

KISSsoft Introduces New Features with Latest Release

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 such 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.

The tooth contact analysis (TCA) is therefore mainly used to reduce noise that 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.

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 ϕ_1) the corresponding gear rolling position ϕ_2 is determined with an iterative calculation (Fig. 1).

The calculation considers the local elastic deformation due to several effects and the corresponding stiffnesses 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 the entire 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_γ in planetary gears or gear pairs. For the tooth root stress, the gear rim factor Y_B according to ISO6336 is also considered.

Before the release of *KISSsoft 04-2010*, the load distribution factors $K_{H\alpha}$, $K_{H\beta}$ for Hertzian pressure and $K_{F\alpha}$, $K_{F\beta}$ for root stress were considered. This has been changed for the enhanced tooth contact analysis.

What's New in Version 04-2010?

With *KISSsoft 04-2010*, the TCA 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 3-D display of results.

Coupling between the slices. For a tooth contact between helical gears, the meshing field is different than for spur gears. 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. 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 (Fig. 2).

$$C_{Pet} = f(C_Z, C_{RK}) \quad (1)$$

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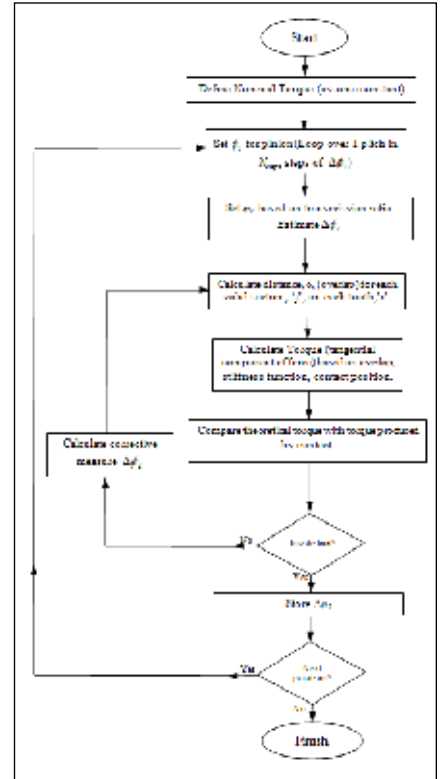


Figure 1—Tooth contact analyses according to Peterson.

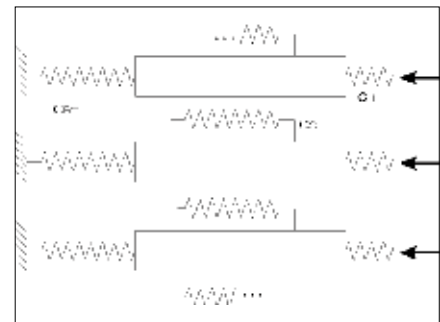


Figure 2—Stiffness model according to Peterson and *KISSsoft 04-2010*.

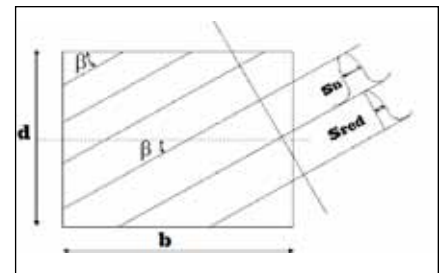


Figure 3—Decreased rigidity on the side borders.

- 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 Hertzian 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

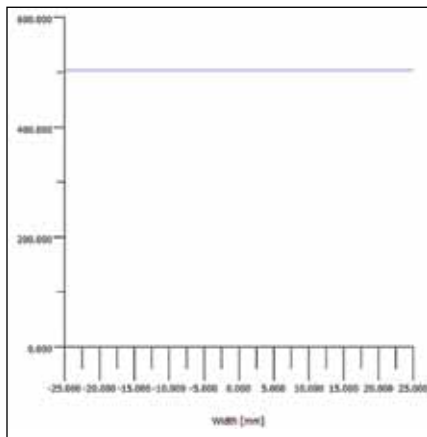


Figure 4a—Previous KISSsoft release shows constant normal force (line load) over face width.

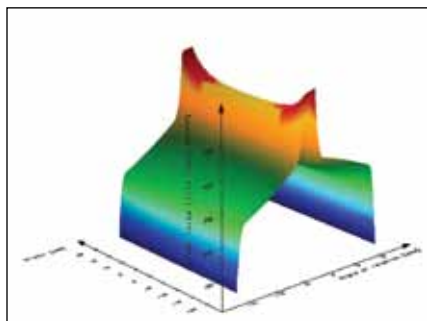


Figure 4b—KISSsoft 04-2010 shows the increased normal force (line load) at edges of contact area.

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 *KISSsoft 04-2010* and the previous release. It is a spur gear (helix angle $\beta = 0^\circ$) with a larger face width for the pinion ($b_1 = 50$ mm) than for the gear ($b_2 = 44$ mm). The supporting effect of the unloaded face area outside the meshing contact causes an increased edge pressure within 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 (Fig. 4a), whereas in *KISSsoft 04-2010*, the normal force at outer ends of meshing contact is increased (Fig. 4b). Note that Figure 4a shows the pinion face width $b_2 = 50$ mm, whereas in Figure 4b only the common face width $b = 44$ mm is displayed.

Decreased stiffness on the side borders of helical gears. For helical gears, the tooth may be cut by the cylindrical bodies (Fig. 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.

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

$$c_{Pet_border} = c_{Pet} \cdot \sqrt{\frac{s_{red}}{s_n}} \quad (2)$$

- 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 04-2010* and the previous release. It is a helical gear (helix angle $\beta = 15^\circ$) with the equal face width $b = 44$ mm. 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 04-2010*, the normal force at the start as well as end of contact is increased.

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 \cos

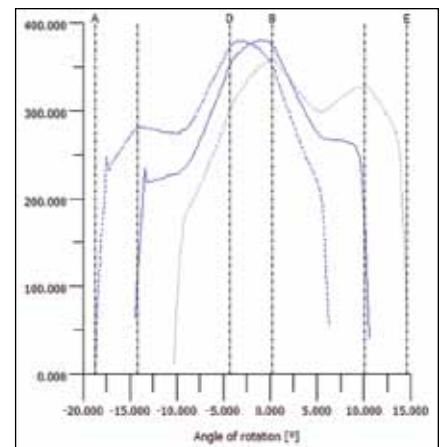


Figure 5a—Previous releases don't show higher normal forces (line load) at ends.

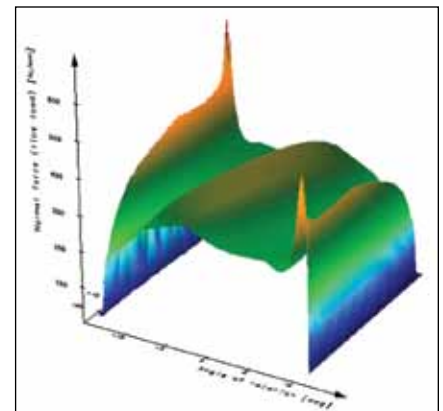


Figure 5b—KISSsoft 04-2010 shows higher normal forces (line load) at start and end of contact.

β , 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 04-2010* and the previous release. It is a helical gear (helix angle $\beta = 15^\circ$) with the equal face width $b = 44$ mm. In *KISSsoft 04-2010*, the tooth stiffness is slightly different from the previous *KISSsoft* release. Since the transmission error is strongly related to the stiffness, the transmission error slightly differs, too.

Load distribution considerations.

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 KH_α , KH_β as well as

KF_α , KF_β (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_f in planetary gears or gear pairs.

In Figure 7, the same gear calculation is compared between *KISSsoft 04-2010* and the previous release. It is a spur gear (helix angle $\beta = 0^\circ$) with the equal face width $b = 44$ mm. It's

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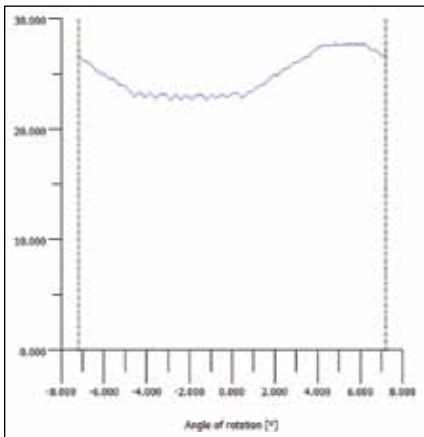


Figure 6a—Previous releases calculate slightly higher tooth contact stiffness.

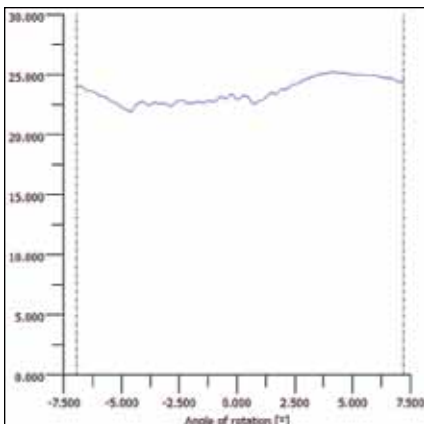


Figure 6b—*KISSsoft 04-2010* calculates slightly lower tooth contact stiffness.

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an overhang design, meaning the gear is outside the bearings. This results in an increased load distribution factor according to ISO standard calculation with $KH_{\beta}=1.27$ and $KH=1.0$.

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 *KISSsoft 04-2010*, flank pressure and root stresses are lower compared to the previous release. For a realistic contact analysis, the gear axis misalignments should be defined with shaft and bearing calculations, i.e., from *KISSsys*.

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 or corner at the beginning of an undercut. Then

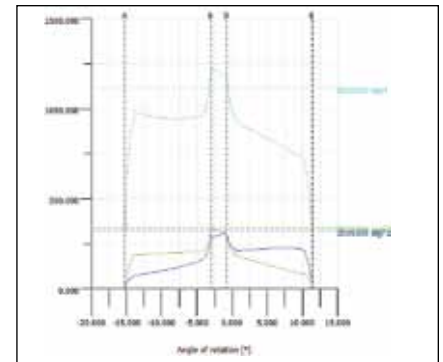


Figure 7a—Previous releases consider load distribution factors in tooth contact analysis.

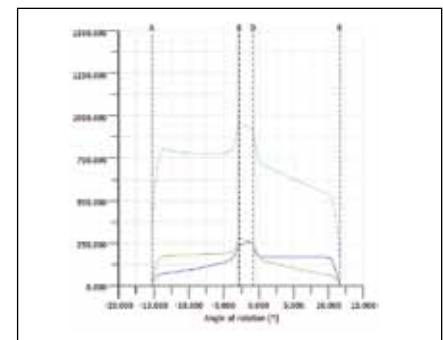


Figure 7b—KISSsoft 04-2010 doesn't consider load distribution factors from standard calculation.

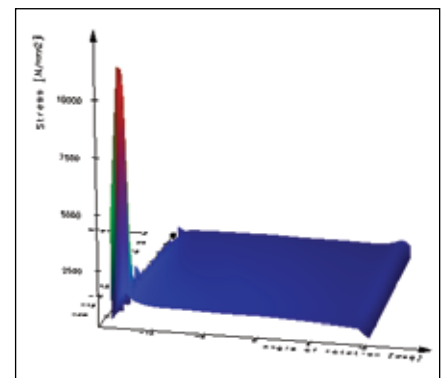


Figure 8a—High pressure peak due to no tip rounding.

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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 remain. *KISSsoft* recommends 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 04-2010* and the previous release. It is a spur gear (helix angle $\beta = 0^\circ$) with the equal face width $b = 44$ mm. In Figure 8a there is no tip rounding applied, whereas in Figure 8b there is a tip rounding of 0.5 mm. The pressure peaks are drastically reduced with the tip rounding.

3-D Display

With *KISSsoft 04-2010* the graphi-

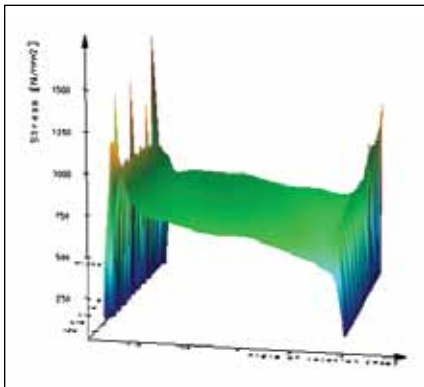


Figure 8b—Much lower pressure with tip rounding of 0.5 mm.

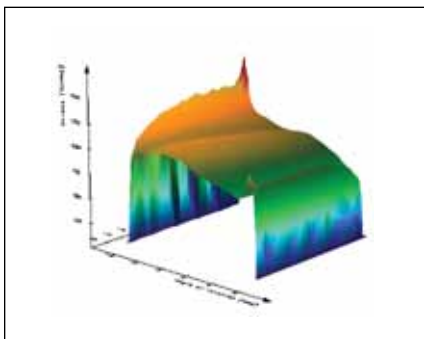


Figure 9—3-D presentation of stress level.

cal evaluation has been enhanced with 3-D graphics. However, the 2-D graphics remain as a good comparison to the previous releases. The 3-D graphics show a three-axis diagram, where the color indicates the stress level. In some cases there may be points where the stress data is 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.

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Guyson Corporation has introduced a 7-axis robotic pressure-blast shot peening system that is designed to support compliance with the most demanding process specifications and to enable automated peening of a wide variety of dissimilar components. The Model RB-10 was developed for technical surface treatment of gears and aerospace components.

The 60 x 60 x 60-inch blast cabinet is mated with a 6-axis robot, such as the FANUC M10iA, as a blast nozzle manipulator. The shot peening machine's rotary table has a diameter of up to 52 inches and is servomotor driven to be controlled as a seventh axis of robotic motion.

Locating hardware is provided to allow interchangeable component-holding fixtures to be positively and repeatably positioned on the turntable.

During the shot peening cycle, the orientation of the component and the motion of the robotic nozzle manipulator are synchronized to precisely replicate the programmed tool path, following the contours of complex-shaped parts, yet constantly and accurately maintaining the required angle of shot impingement, the correct offset of the peening nozzle from the target surface and the right dwell or surface speed to control the cold working process.

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