Adapted part design methods for springback minimization of stamped sheet metal car body components

A Birkert*, F Dreiseitl¹, B Hartmann², T Held¹, O Hetterle¹, M Markin¹ and M Scholle¹

¹ Centre for Metal Forming and Car Body Manufacturing, Faculty of Mechanics and Electronics, Heilbronn University of Applied Sciences, Heilbronn, Germany
² inigence gmbh, Bretzfeld, Germany

* E-mail: arndt.birkert@hs-heilbronn.de

Abstract. In sheet metal forming, both the dimensional accuracy of the manufactured parts and the robustness of the forming process are significantly compromised by elastic springback effects. In contrast to traditional compensation methods mostly relying on modifications of the tool geometry, a part geometry-based approach reduces the amount of springback by appropriately manipulating the part stiffness through slight modifications of the part geometry. To date, part-geometric measures are mostly based on practical experience and trial-and-error processes resulting in a need for simple and effective simulation-based approaches. In the present work three novel simulation-based approaches for a suitable modification of structural component design are suggested relying on unconventional topology optimization, the so-called line-of-force method and shape optimization. First tests with a classical hat profile reveal that the optimized geometries, compared to the nominal one, lead to significant reductions of the springback after stamping and simultaneously to a notable decrease of the process variation when parameters of the forming process are varied slightly. For the adapted topology optimization approach the excellent results are also confirmed when applied to the industry-relevant case of a truck rear panel.

1 Introduction
In automotive industry, the stamping process of complex lightweight car body components of high stiffness is considerably complicated by the occurrence of springback effects compromising the process robustness and the dimensional accuracy of the resulting parts. In daily practice, a bundle of different approaches is employed to handle the elastic springback which can be roughly divided into three groups. The first is tool geometry-based and encompasses all classical and more recent methods of springback compensation [1]; the second is forming-process based aiming at an optimization of the stamping parameters or additional operations [2]; the third is part geometry-based and focuses on slight modifications of the part design itself in order to locally manipulate the part stiffness in a desired way [2]. A distinct advantage of the latter group consists in the fact that cost-intensive tool modifications can potentially be reduced to a minimum while the process robustness with respect to the dimensional accuracy is enhanced [3]. However, the identification of optimized part designs is a highly complex task, the effectiveness of which does not only depend on the exact positioning of the typically used stiffening bead elements but also on their orientation and precise shape [4]. Nevertheless, in industry it...
is still common practice to derive such modifications of the part design according to experience or on a trial-and-error basis leading to time-consuming iteration processes and suboptimal results. Thus, there is a need for simple and effective simulation-based methods allowing to deploy the full potential of part geometric modifications for the improvement of dimensional accuracy and process robustness.

In the present paper three new adapted methods for springback reduction are introduced, all of them following the part geometric approach, where stiffening elements, so-called beads, are applied to the part geometry in order to reduce the amount of springback by increasing the parts stiffness. The first adapted method originates from the field of topology optimization (TO), the second is a variant of the so-called line-of-force (LF) method, and the third origins from shape optimization (SO).

The part geometric approach represents one key element in the whole springback reduction process (figure 1) but cannot completely render subsequent optimizations of the forming process and the tool geometry unnecessary.

![Figure 1. Springback reduction process involving new methods for the part geometric approach](image1)

In sec. 2 the three adapted approaches for springback reduction are explained in detail and applied to and evaluated for a simple test geometry of a hat profile. The impact of the optimized geometries on the variation of the stamping process is then investigated in sec. 3, followed by the application of the TO method to an industry-relevant problem (sec. 4). Results and conclusions are summarized in sec. 5.

2 The three methods in detail

To exemplify the three novel part geometric approaches a simple hat profile is employed using an ultra-high-strength steel, HDT1200M, in order to provoke a large springback effect. In a preliminary forming simulation, hereafter referred to as the base simulation, the springback of the non-optimized part is determined which for the hat profile is up to 19.4 mm measured in normal direction, see figure 2.

![Figure 2. Springback displacement of the hat profile resulting from the base simulation](image2)

![Figure 3. Geometrical specifications of used beads](image3)

Based on this result the methods presented in the following sec. optimize the part design by introduction of stiffening elements. To ensure comparability, both the TO and the LF method use fixed beads with the same dimensions. For the present case of a hat profile, a simple channel-shaped bead structure proved adequate (figure 3); for reasons of feasibility the edges have been rounded. Different from the other methods, the SO algorithm identifies optimal bead geometries automatically with only the maximum bead height being constrained to 4 mm height.
2.1 Topology optimization method (TO)

The aim of TO is to generate an optimal structural component layout for a specified set of boundary conditions, loads and constraints, where typically the compliance of the component is to be minimized (target function) while a minimal total mass shall be retained (constraint). Starting from a primitive initial component design of maximum possible dimensions (design space), the mass density of the part is locally reduced in regions that contribute to the part stiffness only marginally; this process is iteratively performed until the entire mass of the optimized part falls below a given threshold.

The idea of the present paper is to apply classical TO not to some primitive initial design space in order to obtain a very basic design proposal but rather to a finalized geometry with the simulated springback as predefined load case; the result is not used to locally reduce the mass density of the part but rather to identify areas a ‘stiffening’ of which potentially reduces the springback behaviour notably.

For the present investigation, the design space is defined by the geometry of the hat profile and the load case is defined by the springback displacement field shown in figure 2. The target function of the TO is defined by minimizing the compliance; the total mass is constrained to not undershoot 10% of its initial value. Based on this setup, 27 optimization iterations have been performed using Ansys Workbench in order to find the configuration with minimal compliance.

Figure 4 shows the resulting density distribution of a possible design proposal which is used as an indication of where to place stiffening beads. Although five beads are placed along the black-coloured region, see figure 5, the exact number and orientation of the beads are not an outcome of the TO but are chosen to achieve a better comparability with the two subsequent approaches.

![density distribution](image)

Figure 4. Resulting design space after topology optimization

![beads](image)

Figure 5. Adjusted CAD-model of the hat profile based on the topology optimization results

For the CAD model modified according to the bead pattern of figure 5, again the springback has been determined in a forming simulation using AutoForm. As a result, the maximum amount of springback has been moderately reduced by 13% from 19.4 mm to 16.8 mm. Further variations of the bead pattern will be part of future investigations.

2.2 Line-of-force method (LF)

A so-called line-of-force method (LF) has been described in detail by Krauss [5] and demonstrated for a part geometric vibration optimization of an endplate of a car middle silencer. This method consists of three steps. First, defining the load case for the stress analysis, second, calculating a vector field of bending eigenvectors for each element of the finite element (FE) discretization, and third, tracing the ‘lines of force’ using a 4th Order Runge-Kutta Method stabilized by Adaptive Gaussian Windowing [5]. The obtained lines can be used as a guideline for stiffening bead patterns.

In the following the LF method is adopted and modified for the springback optimization of the hat profile, analogously to sec. 2.1. As a slight variation, the load case is defined to be the part’s stress state in the forming simulation while the tool is closed; this stress state should carry more or less the same information as the corresponding springback displacement used in sec. 2.1. However, the forming simulation and the calculation of the springback is performed in AutoForm using a multilayer shell element the stress state of which is, for reasons of practicality, transformed to and approximated in a simple two-layer model used in Ansys Workbench where only the optimization has been carried out.
Thus, the element-wise stress tensors of the upper (L1) and lower layer (L2) are obtained which can be separated into bending and membrane stresses according to Emmrich [6] and Krauss [5], see equations (1); the superimposed stress state is visualized in figure 6.

\[
\sigma_{\text{mem}} = \begin{bmatrix}
\sigma_{xx}^{\text{mem}} \\
\sigma_{yy}^{\text{mem}} \\
\tau_{xy}^{\text{mem}}
\end{bmatrix} = \frac{1}{2} \begin{bmatrix}
\sigma_{xx}^{\text{bend}} + \sigma_{xx}^{\text{mem}} \\
\sigma_{yy}^{\text{bend}} + \sigma_{yy}^{\text{mem}} \\
\tau_{xy}^{\text{bend}} + \tau_{xy}^{\text{mem}}
\end{bmatrix}
\]

\[
\sigma_{\text{bend}} = \begin{bmatrix}
\sigma_{xx}^{\text{bend}} \\
\sigma_{yy}^{\text{bend}} \\
\tau_{xy}^{\text{bend}}
\end{bmatrix} = \frac{1}{2} \begin{bmatrix}
\sigma_{xx}^{\text{bend}} - \sigma_{xx}^{\text{mem}} \\
\sigma_{yy}^{\text{bend}} - \sigma_{yy}^{\text{mem}} \\
\tau_{xy}^{\text{bend}} - \tau_{xy}^{\text{mem}}
\end{bmatrix}
\]

(1)

Figure 6. Separation of the stress state according to Emmrich [6]

The separated stress tensors can now be used to element-wise calculate the principal major bending and membrane stresses \( \sigma_{1|H|}^{\text{bend}}, \sigma_{1|H|}^{\text{mem}} \) and to determine the ratio \( \kappa \) as in equation (2) [6]. Based hereon, regions with predominant bending stress are identified by a ratio of \( \kappa > 1.2 \) following a suggestion of Krauss [5], see figure 7.

\[
\kappa = \frac{\sigma_{1|H|}^{\text{bend}}}{\sigma_{1|H|}^{\text{mem}}}
\]

with:

\[
\sigma_{1|H|} = \max[\sigma_{1,2}]
\]

\[
\sigma_{1,2} = \frac{\sigma_{xx} - \sigma_{yy}}{2} \pm \left[ \left( \frac{\sigma_{xx} - \sigma_{yy}}{2} \right)^2 + \tau_{xy}^2 \right]^{1/2}
\]

(2)

Figure 7. Ratio of bending stresses \( \kappa \) of the hat profile base simulation

In a next step, the eigenvectors corresponding to the major bending stresses are calculated for each element in the local element coordinate system, equation (3), and then transformed to the global coordinate system; the vectors are placed in the centre points of the elements representing a vector field on the discrete surface.

\[
\vec{n}_{1|H|} = \left[ \left( \frac{\sigma_{1|H|}^{\text{bend}} - \sigma_{2|H|}^{\text{bend}}}{\sigma_{bend}^{xx}} \right)^2 + 1 \right]^{-1/2} \cdot \left( \frac{1}{\sigma_{1|H|}^{\text{bend}} - \sigma_{2|H|}^{\text{bend}}} / \sigma_{bend}^{xx} \right)
\]

(3)

Based on this vector field, lines of force are traced starting from predefined initial points and with the local line tangents given by the vectors; the initial points are placed at the boundary lines of predominant bending areas, see the white crosses in figure 7. The initial value problems corresponding to each line are solved by a 4th order Runge-Kutta method using Matlab. In order to stabilize the method, the used vectors are averaged by an Adaptive Gaussian Windowing, as described by Krauss, in which the vectors are averaged in spherical neighbourhoods of fixed radius [5]. The vector field for the hat profile and the resulting discrete support points of the LF are shown in figure 8.

Finally, the obtained support points of the force lines are interpolated by splines in the CAD model of the hat profile and used as a guideline for the positioning and orientation of the beads as shown in figure 9. The number of five bead repetitions is, similar as in sec. 2.1, not a result of the calculations, but is chosen to stay comparable to the other methods.

The subsequent check of the springback, based on a forming simulation with the modified geometry design of figure 9, reveals a maximum springback reduction by 50%, from 19.4 mm to 9.7 mm.
2.3 Shape Optimization method (SO)
Like the above two approaches, shape optimization (SO) can be used to increase the stiffness of shell-shaped structural components for different load cases but, in contrast to the others, node displacements are allowed in normal direction to the finite shell element mesh. SO is also called topography or bead optimization. The capabilities of SO have been demonstrated in Krauss [5] by means of an optimization of a rear silencer. In addition to that SO of draw beads has been used in Shi [7] to ensure part feasibility.

In this sec., SO is adopted for the case of springback reduction and the optimization is performed using XCARAT, a software licensed by FEMopt Studios GmbH. As the software is mostly restricted to load cases with point forces, the load cases considered in sec. 2.1 and sec. 2.2 are reformulated to approximate the springback behaviour by application of scattered point forces to the FE mesh. In this context it was observed that load-carrying nodes that are allowed to move during the optimization can lead to undesired mesh distortion and local stress artefacts rendering the optimization result useless. For this reason, the considered hat profile is subdivided into bands, as shown in figure 11, where the FE nodes of each band are alternately either loaded and frozen or unloaded and movable (design variables) during optimization. Furthermore, boundary nodes are excluded from the set of design variables and a symmetry constraint has been set for the yz-plane.

Now, for the given (approximated) load case the compliance of the part is minimized by variation of the design variables until the change of compliance falls under a given threshold. For the optimization of the hat profile 25 iterations have been carried out and figure 10 shows for each iteration the cumulated compliance reduction and the actual springback reduction obtained from a corresponding forming simulation. Obviously the optimization exhibits a rapidly decreasing compliance and the actual springback reduction is even more pronounced than suggested by the compliance values.

![Figure 8. Bending eigenvector field and resulting lines-of-force](image)

![Figure 9. Beads in the CAD-Model of the hat profile using the adapted line-of-force method](image)

![Figure 10. Comparison of compliance reduction during shape optimization with the springback reduction of associated forming simulations](image)

![Figure 11. Geometry resulting after 25 iterations of shape optimization](image)

The final iteration 25 shows the best result with a reduction of springback of about 90% (19.4 mm to 2.0 mm); the corresponding optimized geometry is shown in figure 11.

For the sake of clarification, it is noted that the load case remains fixed during the iteration and is not modified according to the updated predictions of the forming simulations. However, in the context of
springback optimization this would be a natural enhancement of the procedure and is investigated in forthcoming research.

3 Process Robustness Analysis of the hat profile
The results of the three methods presented in sec. 2 are now compared in a standard forming process robustness analysis using Autoform. The process robustness analysis includes normally distributed noise variables as the initial pressure, lubrication, R-value, blank thickness, x and y positions of the blank and yield stress combined with a correlating tensile strength. This leads to a complexity of 128 simulations for each of the three methods under consideration. Figure 12 shows the impact of the different geometry optimizations on the expected amount of springback (mean value \( \mu \)) and on the process robustness (standard deviation \( \sigma \)).

![Figure 12. Comparison of expected springback and process robustness for the base simulation and the three optimized geometries](image)

The curve on the far-right side of figure 12 is the probability density function for the springback of the non-optimized base simulation. The mean value of the maximum amount of springback \( \mu \) is 19.7 mm and the standard deviation is 1.13 mm. Each of the described methods show a noticeable improvement of springback while a significant process stabilization could only be achieved with SO. For the other two methods the observed process stabilization is less pronounced.

The results of the SO are difficult to compare with the other two methods because here the shaping of the beads is done algorithm-wise and the resulting bead shapes differ a lot from the shape in Figure 3 prescribed for the other two methods. However, this demonstrates that besides the location and orientation of the beads also the precise geometry has a significant influence on the springback.

4 Application of topology optimization to a truck rear panel
Subsequent to the investigations of a rather simple hat profile in sec. 2 and 3, a truck rear panel is chosen as an example of a much more complex car body component in order to demonstrate the capabilities of the new method. The following investigations are confined to the TO method, although forthcoming publications will extend the analysis to the other methods as well.

![Figure 13. Truck rear panel with the calculated springback of the base simulation](image)

![Figure 14. Geometrical specifications of the employed beads](image)

Figure 13 shows the truck rear panel and the calculated springback of the base simulation, i.e. before modification of the geometry, which is up to 10.5 mm measured in normal direction from the nominal part. The stamping process of the truck rear panel is simulated using a single action draw die and a rectangular blank; in a second step the part is trimmed by virtual laser cutting. The material is chosen to be a bake hardening steel HX180BD with 0.8 mm sheet thickness.
4.1 Advanced usage of topology optimization method for a complex car body part

In accordance with the procedure of TO as described in sec. 2.1, the part geometry of the truck rear panel has been optimized to reduce the springback of the part. In a first step a mechanical load has been defined based on the nodal displacements of the forming simulation. Furthermore, the target of the optimization has been defined as to minimize the compliance of the part while the total mass is allowed to decrease down to 10% of its original value. The TO has been realized in 14 iterations and figure 15 shows the resulting design proposal.

![Figure 15. Design proposal for the truck rear panel resulting from the topology optimization](image1)

![Figure 16. Adjusted CAD-model of the truck rear panel based on the topology optimization](image2)

Based on the resulting design proposal the CAD-model of the part has been adjusted by implementing predefined beads within the detected area. The geometrical specifications of the beads are shown in figure 14 and the positioning of the beads on the part can be seen in figure 16.

A forming simulation based on the optimized geometry reveals that the springback of the truck rear panel could be reduced by 86% from 10.5 mm to 1.5 mm.

4.2 Influence of the process-robustness of the topology optimization method

To evaluate the impact of the TO-based geometry adjustment on the process robustness, the parameters of the forming process have been varied exactly in the same way as described in sec. 3 leading to a total of 128 simulations. Figure 17 shows the resulting probability density for the springback of the truck rear panel. The base simulation without any geometrical adjustments exhibits a mean value of the maximum amount of springback $\mu$ of 10.3 mm and a standard deviation of 2.29 mm. In contrast, the results of the adjusted part geometry reveal a noticeable improvement: The mean value of the maximum amount of springback $\mu$ could be reduced to 1.7 mm and the standard deviation could be reduced to 0.71 mm. This demonstrates that, based on the TO method, the process-robustness of the truck rear panel could be improved significantly.

![Figure 17. Process robustness analysis for the truck rear panel based on 128 simulations](image3)

5 Discussion and Conclusion

In the framework of the present paper three novel simulation-based approaches for a suitable modification of structural component design have been suggested which aim both at a robustness enhancement of the component forming process and a reduction of the elastic springback after stamping. The methods are mostly based on classical algorithms from the fields of TO and SO but are, for the first time, used for and adapted to the problem of springback reduction.

The key idea of the adapted TO is to use the mass density distribution resulting from the classical approach as an indication for the positioning of stiffening bead elements of fixed dimensions. In contrast,
the adapted LF method takes the force lines calculated from a bending eigenvector field as a guideline for a stiffening pattern, i.e. the structural component is stiffened along directions of predominant bending stress. Eventually, the adapted SO does not work with fixed bead elements but the nodes of the shell FE mesh are allowed to freely vary in normal direction to the discrete surface during the process of compliance minimization.

When exploring the optimization of a very basic test geometry, namely the hat profile, all three novel approaches deliver promising results. The adapted TO reduces the springback of the hat profile by 13%, the adapted LF method by 50% and the adapted SO even by 90%. Additionally, dramatical forming process stabilization of the hat profile is achieved in the case of SO, where the process variation $\sigma$ has been reduced from 1.13 mm to just 0.16 mm. For the other two methods the observed process stabilization was less pronounced, the reasons of which have to be analysed in forthcoming work.

First investigations of TO for the much more complex rear wall of a truck deliver excellent results demonstrating the capabilities of the new approach also for industry-relevant problems: The springback has been reduced by 86% and the process variation $\sigma$ has been reduced from 2.29 mm to 0.71 mm. However, it must be examined why the method is so effective despite the low geometric determination of the stiffening pattern. So far, the considerations of the truck rear wall are confined to the TO approach. For an extension to the LF method more investigations are needed, particularly to appropriately handle the mostly membrane-stressed part areas. Very first tests and a comparison with the results of sec. 4 suggest that the condition (2) of predominant bending stress has to be relaxed or adapted. For an extension to SO, it is necessary, that the parts stress state in the forming process is approximated with higher quality; the current compromise relying on a point force description needs a revision.

Although the investigations of the present paper are very promising, it must be clear that further development is needed in order to test all methods on a broader variety of complex body parts. Also, for a readily usable industry solution it is inevitable to provide a user-friendly handling, to automate the construction of bead patterns and to reduce the required problem-dependent user interaction to a minimum. Furthermore, in industry practice it may happen that a geometry modification of an optimization is restricted by technical side conditions which have to be met. However, it is obvious that such an easy-to-use and simulation-based method would greatly help to establish the part-geometric approach as a standard tool in the compensation process complex body parts in automotive industry.

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