Design and Optimization of Planetary Gears Considering All Relevant Influences

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Gear Design Process

Light-weight construction and consideration of available resources result in gearbox designs with high load capacity and power density. At the same time, expectations for gear reliability are high. Additionally, there is a diversity of planetary gears for different applications. Gears with one or more stages and with one or more gearbox inputs and outputs are not uncommon. Furthermore, different kinds of teeth exist: e.g., spur and helical gears, and also double-helical gears are doable. For the mounting of shafts and gearings, roller bearings and sliding bearing are used (Fig. 1).

All of these conditions require exceptional and robust design criteria, including maximum load and dynamic loads under different load situations. Experience with drivetrains with stiff foundations and constant, external loads is not directly applicable, due to unique boundary conditions, dynamic excitation of the structure, and changing influences by external conditions (Ref. 12).

The product design process of a gear typically begins with the load calculation, followed by gear and component layout, to the point of structure analysis (Fig. 2).

Only at a test bench, or in industrial use as a component in the whole drivetrain, can the quasi-static and dynamic behavior of the gear in actual conditions be verified. This long chain in the process does not allow for an efficient gear calculation — especially considering the insecurities of the load assumptions — and with that the inevitable, inaccurate stress of the single machine elements and resulting strains.

In these cases the highly precise and, in part, standardized calculations of machine elements can only be applicable as far as the accuracy of the load assumptions allow. Any interactions of the single elements within the stressed gear (e.g., the influence of axle bending on the load dispersion of the gearing) are thereby lost. Furthermore, the gear must —especially with flexible shafts, housing or dynamic excitation — be understood as a sub-system of the drivetrain; only in this way can a realistic load gradient be constructed (Ref. 13).

An evenly balanced calculation model for drivetrains that connects all concerned sub-disciplines (external conditions, drivetrain dynamics, structure dynamics, electrical phenomena and machine regulation) in a comparative model depth does not exist (Fig. 3). And yet, only such a balanced model allowing for all needed conditions can deliver the realistic and reliable statements on dynamic strains needed to make the safe design of drive components possible (Ref. 15).

The resulting problems and damages cannot be fully explained through mere analysis of the single modules. In fact, the essential influences of the surrounding system components must be accounted for and included in the computation. Here arises the

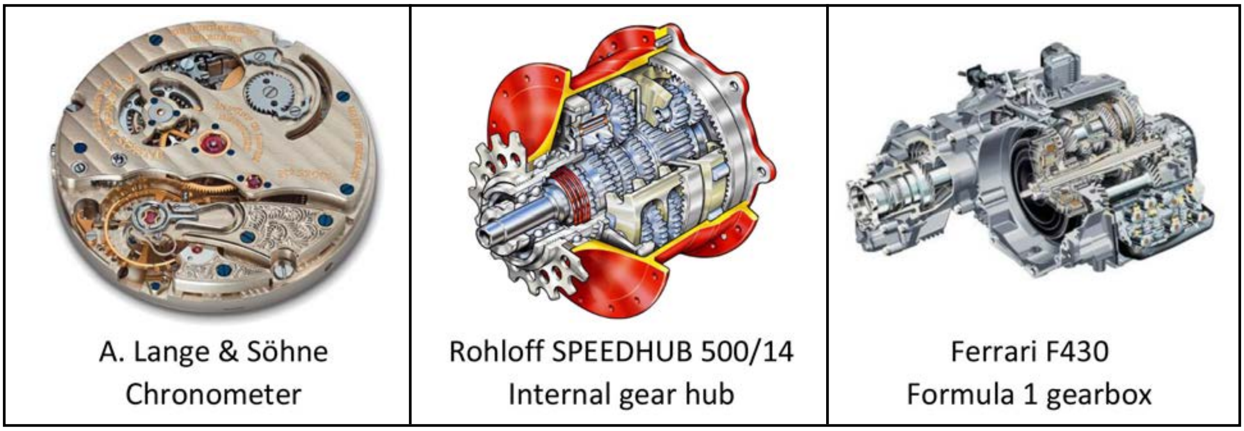

Figure 1 Application of planetary gears (Ref. 13).

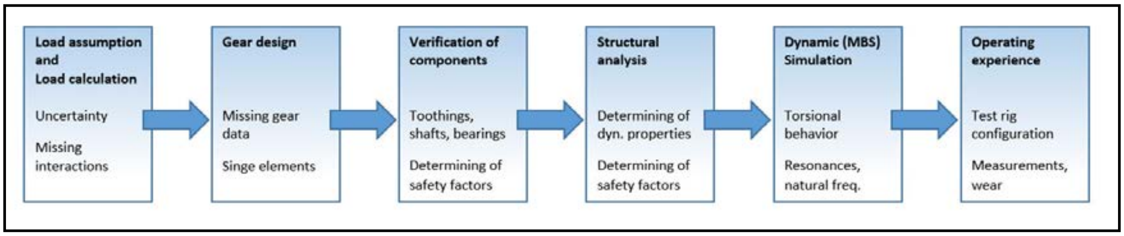

Figure 2 Classic product design process.

real difficulty of finding the necessary system parameters to solve the respective question, which is why the product development process of the future is moving more and more to system analysis, rather than the design of single machine elements. Vital to gear development is continuous — mostly software-supported — analysis, result-conditioning and data maintenance to the point of supervision of the lifecycle of a gear. On one side, all calculations of the machine elements — gear, axle, bearing, axle-hub connection, screw connection, etc. — are to be implemented following the current standards. These must be supplemented through detailed examination of load gradients and load distribution — and to the point of optimization of single target parameters (mass, stiffness).

Gearbox Development and Calculation According to Standards

Especially for design concepts of planetary and spur gearboxes, the newest software development of DriveConcepts GmbH — *MDESIGNgearbox* — is established. This calculation software gives complete product information in the early phase of the product lifecycle (PLC). The calculation cannot replace measurements and test drives, but iteration steps can be reduced economically. The software allows for an intuitive and easy handling in the design process of the entire gearbox — dimensioning of the machine elements (shafts, bearings and toothings) — all according to the existing standards (Refs. 5–7).

For toothing:
- DIN 3990:1987 T1–T6
- ISO 6336:2008 T1–T3, T5 and Technical Corrigendum 1:2008

Future work for toothings:
- Micropitting according to ISO/TR 15144–1
- Scuffing according to ISO/TR 13989 1 and 2, AGMA 925
- Gear mesh efficiency/loss factor H_V and H_{VL}

The shafts of the gearbox are calculated according to:
- DIN 743:2008 T1–T4 and Beiblatt 1, 2
- Different calculations possible for the roller bearings:
- Lifetime L_{H10} according to DIN ISO 281:2009
- Modified lifetime according to DIN ISO 281:2009, Beiblatt 1, 3
- Advanced modified lifetime according to DIN ISO 281:2009, Beiblatt 1, 3
- Lifetime according to ISO/TR 16281:2009

The software enables calculation of the system gearbox in one step, including a complete documentation into a PDF/A document, according to ISO 19005–1:2005 (Fig. 4).

Gear Optimization (Macrogeometry)

The following shows the gear optimization in some case studies:

Load distribution. Next to the load distribution factor $K_{H\beta}$ one of the important tasks of gear development is to optimize

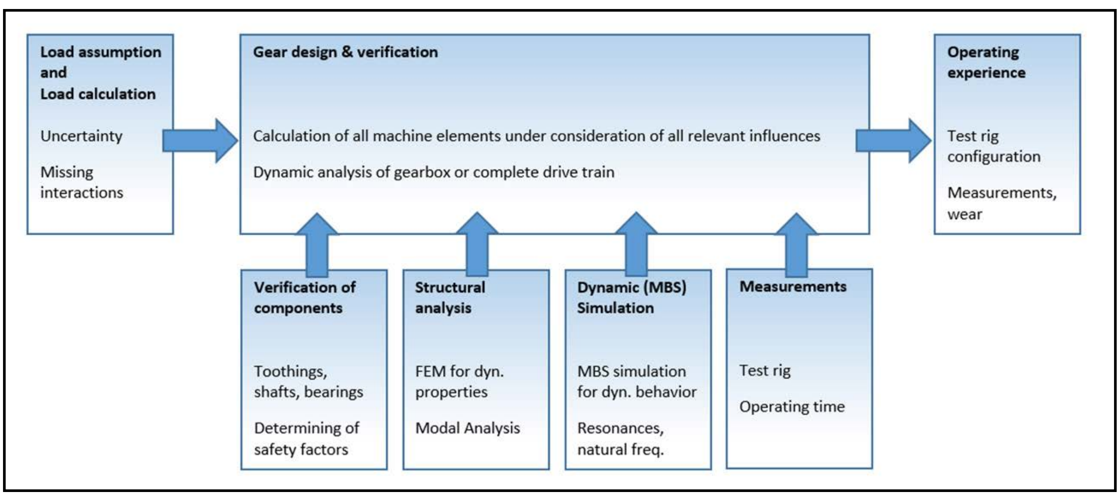

Figure 3 Design process of a gear as a system.

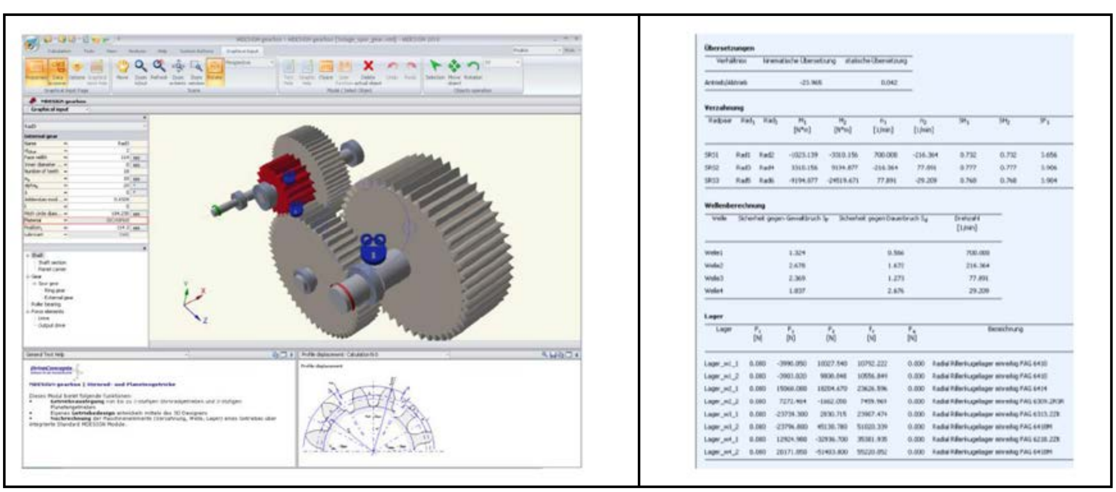

Figure 4 User interface of *MDESIGN* gearbox with *3-D-GearDesigner* and result page (Ref. 10).

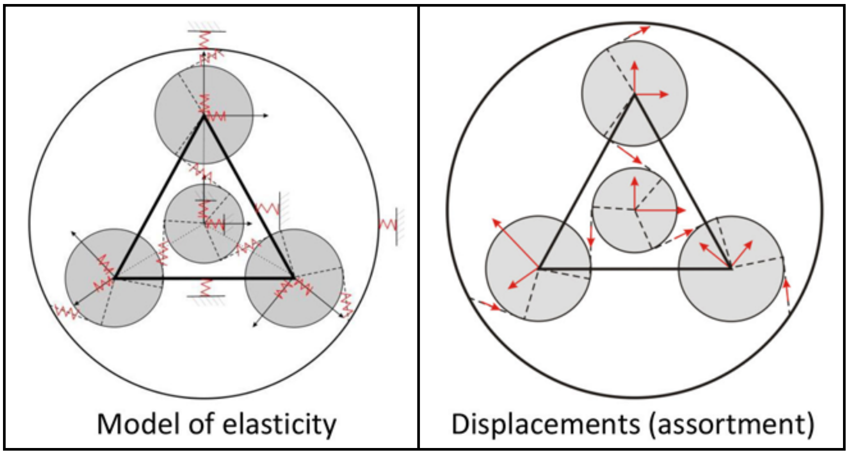

Figure 5 Computation model for load distribution $K_{H\beta}$.

the load distribution of each planet gear. This is done using a pure statistic model that determines load distribution factor *K*. The load distribution factor is defined as "the ratio of the maximum tooth normal force to the median tooth normal force at the speed of zero." Dynamic factors are represented by the factor K_{γ}. The median contact stiffness from the load gradient calculation is used for the analysis, as well as wheel body stiffness (sun, ring gear), bearing stiffness and bearing clearances (sun, planet, ring gear and planet carrier). The following deviations can be accommodated (Fig. 5):

- Single-pitch deviation — sun and ring gear
- Tooth width variations — planet gear
- Center distance deviation and planet carrier pitch deviations
- Displacement — sun, planet carrier, ring gear

The computation of the load distribution allows statements on suitable tolerances or tolerable location variations with exact knowledge of the real load for every single planet. These investigations allow, for example, single parameters to be analyzed with regard to their influence on the load-bearing capacity of the gearing (Fig. 6).

Suitable construction parameters, as well as sensible tolerances of gearings and location variations, can be defined. Research on load distribution (K_{γ} and $K_{H\beta}$) has shown that only a simultaneous optimization of load distribution on flank ($K_{H\beta}$) and planets (K_{γ}) results in an optimal gear (Fig. 7). An effective instrument for a balanced load distribution is the use of optimized, flexible planet gear bearings. The impact is due to the targeted overlapping of bolt and bushing bending, with the goal of minimizing the tilt angle of the planet, which in turn is determined by the deformation of the bushing (Ref. 16).

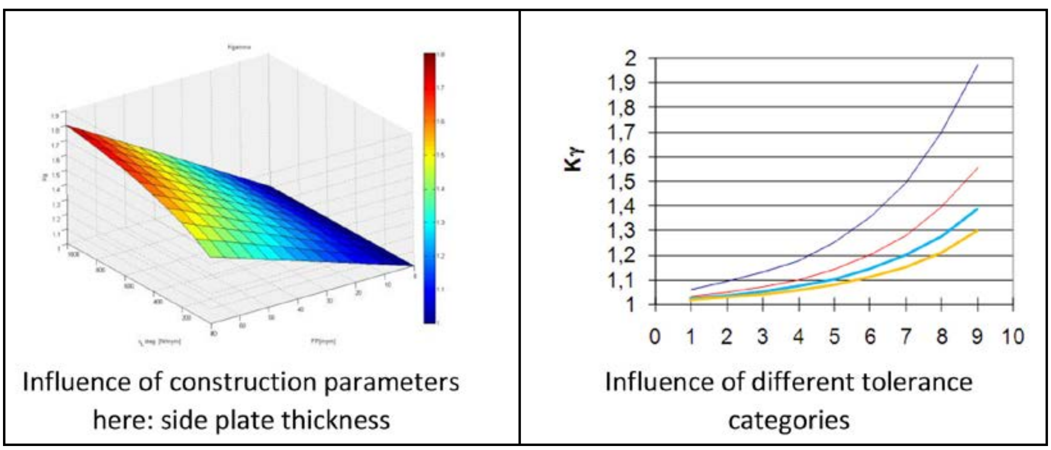

Figure 6 Variation studies for load distribution $K_{H\beta}$

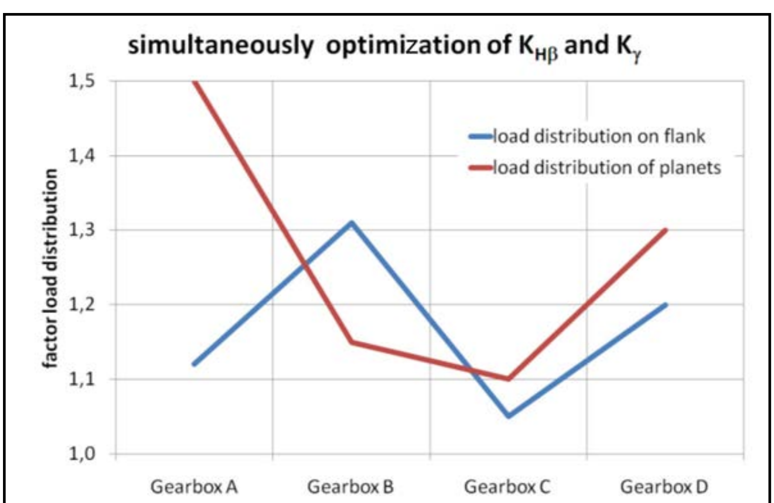

Figure 7 Variation studies for load distribution $K_{H\beta}$ and $K_{H\beta}$.

Stiffness optimization. The optimization of construction parameters with the goal of optimal stiffness of all relevant gear elements is probably one of the most complex development tasks in the design process. Typical is the description of the following variation analysis of a planet mount: "the goal is a design with the least possible mass while retaining necessary stiffness requirements needed in view of the load gradient (Fig. 12)." Both one-sided and two-sided samples can be considered; they can be constructed with a round or optimized outline (triangular, square) (Fig.8).

The geometric parameters to be varied in such a study are shown (Fig. 9). Through the large amount of parameters it is necessary to use software programs

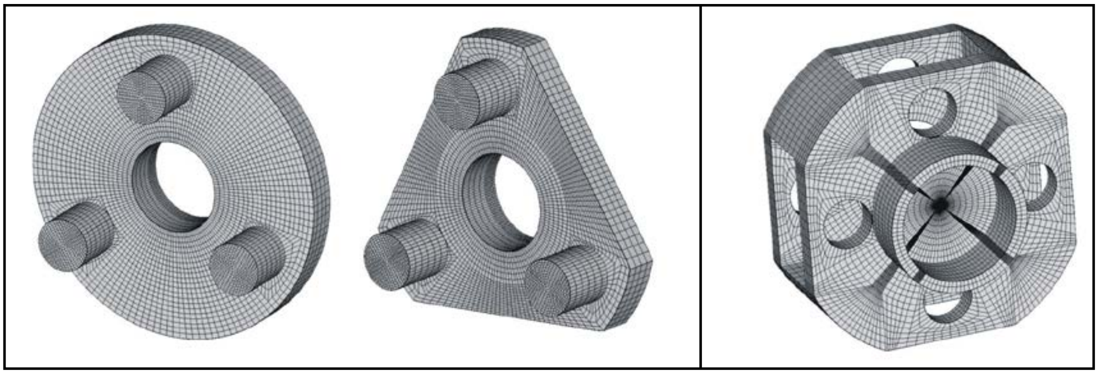

Figure 8 Variants of planet carriers: single-plate (left); double-plate (right) (Ref. 3).

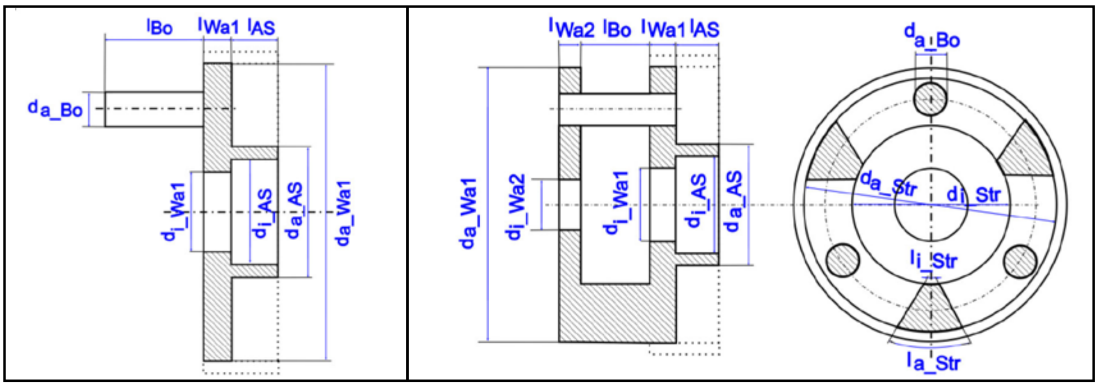

Figure 9 Geometric parameters of planet carriers: single-plate (left); double-plate (right).

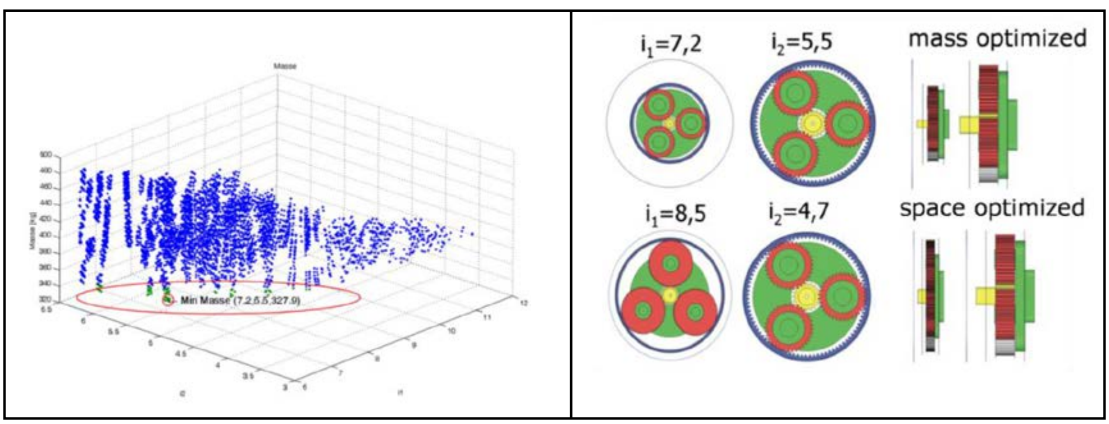

Figure 10 Optimization mass and design space.

with integrated FE solvers for calculating stiffness parameters; only in this way can optimal configurations be found for the entire parameter area.

Mass/design space optimization. Not only has the just-introduced stiffness optimization led the engineer to a number of detail problems; the search for a mass, construction-size optimized gear is a highly complex question due to the number of overlapping influences. Figure 10 shows the field of results of a variation study for a constant, given total gear ratio and defined load.

The investigation can be used for improving present gear solutions, as well as for new designs. Using an existing design as an example, the following shows how great the potential can be (Fig. 11).

At similar dimensions for the ring gear outer diameter d_3 of gear Stage 2, one arrives at a mass savings by adjusting the ring gear diameter for Stage 1 and reducing the tooth width. *MDESIGNgearbox* avoids the over-dimensioning of planet gears by pre-setting safety factors for the gearbox machine elements. The mass of the original is at $m_{ges} \approx 2{,}200$ kg. All generated, optimized solutions arrive at a mass reduction in comparison to the actual gear. The mass, optimized preferred variation is shown (Fig. 12, right).

In this example the mass savings amount to about 25 percent, in respect to the original design. At the same time the optimization of the construction space amounts to 15 percent (Fig. 3). In a second step the consideration of CAD geometry data of housings will be possible. Therefore the software imports a standard geometry format, generates finite element models, calculates stiffness matrices for the housing, and delivers this information to the design process of *MDESIGgearbox* (Fig. 14).

Optimization of microgeometry. The calculation of load distribution in a planetary gear system essentially depends on the helix angle deviation between the contact flanks of the gear

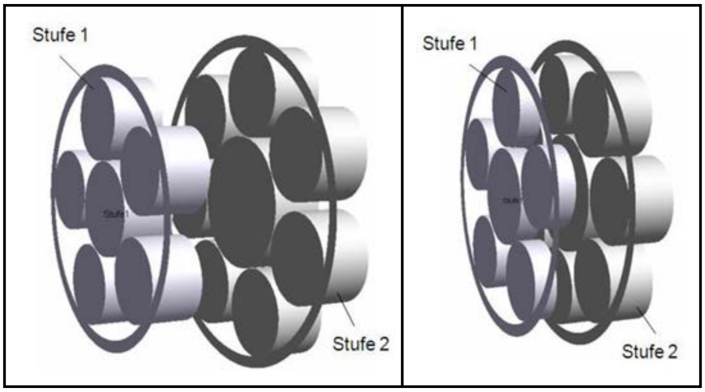

Figure 11 Variation study mass optimization: initial state (left); mass optimized gear (right).

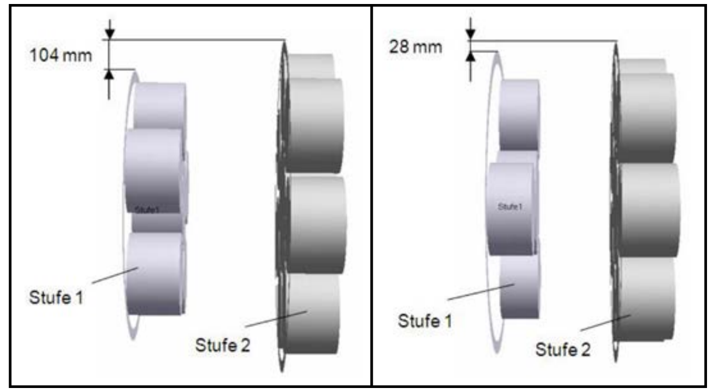

Figure 12 Variation study design space: Initial state (left); space-optimized gear (right).

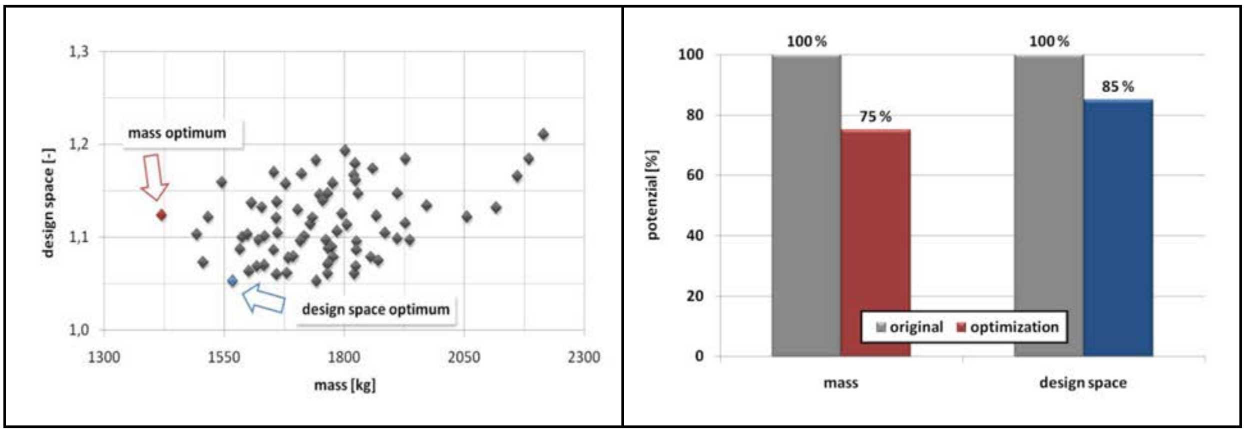

Figure 13 Savings potential: mass (left); design space (right).

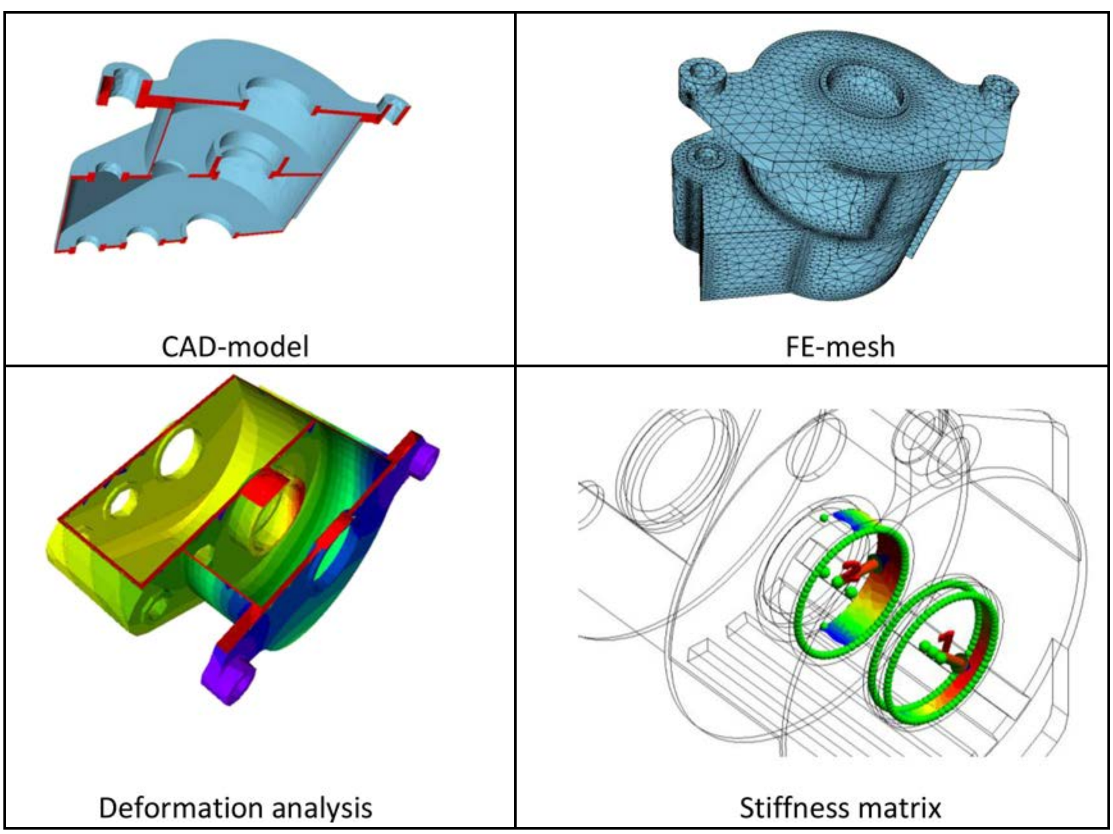

Figure 14 Determining of stiffness matrices in *3-D-HousingDesigner*.

pairs; it can be understood as the sum of different influences. It is assumed that the effects are overlying independently, thus the sum of contact line deviation can be calculated with the single deviations (Ref. 11).

The calculation of single displacements and deformations of all gearbox bodies — especially the planet carrier, the coupling of ring gear and gear wheel bodies and the deformation of teeth — is more complex in planetary gearboxes than in spur gearboxes. To determine the load distribution, the flank deviation for the tooth contact sun/planet and the tooth contact planet/ring gear is calculated by the new software *MDESIGNLVR*^{planet} (Refs. 8 and 9). The result of the calculation is the excessive line load, which is expressed by the factor $K_{H\beta}$. In general, the excessive line load is on the flank side opposite the deviated flank side.

Next to the calculation of the ratio of maximum and middle line load, the software gives detailed information about tooth flank pressure and tooth root stress distribution (Fig. 15).

The flank deviation (FLKM) consists of the following parts:

- Elastic deformation of gear body (ve_{RK})
- Elastic tilting difference of roller bearings /17/ (ve_{WL})
- Torsion deformation of planet carrier (ve_{PT})
- Tilting of planet due to of sliding bearing ($verkipp_{PL}$)
- Effective helix angle modification ($f_{H\beta eff}$)
- Elastic deformation of tooth flank
- Elastic deformation difference of planet carrier bearing
- Deformation of housing

The helix angle deviation for tooth contact sun/planet is calculated by the following equation:

$$FLKM_{1/2} = ve_1 + ve_{2_{1/2}} + ve_{WL1/2} + ve_{PT1/2}\ verkipp_{PL1/2} + f_{H\beta eff1/2} \quad (1)$$

The helix angle deviation for tooth contact planet/ring gear is calculated by the following equation:

$$FLKM_{2/3} = ve_1 + ve_{2_{2/3}} + ve_{WL2/3} + ve_{PT2/3}\ verkipp_{PL2/3} + f_{H\beta eff2/3} \quad (2)$$

ve_1 = deformation difference of sun
ve_2 = deformation difference of planet
ve_3 = deformation difference of ring gear

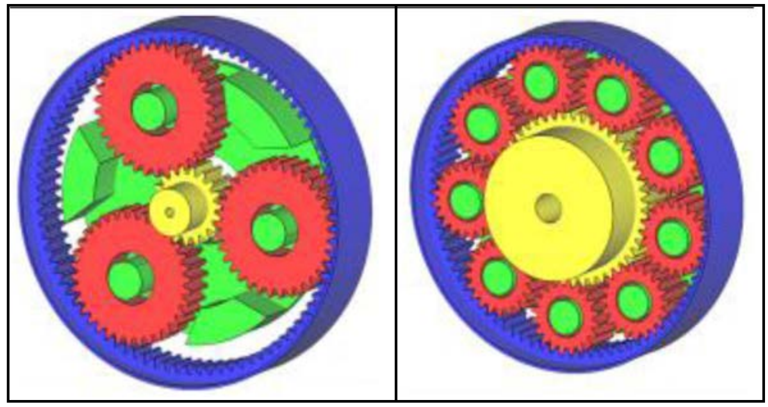

Figure 15 Verification of planetary gear stages.

The deformation is calculated by FE method and is then added to the flank deviation. All parts of the helix angle deviation have to be added as values normal to the flank. The database of the calculation is saved in *XML* format. With this, a structured depositing of design, modification, deviation, load and control data is possible. Furthermore, the program has a project management capability for saving projects, standard examples and more calculation guidelines (Ref. 14).

After input of all necessary parameters: all data are checked, the design models are generated and the FE models for the gears with coupling design and the planet carrier are created. For an efficient calculation it is necessary and reasonable to use drive technology software. DriveConcepts GmbH develops software solutions for drive technology, which is characterized by clear and intuitive handling of all data. In the background, academic-established calculation kernels and consistent, structured interfaces help solve the actual task efficiently.

Case Study

The example of a wind turbine with 2,000 kW output power should show the consequences of different flank modifications with constant load (Ref. 12). The main gearbox consists of one planetary gear stage and two spur gear stages (helical gearing). The detailed parameters of the first planetary gear stage are listed (Fig. 16). The initial state of unmodified gearing under

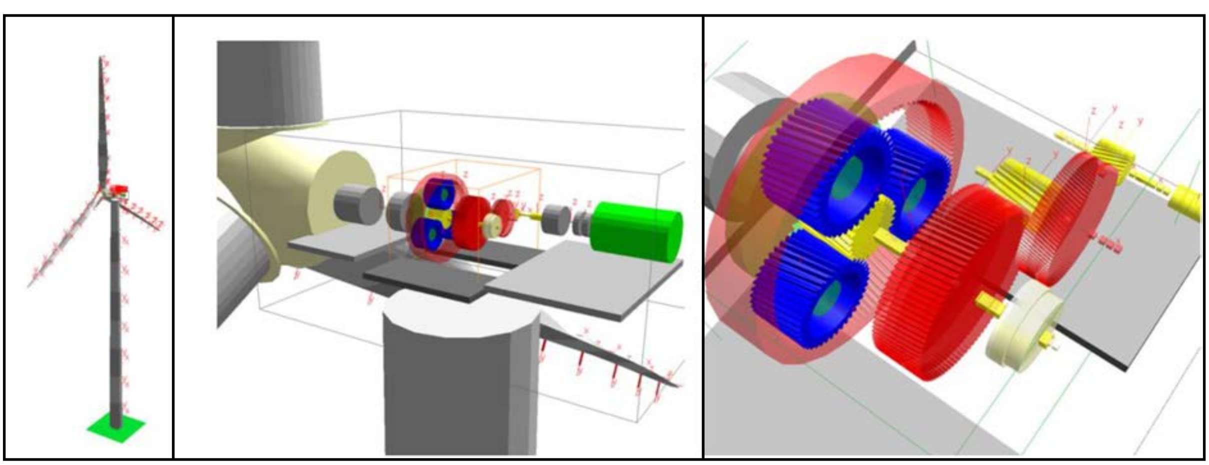

Figure 16 Application case.

module	m	16 mm	face width	$b_{1\|2\|3}$	310 mm
number of teeth	$z_{1\|2\|3}$	20 | 36 | -91	add. modification sun	x_1	0.4
center of distance	a	463 mm	add. modification planet	x_2	0.3156
pressure angle	α	20°	add. modification ring gear	x_3	-1,6429
helix angle	β	8°			

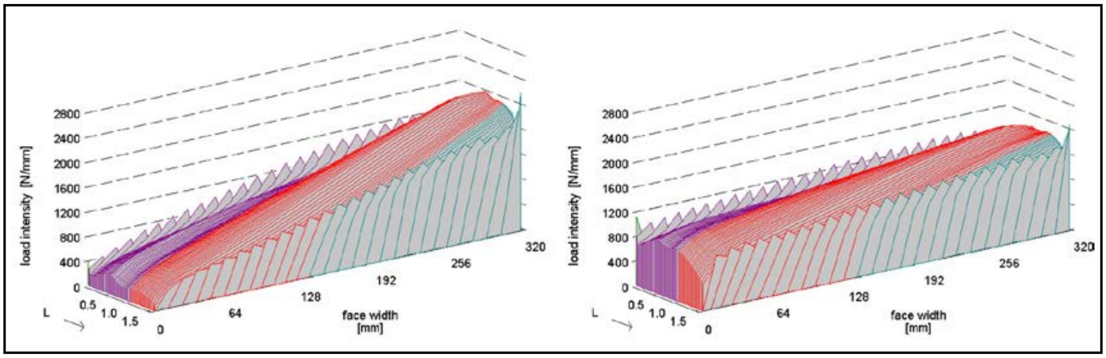

Figure 17 **Initial state (left side); first optimization (right side).**

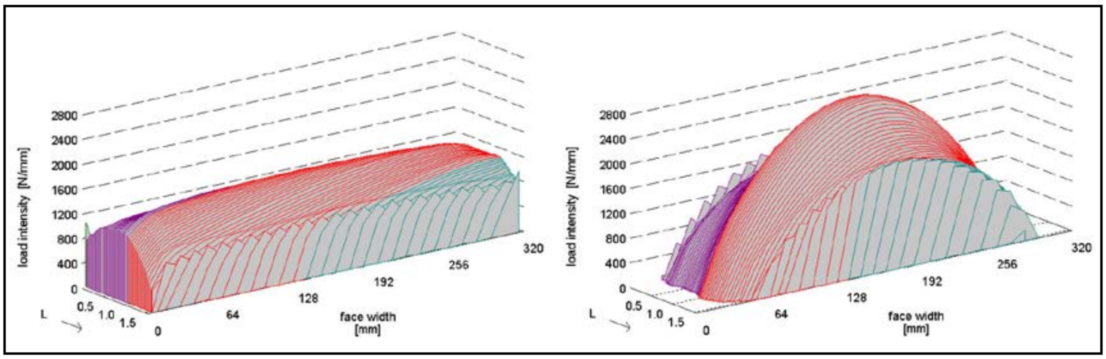

Figure 18 **Final design (left side); bad solution with too much lead crowning (right side).**

nominal load is shown (Fig. 17, left side). In this case the ratio of maximum and mean value of line load is 1.67.

In the first step of optimization with a helix flank modification, the factor can be reduced to $K_{H\beta} = 1.23$ (Fig. 17).

The rest of the unbalanced distribution along the face width — which comes from planet carrier torsion deformation — can be offset with an optimal lead crowning. The ratio of maximum and middle-line load can be reduced to $K_{H\beta} = 1.16$ (Fig. 18, left).

At the right side of Figure 18 it is shown that an oversized lead crowning can also lead to poor conditions. In this case the lead load distribution changes to $K_{H\beta} = 1.98$. The example shows the necessity of the right dimension of macrogeometry and also of used modifications. If these are correct, the lead load distribution $K_{H\beta}$ can be reduced from 1.67 to 1.16; but with unfavorable modifications, the opposite will be the result.

This case study shows advantages of *MDESIGN 2010* with the libraries *LVR, LVR*^{planet} and *gearbox* to develop gearboxes in a very efficiency way.

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