Preference-based Topology Optimization of Body-in-white Structures for Crash and Static Loads

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Abstract

Topology optimization methods are increasingly applied tools for the design of lightweight structural concepts in the automotive design process. Ideally, topology optimization provides the optimum distribution of material within a user-defined design space for a given objective function. In the vehicle design process, two important objectives are to maximize stiffness of components for regular working conditions and to maximize energy absorption in exceptional loading conditions, for instance in crash events. For these objective functions, the Hybrid Cellular Automata algorithm devises efficient structures in case of the separated disciplines, by heuristically aiming for a uniform distribution of energy densities. Recently, it was demonstrated that a concurrent optimization of crash and static load cases can be performed by a linear weighting, in which the user preference is separated from the scaling of the internal energies. In this paper, the approach is applied to the practical example of a vehicle body-in-white design, which is optimized for multiple crash and linear static load cases. By comparing resulting internal energies of different load case settings we demonstrate that the hybrid cellular automata algorithm with scaled energy weighting is capable to find a very good trade-off solution within a single concurrent optimization run.

1. Introduction

One of the current major tasks in automotive development is the design of efficient lightweight vehicle structures targeting low fuel emissions in the spirit of reducing the carbon footprint. Since modern manufacturing methods allow for innovative component design, understanding the optimal distribution of material provides a conceptual layout of the vehicle body structure. With today's given computational power, body designers and engineers profit from advanced numerical optimization approaches to propose and evaluate vehicle concept structures, especially in the face of increasing number of multi-disciplinary and partially conflicting requirements that have to be taken into account [1].

As one of these methods, topology optimization [2, 3] searches for the optimal material distribution in a given design space subject to load and supports. Typically, the design space is discretized, since finite element analysis is required to calculate numerical structure responses. By an iterative increase/decrease of single element material densities and addition/removal of single elements in the design space the material distribution is optimized and a structural concept is obtained that can serve as a starting point in the design process. Depending on the application scenario, various topology optimization targets are expressed as optimization criteria. High stiffness is required for normal operating conditions, taken into account for instance by static loads on the body structure caused by suspension, engine, and seat mount points. High energy absorption is targeted for extreme dynamic loads, as occurring in vehicle crash event. However, while software tools exist for tackling each application scenario on its own, it is still a major challenge to perform topology optimization of structures concurrently for both, static and

dynamic loads, as required by automotive development. One difficulty is posed by different magnitudes of energy produced in the body from normal loads compared to vehicle crash impacts.
Recently, we proposed a topology optimization approach, which utilizes and extends the state-of-the-art Hybrid Cellular Automata (HCA) method [4, 5]. The HCA is a heuristic topology optimization method that targets a uniform internal energy distribution throughout the structure, assuming that in this way an efficient structure is obtained. It has been applied to optimize compliance of linear elastic structures and has also been transferred successfully to the domain of highly nonlinear dynamic loads, such as occurring in crash events [6, 7]. The available software LS-TaSC™ [8] is a commercial implementation, suitable for handling industrial models. Our proposed extension suggests an efficient concurrent optimization for static and dynamic load cases in a multi-load case approach by a scaling of the energy densities. The applicability of this method has been successfully demonstrated in case studies of the authors, first on academic beam models [9], then on a more practical vehicle lower control arm model [10].
In the present paper, we analyze the scalability of the method by evaluating a topology optimization of a vehicle body-in-white structure and show the capability of finding a good trade-off solution for 9 static loads and 2 crash impacts. The paper is organized as follows. In Section 2 we present the mathematical background of the HCA algorithm and the extension of scaled energy weighting. Section 3 illustrates the vehicle body-in-white structure and the load cases, followed by results and discussion in Section 4. Section 5 concludes the paper.

2. Hybrid Cellular Automata with Scaled Energy Weighting

2.1 Problem Formulation

In what follows, we introduce the HCA algorithm and first state the minimum compliance and the maximum energy absorption formulations that are addressed.
The HCA operates on the finite element mesh representation of the design space, in which each element is a binary decision variable. HCA belongs to the class of density-based topology optimization methods [2], for which the binary problem is relaxed and each element is assigned a continuous density variable. A frequently used material interpolation scheme is the solid isotropic material with penalization (SIMP) material model [11], according to which the Young's modulus of the material within an element is determined as:

$$E_i(\rho_i) = \rho_i^p E_0 \ , \qquad (1)$$

where ρ_i is the density of element i, E_0 is the Young's modulus of the full material and p is a penalization exponent.
The minimum compliance problem can be stated as:

$$\begin{aligned} &\min_{\vec{\rho}} c(\vec{\rho}\,) = \vec{u}\vec{f} \\ &\quad s.t.: \vec{K}(\vec{\rho}\,)\vec{u} = \vec{f} \\ &\qquad\quad V(\vec{\rho}\,) = V_t \\ &\qquad\quad 0 < \rho_{\min} \le \rho_i \le 1, i = 1, \dots, N \ , \end{aligned} \qquad (2)$$

with the compliance c, the displacement vector $\vec{u}$ and the load vector $\vec{f}$. In the equilibrium equation, $\vec{K}$ denotes the stiffness matrix. A constraint V_t imposes a target on the volume of the structure. In order to avoid numerical instabilities a minimum density $\rho_{\min}$ is defined.
For crash loads, the linear elastic material model is not applicable, since the structure is usually deforming plastically. In this case, the SIMP interpolation can be extended to a piecewise linear elastic-plastic material model by interpolation of yield stress σ_{Yi} and strain-hardening modulus E_{hi} [6]:

$$\sigma_{Yi}(\rho_i) = \rho_i^p \sigma_{Y0} \, , \, E_{hi}(\rho_i) = \rho_i^p E_{h0} \, , \tag{3}$$

with the yield stress σ_{Y0} and the strain hardening modulus E_{h0} of the full material elements. The maximization of the energy absorption for the crash can be formulated as:

$$\begin{aligned} &\max_{\vec{\rho}} E_{\text{abs}}(\vec{\rho}\,) \\ &\quad s.t.: \vec{r}(\,\vec{\rho}\,, t\,) = 0 \\ &\qquad\quad V(\vec{\rho}\,) = V_t \\ &\qquad\quad 0 < \rho_{\min} \leq \rho_i \leq 1, i = 1, \dots, N \, , \end{aligned} \tag{4}$$

with the total energy absorbed by the structure $E_{abs}(\,\vec{\rho}\,)$ by elastic and plastic deformation at the final time step t=$t_{\max}$ and the residual $\vec{r}(\,\vec{\rho}\,, t\,)$ of the dynamic finite element analysis.
Both problems in the preceding section can be addressed by the HCA approach. The state of each cell is defined by its field variable S_i and its material density ρ_i. The optimization is heuristically addressed by applying a control based update rule aiming for a uniform distribution of the field variable in order to achieve an optimized material utilization. While not strictly minimizing the objective function in a mathematical sense, this nevertheless leads to useful designs from an engineering perspective, based on the idea of fully stressed design. In the design of stiff structures, the elemental Strain Energy Density (SED) is used as field variable [5], whereas in the design of crashworthiness structures, the maximum of the elemental Internal Energy Density (IED) is used [7]. A design update, respectively an update of the cell states is obtained by:

$$\rho_i^{\text{new}} = \rho_i + K_{\text{P}}(S_i - S^*) \, , \tag{5}$$

where K_P is a control parameter and S^* is a set-point for the field variables, which is adapted in each iteration so that the desired volume constraint holds. In order to avoid numerical instabilities and to impose a minimum length scale, smoothing of the field variables using a neighborhood rule and a cell memory is proposed in [6].

2.2 Multi-Load Case Optimization with Scaled Energy Weighting

In this section, we introduce the approach for concurrent topology optimization of crash and static loads. In the optimization of multiple load cases, the objective functions (2) and (4) are replaced by formulations that aggregate compliance or energy absorption for all load cases. Then, in HCA, the field variables of the load cases are combined by a linear weighting:

$$S_i = \sum_{l=1}^{L} w_l S_{il} \, , \tag{6}$$

with the number of load cases L, a weight w_l for each load case and the field variable S_{il} associated with element *i* for load case *l*.

The combination of concurrent static and crash load cases has been considered in [9, 10]. There, a linear weighting of the objectives in (2) and (4) is proposed by applying the HCA methodology for the case of distinguished field variables:

$$S_{il} = \begin{cases} \max_t \mathrm{IED}_{il}(t) & \text{if } l \in L_{\text{crash}} \\ \mathrm{SED}_{il} & \text{if } l \in L_{\text{static}} \end{cases} , \tag{7}$$

with the set of crash and static load case indices, L_{crash} and L_{static} respectively and the time step *t* of the crash simulation. Since both field variable quantities are energies, the summation in (6) is still physically meaningful.

The weights w_l in (6) have a twofold role. Besides expressing user preferences on the load cases, they need to account for different scales of the field variables. The energies occurring in a crash load case are usually orders of magnitudes higher than those occurring for a static load. In order to avoid a dominance of crash load cases, energies have to be scaled. In order to avoid mixing of user preferences and the required scaling of the energies, we propose to split the weight w_l in a user-defined preference factor p_l and a scaling factor s_l:

$$S_i = \sum_{l=1}^{L} w_l S_{il} = \sum_{l=1}^{L} p_l \frac{1}{s_l} S_{il} . \tag{8}$$

The idea of this formulation is to decouple the scaling of the load cases from the preference of the user, so that once a suitable s_l is determined, the choice of the preference is only expressing the trade-off desired by the user. Since the field variables refer to energy quantities we denote the approach Scaled Energy Weighting Hybrid Cellular Automata (SEW-HCA).

The scaling factor is heuristically chosen as [9]:

$$s_l = \frac{W_l^{(\text{full})}}{W_{\min}^{(\text{full})}} , \tag{9}$$

where $W_l^{(\text{full})} = \sum_{i=1}^{N} S_{il}^{(\text{full})} v_i$ is the work of the structure obtained from the analysis of the load cases with the full design space, i.e. when ρ_i=1 for all elements, and the element volume v_i. $\mathrm{W}_{\min} = \min_l W_l^{(\text{full})}$ is the minimum work of all the load cases. The values for $W_l^{(\text{full})}$ are hence obtained by an additional analysis prior to the optimization. They provide an indicator of the different magnitudes of energies involved in the load cases.

Although energy levels might change slightly during optimization, this is usually a better guess than any weights chosen based on user intuition. Concretely, the elemental IED or SED values of load case *l* are scaled by a factor, which is the ratio of the work of load case *l* to the work of load case which from all load cases has the lowest work.

3. Vehicle Body-in-white Model

The focus of this work is the application of SEW-HCA in the vehicle design process. For this purpose, a realistic case study of applying the method to body-in-white vehicle platform design concept was conducted. Figure 1 shows the model of the design space from an early state of a design process. The whole (red) volume is potentially available to contain structural members, for which the best layout is unknown and should be determined by topology optimization.

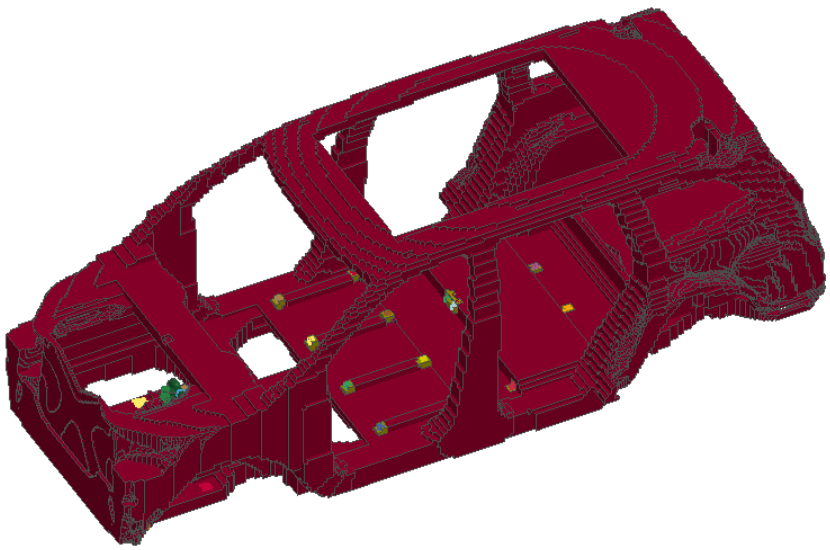

Figure 1: Vehicle body-in-white design space for the case study.

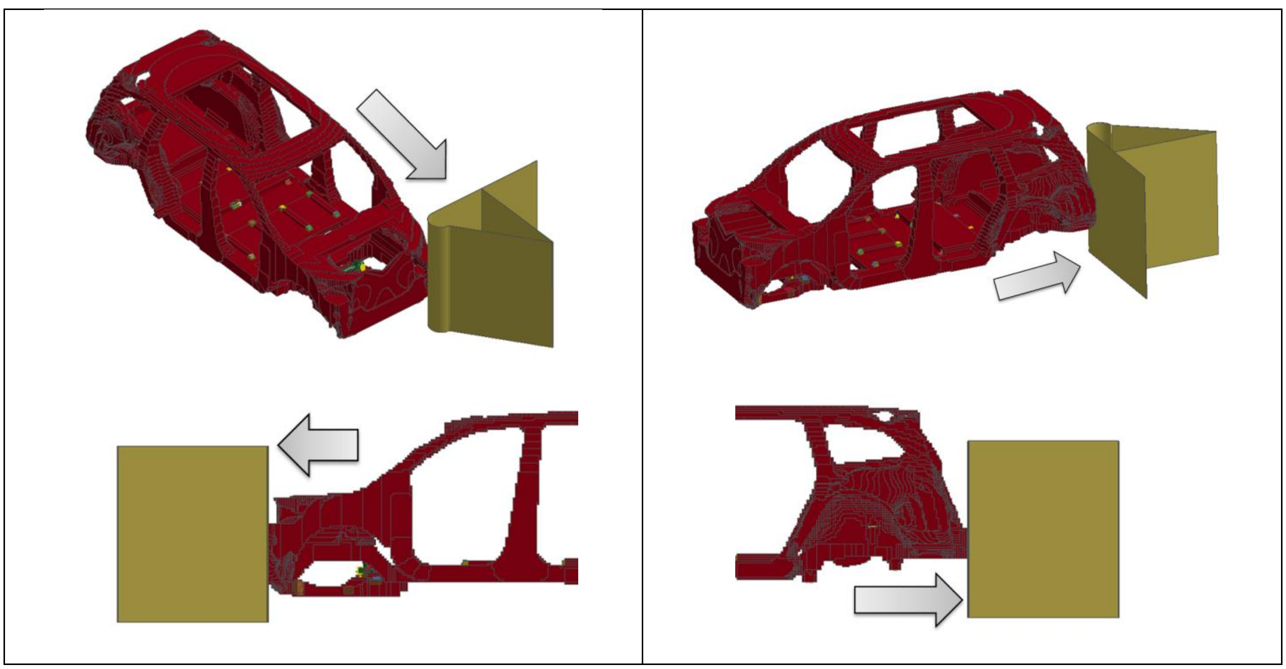

Figure 2: Front (left) and rear crash (right) load cases.

In total, eleven load cases are applied, shown in Fig. 2 and Fig. 3. Concretely, a rear and a front small overlap crashworthiness case are considered, where the vehicle impacts a wall with an initial velocity of 20m/s (rear) and 10m/s (front). These crash load cases are generic and are used to suitably deform the design space in a representative fashion to cover more numerous real world crashworthiness requirements. Additionally, standard operating conditions are considered by nine linear elastic static load cases that can be differentiated into front, seat and rear load cases. In the front, a bending, a lateral, a twist and a sub-frame twist load are applied. In the cabin, loads by the passenger seats in the front and in the rear are applied. In the rear, a bending, a lateral and a twist load are applied. The lateral loads are calculated at the individual suspension body points, that are a representation of the wheel loads with a lateral load applied. With eleven load cases the number of load cases is significantly higher, compared to previous studies with two [9] and three load cases [10].

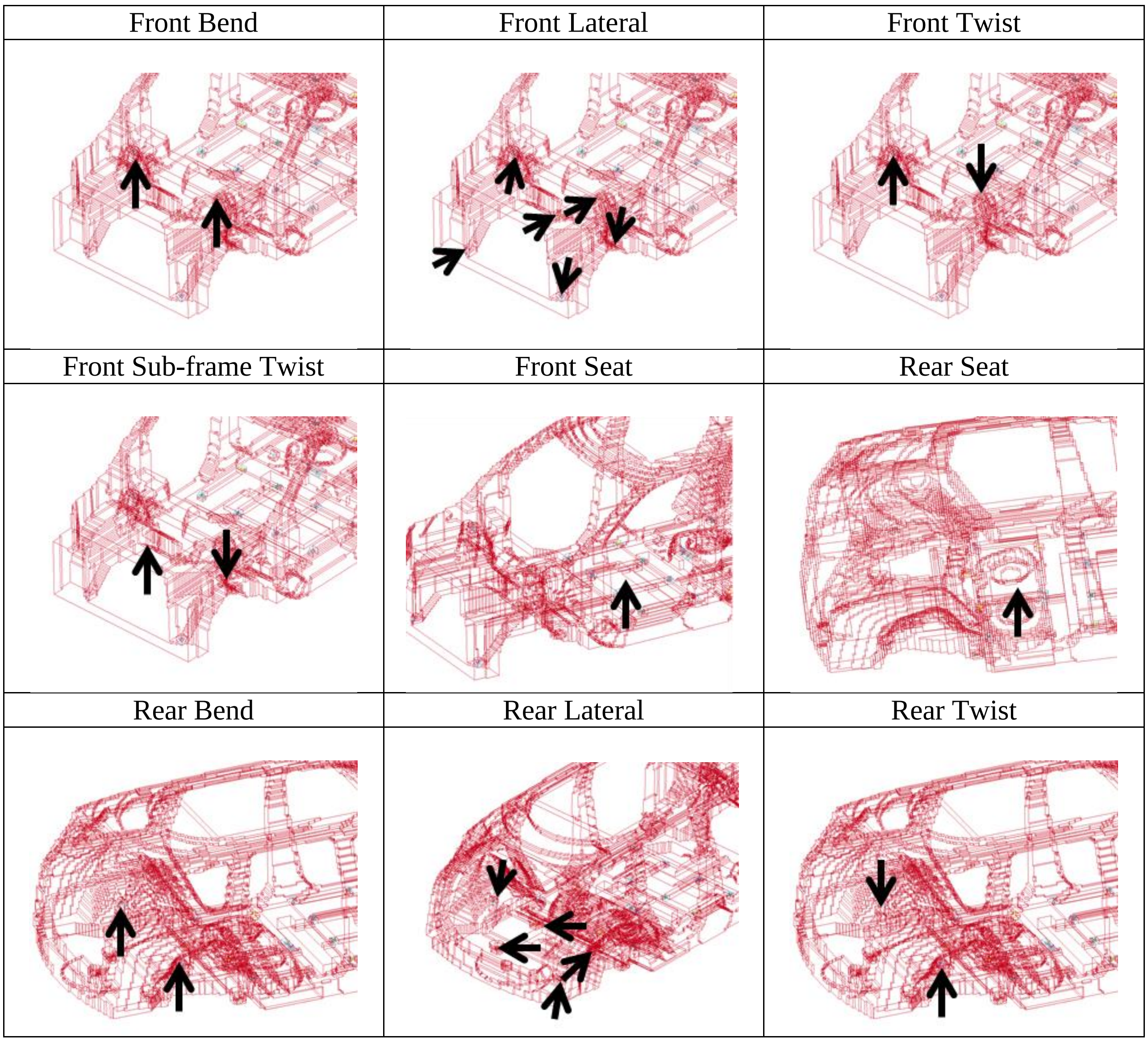

Figure 3: The nine static load cases.

The finite element model of the body structure is developed in LS-DYNA® keyword format and all load cases are analyzed with LS-DYNA. The crash load cases are simulated with the explicit solver of LS-DYNA for 30ms. The static load cases are analyzed an with LS-DYNA implicit. The design space is discretized by 217,520 solid elements with edge length of 20mm. Plastic deformation is captured by a piecewise linear elastic-plastic aluminum material model (keyword MAT_PIECWISE_LINEAR_PLASTICITY).

4. Results

The tool LS-TaSC [8, 9] provides a state of the art implementation of the Hybrid Cellular Automata algorithm for crashworthiness that was introduced in the theory part and is used for the topology optimization. By entering the correct load case weights it can be used as implementation for the SEW-HCA. For the purpose of concurrent optimization it is taken care that the material cards and the element numbering of the design space are consistent for all load cases.
The scaling factors are determined as in (9). Preference is distributed equally between crash and static load cases: $p_{\text{crash}} = p_{1,2} = \frac{0.5}{2} = 0.25,\ p_{\text{static}} = p_{3,4,5,6,7,8,9,10,11} = \frac{0.5}{9} = 0.056.$

HCA Parameter (LS-TaSC Setting)	Parameter Value
Target Mass Fraction	0.3
Neighbor Radius	35.26mm
Move Limit	0.1
Convergence tolerance	0.002

Table 1: Parameter Settings for LS-TaSC.

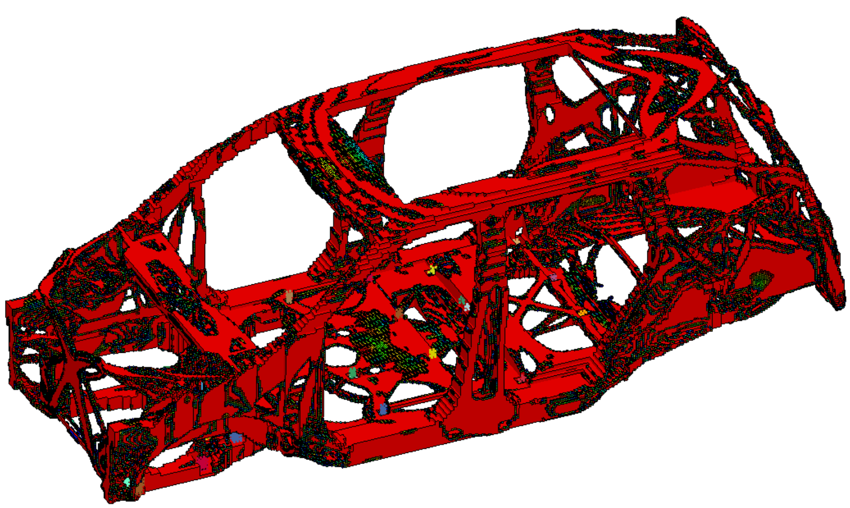

Figure 4: Resulting body-in-white concept from concurrent SEW-HCA optimization.

The parameter settings for LS-TaSC are given in Tab. 1. Typically the mass of a vehicle body structure is much lower than the mass of the considered solid structure when the target of 30%

mass fraction is fulfilled. Yet, in this study, the result is intended to serve as starting point for further design steps, for which structural features can be inspired by the obtained concept, hence the resulting design's mass is considered of minor interest for this effort.
Symmetry of the structure is imposed for the vertical plane along the longitudinal axis. An issue is the handling of minimum density elements. For the explicit crash analysis best practice is a removal of elements and a relatively high minimum density. Yet, for implicit analysis a removal can lead to instability, hence for implicit analysis it is preferred to keep the minimum density elements. In this optimization, a compromise was found by tuning. Concretely the minimum density was set to 0.005 and no elements were removed from the mesh.
On the available computational cluster, when all load cases are analyzed in parallel, this theoretically results in a runtime of roughly 30 minutes per iteration of the topology optimization. The static load cases are run on 96 CPUs, the crash loads are run on 192 CPUs.

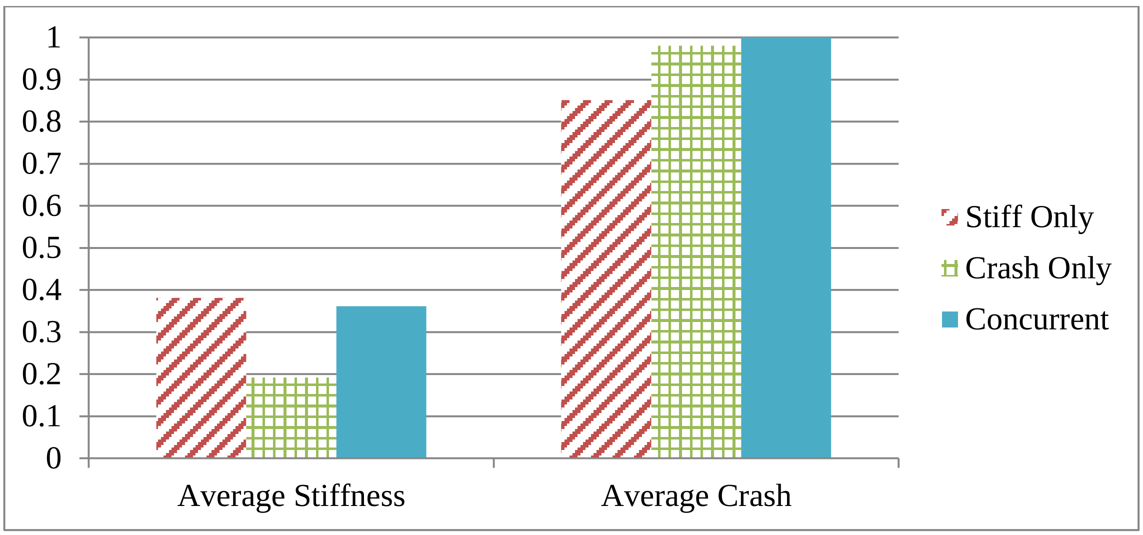

Figure 5: Average stiffness and crash performance of the optimization results

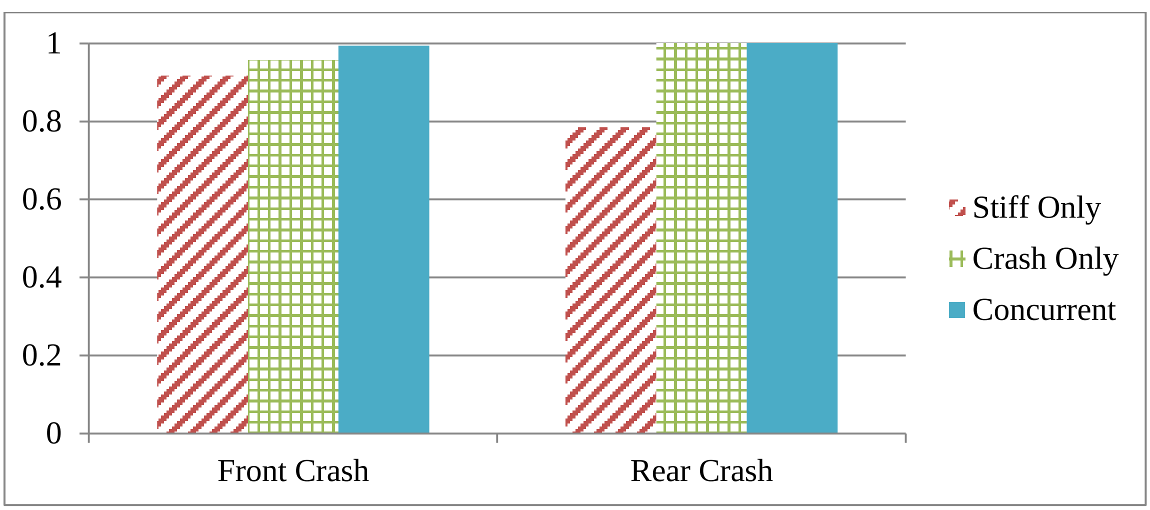

Figure 6: Front and rear crash performance of the optimization results.

Figure 4 shows the structural result, after convergence of the concurrent topology optimization with SEW-HCA. An interesting, organic-looking vehicle body design concept with the targeted mass fraction is obtained. The majority of elements converged to either full material or void and only a small amount of intermediate densities remains, that can be easily post-processed by thresholding. It is possible to identify design features, such as load paths or void volumes, utilizable in the further design process.

As baseline also a single load case optimization is performed for each of the load cases separately. The single load case optimization, ideally should yield the best possible result for the load cases, hence all other performance values are normalized to these baseline values. Besides the single load case optimization, also multi-load case optimizations are performed separately for all stiffness load cases ("*Stiff Only*" results) and for all crash load cases ("*Crash Only*" results), where weights are as well determined with the scaled energy weighting approach with equal preferences. The resulting solution from the concurrent optimization (Fig. 4) with SEW-HCA is referred to as "*Concurrent*".

The objective performance of the crash load cases is measured as the total energy absorbed by elastic and plastic deformations at the final time step. The objective of the static load cases is the compliance, i.e. the work of the structure when the static loads are applied. The compliance values are inverted to the stiffness measure, in order to obtain consistent plots, in which large values refer to better structures. In both cases, the concrete numbers used are the global internal energy values of the final time step of the LS-DYNA glstat output file.

The average performance on all crash and all stiffness load cases is shown in Fig. 5. The *Concurrent* result is the best trade-off of all load cases, since both, *Stiff Only* and *Crash Only* show strong performance drops when applied to the neglected load cases.

Figure 6 shows the results for the crash load cases. If both crash load cases are considered in the optimization, but no static load cases, the performance compared to the baseline drops only slightly for the frontal crash, for the rear crash almost the same amount of energy is absorbed. The *Concurrent* optimization result achieves a slightly higher energy absorption than the *Crash Only* result for the frontal crash. This is explainable if there is similarity of the (ideal) front crash solution with one or several of the (ideal) stiffness solutions. If the frontal crash can utilize material that is put in place by one or several of the static load cases, a better front crash structure will be obtained by coincidence, hence the structural similarity can possibly have a similar effect as choosing a higher preference. The fact that there is only a small performance drop compared to the baseline can be explained by the fact that the total kinetic energy in the system is limited, hence the best energy absorption can be reached also by structures that do not have a uniform energy distribution. The *Stiff Only* optimization performs worse considering the energy absorption, especially for the rear crash case.

Figure 7 shows the results for the static load cases. First, it can be noted that the baselines, as theoretically expected, clearly yield the stiffest structures for the single load cases. As expected, when all stiffness load cases are considered in a multi-load case optimization, the stiffness drops. On average, roughly 0.4 of the baseline's stiffness can be maintained by the *Stiff Only* optimization, as can be seen in Fig. 5. When the *Concurrent* optimization is performed, very similar stiffness values are obtained hence although crash load cases are considered, it is possible to obtain stiffness results almost as good as those of the *Stiff Only* optimization. In the case of the Front Lateral load case, it performs even slightly better, an effect is possibly (again) caused by similarity, in this case similarity of the (ideal) Front Lateral solution with one or both of the crash solutions. As a reference also the stiffness performance of the *Crash Only* optimization result is shown. On average it performs about half as good as *Stiff Only* or *Concurrent.* This validates that optimizing only for crash load cases does not yield the desired stiffness.

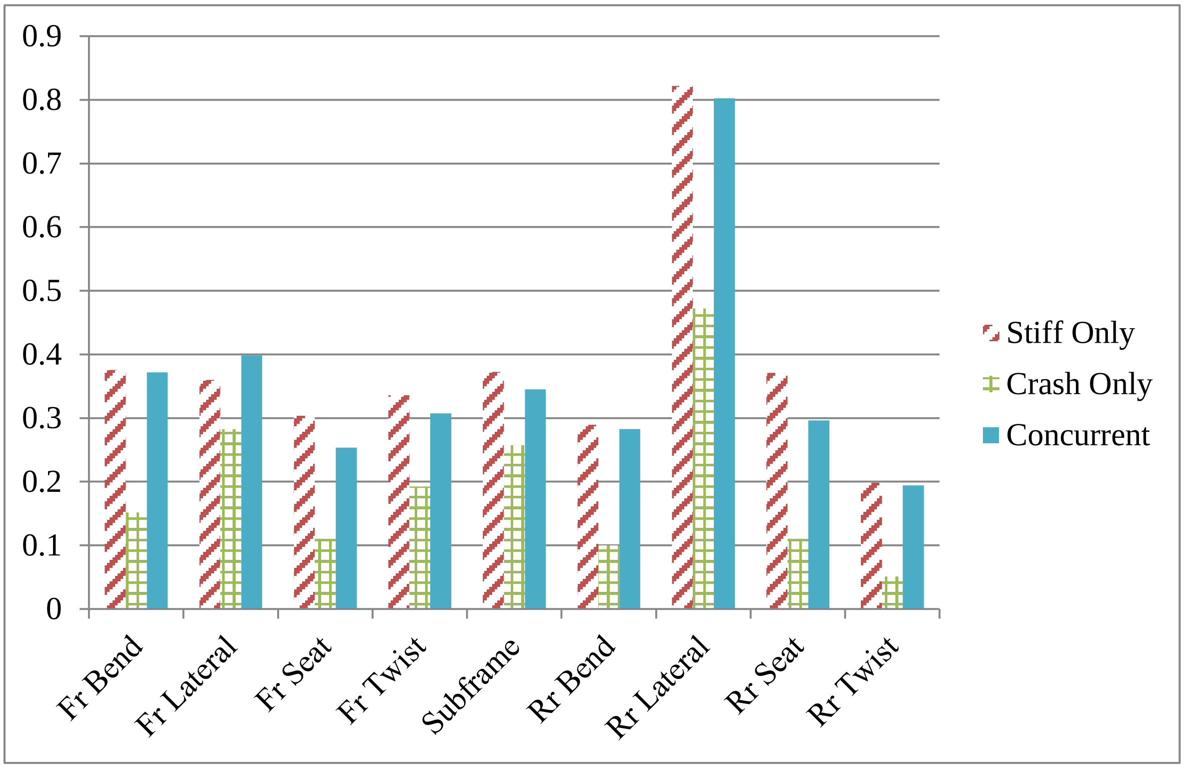

Figure 7: Stiffness performance of the optimization results.

Overall, the SEW-HCA topology optimization approach was successfully applied to the concept design of the vehicle body design space in a concurrent setup of crash and stiffness load cases. All load cases are considered by the optimization and SEW-HCA yields the best trade-off solution, when compared to optimizations that focus either on crash or stiffness load cases separately.

5. Conclusion

In this work, we present a case study of applying the topology optimization of structures subject to stiffness and crashworthiness requirements to a realistic vehicle body structure. In order to consider both types of loads in a concurrent topology optimization a scaled energy weighting approach was applied, that aims at leveling the energy field variables in a Hybrid Cellular Automata approach. It was demonstrated that the method, SEW-HCA, yields a good trade-off result in terms of stiffness and crash energy absorption for all load cases. The trade-off outperforms solutions that only consider one of the two disciplines. Practically, with respect to the large size of the analysis model and the high number of load cases no severe problems were encountered with the proposed method or the applied software tool, and hence these findings encourage the further and more systematic application of the proposed method in the industrial vehicle design process.

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