KISSSOFT
Calculation programs for machine design

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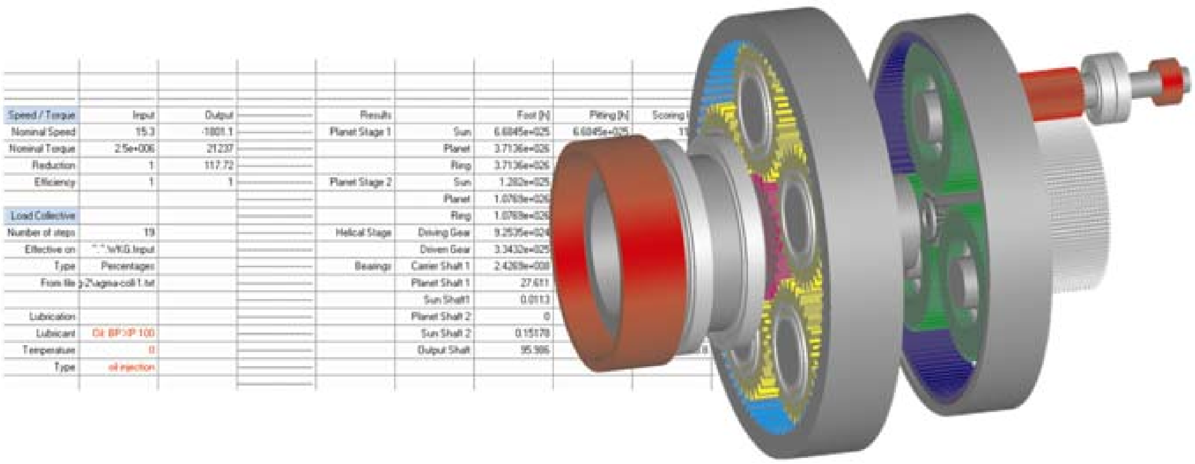


Participating Dialogue between the Partners

Next to the technical problems inherent to Gearboxes for Wind Turbine (WT), the questions of calculation precision, possibility of a simple and reliable exchange of data and the comprehensibleness of the documentation occur with increasing frequency. The indispensable dialogue amongst the gearbox and turbine manufacturers and the technical and the certification experts about the engineering calculations, can only be carried out in a balanced way, when all the partners have a minimum amount of knowledge at their disposal and use standardized methods and tools.

In the design and theoretical verification of Gearboxes for WT, some specific approaches to problems gain relevance in a way that can be considered as atypical for Gearboxes in the Energy Production branch:

- Loads: varying torque, increased load due to vibrations, torque direction changes
- Power: high torque with low number of revolutions, high power density
- Operation: temperature oscillations, cold start, idling, load at shut-down
- Gearbox dimensions: requires lightweight construction, compliant supporting structure

Besides the technical side, some questions on the collaboration between the gearbox supplier and the buyer are frequently asked:

- Calculation procedure precision and its documentation
- Safe data flow within and between the firms involved
- Understandability of assumptions and calculation methods
- Partner agreement on the assumptions and the targeted characteristic values
- Fast buyer control of the gearbox manufacturer calculations
- Interpretation of Norms and calculations in case of controversy

A gearbox calculation methodology that is manageable for both supplier and customer, presupposes, in the first place, a minimum of understanding, knowledge and experience on toothing theory, calculation and behaviour of WT in operation. The methodology must be applicable in a fast and secure way presupposing thus standardized tools, i.e., Software. Because of this, the calculation is now relatively less depending upon the inside know-how of the gearbox manufacturer and has to fulfill the gearbox customer demands for transparency. Furthermore, the Software must be usable for the whole procedure of layout, optimization, verification, documentation and study as, for instance, during revisions and/or training. Since this procedure spreads itself over several firms, the standardized Software may be used as a means of achieving a safe exchange of data.

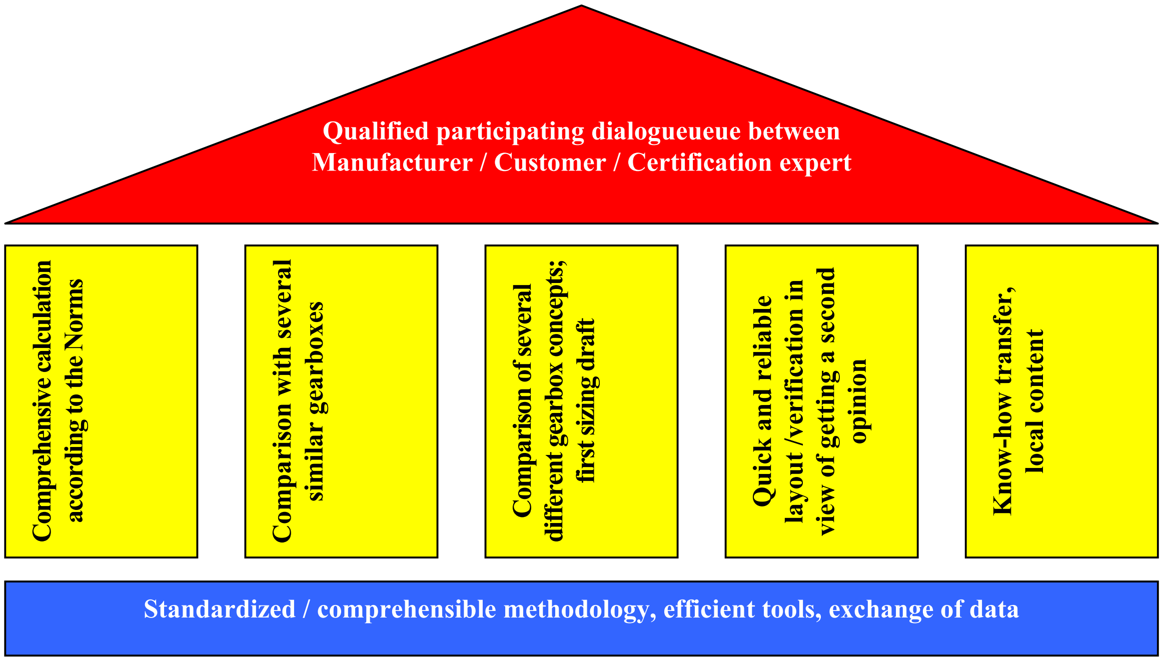


Fig. 0-1 Use of a standardized Calculation Procedure

Not all partners can afford the high costs involved with detailed analyses such as FEM (Finite Element Analysis) or vibration simulation. The idea behind this article is to describe, by means of an example of toothing calculation, how it is possible to get a qualified assessment about the toothing using user-friendly tools at a justifiable cost.

1- Gearbox Concepts

1.1 Current Gearbox Concepts

The following picture shows an overall view of the current gearbox concepts. At present, the standard is specially the one- and two-stage planetary gears (connected afterwards with two or one helical gear stages). An increase in power implying a larger number of gear-meshes results in a cost increase of the toothing calculation and further underlines it’s importance.

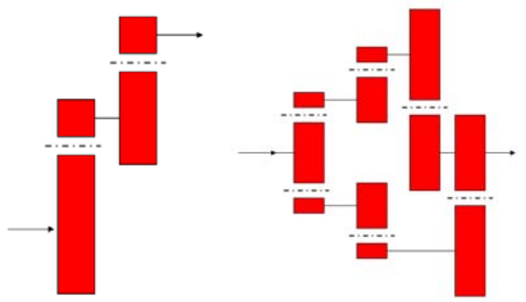
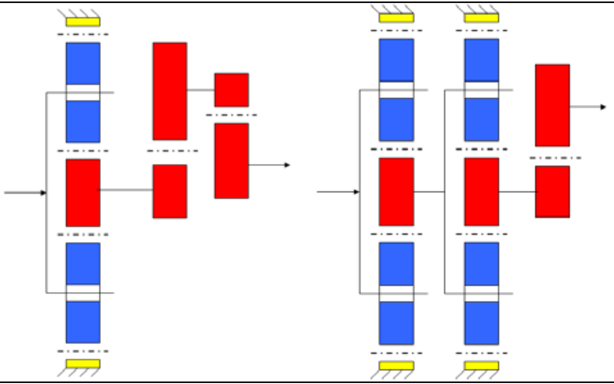
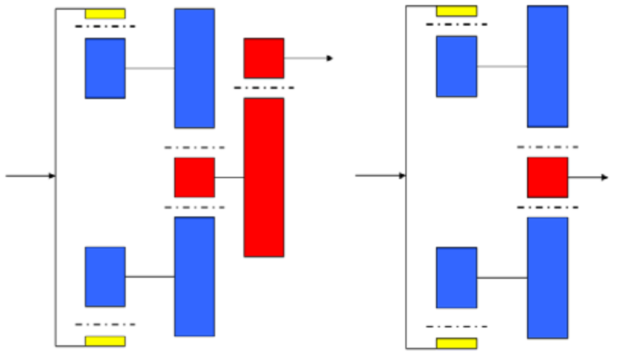
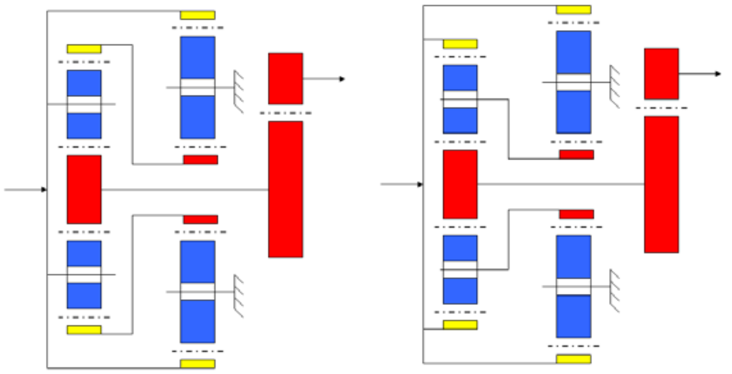
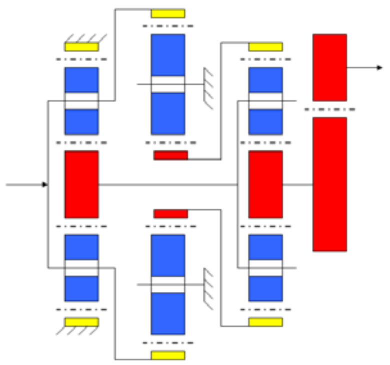
Helical gear gearbox, also power-split		Also for power-split solutions with several generators
One / two Planet stages Two / one Helical gear stages		Drive by the planet carrier PTO by the sun The ring is mostly in direct connection with the housing Helical helical gear stages
Planet stages With or without Helical gear stages (i. e. left: Wind Turbine/Renk, right: Multibrid/Renk)		Ring gear not part of the housing Favourable concerning bearing lubrication
Planet coupled gearbox (e.g. Left DPPV/MAAG)		Internal load distribution to the stages Ring gear not part of the housing
Planet differential gearbox (e. g. Bosch Rexroth,)		Third stage as overlapped stage Costly Construction Costly bearing

Fig. 1-1 WTI Gearbox Concepts

1.2 Trends

Occasional peak loads will be decisive for determining the size of the Wind Turbine Gearbox. We will consider some concepts that limit the acting torques in the gearbox. We can consider here a planetary set as a differential gearbox superimposed to the fast running stage and with the support moment at the ring hydrostatically limited (Henderson Gearbox). More complex are some concepts that, through a CVT gearbox, also maintain a constant output speed as much as possible. These can be designed either as hydrodynamic (Voith Win Drive) or electrical additional drives.

Other concepts go by that helical gear gearboxes must be used with multiple generators. The gearbox takes place through a central wheel, which meshes with the various pinions. The generators are either directly connected to the pinion shafts (e. g. Clipper Concept) or the power of several pinions is split over two generators using helical gear stages (e. g. Multi-Duored Concept).

2 Gearing Verification of WTI Gearboxes

2.1 Leading Verification Methods

The gear verification is carried out according to the valid Norms and Directives currently in force. Following, we will assume that these calculation methods are known and will point out specific methods that deviate from the Norm and are meaningful for WT gearing.

Tooth Root Strength

As a rule, Norms underestimate the contact ratio influence, especially where it is typical to require a high precision manufacturing –WT Gearboxes. The following figures show the actual stress conditions (Flank – light blue; Root 1 – Violet; Root 2 – red) as compared to the results using the gear-mesh calculation according to DIN 3390. The tooth root stress is especially overestimated. Following figures show the comparison between the tooth root stress according to the Norm and the same stress taking into consideration the load distribution. Furthermore, this comparison clearly shows the influence of the pitch error on the stress conditions.

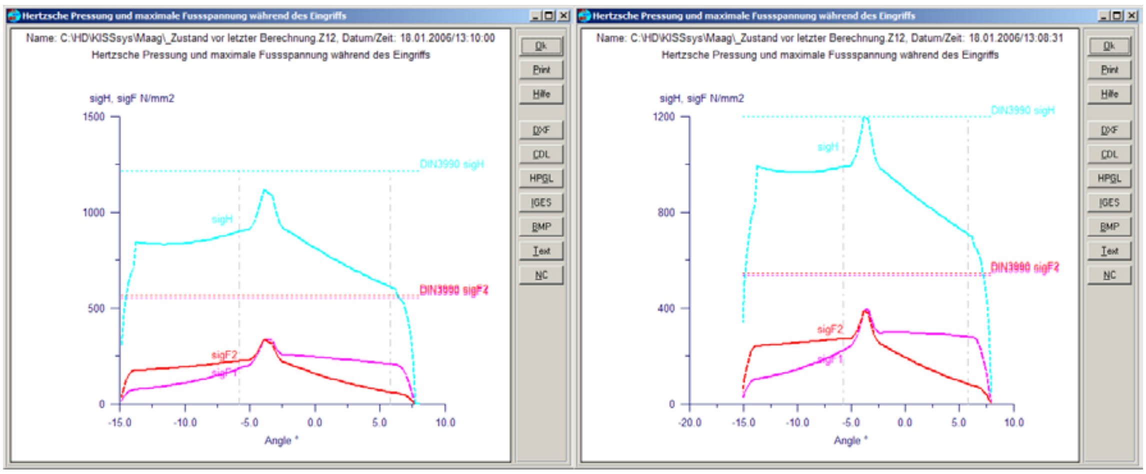


Fig. 2-1 Influence of the Load Distribution over several Teeth on the Stress Levels. Left for Quality 7; Right for Quality 5. Final Stage of a 3.6 MW WTI ε_α=1.73.

Following modifications for the calculation of the ring gear strength are recommended:

- Calculation at the 60º Tangent contact point taking into consideration the root contour actually produced by the tool as, for instance, suggested in the present revision (FDI) of the ISO 6336 (or the VDI 2737) (left figure below)
- Calculation of the exact tooth profile based upon the generation simulation with the tool to calculate the YF and YS along the entire fillet ("graphical method", right figure below)
- Taking into consideration the ring gear's wall thickness

Comparing both figures shows that the calculation in accordance with the at the time valid DIN or ISO Norm versions strongly underestimates the ring gear strength.

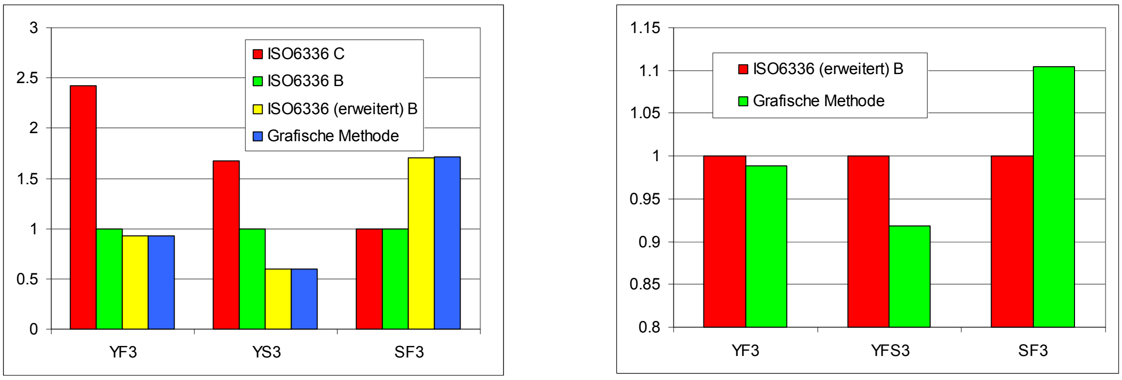


Fig. 2-2 Influence of the Calculation Methodology on the Ring Gear calculated Tooth Root Safety.
Left: Reference Tooth Profile 1.25/0.38/1.00;
Right: Reference Tooth Profile 1.40/0.38/1.25

Tooth Flank Strength

Because of the flank's pressure charge, a shear stress distribution builds up in depth with a maximum under the surface. In case of overload, the increased shear stress results in cracks under the surface that lead to 0.5 to 1 mm material deterioration (pitting). During the case hardening of the flank, the hardening depth must exceed the one of the maximum shear stress peak depth. On the other hand, out of cost reasons, the case hardening depth must be as small as possible and that is where the shear stress distribution calculation that follows becomes interesting.

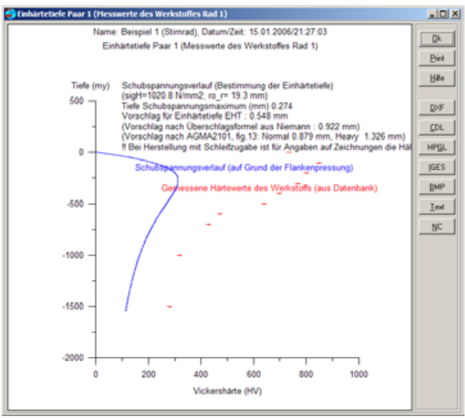


Fig. 2-3 Hertzian Stress Distribution inside the Toothing

Scuffing

The ISO 6336 offers no solution for scuffing; it is usual here to use DIN 3990 Part 4. The AGMA 6006 as well as the GL guideline demand that the FZG lubricant scuffing load level for the calculation be reduced one unit. Nevertheless, because modern oils with EP (extreme pressure) additives have a scuffing load > 12 and the calculation methodology considers only maximum values of 12, this limitation is in practice less relevant. The calculation of the safety against scuffing is based upon an estimated temperature at the tooth line of contact, which depends upon the pressure and the sliding speed. Scuffing appears mainly on the tip and root areas where high relative speeds exist. In order to reduce the contact stress in these gear mesh areas, it is necessary to carry out tip- and/or root reliefs; the following figure shows their influence in the contact temperature.

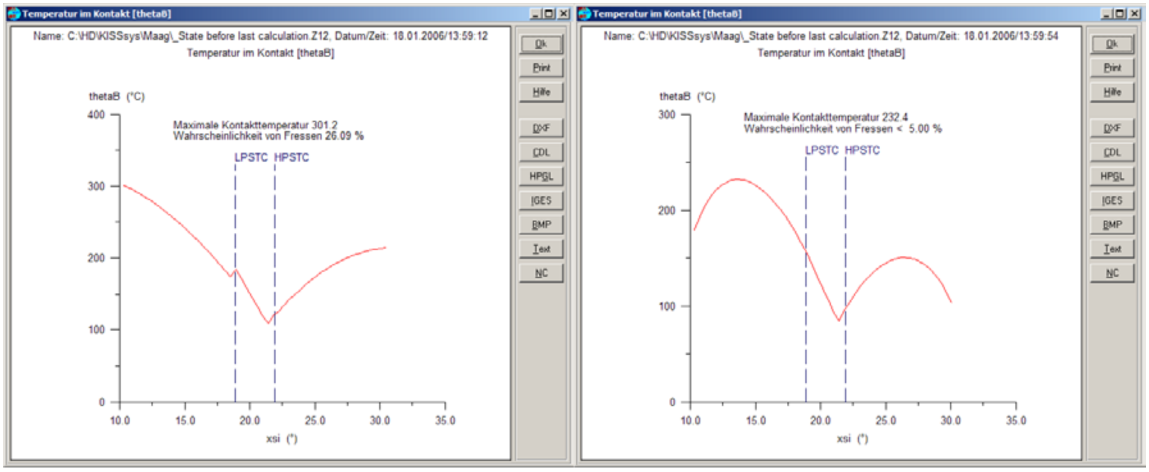


Fig. 2-4 Flash Temperature Reduction from 300 to 230 ºC because of Tip Relief. End Stage of a 3.6 MW WTI. Calculation according to AGMA 925-A03.

Micro pitting

The friction between the tooth flanks due to insufficient lubrication, high load or unfavourable operational parameters, will also increase the friction due to tooth contact roughness (mixed friction μ=0.2-0.4 as opposed to viscose friction μ=0.05). Because of the higher friction stress, a shear stress develops at the tooth surface that can go beyond a critical value even at a still uncritical torque for creating pitting. This overstraining leads to material erosion at the surface (failure depth circa 10-20 µm over an area of about 20x100 µm) and to a grayish couloring of the tooth surface. The resulting profile error leads to a higher tooth stress ($K_{H\alpha}$, $K_{F\alpha}$, K_v, noise increases) and the risk of pitting also increases. A measure against the micro pitting risk is the specific lubricating film thickness λ, the quotient between the lubricating film thickness and the roughness of the toothing surface. The calculation is carried out in accordance with AGMA 925 or the FVA Work Sheets 54 / 259; there is also an ISO Norm in preparation.

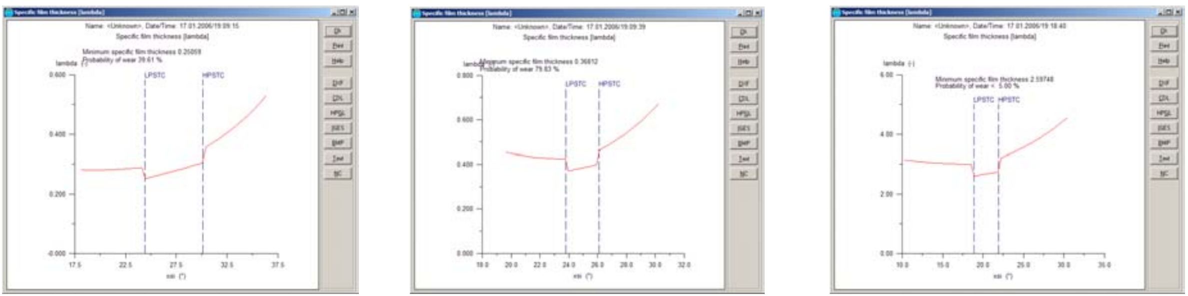


Fig. 2-5 λ-Values for the 1st and 2nd Planet Stages (Sun Planet Meshing Gear) and Helical Gear Stage for a 3.6 MW WT Gearbox

Static Verification

We must also carry out the static strength verification for toot plastic deformation or fracture. Since the classical static verification is not covered in the Norms, it is frequently considered as fatigue verification with a cycle factor = 1.

However, verification for yield- and fracture limits is more meaningful.

Required safety factors

The required calculation safety factors are specified in the Norms. Determining the really required safety factors is extremely difficult and asks for a comprehensive field experience. The knowledge of these required safety factors constitute an enormous capital for the gearbox manufacturer because they can only be acquired not bought.

Norm	Verification in accordance to	SF: (Fatigue / Static)	SH: (Fatigue / Static)
AGMA6006	AGMA2101-C95	1.0	1.0
AGMA6006	ISO6336	1.56	1.25
GL Norms	ISO6336	1.5 (1.4)	1.2 (1.0)
Danish WT Cert. Sch.	ISO6336	1.45	1.2
IEC61400	ISO6336	1.56	1.25

2.2 Transmission Error, Meshing Shock

The environment demands that WT have a low noise profile. The vibrations generated by the gearing are a considerable source of noise. Thus, the target must be, on one hand, to maintain the transmission error as low as possible and, on the other hand to reduce the meshing shock. This is why the layout of the tooth height and the tip relief (amount, type and height) is of utmost importance.

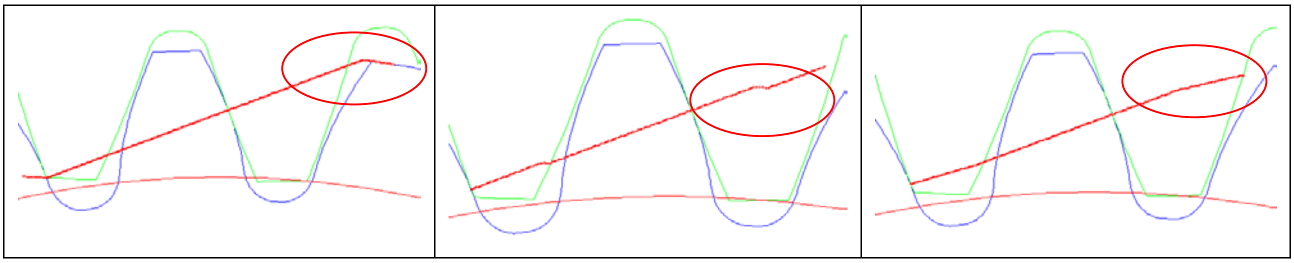


Fig. 2-6 Line of Contact under Load. End Stage of a 3.6 MW WTI:
Left: Premature Meshing in an uncorrected Toothing (Inflection in the Line of Contact).
Center: Toothing with a shorter linear Tip Relief (6.2 µm and 0.5 µm).
Right: Toothing with progressive Tip Relief (9.3 µm and 7.5 µm).

On the left figure, the premature meshing (prolongation of the line of contact) leads to a so-called meshing shock that can be reduced by means of a tip relief and, afterwards, the line of contact will not show the characteristic prolongation anymore. However, for a linear relief, one can see this unstability in the line of contact (center figure) but this abrupt change in the angle of rotation can lead to vibrations. We can avoid this with a progressive relief (right figure).

2.3 K Factors

Application Factor K_A

For the gearing layout, a load spectrum can be easily replaced by a nominal load and an application factor K_A. The calculation of K_A can be done, e. g. in accordance to DIN 3990, Part 6, Method III or to AGMA 6006 as follows:

$$K_A = \frac{T_{eq}}{T_n}\,,\; T_{eq} = \left(\frac{\sum_i n_i * T_i^p}{\sum_i n_i} \right)^{1/p}$$

p: Wöhler Line Slope, n_i: Load Change Stage i, T_i: Torque in Stage i

For a gear layout, the equivalent torque calculation with the derived K_A, does not consider any horizontal part of the Wöhler line, being thus conservative. A finer procedure is for instance described in [2].
Since the Wöhler line slope p changes for different materials, type of treatment as well as for root and flank, separate K_A values (one for each p) must be determined. This is not very practical. That is why one should not use an application factor for the verification but a damage accumulation method (i. e. in accordance with DIN 3990, Part 6) for fatigue analysis.

Load Distribution Factor

Due to manufacturing tolerances and deformations, the load distribution between the load paths (Planets) is not equally balanced. Therefore, for the calculation of planetary sets the load has to be multiplied by a K_γ factor. The data values for WT strongly differ according to each source, see Fig. 2-7, bellow. In order to improve the load distribution between the planet gears it is possible to equip independent planetary set elements (sun, ring, and planet) with elastic or floating bearings, for instance, floating sun shaft or elastic ring bearing. Another possibility is to use a flexible planet bearing (Flexpin, see [1]) as an elastic element. The Flexpin allows a planet radial or peripheral alignment. The effect of this flexible planet bearing can be seen in the chart bellow as a difference between the lines "MAAG without Flexpin" and MAAG with Flexpin" which are based partly on measurements, partly on experience. The $K\gamma$ values proposed by MAAG will be included in the new AGMA 6123.

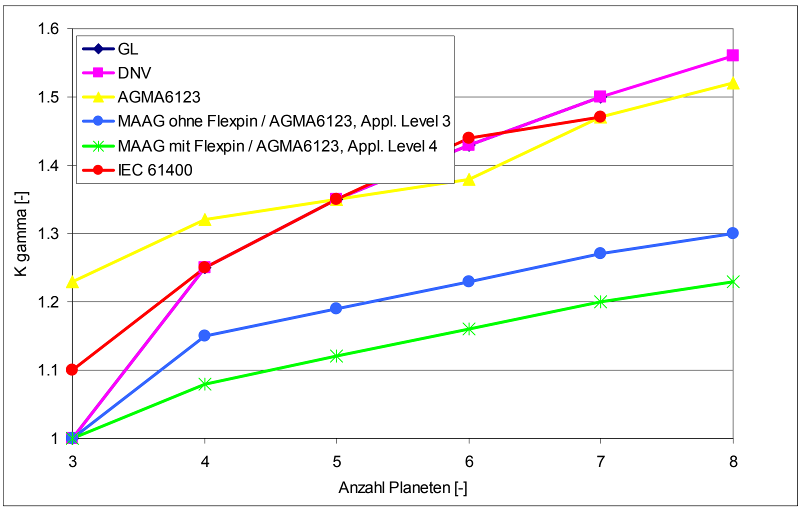


Fig. 2-7 Load Distribution Factor in Function of the Number of Planets

Measurements show that $K\gamma$ values sink with increasing loads because the manufacturing tolerances in comparison with the load-dependent deformations loose importance. However, they are costly and only allow conclusions for the product $Kv*K\gamma$. The approximate values given above should only be taken as top limits. Because of this, since one can reach low $K\gamma$ values, solutions with more than three planets are anyway economical. For instance, with five planets and a measured $K\gamma=1.12$, 95 % of the torque can be transmitted as well as for seven equally wide planets and a $K\gamma=1.50$ (conservative assumption / regulations). This, again, allows higher stage ratios or offers more room for the layout of the planet carrier.

Face Load Factor

Face Load Factor K_β describes the distribution of the peripheral load over the tooth width and its effect upon the flank- ($K_{H\beta}$) and root load ($K_{F\beta}$). Further to the simplified calculations as described in the Norms, the directives for WT Gearboxes ask for a detailed numerical calculation of the load distribution when a required minimal value ($K_\beta \geq 1.15$) should be reduced for the rating. For that, there are several calculation programs such as LVR, Rikor, Planlorr or LDP available.

There is a way of achieving a uniform load on the tooth width with a toothing correction. Generally, the face load factor for thinner toothing is better than for a wider one. We mention here again the Flexpin because it prevents planet tilting under load (through K_β directly reduced) and, by means of a favourable $K\gamma$ value, allows the use of five or seven thinner planets (with a lower K_β). These self-centering systems also allow using a still lower $K_\beta<1.15$ in the calculation.

K_β values greater than 1.30 lead to unacceptable contact patterns that in practice (in test run-up) also must be recognized. That is, that realized gearboxs show a K_β between 1.10 and 1.30. The face load factor is in practice (especially under load) in most cases lower than the calculated results determine.

Transverse Factor

The transverse factor K_α considers the load increase on the flank ($K_{H\alpha}$) and on the root ($K_{F\alpha}$) due to pitch errors and irregular load distribution over several teeth in meshing. The calculation is carried out in accordance to ISO 6336. The toothing quality required for WT Gearboxs (e. g., according to GL directives, minimum quality 6 for external toothing) allows the use of $K_\alpha=1$. Since the face load factor K_β has a strong influence on the stress behaviour, an alternative 3D contact analysis is required for the precise calculation of the product K_β* K_α resulting in a combined factor $K_{\alpha\beta}$.

Dynamic Factor

The dynamic facto Kv considers the load increase on the flank (K_{Hv}) and (K_{Fv}) on the root because of toothing rigidity variations during meshing. Should the frequency of the rigidity variations / number of revolutions be directly comparable to the meshing inherent frequency, so will appear dynamic additional stresses. The calculation is carried out in accordance with ISO 6336, Method B and differentiates between three speed areas (sub critical: N>>1, critical: N~1, overcritical: N>>1) that are defined by the reference speed N (pinion speed n_1 compared to the resonance speed n_{E1}) where N considers the reduced toothing mass m_{red} and the toothing rigidity c_γ.

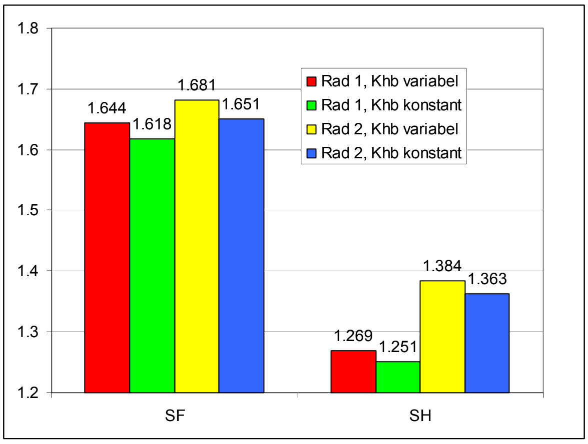
$$N = \frac{n_1}{n_{E1}} = 2\pi n_1 z_1 \sqrt{\frac{m_{red}}{c_\gamma}} \quad , \qquad K_v = f(N)$$

Due to the low speed, WT will be operated in the sub critical zone and the Kv values will be correspondingly set. In the WT Certification Directives, as well as in the appropriate Norms, a value of Kv =1.05 or higher is requested. Should a lower value be used, measurements or a detailled calculation is required.

2.4 Calculation with Load Spectra

The toothing verification calculation is carried out with Load Spectra. These will be calculated determined from the wind speed charts for several wind mean speeds by means of dynamic simulation or from measurement time charts for torques and speeds at the rotor shaft. These time charts will then be classified by speed and torque in a load duration distribution spectra. We have to stress that, the speed practically does not vary and that it enters the service life calculation in a linear way. The classification should thus orient itself in direction of the torque chart.

There is a question whether K factors, especially K_β, should remain fixed for every load stage (for instance as requested in [2] / [3]) or be separately calculated for each load stage (according to [8]). The following comparison (see also [11]) shows that, however, the calculated toothing safety factors only slightly differ from each other when instead of a constant $K_{H\beta}$=1.26 (calculated for the nominal load) they are separately calculated for each load stage.



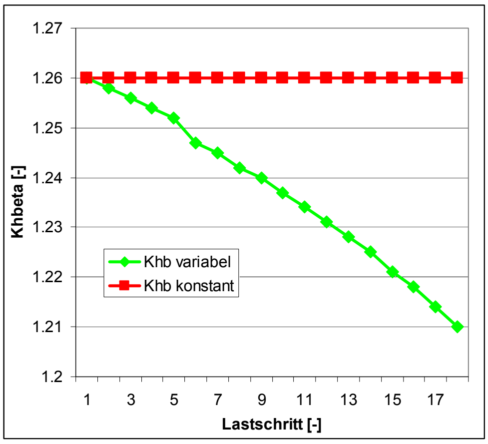


Fig. 2-8 Comparison between Calculations with variable and constant K_β.
End Stage of a 3.6 MW WTI, L=20.000 h, ISO 6336, B.

Frequently, it can be observed that only a few steps in the load spectra generate the majority of partial damage. It is then meaningful to further subdivide and identify these relevant damaging load levels. Often, it is here the question of the influence of load levels with very high loads (close to the static strength) but with a low cycle numbers (low cycle fatigue). DIN and ISO also likely underestimate the influence of load bins with high cycle umbers but lower loads (high cycle fatigue). While using DIN 3990 and ISO 6336, it is thus meaningful, to modify the Wöhler line with the Haibach modification (no fatigue limit zone, see Line b) in the left part of the figure bellow) and recalculate in order to determine a lower conservative limit.

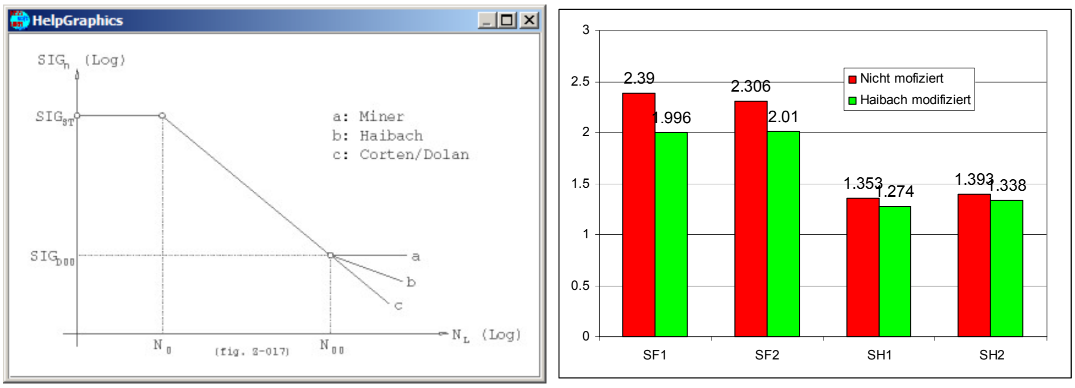


Fig. 2-9 Haibach Modification Influence in the calculated Safety Factors; not "modified" according to Line a); "Haibach modified" according to Line b)

3 Summary

3.1 Investment in Fundamentals

In view of the background of the often-occurring damages in WT Gearboxes, it is of vital interest for all participants to maintain a qualified and balanced dialogue. An aspect of this dialogue is the calculation of the verification of the toothing strength that was shown in this article. Here, it is necessary to invest in education and training, tools and procedures in order to balance the information gradient between gearbox manufacturers and customers.

3.2 Calculation Procedure Safety

The administration of many individual calculations, the use of different tools und the necessity to use identical data in the engineering analysis, drawing an fabrication leads to a highly tiresome task in the management of data. The target should thus be to manage with the appropriate tools all WT necessary gearbox calculations (toothing, bearings, shafts, joints) integrating them in a comprehensive model. This will also allow for the fast verification of the complete gearbox for changed input data, e. g. customer-specific load spectra as well as back tracing of field experience according to the spirit of Know-how management. The integration of all engineering analysis data into a single dataset can also be achieved with respective software.

3.3 Questions on the appropriate Gearbox Concept

At present, there is no comparison of the gearbox concepts mentioned above concerning calculation verification, construction details, production, costs and operational behaviour available and they must be internally prepared inside the Firm. They will constitute the basis for a Risk Estimation before Investment Decisions. For the buyer of gearboxs, it must be rather interesting to have a supplier-independent standardized methodology, with the corresponding tools, allowing him to compare in a fast and safe way different gearbox concepts or gearboxs from different suppliers.

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