Weight Optimization of a Gear Wheel Considering the Manufacturing Process and Cyclic Symmetry

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Abstract

In vehicle design minimizing the weight of a component is doubly valuable. Beside the saved material the lighter design helps to reduce fuel consumption. Today, it is standard practice to use a combination of Finite Element Analysis (FEA) and numerical optimization methods to develop lightweight components. There are different methods to solve the optimization tasks. Topology optimization is used to find a first layout of the component geometry. In the design process this layout can be transformed to a CAD model. With a subsequent shape optimization extreme stress values can be reduced and stress distribution is smoothed.

The paper shows the procedure for lightweight design of a gear wheel for a truck transmisssion in several steps. It also provides short information about a very easy way to specify shape basis vectors in the shape optimization as an example for advanced modelling capabilities. In addition, information about costs and the realised benefits are presented.

Kurzfassung

Die Minimierung des Bauteilgewichts ist im Fahrzeugbau doppelt wertvoll. Neben dem eingesparten Material trägt eine leichtere Konstruktion auch zur Senkung des Kraftstoffverbrauchs bei. Um eine gewichtsreduzierte Konstruktion zu entwickeln, bedient man sich heute oft der Kombination aus Finite-Elemente-Methode (FEM) und rechnergestützten Optimierung. Für die Optimierungsaufgaben stehen mehrere Methoden zur Verfügung. Mit der Topologie-Optimierung wird ein Entwurf für die grobe Form des Bauteils ermittelt. Dieser wird dann konstruktiv in ein CAD-Modell überführt. In der nachgeschalteten Form-Optimierung findet das Feintuning statt. Es werden lokale Spannungsspitzen abgebaut und eine gleichmäßigere Auslastung des Bauteils erreicht.

Dieser Beitrag zeigt das Vorgehen in einzelnen Schritten an dem konkreten Beispiel eines Zahnrades aus einem LKW-Getriebe. Am Beispiel der Definition von Basisvektoren für die Formoptimierung, enthält er auch einen Hinweis auf das deutlich vereinfachte Arbeiten mit den verschiedenen Optimierungsmethoden. Er gibt auch Auskunft über den benötigten Aufwand und den erzielten Gewinn.

1 Introduction

There are a lot of reasons for an economical use of material in vehicle design. At the same time lightweight designs often have to fulfil even higher requirements due to static, dynamic or thermal loads.

Today, FEA is mostly used for simulation and evaluation of design strength. The need to fulfil weight and strength requirements at the same time leads to an optimization task in the simulation process.

One approach is to apply FEA with integrated numerical optimization methods. Therefore, topology optimization to the search for a suitable layout with minimum weight and shape optimization to fulfil the strength requirements are of a special interest.

Because of its comprehensive and sometimes complex data input optimization of parts was mostly subject to specialists. This situation has changed in the past few years. Now, the optimization model is much more easily defined and visually validated using advanced graphic tools. In addition, optimization algorithms were improved a lot and they give good results after just a few iteration loops. Due to these progresses and further increasing requirements from the design of parts, numerical optimization is used more frequently and is going to become a standard design tool.

During topology optimization those finite elements of a previously specified design space are selected which are necessary to achieve among others stiffness and weight targets for given loads and boundary conditions. Therefore, load case dependent design limits have to be specified for result quantities like displacements, stresses, or reaction forces. In addition, restrictions from the manufacturing process can be imposed like release directions from casting processes or symmetry conditions. Then, the optimization results in a new layout of the part geometry. This step is most efficient in an early stage of development. There, the highest possible reduction of weight can be achieved.

After the design engineer has transformed the new layout to a new CAD model, a subsequent parameter or shape optimization is performed in order to improve the mechanical behaviour of the design. The parameter optimization is used to optimize e.g. plate thickness, cross sections of beams or stiffness values of a support. The shape optimization reduces stress peaks and leads to a smooth stress distribution.

In the following the discrete optimization steps are applied on a real design. For confidentiality reasons the presentation of results is made using normalized values.

All computations are made with the FEA software PERMAS from INTES, where the optimization model was generated using VisPER (Visual PERMAS) which was also used to post-process results.

2 Definition of task

The weight of a gear wheel of a truck transmission shall be reduced. Possible savings are assumed at the region between the hub and the gear rim. The gear wheel has to be forged or cast.

At first the current design of the gear wheel is analysed. Subsequently, a FEA-based topology optimization is performed for the loaded gear wheel. Due to revolving load the geometry has to be cyclic symmetric. Undercuts are not allowed by the manufacturing process. The result of this step is a new design draft.

The transformation of the draft into a CAD model is done by a design engineer. With the same loads and boundary conditions as in the topology optimization a shape optimization is performed for the new gear wheel. There, cyclic symmetry must be retained. This step is to decrease stress maxima and to smooth strains.

3 Model description and initial state

The model description is illustrated using the initial design. The following illustrations can be used for all models, because the same loads and boundary conditions are used for all design steps.

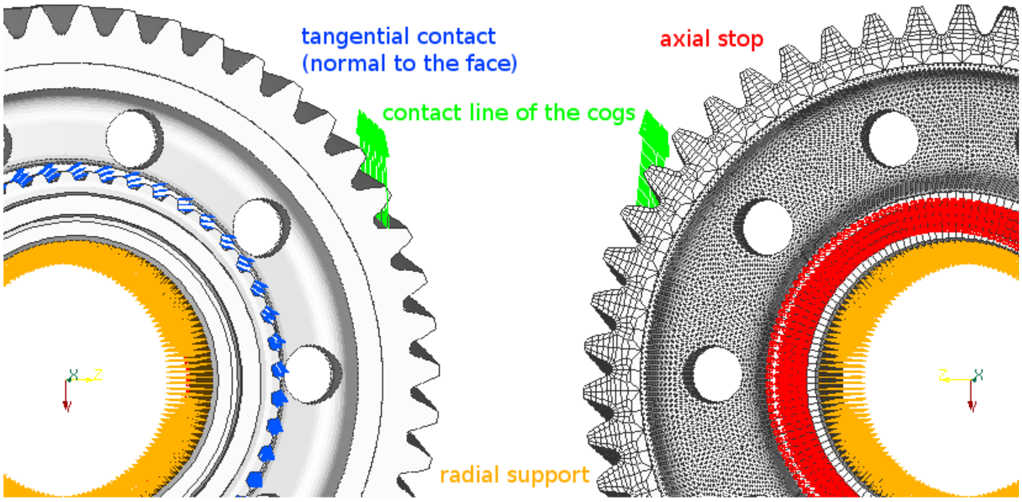

Fig. 1: Model description, supports, and loads (right picture with FE mesh)

The gear wheel is loaded at single cogs of the outer gear rim with forces normal to its face (green arrows). Each load of one cog is specified in a separate load case. Due to cyclic symmetry, it is sufficient to load the cogs of one sector. The needle roller bearing / shaft is replaced by a radial contact definition at the inner surface of the hub (orange arrows). The axial stopper is simulated by a contact definition normal to the face (red arrows). The traction with the sychronizer ring is carried out as contact normal to the tooth flank (blue arrows).

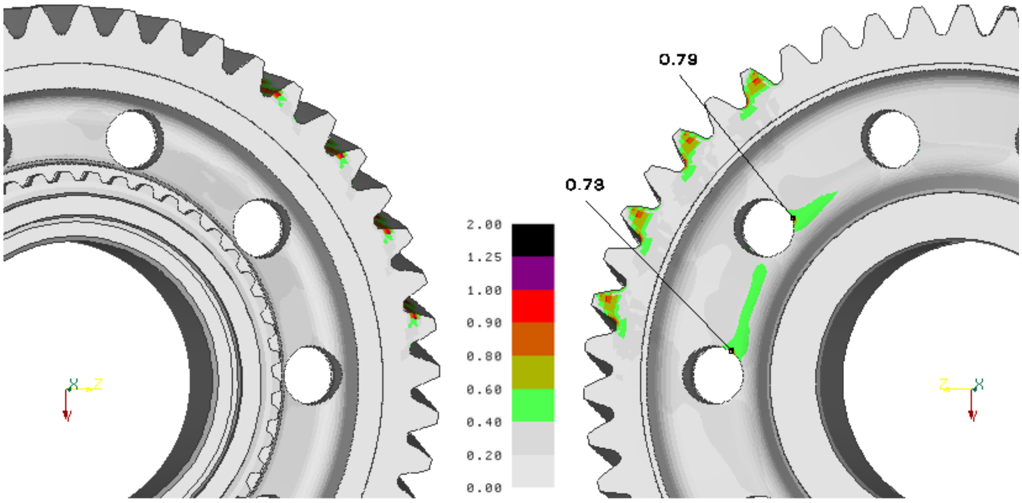

Fig. 2: Initial state: Maximum v. Mises Stress of all load cases (normalized to the limit value).

To evaluate the initial geometry the maximum von Mises stress for all load cases is determined by a static analysis. The values are normalized to the given stress limit. The very local maximum is 79%. But most of the region between hub and rim shows stress values less than 40%. This indicates potential weight savings.

4 Topology Optimization

The first step of a topology optimization is the definition of a design space. It denotes the region where geometrical modifications can be made. Furthermore, design constraints and a design objective has to be specified.

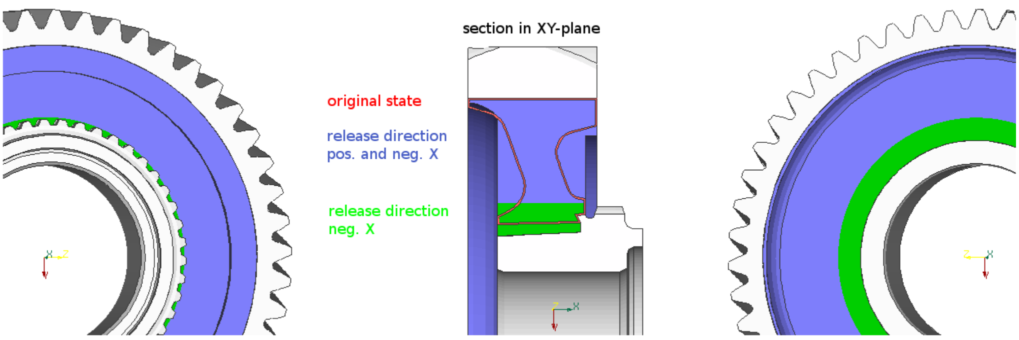

Fig. 3: Model description for topology optimization, design space in blue and green

The design constraints for the topology optimization are:

- maximum von Mises stress at the surface between design space and gear rim
- cyclic symmetry with seven sectors and planar symmetry within each sector
- maximum weight

The highest possible stiffness is used as design objective. So, a new draft with maximum stiffness for all load cases under a given weight is sought.

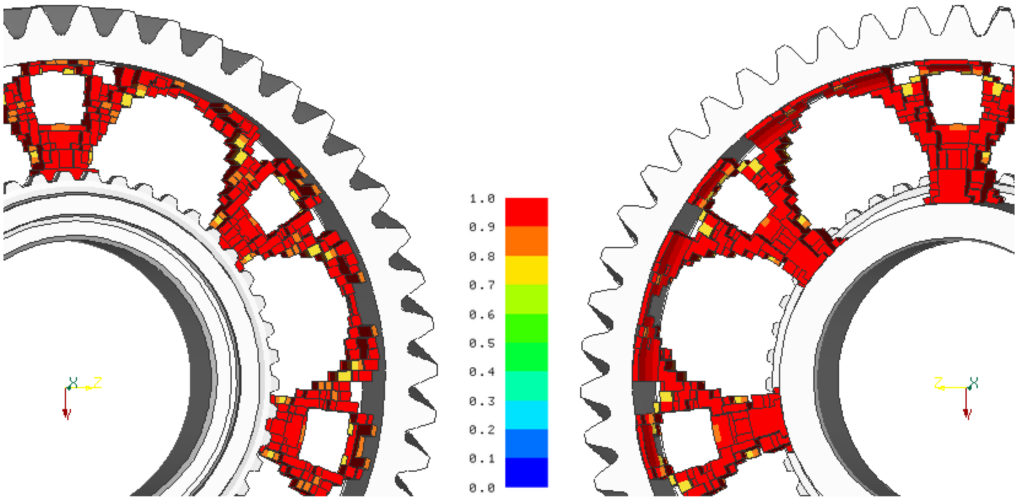

Fig. 4: Filling ratio as primary result of a topology optimization.

The primary result of a topology optimization is the filling ratio. It gives information about the contribution of each element. The filling ratio is equivalent to the computed mass. There is a cubic relation between stiffness and mass. E.g., an element with a filling ratio of 0.5 has 50% of the initial mass and about 13% (0.5*0.5*0.5) of the initial stiffness. The picture (Fig. 4) shows all elements with a filling ratio of 70% or higher.

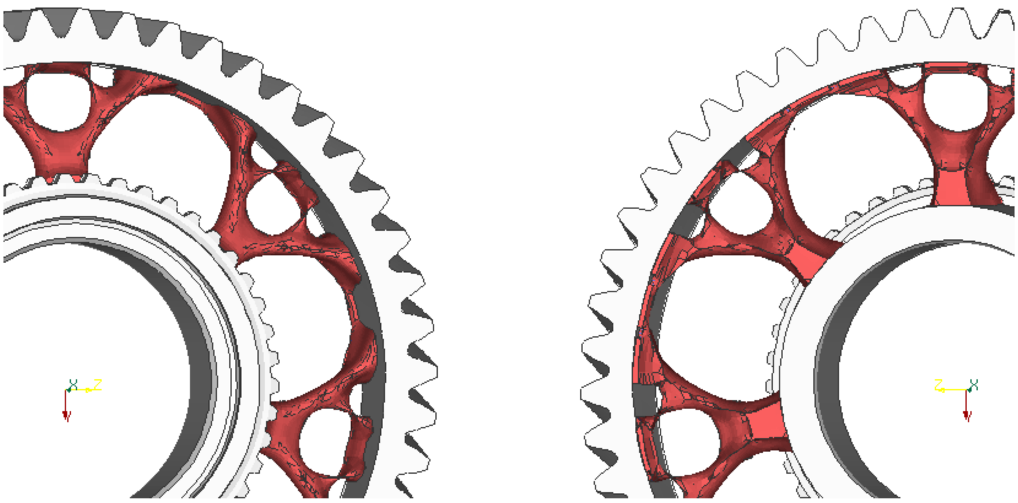

Fig. 5: Smoothed hull of the draft design

During post-processing it is possible to generate a smoothed hull of the design space which is based on a specific the filling ratio. The hull is forwarded to the design engineer.

5 Shape Optimization

After discussing the results of the topology optimization a forged construction was preferred. This decision implies that all holes have to be drilled or stamped. Therefore the holes have to be accessible from both sides of the gear wheel. Holes close to the hub are no longer permitted, because they are not accessible from the side with the small gear rim. The new design geometry is used for the following shape optimization. A static analysis for all load cases is performed and evaluated for this model.

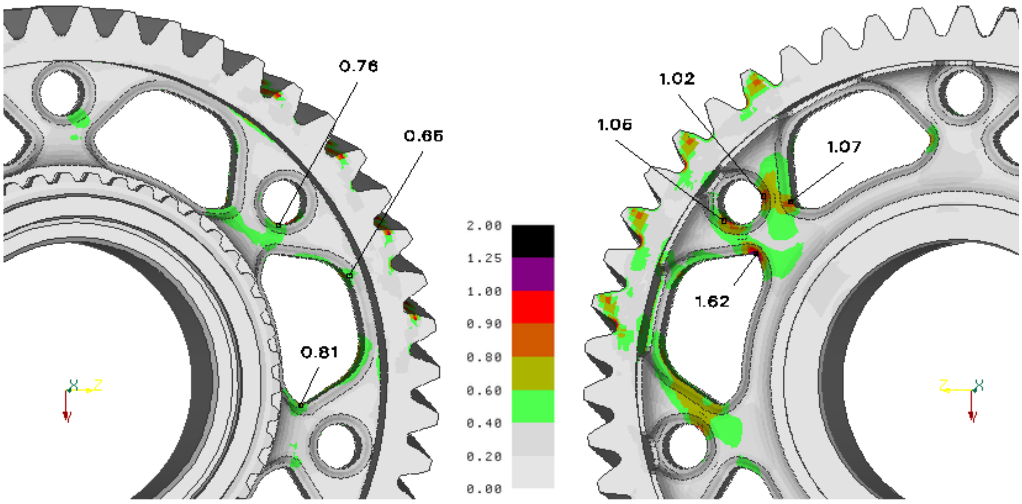

Fig. 6: Maximum von Mises stress of all load cases (normalized to the limit stress value) before shape optimization.

The stress distribution shows a local maximum of 62% beyond the limit at a fillet of the bigger hole. At the smaller hole the stress limit is exceeded by about 5%.

The input of a shape optimization requires the definition of shape basis vectors. The linear superposition of these vectors defines the possible modification of geometry. Therefore the shape basis vectors are a fundamental input. Design constraints include a stress limit for the optimized region and its neighbourhood. The weight is used as design objective.

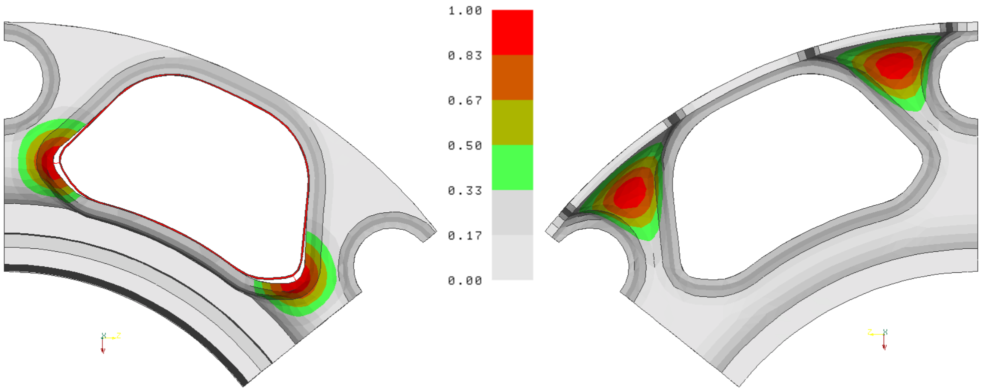

Fig. 7: Examples of shape basis vectors: left- changes of the border; right - modification at a fillet

The definition of the required shape basis vectors has been highly simplified. Today, it is enough to select a few nodal points which can move in a given direction. The easiest and most frequently used way is to define a surface and apply its normal direction. The shape basis vector on the left picture (Fig. 7) is defined with six nodal points at the inner surface of the larger hole, the right one with two nodal points and the surface around them. The program is able to calculate shape basis vectors with this information. They can be controlled graphically before the full analysis is started.

In case of the gear wheel 18 shape basis vectors are defined. They describe possible geometry changes at the surface of the holes as well as their axial position. Others allow modifications of fillets and member thicknesses.

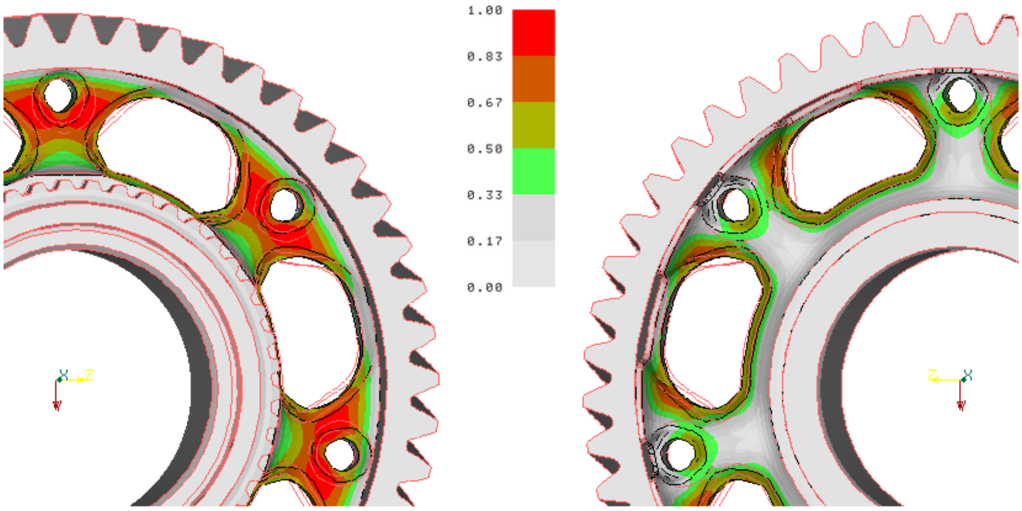

Fig. 8: Geometrical modification normalized to its maximum, display is unmagnified, front and back side.

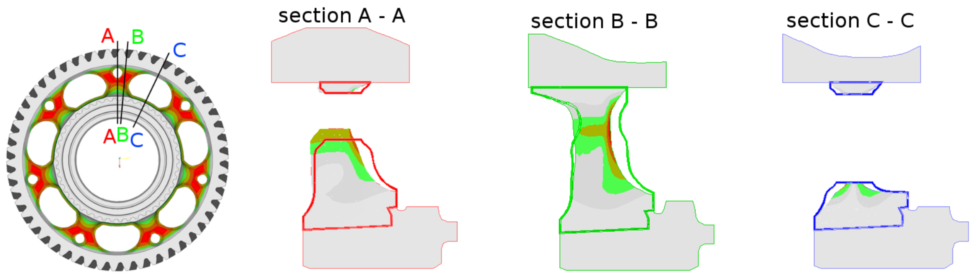

Fig. 9: Geometrical modification normalized to its maximum, display is unmagnified, cutting sections

Fig. 8 and Fig. 9 show the unmagnified shape modifications. The shape optimization leads to a more constant corner at the border of the larger hole. Its area increases slightly. Therefore the area of the small hole decreases. The cutting sections (Fig. 9) illustrate the reduction of member thickness.

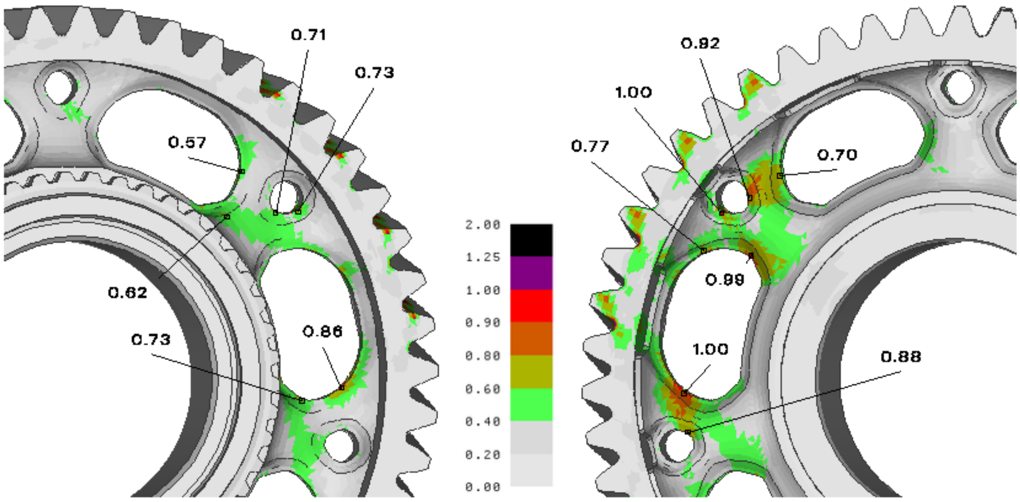

Fig. 10: Modified geometry after the shape optimization: Maximum von Mises stress of all load

For checking the validity a static analysis is performed with the modified geometry. The stresses do not exceed the given limit in any case. The smoother surface of the larger hole reduces the stress peak from 162% to the limit of 100%. The stress field of the optimized region is smoother as well.

6 Costs and Benefits

The development departments have to take care of their budgets and have to justify the use of new design methods. In order to check the effect of using optimization methods for the design of the gear wheel the required working effort is compared with the weight reduction.

	working days	total mass normalized	mass of design space normalized
Initial geometry: - mesh FE model based on CAD data - define loads and constraints - static analysis - evaluation of the results	3	100.0 %	100.0 %
Topology Optimization - mesh design space - estimation for 5/6/7 sections with different design constraints	7	62.5 %	20.0 %
Transformation to a new CAD model	1	78.1%	53.3%
Shape Optimization - mesh the optimized region - define and select useful shape basis vectors - final evaluation of results	5	74.8 %	46.7 %

The maximum weight reduction was achieved during topology optimization. Most of the gain is lost with the decision to forge the gear wheel. Mainly the region close to the hub, which is not accessible from both sides, is responsible for the added mass during transformation to a new CAD model. Although shape optimization has to decrease the stress peak from 162% to the limit, a further reduction of mass was still possible.

The mass reduction is about 25% all over the gear wheel. Considering just the design space, the mass reduction is more than 50%. The effort needed for the complete optimization process is about 16 working days. This is very few, especially if you think of the dimensions of a truck transmission and the number of produced pieces is taken into account. The low time effort is mainly caused by the improved modelling tools, the improved algorithms and the growing experience in numerical optimization.

And, of course, a good cooperation of design engineers and analysts is an essential success factor for such a project.