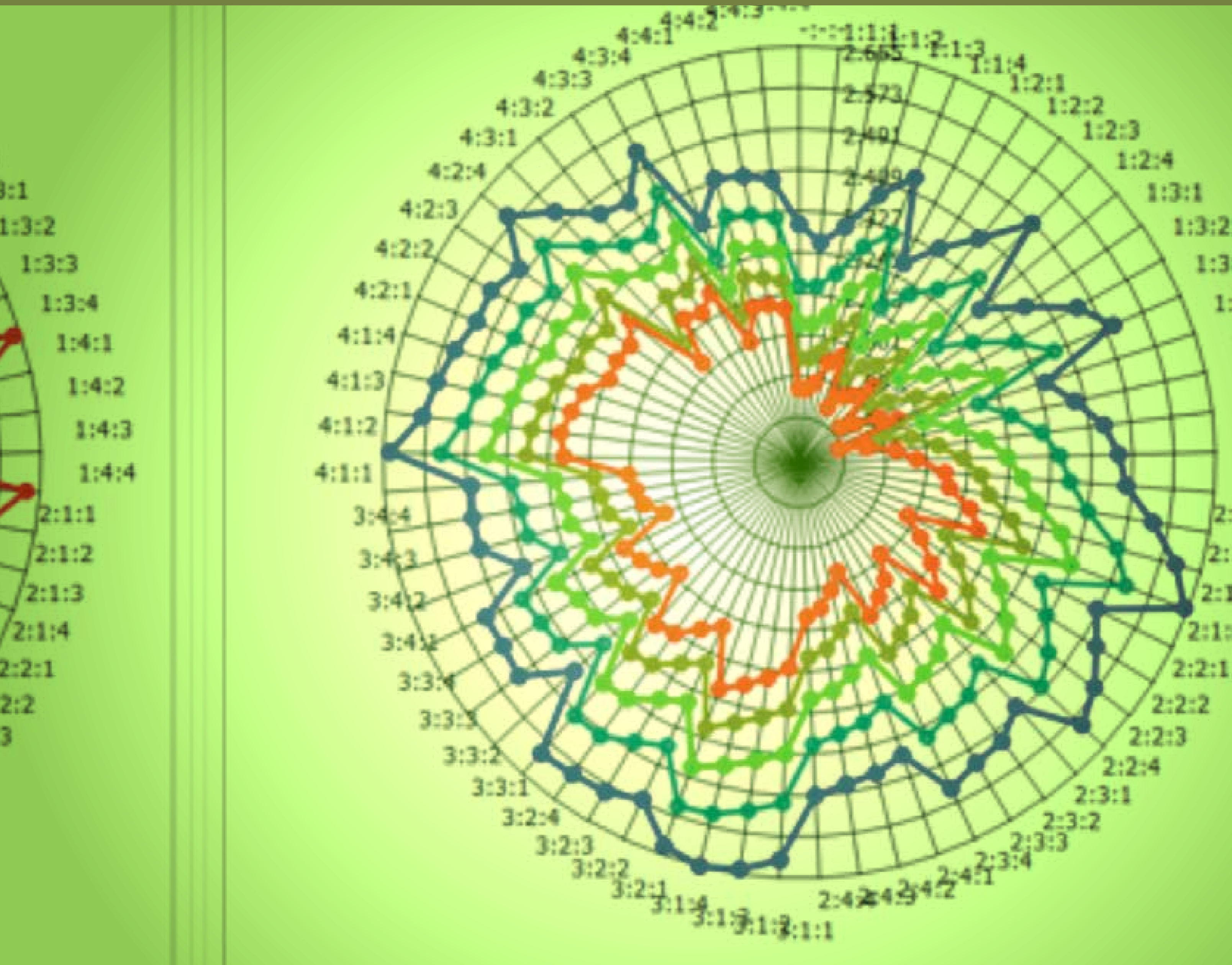
Contact Analysis for Planets in KISSsoft

A precise analysis of the contact ratios in the meshing of gears is essential, whether you are designing new gears or assessing existing gear units.

By Hanspeter Dinner, Ioannis Kaliakatos and Benjamin Mahr



KISSSOFT'S CONTACT ANALYSIS SOFTWARE IS WIDELY USED BECAUSE IT CALCULATES A WIDE VARIETY OF CHARACTERISTICS OF INTEREST FOR GEARS THAT ARE UNDER LOAD. THESE INCLUDE:

- THE NORMAL FORCE THAT OCCURS DURING MESHING;
- THE ACTUAL PATH OF CONTACT UNDER LOAD;
- THE TRANSMISSION ERROR, STRESS CURVES, FLASH TEMPERATURE, SPECIFIC SLIDING, POWER LOSS, HEAT GENERATION, LUBRICANT GAP THICKNESS, REQUIRED SAFETY AGAINST MICRO PITTING, WEAR, AND OTHERS.

To achieve this, the simulation covers the actual bearing and shaft deformation (both bending and torsion) along with, of course, the microgeometry of the toothing. The deformation of the planet carrier can also be taken into account by transferring the misalignment of the planet pin from a separate FEM calculation to the software.

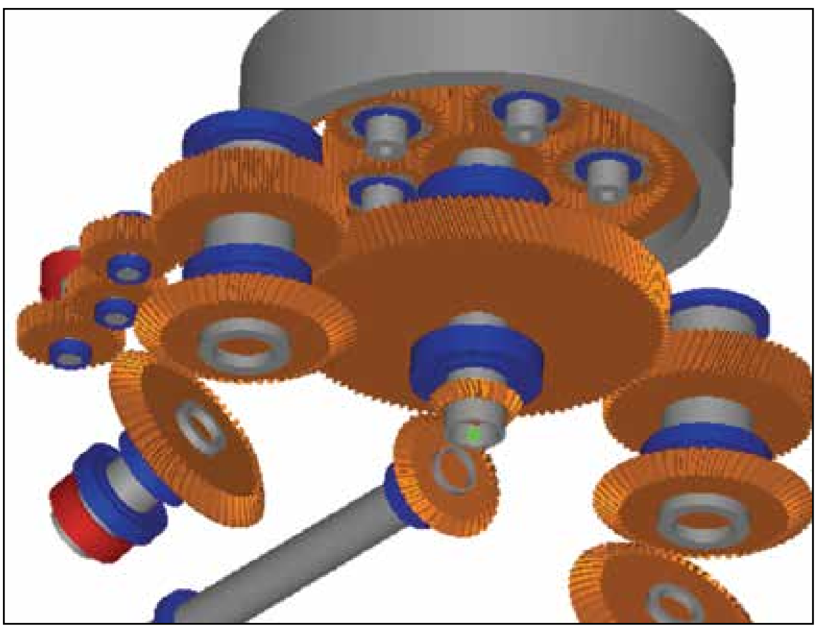


Figure 1: Helicopter gear with a planetary gear set.

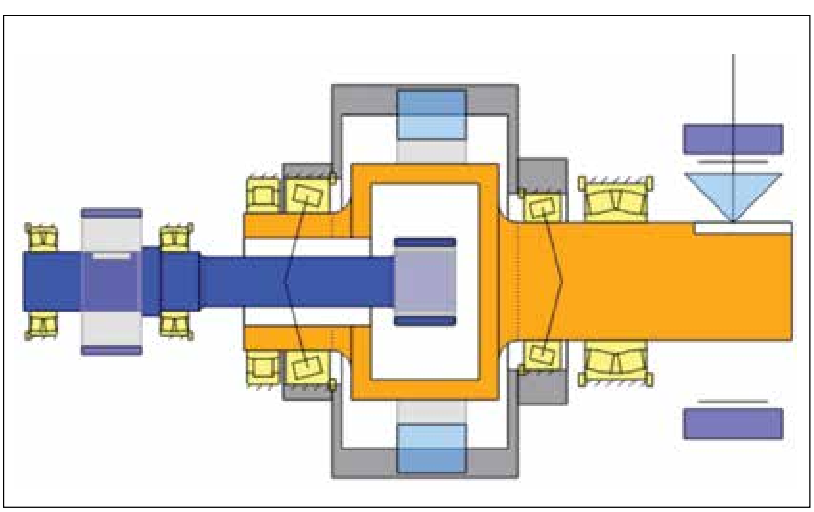


Figure 2: Shaft calculation with coaxial shafts.

SPECIAL FEATURES FOR PLANETARY STAGES

The contact analysis for planetary systems takes the system equilibrium into account in every meshing position. In addition, the face load distribution coefficient $K_{H\beta}$ and the static transmission error are also calculated for the entire planet gear. The interplay between the load distribution on the planet and its tilting on the planet bearings, which are subject to clearance, is resolved iteratively—until a state of equilibrium is achieved.

All contacts can be displayed simultaneously, either across three teeth or over a single tooth, showing how the system reacts and to compare the results of the calculation and the test outcomes.

Calculating load distribution between planets is unique for planet gears with more than three planets. In this case, the positioning errors of the planet bolts are taken into account.

LINK TO THE SHAFT CALCULATION

A new feature in Release 03/2013 allows KISSsoft shaft files to be entered directly in the contact analysis for planets so that the alignment and deformation of the tooth flanks can be determined. This takes into account the following deformations:

- misalignment of the sun, the planet carrier, and the internal gear
- torsion of the planet carrier as the tilting of the planet pin
- tilting of the planet on the planet bearings
- torsion of the sun shaft and the internal gear

If the stiffness of the housing or the bearing is defined in the shaft calculation (or calculated, as is the case for bearing stiffness from the internal geometry), these effects are included in the contact analysis calculation. A shaft file that involves a number of coaxial shafts can also be used.

The link to the shaft calculations takes place automatically in KISSsys, the KISSsoft system add-on. As a result, the behavior of the entire system can be taken into account without

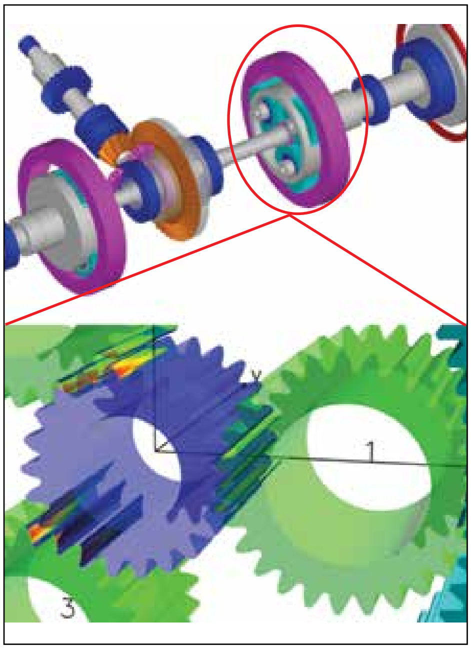


Figure 3: Display of gear units with KISSsys and automatic linkage of the shafts with the KISSsoft contact analysis module.

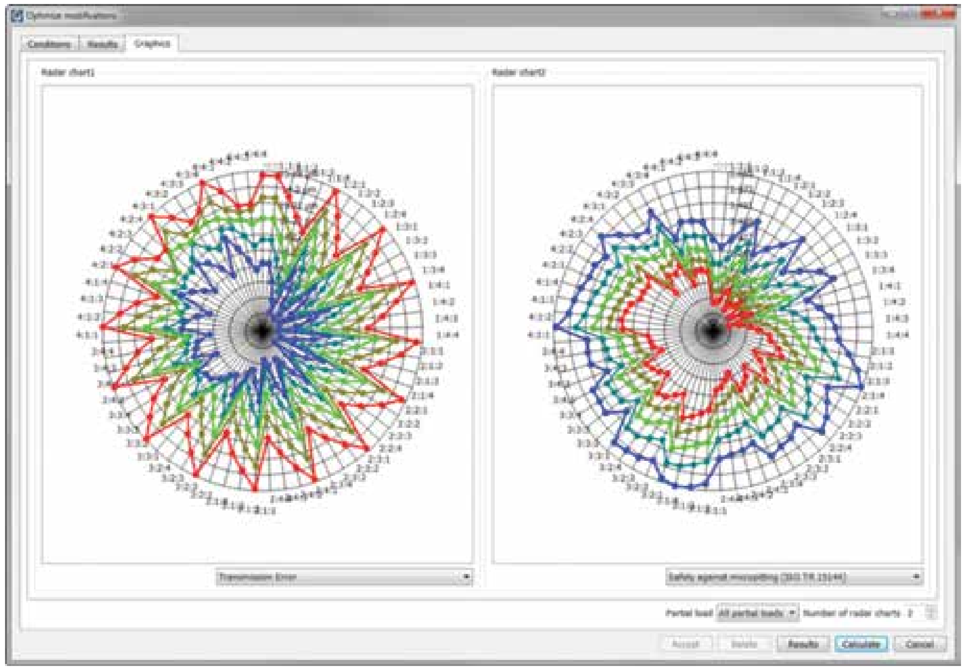


Figure 4: Display of radar charts for different criteria.

any significant additional time or effort (Figure 1). Alternatively, data about the deviation error and inclination errors of axis of shafts or planets can be predefined manually in the contact analysis, in the same way as before.

SIZING AND OPTIMIZATION OF PROFILE AND TOOTH TRACE MODIFICATIONS

Two assistants will help you size profile and tooth trace modifications. The tooth trace modification assistant suggests parallel or conical helix angle modifications, which the user can then accept if suitable.

Microgeometry optimization can now also be performed for planets and varied through all possible combinations of profile and flank modifications in a parameter range predefined by the user. This process runs a complete contact analysis for every variant, and also has the option of using different loads. The results of all the load and geometry combinations are listed in a number of different reports, which can easily be compared with each other.

The user can then select the best possible variant on the basis of the calculated transmission error, the face load factor KHβ, the safety against micropitting, efficiency, etc. Radar charts can display a number of results for several load levels at the same time.

In practice, it usually happens that different values are calculated for the optimum profile and flank modifications for different criteria. This new calculation in KISSsoft gives the users maximum clarity and provides them with sound basis for making decisions about optimizing toothing.

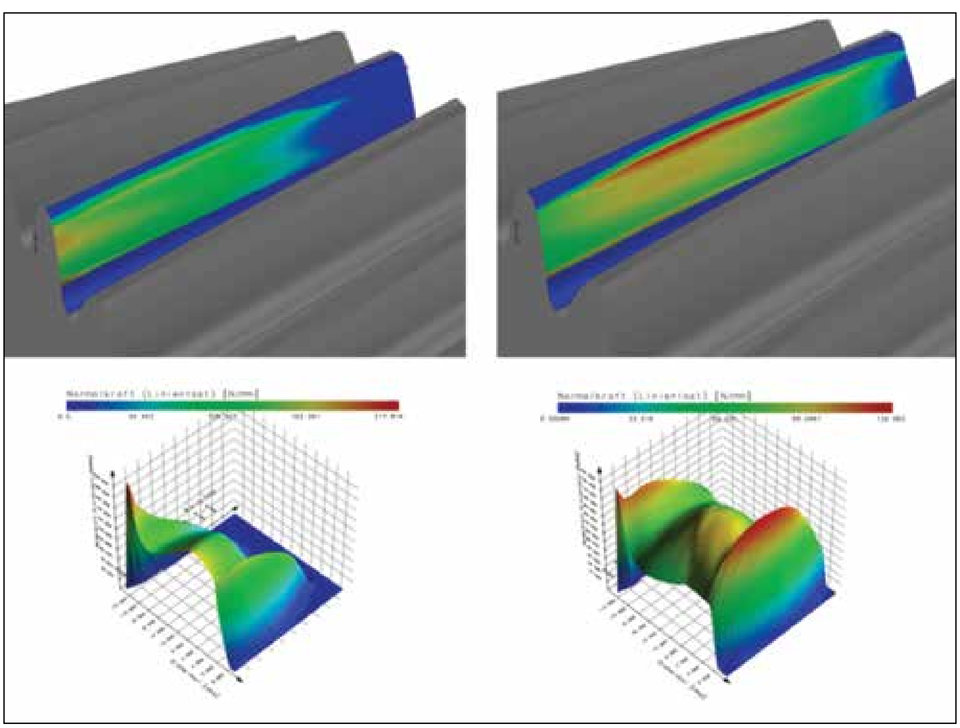


Figure 5: Contact pattern from contact analysis, before ($K_{H\beta}$ = 1.51) and after optimization ($K_{H\beta}$ = 1.15).

COMPARISON OF DIFFERENT CONTACT ANALYSIS PROGRAMS

To achieve a precise analysis of your new gears or existing gear units, modern analysis programs must take into account the effect of bending, torsion, and mounting misalignments of the shafts, the reduction or deformation of the bearing's internal geometry, and the deformation of the teeth in the actual meshing. This article discusses the theoretical basis of contact analysis according to Weber/Banaschek [1] in combination with a section model for both straight and helical cylindrical gear toothing that is used as the basis for contact analysis in KISSsoft [2].

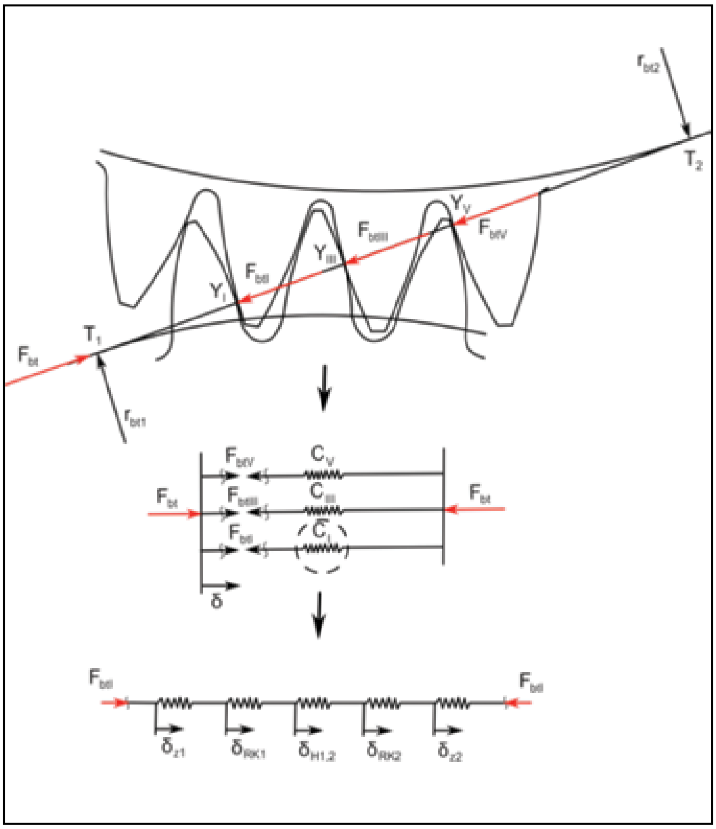


Figure 6: Stiffness model of an individual disk according to Petersen.

THE NATURE OF THE PROBLEM

The load involved in a meshing action causes the tooth to deform, the gear body to bend, and the tooth contour to be flattened by the Hertzian pressure created in the contact zone. This deformation effect is distributed across all the pairs of teeth involved in the meshing action. The deformation causes "elongation" in the length of the path of contact along the tip circles of the gears – and also causes the point of contact of each pair of teeth in the meshing to shift along the length of the path of contact. This effect causes an increase in the transverse contact ratio and may also give rise to an unwanted start and end of contact. This start and end of contact can be compensated for by the corresponding profile modifications (tip and end relief).

The varying number of teeth in contact also causes the normal force curve to differ during the meshing. Deflection, torsion and misalignment of the shafts lead to non-uniform load distribution across the facewdith. This may result in a less than ideal contact pattern and therefore reduce the service life of the gear. An accurate contact analysis that takes all these factors into account along with any profile corrections is required to analyze this behavior in greater detail.

STIFFNESS MODEL FOR A DISK

The stiffness analysis for helical gear teeth defined by Weber/Banaschek is for an infinitesimal thin disk. So far a helical gear has to be extended by the addition of a disk model. In this case, both the stiffness and the shift of the contact point along the path of contact is calculated for each of the n disks individually.

The stiffness model defined by Weber/Banaschek for an individual disk is calculated from the total deformation δ along the path of contact. Here, δ is the total deformation:

$$\delta = \delta_{z1} + \delta_{RK1} + \delta_{H1,2} + \delta_{RK2} + \delta_{z2}$$

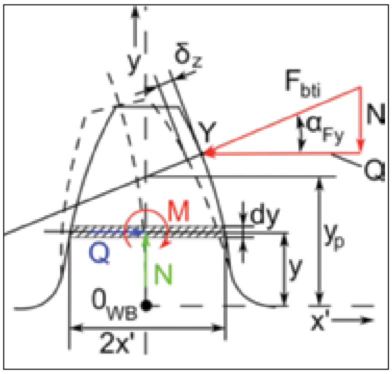


Figure 7: Bending of the tooth.

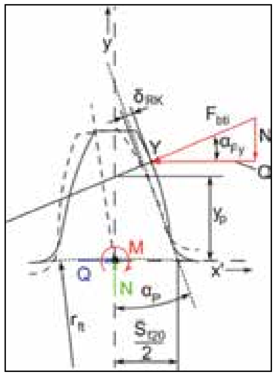


Figure 8: Deformation of the gear body.

The deformation of the tooth δ_Z is calculated from:

$$\delta_z = \frac{F_{bti}}{b}\cos^2(\alpha_{Fy})\frac{1-\nu^2}{E}\left[12\int_0^{y_P}\frac{(y_P-y)^2}{(2x')^3}dy + \left(\frac{2{,}4}{1-\nu}+\tan^2(\alpha_{Fy})\right)\int_0^{y_P}\frac{dy}{2x'}\right]$$

where the integrals $\int_0^{y_P}\frac{(y_P-y)^2}{(2x')^3}dy$ and $\int_0^{y_P}\frac{dy}{2x'}$ can only be resolved numerically. 8

The deformation of the gear body δ_{RK} is calculated from:

$$\delta_{RK} = \frac{F_{bti}}{b}\cos^2(\alpha_{Fy})\frac{1-\nu^2}{E}\left[\frac{18}{\pi}\frac{y_P^2}{\bar{s}_{f20}^2} + \frac{2(1-2\nu)}{1-\nu}\frac{y_P}{\bar{s}_{f20}} + \frac{4{,}8}{\pi}\left(1+\frac{1-\nu}{2{,}4}\tan^2(\alpha_{Fy})\right)\right]$$

The Hertzian flattening δ_H is calculated from:

$$\delta_H = \frac{F_{bti}}{\pi \cdot b_g}\left[\left|\frac{1-\nu_1^2}{E_1}\ln\left(\frac{b_H^2}{4t_1^2}\right)+\frac{\nu_1(1+\nu_1)}{E_1}\right| + \left|\frac{1-\nu_2^2}{E_2}\ln\left(\frac{b_H^2}{4t_2^2}\right)+\frac{\nu_2(1+\nu_2)}{E_2}\right|\right]$$

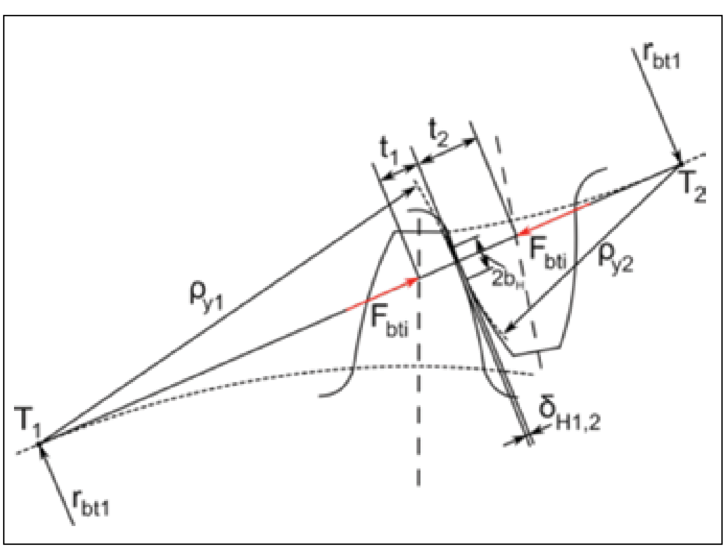


Figure 9: Hertzian flattening in the contact zone.

"The load involved in a meshing action causes the tooth to deform, the gear body to bend, and the tooth contour to be flattened by the Hertzian pressure created in the contact zone."

with

$$b_H = \sqrt{\frac{4}{\pi}\frac{F_{bti}}{b_g}\frac{\rho_{y1}\cdot\rho_{y2}}{\rho_{y1}+\rho_{y2}}\left(\frac{1-\nu_1^2}{E_1}+\frac{1-\nu_2^2}{E_2}\right)}$$

where b_g is the total width of gears 1 and 2.

According to the Formula above, it is possible to calculate the stiffness at each contact point at each disk and build a reduced linear equation system.

CONCLUSION

This method of analysis is one way of determining contact ratios during meshing. The stiffness model defined by Petersen makes it possible to precisely calculate the total meshing stiffness as well as the shifting of the point of contact along the path of contact. The use of a disk model extends the analysis option, enabling it to be used for helical gear teeth or cylindrical gears with uneven facewidth. The KISSsoft functions therefore enable designers to calculate and evaluate the contact pattern, the normal distribution of force along the tooth flank, or its progress along the length of the path of contact, and the distribution of Hertzian pressure, in conditions that are as close to real life as possible.

BIBLIOGRAPHY

[1] Weber, C. und K. Banaschek: Formänderung und Profilrücknahme bei gerad- und schrägverzahnten Rädern. Schriftenreihe Antriebstechnik, Heft 11, Vieweg und Sohn, Braunschweig 1953.

[2] KISSsoft, www.kisssoft.ag

ABOUT THE AUTHOR:

Benjamin Mahr was trained as mechanical engineer at Hochschule Darmstadt Germany. He started his career with his bachelor thesis at the KISSsoft AG Development Department in 2011. Today he is the main developer of the contact analysis of cylindrical and planetary gears and is involved in KISSsoft trainings as a teacher. Ioannis Kaliakatsos holds a M.Eng. in mechanical engineering from the National Technical University of Athens, Greece, with a specialization in robotics and automatic control. Having a passion for technical programming, he joined the KISSsoft team in early 2010 and has since been the main developer behind shaft analysis and bearing nonlinear stiffness. Hanspeter Dinner was trained as a mechanical engineer at the Swiss Federal Institute of Technology, Switzerland and the National University of Singapore. After joining KISSsoft AG as a support engineer, he became head of sales. He started his own consultancy company six years ago, working mostly in wind gearboxes and other heavy duty applications. He is a member of the Swiss Standardisation Committee on gearing and much involved in gearbox design, failure analysis, test witnessing and engineering analysis on a global basis.