Sizing of Profile Modifications for Asymmetric Gears

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Introduction

Lately, the use of asymmetric gears in automotive and other applications is an upcoming trend, though few applications are known to have asymmetric teeth. However, an increased interest in asymmetric gears can be seen (Refs. 2, 12–13). Many companies have started to design and test such applications.

The pressure angle in the normal section of an asymmetric gear is different for left and right flank. This can be used as an advantage compared to symmetric gears, since higher pressure angle increases the pitting capacity (Ref. 4). For gears rotating mostly in one direction, the loaded flank of the tooth can be designed with a high pressure angle and the non- (or seldomly) loaded flank with a lower pressure angle. For a given gear pitch, a low pressure angle on one flank permits the loaded flank to have a higher pressure angle compared to a symmetric gear (while avoiding a pointed tip, for instance). The main benefit of asymmetric gears over symmetric is higher load capacity for a given design (Ref. 4). The main drawback is a more complex manufacturing process (for steel gears) and related costs.

Currently there is no standardized method available for the geometry and strength calculation of asymmetric gears. For the calculation of bending strength, a method proposed by Langheinrich (Ref. 3) can be used, which is a combination of an analytical (based on ISO 6336) and an FEM approach. For the calculation of flank strength, the ISO 6336 calculation can be applied. Other gearing characteristics such as efficiency, micropitting, and scuffing can be checked based on the same calculation methods as for symmetric gears.

When an asymmetric gear design is evaluated, the following must be carefully analyzed:

- The potential gain in power capacity
- The eventual increase of the manufacturing costs
- The difference in noise/vibration behavior

For asymmetric gears, just like symmetric gears, profile and lead (flank line) modifications are important sources of improvement for torque capacity and noise/vibration behavior, as well as other characteristics. Due to the different profile of the left and the right flank, different modifications must be applied on both flanks. This again makes the manufacturing process more complicated. It's often possible to apply modifications to only the higher-loaded flank, but then noise performance of the lower loaded flank may be unsatisfactory.

Profile and Lead Modifications

The application of tooth modifications for asymmetric gears is like the technique used for symmetric gears. Modifications are mainly used to compensate for tooth and shaft deflection and manufacturing errors. Lead modifications are used to compensate for deflection of shafts, bearings and housing deformation. Profile modifications compensate primarily for tooth bending to avoid contact shock (premature start of meshing contact) and to reduce the transmission error (PPTE). Additionally, modifications may be used for other purposes, such as the reduction of Hertzian pressure in areas of the tooth flank where high pressure may cause issues.

Lead modifications are applied to compensate for shaft bending and torsion, misalignments due to manufacturing errors, bearing clearance, deformation and housing influence. Optimal lead modifications will normally increase the torque capacity of the gearbox due to a more even load distribution along the flank, thus reducing the face load factor $K_{H\beta}$. Typically, a helix angle modification is applied to compensate for shaft misalignments and crowning to compensate for general manufacturing errors and torsional effects.

An efficient layout technique to design modifications is to subdivide the task in three steps:

The first step is to apply the lead modifications for optimal load distribution over the face. Once the optimum modification is defined, the second step is the profile modifications. Profile modifications are more difficult to define and optimize, due to the different, and sometimes contradictory, requirements that must be fulfilled.

Various effects such as lower contact temperature, higher efficiency, smooth normal force distribution, and higher

Table 1 Symbols

Symbol	Description	Units
b	Face width	mm
c′	Maximum tooth stiffness per unit face width (single stiffness) of a tooth pair	N/mm/µm
$C_{\gamma\beta}$	Mean value of mesh stiffness per unit face width, secant value used for KHp	N/mm/µm
$f_{P(i)}$	Single pitch deviation of the paired gear	µm
$C_{\alpha a}$	Tip relief	µm
F_{bt}	Nominal transverse load in plane of action	N
$K_{H\beta}$	Face load factor (1S06336)	-
L	Length on involute	mm
L_{ca}	Length of the tip relief	mm
α_n	Normal pressure angle.	°
ε_α	Contact ratio under load	-

micro-pitting resistance may be achieved. The third step is to recheck the lead modification. But normally the load distribution over the face is not strongly changed by profile modifications, and so no or small adaptations are sufficient.

Load distribution over face calculation according to ISO6336-1, Annex E. For the application of the lead modification, a procedure, as described in ISO 6336-1, Annex E (Ref. 7), can be used for a fast and straightforward design of an optimum lead modification (Ref. 8). The procedure is relatively simple, compared with a full LTCA (see next section), and considers shaft misalignment due to bending, torsional deformation and manufacturing errors. Bearing stiffness/offset and housing deformation are also included (Fig. 1). Basically, this method is a one-dimensional contact analysis, providing the load distribution along the operating pitch diameter.

The method in ISO 6336 is not designed for asymmetric gears. But the only value, influenced by the asymmetric tooth form, is the meshing stiffness [$c_{\gamma\beta}$]. This value can be obtained with a LTCA calculation for the asymmetric gear pair or with the slightly adapted (Ref. 3) formulas of ISO 6336-1 for single stiffness c' and meshing stiffness $c_{\gamma\beta}$ (Table 2).

Loaded tooth contact analysis. The detailed effects of modifications must be checked with a complex calculation procedure, the loaded tooth contact analysis (LTCA). The aim of LTCA is to evaluate the gear mesh under load. For a consistent number of steps over one pitch during the rotation of the gears, the contact between all teeth under load is calculated. For the calculation of tooth deformation, a tooth stiffness model is required. An analytical model for tooth deformation was presented by Weber & Banaschek (Ref. 1), where gear deformation is divided into three main components:

- Gear body deformation
- Tooth bending deformation
- Hertzian flattening

Based on this method, an analytical stiffness model can be created. A loaded tooth contact analysis can then be performed based on the tooth deformation, shaft misalignments, manufacturing errors (e.g. pitch error) and a defined partial load for the calculation (Fig. 2). The results of LTCA provide important

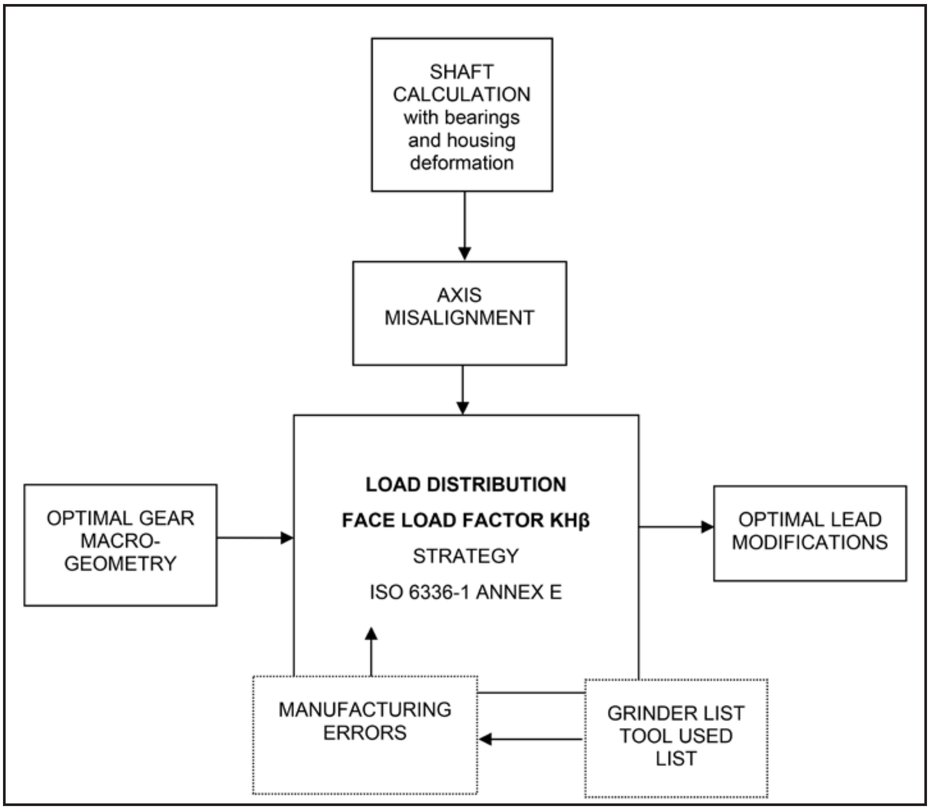

Figure 1 Optimization of face load distribution with lead modifications.

Table 2 Mean stiffness according ISO6336 and LTCA

	Mean value for single stiffness c′		Mean value for mesh stiffness cep	
All values in N/mm/µm	ISO 6336	LTCA	ISO 6336	LTCA
Symmetric gear, α_n 24°	13.6	10.8	15.5	19.4
Asymmetric gear, right flank, α_n 34°	14.8	12.6	15.1	19.8
Asymmetric gear, left flank, α_n 14°	12.3	9.9	17.5	21.3

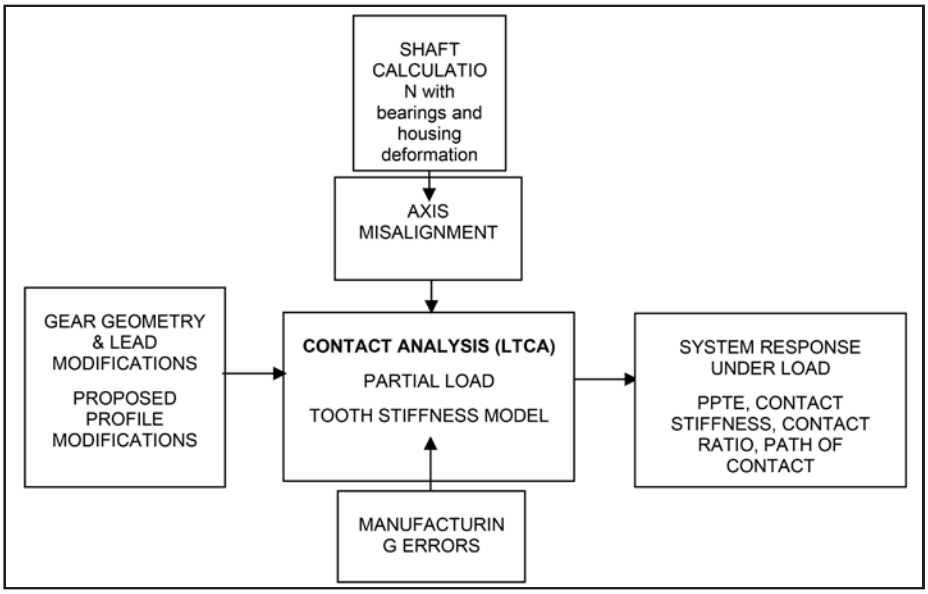

Figure 2 Loaded tooth contact analysis.

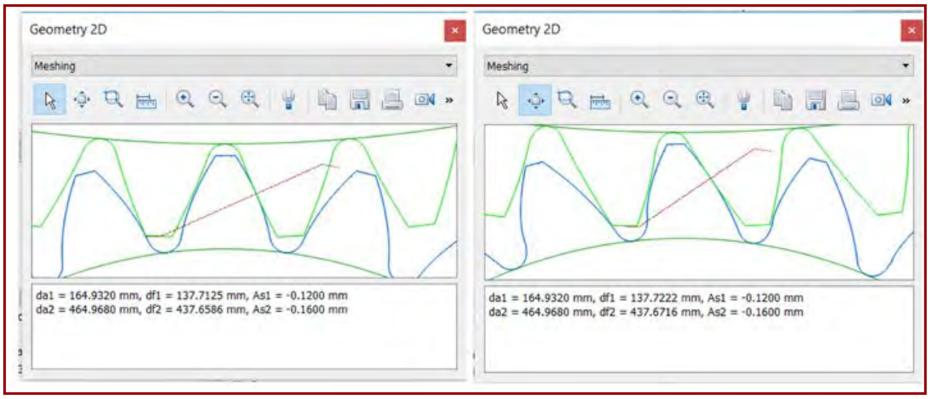

Figure 3 **Tooth form of a symmetric gear pair (pressure angle $\alpha_n = 24°$) and an asymmetric pair (pressure angle left/right $\alpha_n = 14°/34°$).**

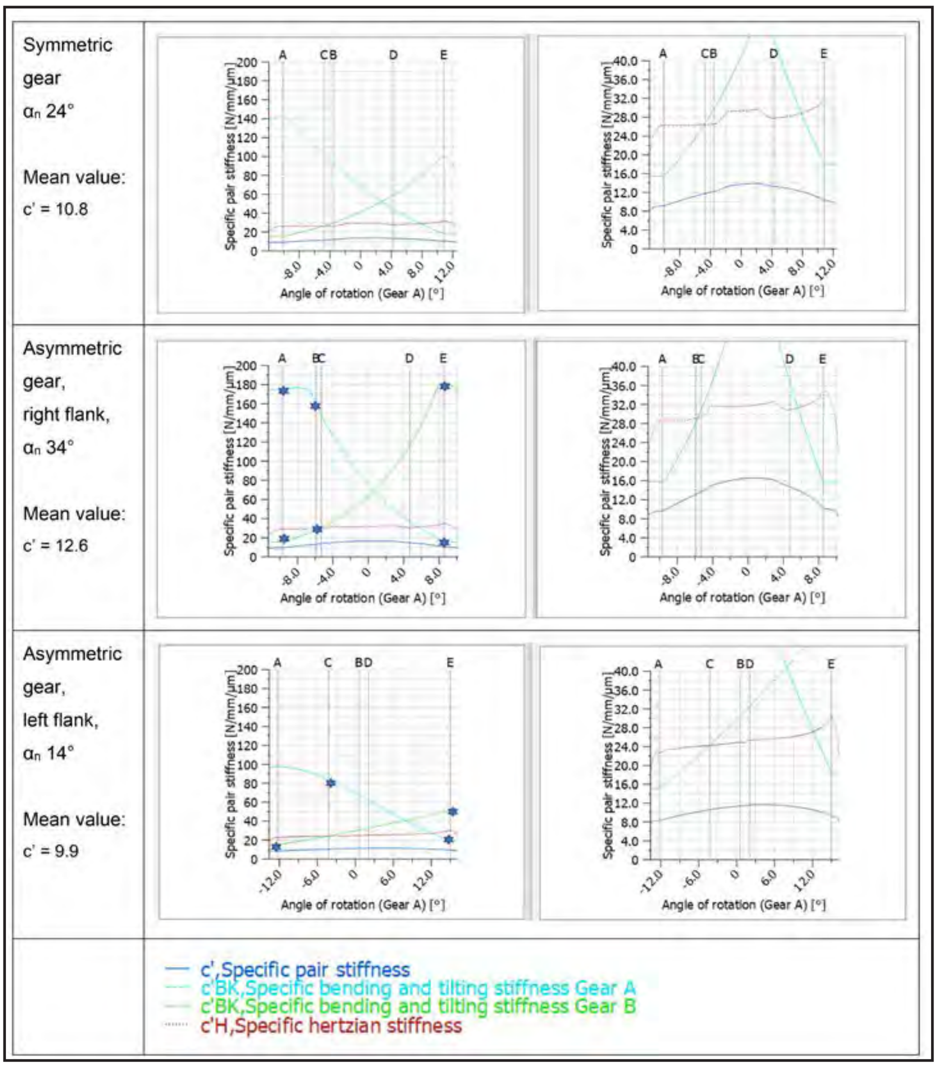

Figure 4 **Mesh stiffness c' of a single tooth pair over a meshing cycle, bending stiffness for a symmetric and an asymmetric gear pair; graphs on the left and rig, 0 to 40 (right). Indicates stiffness value calculated by FEM.**

parameters for noise characterization and optimization:

- Transmission error;
- Amplitude spectrum of the transmission error;
- Force excitation;
- Path of contact under load, etc.

The semi-analytical process according Weber & Banaschek (W&B) (Ref. 1) is very efficient and reliable, used by some of the best-known software programs (*as RIKOR;* Ref. 11) and others) for symmetric gears. The method describes the deformation of a tooth as a combination of four phenomena: The bending and the shear deformation of a one-sided clamped beam, the Hertzian flattening in the meshing contact and the rotation of tooth in the gear body. An LTCA calculation can also be performed with an FEM tool, but every LTCA result requires a large number of individual FEM calculations (to calculate different contact positions during a gear mesh) and is very time consuming.

As it is not easy to directly find a modification set providing a good solution, many variants must be checked. Therefore, a consistent amount of LTCA calculations is needed. Doing this with a FEM-based method is not recommended. Kapelevich (Ref. 2) proposes a partially FEM based method to speed up calculation time, but only for spur gears.

As the W&B method is accepted, quick, and well documented, the author, together with Mahr & Lang-Heinrich (Ref. 5), decided to adapt the method for asymmetric gears. The method is implemented in the contact analysis of *KISSsoft* (Ref. 9) and compared with FEM results. This now permits the study of the behavior of asymmetric gears with applied modifications.

Contact Analysis for Asymmetric Gears

Single tooth pair stiffness and meshing stiffness. The tooth stiffness of an asymmetric gear is quite different from symmetric gears. As previously mentioned, tooth stiffness is composed of tooth bending, gear body deformation and Hertzian flattening. The primary difference originates from the tooth bending. Hertzian flattening is different for the left and the right side of the asymmetric tooth due to the different flank curvatures, but identical to a symmetric tooth having the same pressure angle. Gear body deformation is similar to that of a symmetric tooth, with the exception of the load application angle.

Figure 3 shows a symmetric and an asymmetric gear pair with tooth numbers 25:76 (case carburized spur gear with module 4.0 mm, face width 44 mm, torque on pinion 1,600 Nm, speed 441 rpm). For these gears Figure 4 shows the single stiffness *c'*, which is the stiffness of a single tooth pair over the contact path. Also, the components of *c'* are displayed, the stiffness due to Hertzian flattening and due to the tooth bending and gear body deformation (tilting) of both gears. The stiffness for bending of the right flank with the higher pressure angle is *two to three times* higher than the stiffness on the left flank! Still, the single tooth pair stiffness c' is not so different: *c'* is just 20% lower on the lower pressure angle side. To verify the results, at some meshing positions the bending stiffness was also calculated by FEM. These points are indicated by a blue star in Figure 4.

The mesh stiffness, $c_{\gamma\beta}$, is the mean value of the stiffness of all the teeth in a mesh. For the determination of the face load factors $K_{H\beta}$ and $K_{F\beta}$, a line is constructed in the load-deflection graph between the origin and the pertinent load point used for evaluation of $c_{\gamma\beta}$. The mesh stiffness $c_{\gamma\beta}$ is displayed (Fig. 5). Although the stiffness for tooth bending is quite different for the left flank versus the right flank, the mesh stiffness is only 7% higher on the lower pressure angle side. It should be noted that *c' is lower, but is* $c_{\gamma\beta}$ higher, on the lower *pressure angle side*! This is due to the increase in profile contact ratio, ε_α, which is greater than two on the lower pressure angle side; therefore, at least two teeth are always in contact under load, increasing the total stiffness (Fig. 5).

As discussed earlier, the mean mesh stiffness $c_{\gamma\beta}$ is needed for the calculation of the load distribution over the gear face. The formulas provided in ISO 6336-1 (Ref. 7) for symmetric gears, adapted for asymmetric gears by Langheinrich (Ref. 3, Chapter 7), give relatively good results, compared with results obtained with the more sophisticated LTCA calculation. Therefore, for a first approach, the ISO 6336 stiffness values are practical.

Contact analysis. The stiffness model is an important component of the LTCA method. For this analysis, the gear is cut into several transverse sections and the stiffness is calculated for these slices. For a spur gear with lead modifications, or a helical gear, the beginning and end of contact of the slices is dependent upon the position of the slices along the tooth width. The total stiffness is calculated by integrating the stiffness functions for the slices over the width, with increasing delay at the begin of contact. In the simulation of the meshing the deflection of the teeth is introduced by the normal force applied divided by the stiffness.

Since the point where the force is applied varies in the height direction, the stiffness depends on the meshing position. Further, if an additional pair of teeth comes into contact the stiffness increases sharply, so the deflection of the first pair of teeth is reduced. To find the correct point of contact, an iteration must be performed.

Basically, the stiffness is the only part where a LTCA method for symmetric gears must be adapted for asymmetric gears. Clearly the sense of rotation is now significant, as this decides which flank is in contact.

Sizing of a Tip Relief Modification

The final step in any gear design is the definition of the profile modifications. As mentioned before, different features such as noise, contact temperature, efficiency, micro-pitting or scuffing can be improved with well-sized profile modifications. The reduction of noise/vibration generation is based on the following strategy:

- Eliminate contact shocks at the beginning and at the end of the mesh
- Reduce the amplitude of the transmission error (PPTE)
- Reduce the second and higher order of harmonics of PPTE to become as close to zero as possible

In ISO 21771 (Ref. 10) various modification types are defined. Typically, a tip relief (Fig. 6) on both gears is applied to reduce gear noise. The amount of tip relief $C_{\alpha a}$ is adjusted to eliminate contact shocks and the tip relief roll length L_{ca} is chosen to minimize PPTE.

In Niemann's book (Ref. 6), a useful suggestion for a reasonable tip relief is documented (Table 3). The values for $C_{\alpha a}$ depend on the gear stiffness numbers (*c'* for spur and $c_{\gamma\beta}$ for helical gears) and the single pitch deviation f_p. Based on a long

Symmetric gear, α_n 24°	Asymmetric gear, right flank, α_n 34°	Asymmetric gear, left flank, α_n 14°
Mean value for single stiffness: c' = 10.8 N/mm/μm	c' = 12.6 N/mm/μm	c' = 9.9 N/mm/μm
Mean value for mesh stiffness: $c_{\gamma\beta}$ = 19.4 N/mm/μm	$c_{\gamma\beta}$ = 19.8 N/mm/μm	$c_{\gamma\beta}$ = 21.3 N/mm/μm
Contact ratio under load ε_α = 1.70	ε_α = 1.49	ε_α = 2.07

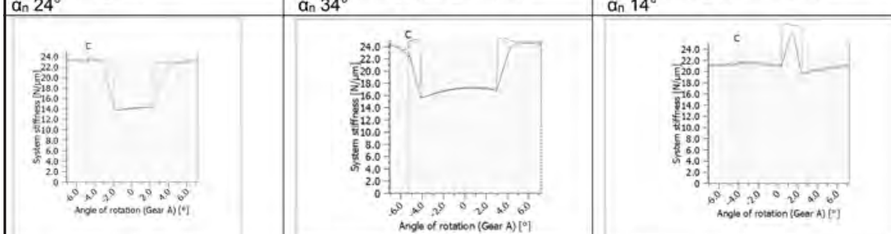

Figure 5 Stiffness (of all the teeth in a mesh) (blue dashed line: tangent stiffness $c_{\gamma\alpha}$; red line: secant stiffness $c_{\gamma\beta}$).

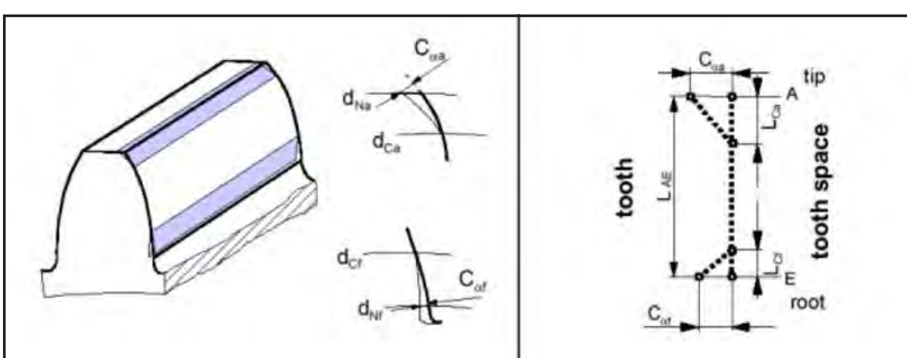

Figure 6 Tip and root relief (linear) (Nomenclature acc. ISO6336 (Ref. 10)).

experience with these propositions, it can be confirmed that for a first guess the values are appropriate; this applies also for asymmetric gears. With the correct amount of tip relief $C_{\alpha a}$, the pressure at the tip diameter should reach zero — and thus eliminate contact shock.

The length of the tip relief Lca has a great effect on the reduction of the peak-to-peak transmission error; PPTE is a valuable parameter for noise optimization. The Fourier transformation provides the orders of harmonics and allows the evaluation of excitation frequencies. From the transmission error and the contact stiffness, it is possible to derive the excitation force (EF), which allows the comparison of different geometric solutions in terms of vibration excitation.

In literature (Ref. 6), short and long modifications are proposed. The contact of the gears on the path of contact starts at point A and ends at E, going over meshing point C and the single contact points B and D as defined in ISO 21771 (Ref. 10). The short modification is applied from A to AB, respectively from DE to E. The long modification is applied from A to B, respectively from D to E.

As a general rule, which gives good results in many cases, the short modification increases the torque capacity of a gear set because high pressure and scoring risk in the tip area is reduced, but no noise improvement will be achieved (PPTE is not reduced). The long modification in most cases reduces PPTE significantly, but also reduces torque capacity.

Profile modifications can be defined by many different parameters. Tip relief is the easiest modification to apply in manufacturing, so in this paper for the sizing of the modification, tip relief is used. The main parameters are the amount of tip relief $C_{\alpha a}$ and the length L_{ca} of the modification. The form of the relief, if linear, arc-like or crowned, has also a significant, but less dominant, effect. So, for the sizing of an optimum profile modification, mostly $C_{\alpha a}$ and L_{ca} are crossed-varied.

General Procedure for the Sizing of Profile Modifications

Optimization of profile modifications in a case-by-case manner is extremely time-consuming and demanding. Results of an LTCA are not easy to evaluate. Comparing results of different LTCA calculations with slightly changed modifications is even more challenging.

Knowing this problem, a concept was developed, in partnership with a German gear company, the "modification sizing" tool. The basic idea is to systematically vary properties of an unlimited number of modifications. The possibility to cross-vary properties of individual modifications (such as tip relief to length of modification) is also available. With this, a certain number of variants with different modifications are defined. Then for every variant a full LTCA is performed and all relevant data is stored. This can be time-consuming if hundreds of variants are analyzed, but the process is fully automatic. To be able to provide such a process and still get a solution with hundreds of variants in a reasonable time (such as one hour), it is so important to use the modified Weber & Banaschek approach for the LTCA of asymmetric gears.

To explain the procedure, an example is discussed which was made for a US/Italy-based company. The main issue of a gear transmission was noise/vibration, but the Hertzian pressure was just at the limit at higher torques. The agreement made was that the noise had to be minimized at mean torques, but the pressure at high torques should not be increased.

Such problems are typical. Often noise problems must be optimal at the most typical load conditions, usually close to the mean load; but stress parameters must be optimal at peak loads. It is not possible to get optimum performance with modifications for every torque level, because tooth deformation is torque-dependent. But it also must be verified, that a modification with good results at mean torque does not perform poorly at the lower and higher torques.

The asymmetric gear pair discussed here has tooth number 30:43, module 4 mm, helix angle 17° and a pressure angle on the right flank of 31° (face width 40 mm, torque on pinion 1,682 Nm, speed 2,180 rpm). The actual lead modifications were checked with good results, and therefore not changed. For the profile modification it was decided, due to manufacturing restrictions, to apply only a tip relief. Then the modification variation process was used. Tip relief $C_{\alpha a}$ is varied in 6 steps, from 18 to 48 µm, and the modification length factor L_{ca}* (in module) also in 6 steps, from 0.5 to 5.5. All the values are cross-varied, therefore 36 variants are generated. Additionally, as a reference, the variant with no tip relief (declared number 0) is calculated. Both gears were synchronized, so in every variant both gears have the same profile modifications. This is a reasonable decision, otherwise 36*36 variants would have been generated (Fig. 8). More importantly, every variant is run at three torque levels (declared as min,

Table 3 Proposal for tip relief according Niemann (Ref. 6)

	Start of contact (for the gear, if pinion is driving)		End of contact (for the pinion, if pinion is driving)	
Tip relief	$C_{\alpha a}$ (min)	$C_{\alpha a}$ (max)	$C_{\alpha a}$ (min)	$C_{\alpha a}$ (max)
Spur gear	$\frac{F_{bt}}{c'b}+f_{p(i)}$	$\frac{F_{bt}}{c'b}+2f_{p(i)}$	$\frac{F_{bt}}{c'b}$	$\frac{F_{bt}}{c'b}+f_{p(i)}$
Helical gear	$\frac{F_{bt}}{c_{\gamma\beta}b}+\frac{f_{p(i)}}{2}$	$\frac{F_{bt}}{c_{\gamma\beta}b}+\frac{3f_{p(i)}}{2}$	$\frac{F_{bt}}{c_{\gamma\beta}b}$	$\frac{F_{bt}}{c_{\gamma\beta}b}+f_{p(i)}$

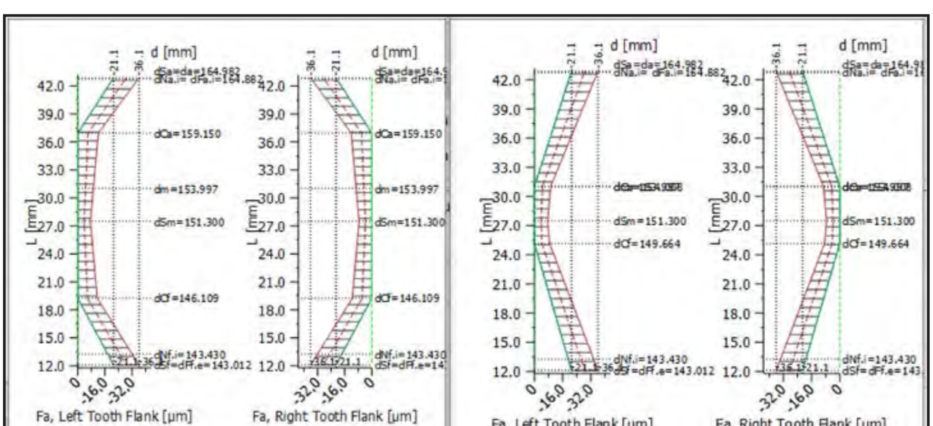

Figure 7 Profile diagram of a short (right) tip and root relief and a long relief (left).

mean and max). This allows the analysis of the results depending on the torque level. With 3 torque levels and 36+1 variants, a total of 111 LTCA calculations are performed.

The main LTCA results of every variant are collected in a list (in csv-format) and presented in different graphics. It is not easy to extract the optimum solution from such a large amount of data! In Figure 9 the spider diagram technique is used to give an overview of the most pertinent data. The solution 0 considers only the lead modifications (no profile modification). The solutions from 1–6 have equal $C_{\alpha a}$, the smallest value (18 µm), while the length of the modification L_{ca}* vary from 0.5 to 5.5. The solutions from 7 to 13 have a $C_{\alpha a}$ increased by 5 µm, corresponding to the profile slope deviation $f_{H\alpha}$ tolerance, and the same behavior for the length factor.

The use of the profile slope deviation $f_{H\alpha}$ tolerance as a step between different $C_{\alpha a}$ values tested is a clever strategy. If, for example, variant 11 is selected as optimal, then variant 5 and variant 17 have the same modification length, but 5 µm more material (variant 5) or 5 µm less material (variant 17) on the profile, therefore showing what happens if the manufactured tooth is at the min or max limits of the profile slope deviation.

The result overview in Figure 9 shows the profile contact ratio ε_α under load, together with the theoretical ε_α (orange circle). If the profile contact ratio under load is bigger than the theoretical value, than noise due to a contact shock will be generated. So, solutions for mean load (green line) outside of the orange circle should be avoided. The PPTE graph shows clearly which solutions have low values for all torque levels, therefore the choice for a variant with good performance can be made. The Hertzian pressure display shows as a red line the Hertzian pressure at the maximum torque. Solution 0 indicates the actual value (approx. 1,200 N/mm^2), which should not be increased.

Which solution is best? The first step is to check the contact ratio ε_α at the medium load. Solutions 13, 19, 25 and 31 are close to the theoretical contact ratio (orange curve) and have a good PPTE, but the Hertzian pressure is too high; solutions 2, 8, 14, 20 are better.

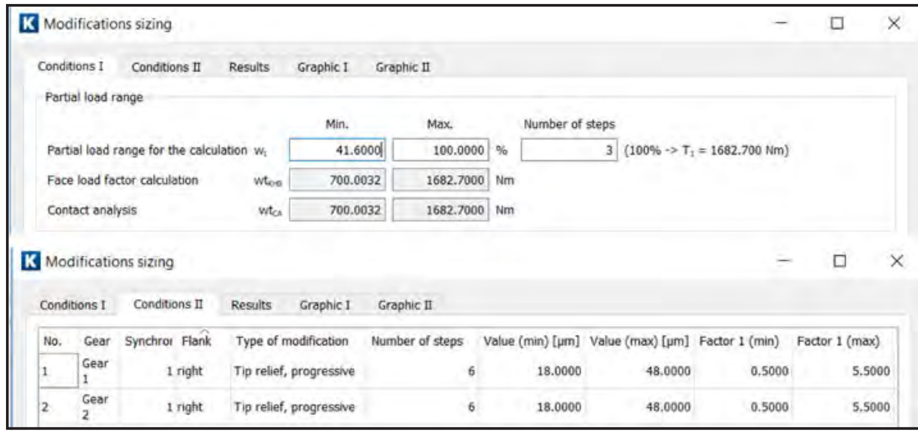

Figure 8 Process input data for the modification variant generator (Ref. 9).

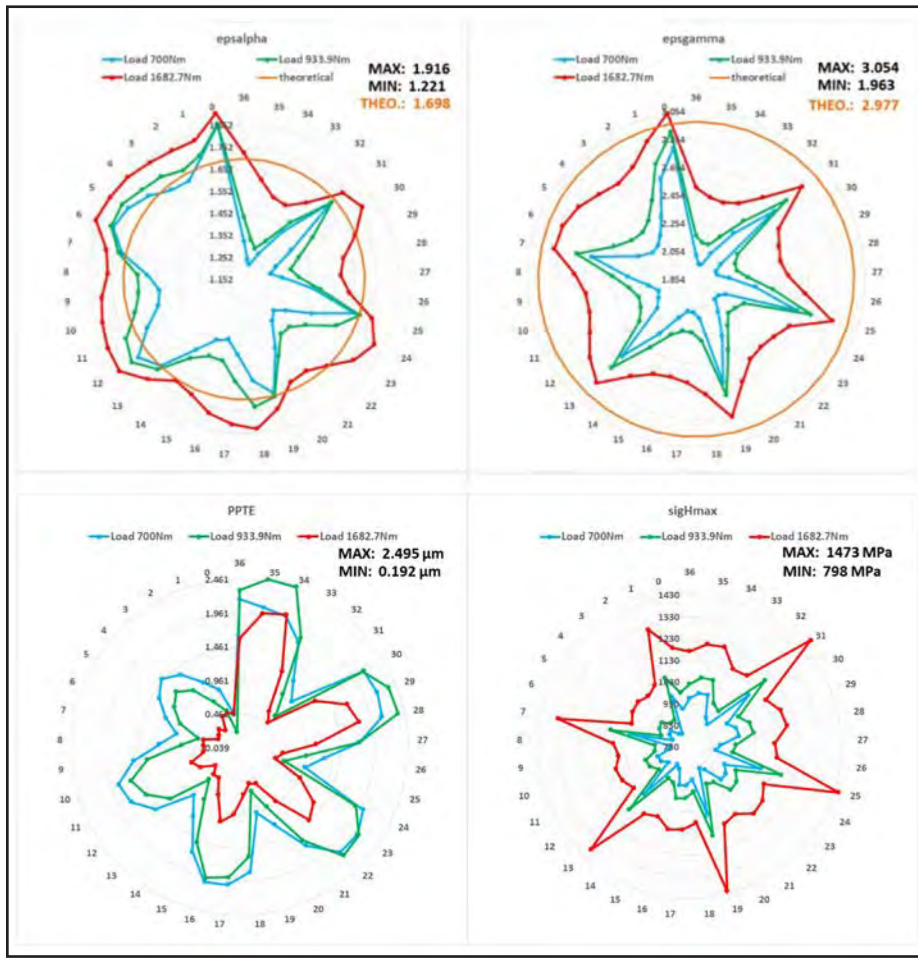

Figure 9 Overview of the main results for 37 profile modification variants at 3 torque levels; profile contact ratio under load ε_α, total contact ratio ε_γ, PPTE and maximum Hertzian pressure.

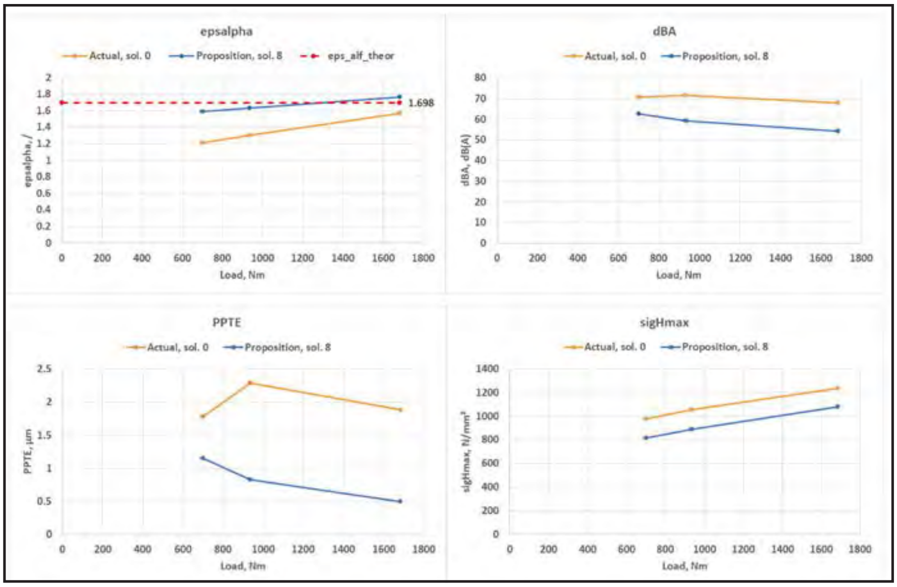

Figure 10 Difference between the gear pair with the actual modifications and the new solution.

Conditions I | Conditions II | Results | Graphic I | Graphic II

No.	Gear	Synchronize with no.	Flank	Type of modification	Number of steps	Value (min) [µm]	Value (max) [µm]
1	Gear 1	1	both	Pressure angle modification (value)	1	6.0000	6.0000
2	Gear 2	2	both	Pressure angle modification (value)	1	-6.0000	-6.0000
3	Gear 1	3	both	Helix angle modification, parallel (value)	2	-7.0000	7.0000
4	Gear 2	4	both	Helix angle modification, parallel (value)	2	-7.0000	7.0000

Figure 11 Inputs for the simulation of manufacturing errors to check the 'stability' of a proposed solution.

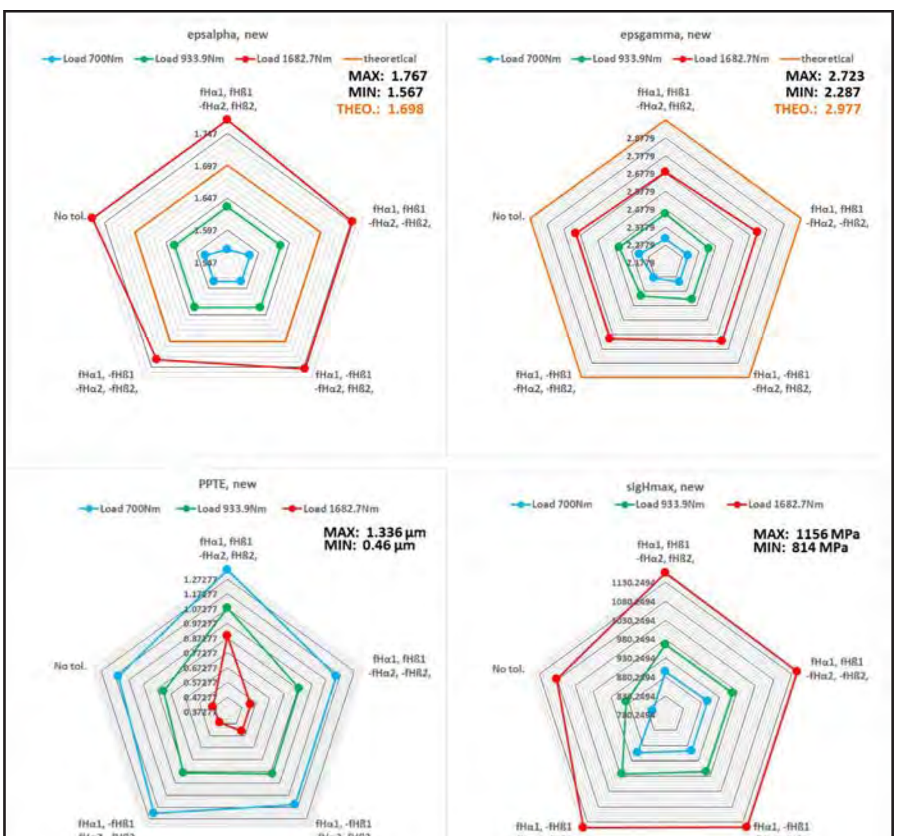

Figure 12 Simulation of manufacturing errors to check the 'stability' of a proposed solution. 'No tol' on the right side is the theoretical solution without any error. Displayed is the profile contact ratio under load ε_α, total contact ratio ε_γ, PPTE and maximum Hertzian pressure.

Comparing these 4 solutions, solution 8 can be a "good compromise" for min, mean and max load. The solutions 2 = (8–6) and 14 = (8+6) have a similar PPTE and an ε_α a bit higher (solution 2) or a bit smaller (solution 14). For this gear pair, solution 8 was chosen.

Then the procedure must be repeated for the left flank. The left flank, in the discussed project, is less loaded and seldom used, so no improved modifications were needed. In Figure 10 the main parameters of the actual gear set are compared with the optimized new solution. The difference is significant. The same procedure was then repeated for all gear pairs in the transmission. Meanwhile, the modified gears could be tested with a prototype on the test fixture. The first results are very satisfactory. We are looking forward to getting permission to publish these results in our next paper.

Influence of Manufacturing Tolerances

Grinding of asymmetric gears is challenging (Ref. 4). It is tougher to achieve the same quality that can be achieved for symmetric gears. Therefore, it is very important to consider manufacturing tolerances in the selection process of modifications.

As explained in the previous section, a specific modification should be as 'stable' as possible for torque variation. This is already a challenge, but the influence of manufacturing tolerances makes this more difficult. Tolerances, depending on the required gear quality, may be substantial compared with the amount of the proposed modification. What if we find a good modification with a tip relief of 5 µm, but profile deviation according the quality of ±10 µm?

Clearly, for good success a high gear quality is needed. Nevertheless, a check if the tolerances may cancel any benefit coming from the selected modifications is recommended. So the solution, as found in the previous section, must be checked for stability of the main parameters when profile and lead errors are added.

To consider the manufacturing tolerances, again the "modification sizing" tool can be used. This time the modifications of the previous solution 8 are kept constant, but additionally profile and helix angle modifications are varied to simulate manufacturing errors (Fig. 11).

Figure 12 shows the results. The solution without errors "No tol" is compared to variants with positive and negative errors. The contact ratio under load ε_α has a minimum of 1.57 and a maximum of 1.77. The largest PPTE increases from 1.15 to 1.34 µm. The Hertzian pressure increases from 1082 N/mm² to 1,156 N/mm². Overall, the increase in the range of 10% of the critical parameters is acceptable. The proposed modifications can be considered appropriate for the manufacturing process.

Conclusion

The layout of modifications, particularly profile modifications, is difficult. It is challenging to find a good solution also with high professional knowhow. Therefore, the parameter variation technique as described in this paper is a useful procedure.

The verification of the effects of a modification must be made by LTCA, which is a time-consuming calculation process. To permit a parameter study, which demands a large number of variants, a reliable but fast method is preferred. The Weber-Banaschek approach was recently adapted for asymmetric gears to allow such efficient analysis.

The procedure to obtain optimized modifications of a gear set is applied on a gear drive with noise issues. The procedure is discussed step by step on one of the gear sets. The results are very satisfactory compared to the original design. Low-, mean- and high-torque PPTE was reduced by 40% or more, contact shock was eliminated and the *db*(*A*) value was reduced by nearly 10 *dB*(*A*). The Hertzian stress was also consistently reduced. Finally, the proposed solution with manufacturing tolerances is verified to ensure that the selected modifications are appropriate for the manufacturing process.

For more information.

Questions or comments regarding this paper? Contact Ulrich Kissling at *ulrich.kissling@kisssoft.ag.*

References

1. Weber, C.and K. Banaschek. 1953, Formänderung und Profilrücknahme bei gerad- und Schrägverzahnten Rädern; Schriftenreihe Antriebstechnik, Heft 11, Friedr. Vieweg & Sohn, Braunschweig.
2. Kapelevich, A.L., and Y.V. Shekhtman, Y.V. Flank Profile Modification Optimization for Spur Asymmetric Gears, International Gear Conference Lyon, Chartridge Books Oxford, p. 66.
3. Langheinrich, A. "Geometrie, Beanspruchung und Verformung asymmetrischer Stirnradverzahnungen," Dissertation TU München, FZG-Nr. 201/2014, 2014.
4. Kissling, U., and A. Pogacnik. 2018, *Design of Aymmetric Gears — Potential and Limits*, International Gear conference Lyon, Chartridge Books Oxford, p. 1256.
5. Mahr, B., and A. Langheinrich. 2019, "Derivation of Tooth Stiffness of Asymmetric Gears for Loaded Tooth Contact Analysis," International Conference on Gears, Munich.
6. Niemann, G. and H. Winter. 1985, "Machine Elements Design and Calculation in Mechanical Engineering": Vol 2: *Gears*, ISBN 0-387-11149-2, Springer publisher, Berlin.
7. ISO 6336, 2006. Calculation of Load Capacity of Spur andHelical Gears, Part 1,2,3 and 6.
8. Kissling, U. 2017, Flankenlinienkorrekturen — eine Fallstudie, DMK 2017, ISBN 13:978-3-86844-940-2, Dresden.
9. *KISSsoft* gear calculation software, 2019, *www.KISSsoft.com.*
10. ISO 21771. 2007, Gears — Cylindrical Involute Gears and Gear Pairs — Concepts and Geometry.
11. *RIKOR*: LTCA calculation software of Forschungsvereinigung Antriebstechnik e.V, Frankfurt.
12. Sahin, B. and N. Yildirim, a.o. 2018, "Reduction of Tooth Root Bending Stress by Employing Asymmetry both in Tooth Profile Shape andTooth Fillet Form — Dual Asymmetry," International gear Conference Lyon, Chartridge Books Oxford.
13. Chao, C., D. Bestle, D. Kruger and B. Meissner. 2018, *Increasing the Tooth Root Capacity of Symmetric and Asymmetric Gears by Optimizing the Tooth Root Fillet Shape*, International Gear Conference Lyon, Chartridge Books Oxford.

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