A holistic approach to lightweight design of multi-component gearwheels

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A holistic approach to lightweight design of multi-component gearwheels

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Abstract. Multi-component gearwheels consist of a gear ring, that carries the teeth and a wheel body, that connects gear ring and shaft. Both parts are joined by a press fit after the gear ring’s heat treatment, what offers the possibility to choose a lightweight design for the wheel body independently from the gear ring. This paper focuses on a wheel body made by fine blanked sheet metal layers, which are stacked on top of each other until the gear ring width is reached. The manufacturing method influences the gearwheels overall behavior, as fine blanked edges are not rectangular. Lightweight design of the wheel body is a main objective. Again, this influences the gearwheel behavior, as the stiffness of the wheel body changes. To meet the lightweight objective while still fulfilling the mechanical needs of a gearwheel, a holistic development approach is used. In every step of that development process, it is checked, if a design proposal matches the mentioned requirements. The experiments show promising results for the further development of multi-component gearwheels.

1. Introduction
Lightweight design is an important driver of innovation in the automotive industry [1]. So far, the car body is the major component affected by lightweight design. Several research initiatives have focused on materials, design and properties of car bodies [2]. There will be further improvements in this area, but the costs for new lightweight activities will increase. The lightweight potential for the drive train and chassis have not yet been investigated in detail. Because they account for around 41% of a cars weight [3], this paper focuses on the challenges in designing lightweight parts for the drive train. A gearwheel is chosen as an example here because of the challenging load distribution. In its inner area, loads are considerably lower than underneath the teeth (Figure 1 left). Current gearwheels in automotive transmissions are often designed as web wheels to reduce weight and to address the issue of uneven load distribution (Figure 1 right). Loads at the tooth root rise with an increase in the torque that the gearwheel transmits. These loads are limited by the materials yield strength, whereas the load in the gearwheels inner area is still within a range that lower steel grades can resist. This accounts for the solid gearwheel as well as for the web wheel. Today, uneven load distribution is not only a mechanical issue, but an economical one as well. Ultra-high strength steel is considerably more expensive than a low-grade steel and therefore the multi-component design for gearwheels is proposed [4, 5]. A gear ring, which carries the teeth and a wheel body that connects the gear ring and shaft form the multi-component gearwheel. Both parts are manufactured separately, which allows to define their properties according to load and cost limits independent of each other.
This contribution presents the design and manufacturing procedure for a wheel body, which is manufactured by fine blanking, stacking and bonding sheet metal (fine blanked wheel body). A multi-component gearwheel with a deep-drawn wheel body was presented in detail in [4].

2. Multi-component gearwheel

In this case, a multi-component gearwheel consists of two main parts, the gear ring and the wheel body (Figure 2). Both parts are joined by press fit directly after the gear ring’s heat treatment [6]. The main reason for dividing a solid gearwheel into two parts, and hence increase its complexity, is to save weight. This leads to the objective to reduce the gearwheel reference weight (617 g) by at least 25%. Reference weight, root and tip diameter and gear shaft geometry (DIN 5480-WA h30x1x28xh6x9e) are derived from the solid reference gearwheel in Figure 1 left.

The fine blanked wheel body consists of nine sheet metal layers (thickness = 1.5 mm) stacked on top of each other to reach the gear ring width. Four fasteners hold the sheet metal layers in place. Each sheet metal layer’s outer circular geometry is fine blanked, whereas the inner geometries are laser cut to save tool costs. Figure 3 shows the geometry of one sheet metal layer. In terms of an industrial process, fine blanking the outer and gear shaft geometry is feasible. The inner geometries might be shear cut because there is no demand for high surface quality in this area. A detailed description of the fine blanking process parameters and the resulting sheared edge quality can be found in [5]. The materials used for the fine blanked wheel body are the dual-phase steels HCT780X, HCT980X and HCT980XG (Table 1).

The gear ring is a thin ring that carries the teeth. This geometry leads to challenges within the gear ring’s heat treatment because large deviations occur. Therefore, manufacturing, heat treating and joining both parts is done by the IWT, Bremen [6]. The authors use a test ring (Figure 4 right), which is easy to manufacture, to join it with the wheel bodies and to test the assembly. Its outer diameter equals the tip diameter of the gear ring and both parts’ inner
Table 1. Material parameters.

| Parameter           | Unit | HCT780X\textsuperscript{a} | HCT980X\textsuperscript{a} | HCT980XG\textsuperscript{a} | 18CrNiMo7-6\textsuperscript{b, c} |
|---------------------|------|-----------------------------|-----------------------------|-----------------------------|-----------------------------------|
| Young’s modulus     | GPa  | 210                         | 210                         | 210                         | 210                               |
| Density             | t/m\textsuperscript{3} | 7.8                         | 7.8                         | 7.8                         | 7.8                               |
| Poisson’s ratio     | [-]  | 0.3                         | 0.3                         | 0.3                         | 0.3                               |
| Yield strength      | MPa  | 468                         | 583                         | 712                         | 1110                              |
| Tensile strength    | MPa  | 780                         | 980                         | 980                         | 1470                              |

\textsuperscript{a}material data provided by ThyssenKrupp Steel Europe  
\textsuperscript{b}material data taken from [7, p. 439]  
\textsuperscript{c}hardened

diameters match as well. The gear and test ring consist of the case hardening steel 18CrNiMo7-6 and their inner geometry serves as a boundary condition for the wheel body design. In the case of the gear ring, the 18CrNiMo7-6 is hardened, and in the case of the test ring, it is unhardened. Material data for the unhardened 18CrNiMo7-6 have not been obtained yet.

3. Design strategy

The main objective of the design process is to deliver a lightweight structure without the need to conduct time-consuming optimizations. Fulfilling this task is achieved by taking the manufacturing process into account while designing the part. This leads to a considerable decrease of producable wheel body geometries due to the manufacturing process’ limitations.

Because fine blanking uses a translational tool movement, the wheel body design can be changed in the radial or tangential direction only (see Figure 4 for coordinate system). The axial direction is parallel with the tool movement, and therefore geometry changes in this direction are hardly possible. With these limitations, the result needs to be a kind of spoke structure that is able to support the applied loads without buckling. Designing this spoke structure is the first step and dimensioning is the second one. For the first step, Mattheck’s torque anchor method [8] is used. It describes an analytical method for designing the load path between a torque anchor (gear shaft) and a force acting tangential to it (Figure 3). From every force application point (loading point), two load paths can be drawn toward the torque anchor. Putting all load paths on top of each other leads to the final design. In the case of the wheel body, where the loading points are evenly positioned around the outer geometry, this leads to the spoke structure shown.

![Figure 3](image-url)
in Figure 3. To connect the wheel body’s nine sheet metal layers, the ideal design is altered at four spots to provide round fastener holes. Increasing the number of loading points (max. no. of loading points = no. of teeth) results in smaller holes within the wheel body. This is critical for cutting because the necessary tools become small and complicated in manufacturing. Decreasing the number of loading points leads to bigger holes that are counteracting the structure’s radial stiffness needs. Radial stiffness influences the transmittable torque (see Section 6). Working with twelve loading points is chosen as a compromise between stiffness and producibility. In the second step, the spokes designed in the first step are iteratively increased in width. After every iteration, the loads within the part are determined in a static simulation. The iteration process stops when no loads occur above the materials yield stress. Eliminating peak loads by making minor geometry changes follows the last iteration.

4. Numerical setup

The software used for the elastic-plastic numerical investigations is Abaqus 6.12-3 with the integrated standard (implicit) solver. Figure 4 shows the two models and the cylindric coordinate system used. Model one includes the gear ring with teeth (Figure 4 left) and model two includes the test ring mentioned in Section 2. Both models serve different needs. The first model is used to investigate the gearwheels behavior under tooth forces (including radial and axial forces) derived from the reference gearwheel. These forces represent the point under dynamic load where the gearing fails. Model two is used to investigate the conditions of the test bench that the experiments are conducted with (see Section 5). It is used to validate the simulation and to improve the model one. External load is applied via a displacement boundary condition rather than forces. This allows to simulate beyond the point where the press fit fails. There are two kinematic couplings active in both models. RP center and the gear shaft on the inside of the wheel body are coupled, locking the axial and tangential translation. This allows the gear shaft geometry to deform in the radial direction. RP load and the corresponding tooth flank of the gear ring or accordingly the ledge of the test ring are coupled locking all translations. Both reference points act as control points for these couplings.

![Figure 4. Numerical models used to evaluate the behavior of the multi-component gearwheel and cylindric coordinate system.](image)

| Part          | Gear ring  | Wheel Body | Parameter | F_a | F_r | F_t | u_t |
|---------------|------------|------------|-----------|-----|-----|-----|-----|
| Element type  | C3D10M     | C3D8R      | Value     | 5263 N | 4395 N | 8858 N | 0.2 mm |
The material data used for the sheet material is obtained from ThyssenKrupp Steel Europe. From [7, p. 439], the yield curve for the hardened 22MnB5 is used to model the material behavior of the 18CrNiMo7-6. Tensile and bulge tests for the sheet material show minor differences compared to the material data obtained from ThyssenKrupp. Compression tests with specimens made of the gear ring material in hardened and unhardened conditions show that there are small deviations between reality and the numerical model regarding the hardened 18CrNiMo7-6 and large deviations regarding the unhardened 18CrNiMo7-6. Currently, material models based on the tests mentioned above are implemented in the numerical model.

Both models contain two steps. In the first step ‘press fit’ two boundary conditions are applied. Translations in the axial and tangential direction at the inner surface of the gear/test ring and the outer surface of the wheel body are permitted. Therefore, radial deformation due to the press fit is not constrained by boundary conditions. Interactions between the parts are modeled using the penalty algorithm and friction coefficient of $\mu = 0.12$. The friction coefficient is derived from the torque experiments (see Section 7) with HCT980XG, including the lightweight structure. Between the gear ring and wheel body, there is interference needed for a press fit. This interference is modeled geometrically and therefore the interaction formulation between those parts includes the removal of overclosure. In the second step ‘load’, tooth forces or the displacement are applied to the RP load. Boundary conditions from the first step are deactivated and a new one - clamping RP center - is created. Usually the simulation ends once the step has concluded, although in model one it might end earlier due to failure of the press fit. Failure of the press fit causes an insufficient constraining of the gear ring and thus a decreasing time step. If the time step needs to be decreased more than twice, the simulation ends. The maximum moment acting on RP center around the axial direction is ultimately identified. To determine the time step to take the maximum moment from, the contact status within the press fit area is investigated. It can have three different states: open, slipping and sticking. After the press fit step, most areas in the press fit are in the sticking contact status. The status changes to slipping once the load starts increasing. The point where there are no longer any sticking areas within the press fit, is defined as the point of failure. The time step just before this happens is used for identifying the maximum moment the multi-component gearwheel can transmit. [4]

5. Experimental setup

Figure 5 shows the test bench. Connecting the whole test bench with a tensile testing machine that moves the upper plate and measures the forces needed completes the experimental setup. The objective of the experimental setup is to test the press fit under loads that are similar to the real load on a gearwheel. A column mount is the platform that all other components are mounted on. Upper and bottom plates are guided by the columns that ensure reproducible measurements. A test wheel consisting of test ring and wheel body is joined with a shaft. To hold this assembly in place during testing, the shaft is supported by two shaft holders that are mounted on the base plate. A parallel key (not shown in Figure 5) connects the shaft and shaft holder to suppress rotation around the shaft axis. The punch applies the load on the ledge of the test ring. From the point of first contact, the force needed to move the upper plate downward increases over time. As soon as the press fit fails, the force drops immediately and the test ends. The maximum torque is calculated from the maximum measured force and the lever arm.

6. Numerical results

Initially, the numerical model with the gear ring is used. The investigated wheel body geometries are those used within the experiments. One includes the lightweight structure (LWS) shown in Figure 3 and one has the four holes for the fasteners only (no LWS). Figure 6 shows cutting edges taken into account. The dotted line represents the contour of an ideal sheared edge and the black line the real-edge model [5]. The real-edge model is based on eight measurements
Figure 5. Test bench for testing the multi-component gearwheels.

(gray lines) of the sheared edge of a fine blanked specimen. It does not match the real sheared edge exactly, as the thickness of one sheet metal layer is only meshed with three finite elements. This makes an abstraction necessary. Increasing the mesh quality would lead to an increase in elements within the contact area and result in immense simulation costs.

Figure 7 shows the influence of the wheel body geometry and sheared edge on torque, relative weight and torque-to-weight ratio. The lightweight objective is clearly met, as the multi-component gearwheels including the lightweight structure weigh 67% of the reference gearwheel. Conclusions regarding the influence of the lightweight structure cannot be drawn for model one. The reason why is, that the calculation ends when a torque of 400 Nm has been reached, which is what gearwheels with both wheel bodies do. In contrast, the influence of the sheared edge on the transmittable torque is obvious. A numerical model using the ideal edge delivers a maximum torque that is 91 Nm lower than a comparable model which uses the real-edge model.

To investigate the influence of the wheel body geometry and the wheel body material on the torque, the numerical model with the test ring is used. Figure 8 shows the results. Due to the displacement boundary condition that moves the test ring around the wheel body (see Section 4), torques higher than 400 Nm are applied to the gearwheel. As mentioned in Section 3, the wheel body stiffness in the radial direction influences the transmittable torque. Therefore, gearwheels without the lightweight structure fail at higher torques than those including the...
lightweight structure. The wheel body material has no visible effect.

7. Experimental results
The experimental torque values were determined by averaging the results of six specimens. In general, the experiments confirm the findings of the numerical investigations (Figure 8). There is no visible effect of the wheel body material on the torque, and the torque increases with increasing wheel body stiffness in the radial direction. Regarding the wheel bodies with the lightweight structure, numerical and experimental results match well while the results for wheel bodies without lightweight structure differ. This might be explained with different contact conditions within the press fit area. These conditions are influenced by the friction value, the contour of the sheared edge, the wheel body stiffness in the radial direction and the test ring material. The edge contour that results after the fine blanking process is influenced by the part stiffness [9]. It is observed that the sheared edge contours of the two wheel bodies are not

![Figure 7](image-url)

**Figure 7.** Influence of wheel body geometry and sheared edge contour on torque, relative weight and torque-to-weight ratio. Results are obtained using the numerical model with the gear ring.

7. Experimental results
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![Figure 8](image-url)

**Figure 8.** Influence of wheel body geometry and wheel body material on torque, relative weight and torque-to-weight ratio. The numerical values are obtained using the numerical model with the test ring and the real-edge model.
the same. The edge contour for the wheel body without the lightweight structure needs to be implemented in the numerical model. Furthermore, the test ring material is not hardened, but the material model is based on hardened steel. This might explain why the deviations between simulation and experiments for the HCT980XG without lightweight structure are larger than those for the other materials. In the experiments, yielding is observed on the inside of the test ring, which does not occur in the simulation. In addition to that, the influence of the test bench’s stiffness on the results needs to be further investigated [1].

8. Conclusion and Outlook
The experiments show promising results regarding the absolute torque the press fit transmits. Because these experiments are static tests so far, several multi-component gearwheels are being taken to endurance trials at the Gear Research Centre in Munich. This way, their dynamic loading capacity will be validated. Furthermore, the agreement between numerical calculations and experiments needs to be improved. As mentioned above, new edge models for the other material-wheel body combinations will be implemented. The material models for all materials used will be matched with the results of the tests that were conducted at the author’s institute. The numerical results will make it be possible to evaluate the load capacity of a new wheel body structure without further experiments, whereas the results from the endurance trials will help determine the applications the multi-component gearwheels might be useful for in future.

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