tooth form in normal section. In earlier KISSsoft-versions the tooth form was based on the transverse section multiplied by the factor cosβ, which is a less accurate procedure. Therefore the results slightly differ between this and older releases.

In figure 6, the same gear calculation is compared between KISSsoft release 04-2010 and the previous release. It is a helical gear (helix angle β=15°) with the equal face width b=44mm. In the KISSsoft release 04-2010, the tooth stiffness is slightly different to the previous KISSsoft release. Since the transmission error is strongly related to the stiffness, also the transmission error slightly differs too.

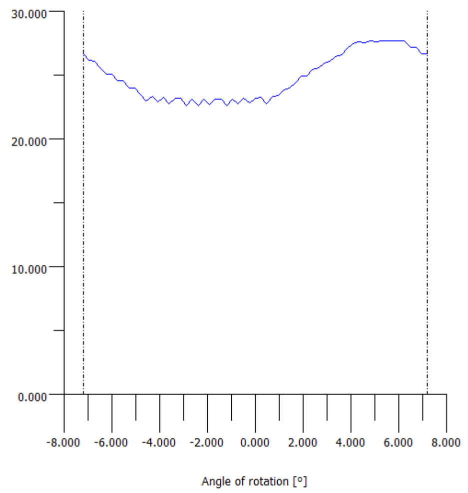

Fig. 6a: previous Releases calculate slightly higher tooth contact stiffness

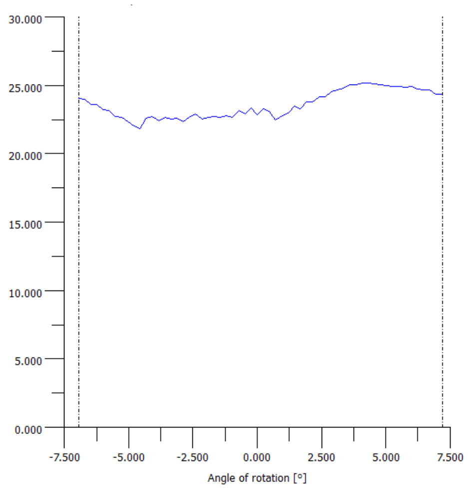

Fig 6b: KISSsoft 04-2010 calculates slightly lower tooth contact stiffness

2.4 Load distribution factors not multiplied any more to stresses

In previous KISSsoft releases it was not possible to consider any unequal load distribution correctly since the slices were not coupled. Therefore the load distribution was added taking the factors $K_{H\alpha}$, $K_{H\beta}$ as well as $K_{F\alpha}$, $K_{F\beta}$ (ISO) and K_M (AGMA) from the standards calculation. These were multiplied to the stresses from the tooth contact analysis. In KISSsoft 04-2010 these factors are no longer used. However, the displayed stresses are still multiplied with the application factor K_A, dynamic factor K_V and load distribution factor $K\gamma$ in planetary gears or gear pairs.

In figure 7, the same gear calculation is compared between KISSsoft release 04-2010 and the previous release. It is a spur gear (helix angle β=0°) with the equal face width b=44mm. It's a overhang design, meaning the gear is outside the bearings. This results in a increased load distribution factor according from ISO standard calculation with $K_{H\beta}$ =1.27 and $K_{H\alpha}$ =1.0.
The tooth contact calculation is done without considering any misalignment of the gear axis, values for deviation error and inclination error are set as 0. In the KISSsoft release 04-2010, flank pressure and root stresses are lower compared to the previous release. For a realistic contact analysis the gear axis misalignments should defined with shaft and bearing calculations, i.e. from KISSsys.

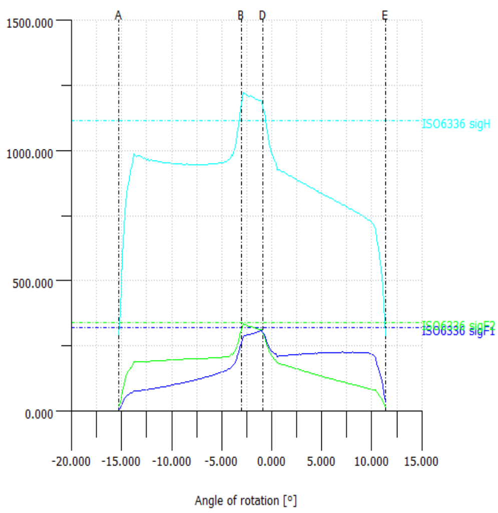

Fig. 7a: previous releases consider load distribution factors in tooth contact analysis

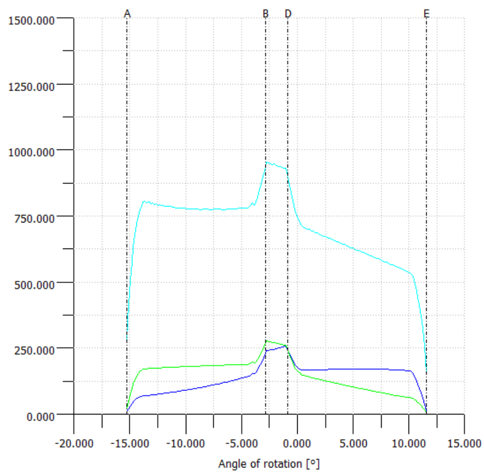

Fig 7b: KISSsoft 04-2010 doesn't consider load distribution factors from Standard calculation

2.5 Calculation of Hertzian Pressure

The calculation of the Hertzian stress is based on the Hertzian law in the contact of two cylinders. This gives realistic results in most situations. However a problem is encountered when the contact is on a corner of the flank, i.e. corner at the tip diameter, corner at the beginning of a linear profile modification, corner at the beginning of an undercut. Then the radius of curvature becomes very small, which results in a high peak of Hertzian stress calculation. This is not a realistic issue, because the part of the flank near to the corner will be joined in the contact. An algorithm, checking the joining flank parts and increasing the radius of curvature is implemented. However, it may be that high peaks still remains. Then we recommend adding a realistic radius to the corners and using circular profile modifications instead of linear.

In figure 8, the same gear calculation is compared between KISSsoft release 04-2010 and the previous release. It is a spur gear (helix angle β=0°) with the equal face width b=44mm. In fig. 8a there is no tip rounding applied whereas in fig. 8b there is a tip rounding 0.5mm. The pressure peaks are drastically reduced with the tip rounding.

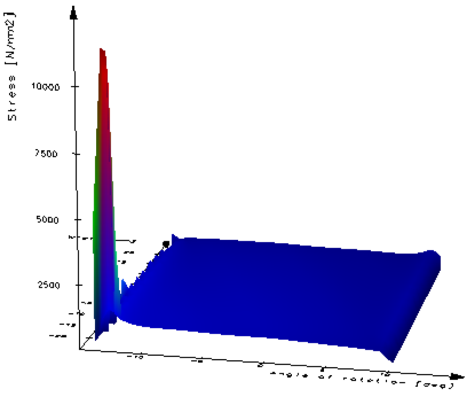

Fig. 8a: high pressure peak due to no tip rounding

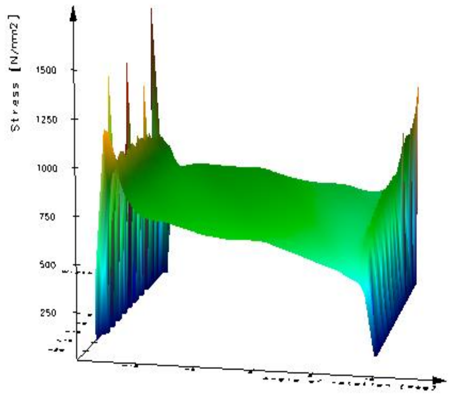

Fig 8b: much lower pressure with tip rounding 0.5mm

2.6 3D display

With KISSsoft release 04-2010 the graphical evaluation has been enhanced with 3D graphics. However, the 2D graphics remain as a good comparison to the previous releases. The 3D graphics show a 3 axis diagram, whereas the color indicates the stress level. In some cases there may be points where the stress data might be missing. In such cases the colors are interpolated directly between two neighboring stress data values. This may result in unequal color display. Figure 9 shows an example of this effect at start and end of contact.

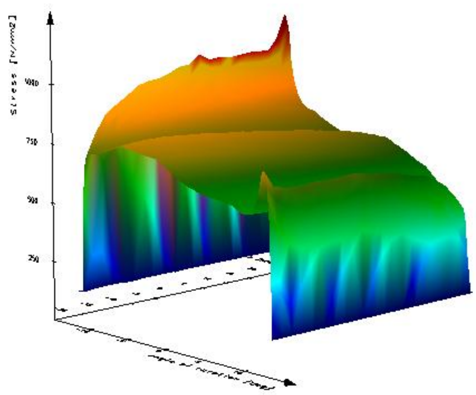

Fig. 9: 3D presentation of stress level