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# Optimization Approach for Crashworthiness of Vehicles Based on Physically Defined Equivalent Static Loads for a new Topology

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## ABSTRACT

Optimization Approach for Crashworthiness of Vehicles Based on Physically Defined Equivalent Static Loads for a new topology. The new method is based on principle energy considerations inspired from the current design process in modern automotive product development. The potential of the vehicle concept to absorb kinetic energy can be estimated at the very beginning of the design process by the free crash lengths in the different areas of the vehicle and estimates of average forces required in the specific parts of the car body at particular crash phases. Here it is important to determine appropriate force distributions and the corresponding load paths through the whole structure for all relevant crash load case. Depending on the vehicle type, a knowledge base is derived for the spatial and temporal distribution of force levels, which is then used in a temporal sequence of static topology optimizations. Standard topology optimization methods based on linear finite element methods are used in a time sequence where the loads are applied for certain time periods in special areas of the package assigned to the structure for different phases of the crash. For this it was necessary to derive distinct deformation phases characteristic for each crash load case and corresponding equivalent static loads, which reflect the actual transfer of loads through the structure.
**Keywords :** Crashworthiness, Topology Optimization, Equivalent Static Forces, Simplified Crash Modeling

## I. INTRODUCTION

### TOPOLOGY OPTIMIZATION FOR CRASHWORTHINES

**Mathematically defined nodal Equivalent Static Loads (ESL)** The method presented here may be regarded as a topology optimization for crashworthiness based on equivalent static loads (ESL). In contrast to the publications based on the original work on ESL (introduced in [1,2] and further developed in [3-6]), the method presented here is based on physically and not mathematically defined equivalent static loads. Park's group proposed to derive the ESL from non-linear displacements $u_n(t)$ obtained by a fully non-linear FEM crash computation. Here $u_n$ represents the 3D time dependent field of nodal displacements. For a pre-defined set of times $t_i$, $i = 1...m$, the equivalent static (i.e. linear) loads $f_{eq}(t_i)$ are generated by multiplying the nonlinear displacements with the linear static stiffness matrix $\boldsymbol{K}_{lin}$. Thus, for each time $t_i$ we get equivalent force vectors for each node of the FEM mesh. The optimization is then realized by a double loop approach where the inner loop uses the ESL to optimize the linear and static FEM problem and the ESL are updated after the inner optimization is finished by a new non-linear computation based on the optimal design variables obtained in the inner loop. In case of dynamic problems, the inner loop optimization must use $m$ linear FE computations. Because this approach is still very new, further research is needed. Crash simulation is normally based on the deforming geometry, i.e. it is questionable if the undeformed stiffness matrix used in this approach is the appropriate choice. Furthermore it is assumed here that the linear optimizations (inner loop) point in the direction of the non-linear optimum. This might be only justifiable for problems with small non-linearities. The approach is very sensitive to the nodal equivalent forces, a study in the frame of an ongoing MTech thesis [7] showed that this approach is difficult to apply in cases where the FE geometry is re-meshed or in cases with high non-linearity [8]; a mapping strategy failed and needs further improvement. For topology optimization, it is a challenge to take into account that some elements are deleted in the optimization process (inner loop) and that the nodal equivalent forces defined on these elements need to be re-distributed. Concerning crash, further studies are required how to include dynamic criteria into the static FEM computations. If for example a structure should be improved with respect to the injury risk of pedestrians, the head injury criteria (HIC) should be regarded, which is derived from a sliding window integration of the acceleration in the centre of gravity of the free flying head form, e.g. [7]. Is it possible to define this in the frame of the mathematical ESL method? These issues and the very detailed nodal ESL (topology optimization is used in the early design phases) with its lack of physical interpretation of these loads may motivate the search for a more robust and more flexible approach.

**Hybrid Cellular Automata (HCA)**

A second approach recently published is based on hybrid cellular automata (HCA), first presented for crashworthiness in [9,10]. This approach uses a regular grid of material cells with a certain set of characteristic values, which are updated considering similar information from the neighbor cells via particular update rules (principle of cellular automata). To reduce the enormous numerical effort, the local neighbor data is enriched by global information obtained via FEM (principle of hybrid cellular automata). Hence this method is in a certain way similar to the approach mentioned above; the design space is filled with 3D elements (voxels). Then the model is updated by successive deletion of voxels via a modified density approach adapted to non-linear material modeling. In some cases elements can be reactivated to recover feasible solutions. Compared to the mathematical ESL method mentioned above, the HCA is based on non-linear explicit FEM (non-linear with respect to material, geometry, contact etc.) and does not use linear computations. Objective of this gradient-free method (see [11] for a discussion) is to achieve a fully stressed design. A gradient-based method seems currently not possible because of the inherent numerical noise and the non-availability of gradients in non-linear crash simulations, e.g. [12]. In the case of crashworthiness a homogeneous distribution of plastic deformation energy (more exactly the internal elastic and plastic energy density without rebound) is taken as objective. A detailed discussion of the appropriateness of the homogeneity objective is still missing. The well established analytical theory for estimation of energy absorption of thin-walled structures, e.g. [13-15], shows clearly that the deformation is driven by the formation and movement of local hinges or hinge lines. This would mean that a good performance is not related to homogeneous distribution of plastic deformation energy in case of plastic buckling. The regional strain energy formulation proposed by [16] may be an alternative, although the definition a-priori of the energy absorption zones has to be discussed further, see as well [17]. Additionally, the attempt to model crash by bulky components using 3D volume elements is not always representing the behavior of typical automotive structures in impacts. Most of the structural parts of current models used in industry consist of thin walled panels or members with hollow cross-sections. A typical plastic deformation mode is due to plastic buckling, which is difficult to obtain by voxels. Here, a very fine mesh (high computational effort) is required or the method in general might fail to reproduce the correct behavior. Hence the applicability of the original HCA approach is restricted to cases where these two remarks are not relevant, i.e. in cases, which are driven by bending and not plastic buckling, and in cases where more bulky structures are analyzed. To overcome the first difficulty, studies have been published on the usage of the HCA for problems formulated with shell elements. Mozumder transfers the approach of [9] to topometry optimization, where a shell-based sheet metal structure is optimized with respect to the optimal material thickness distribution under a dynamic loading [18]. Hence a specialized (i.e. 2D) problem is solved here, which is nevertheless very interesting for a group of applications. More recently, [19] published a modified HCA approach, which solves a real 3D topology optimization problem. Here the original design space is defined by hollow boxes with thin walls and thin inner reinforcements. Contrary to the standard HCA approach, complete walls/reinforcements are successively deleted in the process. By this approach, the plastic buckling of thin walls is considered and plastic deformation modes similar to those encountered in car bodies are possible. This adaptation of the HCA to shell structures combines the ground structure approach (see next section) with the density/homogenization approaches. These two modifications of the HCA seem to be promising although they do not solve the problem of the homogeneity objective. Hence these methods might be more applicable for local topology optimization tasks where the topology of a component of the car body is improved (e.g. cross-section). As a summary of this part of the state-of-the-art in crash topology optimization, it can be concluded that we need different methods for the different fields of topology optimization. Mathematical ESL and HCA seem to be better suited for component topologies whereas a more global approach valid for the derivation of complete vehicle concepts is needed.

**Topology optimization by a single set of static loads**

There are a larger number of publications using static loads to represent crash load cases, e.g. [20-22]. All more or less address the full car problem and not the component topology optimization. Christensen et al. [23] presented a detailed study on new car concepts based on this idea where linear elastic FEM and commercially available tools for topology optimizations are used considering six crash load cases and some variations in the load conditions. Due to the completely static character of the investigation inertia effects have not been included. An example of the static loads considered in this study and the result of the topology optimization is given in Figure 1. A comparable result was derived in the Future Steel Vehicle project (www.futuresteelvehicle.org) where Evolutionary Topology Optimization as a special case of evolutionary structural optimization (ESO, e.g.[24]) was used to derive the car body concept, see Figure 2. Regarding these results for a full car body topology optimization based on a single set of static loads, it remains an open questions how to represent the different phases of the impacts during a crash test. The loads should change over time taking into account the different contact situations. New forces are generated for example in a frontal offset test when the wheel impacts on the rocker or the engine hits the firewall. Which forces should be applied in which crash phase? Hence this approach needs modifications which are discussed in the following sections.

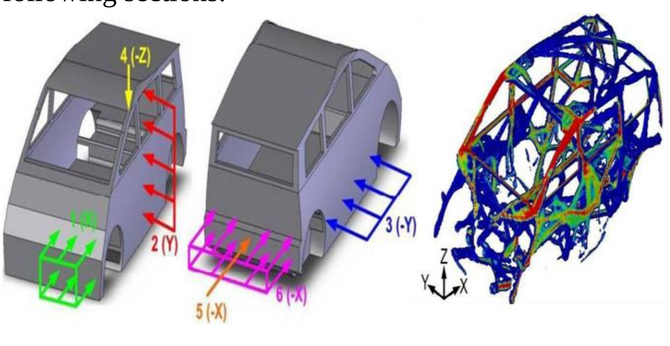


**Figure 1:** Equivalent static loads and optimization result [25].

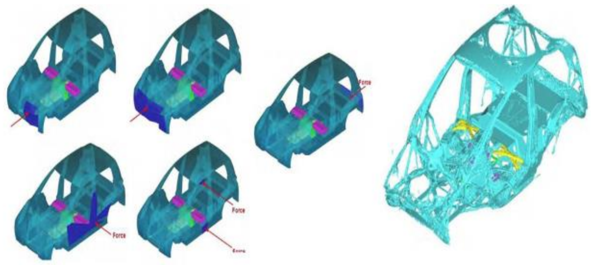

**Figure 2:** Equivalent static loads and optimization result for the Future Steel Vehicle.

The analysis in this first part of the paper of existing topology optimization approaches led to the insight that the following two types of topology optimizations are useful for crashworthiness problems:

Type 1: Topology optimization for full vehicle problems identifying load path concepts in the very early design stages. Here approaches based on a single set of static loads are proposed in the literature.

Type 2: Topology optimization for component problems identifying local topologies like cross-sections or reinforcements. Here approaches based on the mathematical ESL method or the non-linear HCA method were published.

**Other approaches for crash topology optimization**

To finalize the state-of-the-art, a rough overview of other available methods for crash topology optimization is given. To the author's knowledge, the first published work on topology optimization for crash is the work of [26,27]. They used a homogenization method where the size of holes in the cells was adjusted to fulfill the objective of maximal internal energy at the final time step. Heuristic weighting factors were used for modification of the objective and non-linear FEM computations are employed to compare the different results. This approach resembles in a certain manner the methods discussed above (density approaches) although the heuristic rules made it difficult to generalize the approach. A better justification of the structural update and the weighting factors might be needed. Then Soto [28,29] presented a heuristic methodology that as well did not require sensitivity information and was based on non-linear FEM. By prescribing distributions of plastic stress and strain, a heuristic rule was used to vary densities for optimal design. Continuing this work, Forsberg et al. [30] discussed two alternatives for crash topology optimization based on successive element elimination or thickness reduction to achieve a homogeneous internal energy density distribution (again: is this the correct objective ?). Here non-linear effects (plasticity) were considered via true non-linear explicit FEM (only 2D plane stress problems were published in the original paper). Again it is a sensitivity-free approach. Compared to the HCA approach, no neighborhood information is needed here. The examples chosen in the paper are cases where the deformation is mainly due to bending (tension / compression problems as stated by the authors) and no local buckling occurs. Hence the validation of these approaches for the axial load case of a member is still missing. In the bending cases, the structure mainly needs strong load paths to resist the impact forces and no energy absorption due to buckling of thin-walled shell structures is required and modeled. Contact is not an issue here. Hence - as shown in the paper - the topology results are rather comparable between the linear elastic and the non-linear elasto-plastic case.

A rather different approach to derive a frontal structure of a car body was based on the so-called ground structure approach proposed by Pedersen [31-34], where the relatively simple design space was filled initially with beam structures

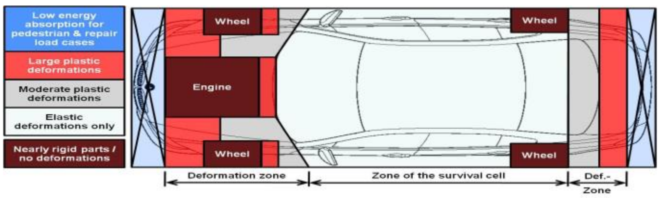


**Figure 3:** Principal crash deformation zones

which were then successively deleted. This may be of interest for principal studies but it might be not flexible enough to be applied in realistic industrial environments where the design space is more complex and the design freedom at the beginning is not sufficiently big to justify this approach. Other groups published comparable results where a discrete beam structure is used to derive concepts in the early design phases. More publications are not provided and discussed here because this paper focuses on continuous and not discrete modeling. Further approaches to crash topology optimization can be found for example in the works of Schumacher and co-workers where a bubble method and a graph method are analyzed. In addition some parameterization methods used for concept design offer the possibility to optimize at the same time size, shape and topology. In particular, the implicit parameterization technique implemented in the software SFE CONCEPT (www.sfe-berlin.de) enables fast shape and topology modifications, which can be used in optimization approaches.

## NEW TOPOLOGY OPTIMIZATION APPROACH FOR CONCEPT DESIGN

### Principal idea

The product development process (PDP) in the automotive industry can be roughly divided into two main phases. In the first phase, the "Concept Development", several concepts are evaluated until the design is frozen and the second phase, the "Series Development", is realised. Topology optimization of the first type discussed in the section above (the full body problem) addresses the question how to derive ideas for good concepts in this very early stage of the PDP while the second type on component level might be used slightly later when the main structural load paths are already decided. The starting point for the first is often the definition of a coarse package, which is determined by the type of the car concerning usage (SUV, convertible, sedan, etc.) or motorisation (electric, hybrid or conventional). This can already be quite complex when vehicle families have to be considered in the optimization respecting constraints from platform considerations. In this package, the basic dimensions (wheel base, overall length, engine size and position, etc.) are defined, which leads to the identification of the principal deformation zones of the vehicle (as shown in Figure 3):

Zones with low force levels for energy absorption in pedestrian or repair crash tests.
Zones with high energy absorption due to available deformation space (free crash lengths).
Zones with moderate energy absorption capability due to limited deformation ability.
Zones with no energy absorption capability because of more or less rigid structures.

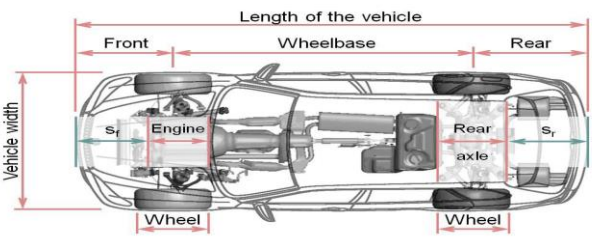


**Figure 4:** Free crash lengths *sf* and *sr* (simplified).

These zones lead to the rough estimate of free crash lengths *sf* in the front and *sr* in the rear (Figure 4), where most of the energy can be absorbed. It is a sum of the high energy zones in Figure 3. This quantity is one of the main parameters for concept design. Multiplied with an average force in the corresponding structural part it leads to an estimate of the energy absorption capability in this area of the structural concept. This energy should be equal to the kinetic energy from the crash test definitions, for example *E*kin = *mv*2/2 with *m* as the total vehicle mass and *v* as the test velocity. It should be noted here, that this approach is too coarse and needs refinement, which should be case specific and is given in the next section for the rear impact example.

**Physically Defined Equivalent Static Loads for the Rear Impact**

In a first step, the kinetic energy of a rear impact crash test should be determined, which is here based on the definition of the FMVSS 301 test (see www.nhtsa.gov/cars/rules/import/FMVSS), which assesses the integrity of motor vehicle fuel systems to reduce fire risk and related fatalities and injuries. A moving deformable barrier (MDB, mass *m*b = 1386 kg impacts the vehicle at rest with an offset of 70% and a velocity of 0 *b v* = 80 kph as shown in Figure 5. The corresponding kinetic energy of the barrier before the impact is 0 *b,kin E* = 338 kJ. Both kinetic energies after the impact have to be estimated based on the experience from predecessors and on the planned vehicle mass. In the example regarded here, the in-house data-base of BMW shows that a total vehicle test mass (including cargo, luggage and occupants according the official test definition) of *m*v = 2645 kg was in

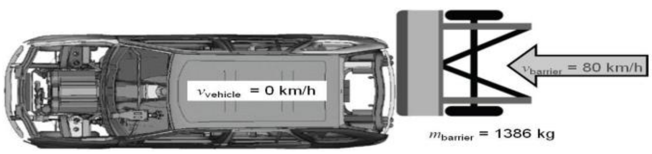


**Figure 5:** Rear impact test as defined in FMVSS 301.

The past often related to a barrier velocity after the impact of *v*b = 0.3 0*b v* = 24 kph, i.e. the kinetic energy of the barrier after the crash is *b kin E* , = 30.8 kJ. The kinetic energy of the same vehicle after the impact is roughly *v kin E* , = 105 kJ (final vehicle velocity of *v*v = 32 kph). The deformation of the barrier consumes approximately *v def E* , = 20% 0*b,kin E* = 67.6 kJ of the total energy.

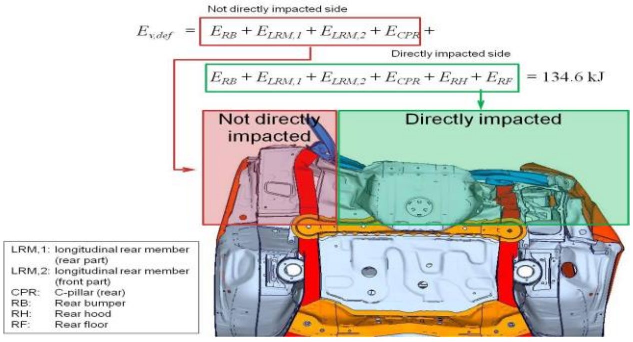


**Figure 6:** Main vehicle parts for the offset rear impact.

This energy is consumed by the different parts of the rear. Its distribution needs to be clarified, i.e. we have to determine the corresponding pairs of deformation lengths (taken from the package concept as shown in Figure 7) and force levels. For this, the car body structure has to be divided into their main parts as shown in Figure 6. It should be noted that due to the offset of the barrier impact, the left and the right vehicle sides are treated separately.

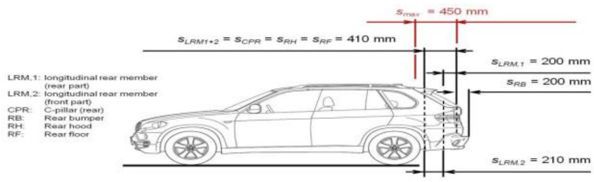


**Figure 7:** Free crash lengths for the rear impact.

The term related to the bumper is determined by the repair test (e.g. the AZT test of the Allianz Zentrum der Technik). Here a low-speed impact is used with a kinetic energy.

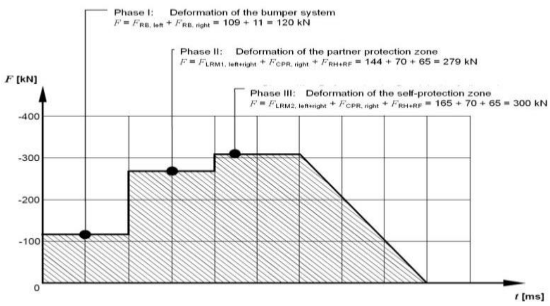


**Figure 8:** Crash phases for the rear impact.

For the topology optimization via physically derived equivalent static loads, the average force levels from this table are taken, which have to be distributed over certain areas of the design space. It is recommended to also consider loads on the not directly impacted side (e.g. 10% of those of the impacted side on the main structures). In contrast to the approaches from the state-of-the-art (Section 1), it is crucial for the success of topology optimization for crash to consider the different crash phases in time. The loads of Table 1 should not be applied in one single static computation. The rough time dependency of these forces is given in Figure 8 and should be used in the topology optimization. Finally, the geometrical distribution of these forces for each phase has to be determined. Because the global vehicle package is normally quite restrictive, the flexibility in topology design is limited. In the example regarded here, two main areas for energy absorption were pre-defined (area 1 and 2). Figure 9 summarizes the areas for crash phase 2.

**Topology optimization of the example**

To validate the approach via the newly defined equivalent static loads, a reference car model is taken where the rear part is replaced by volume elements (voxels) for the topology optimization. To assure that in this space the structure is roughly distributed equally in axial direction, three design spaces are defined with an objective of 25% final volume (Figure 10). Because a standard static topology optimization is used, the compliance of the total structure is minimized. In addition minimal member size is respected and symmetry of the design is enforced. The results are given in Figure 11.

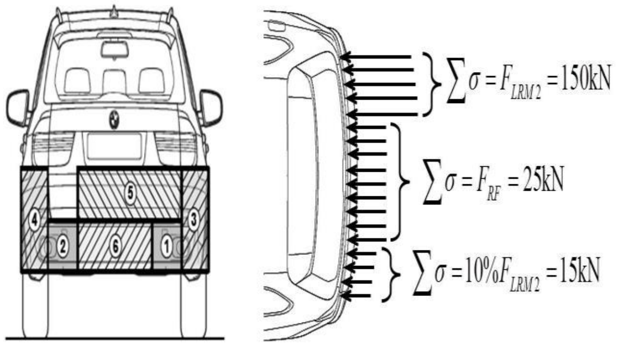


**Figure 9:** Load distribution for the rear impact (phase 2),

The new design (after interpretation and realization) showed a better performance in the FMVSS 301 rear impact than the reference design taken from the standard series development process (Figure 12), i.e. the plastified zones appeared more in the rear and showed higher energy absorption, which is more conform with the design rules. In particular, the plastification in the area of the rear axle occurring in the development stage of the reference design was avoided. Hence the risk of damage in the tank area and therefore of fuel spillage is reduced by the topology optimization. By establishing a new load path, all relevant force levels were achieved leading to a fulfilment of the requirements like door opening after the crash, low intrusion in critical areas and lower injury risk (whiplash).

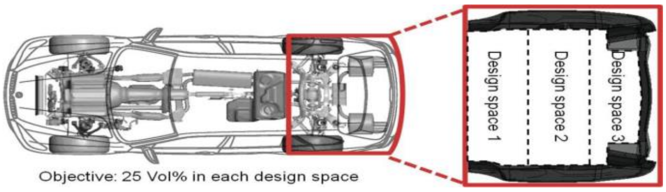


**Figure 10:** Three design spaces for the validation example.

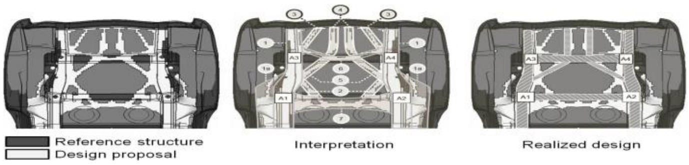


**Figure 11:** Optimized topology compared to the reference design and its interpretation.

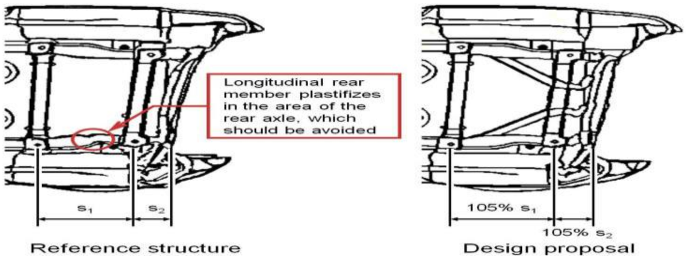


**Figure 12:** Performance of the optimized topology compared to the reference design.

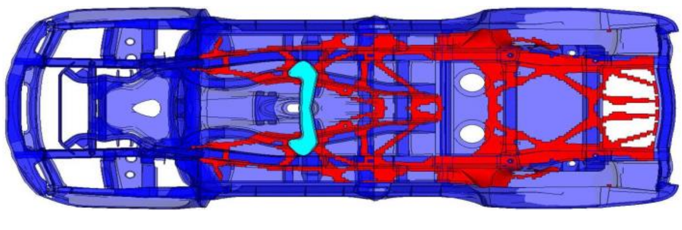

**Figure 13:** Topology optimization result (front, side and rear impacts).

## II. CONCLUSIONS

A new approach to define equivalent static loads for crashworthiness was presented and used in a topology optimization for the high-speed rear impact of an industrial-sized example. Compared to other approaches (Mathematically defined ESL or HCA method), this method seems to be able to address better full vehicle problems. In addition it can also be used to optimize a part of the vehicle embedded in the full model, which is an advantage reflecting the need to integrate these methods and tools into modern product development processes. Compared to recently published studies also based on standard topology optimization methods and linear FEM, the work presented here uses physical considerations to derive several sets of static load distributions in space and time, which can then be used sequentially for topology optimization. In an example for the rear impact (FMVSS 301), it was shown that this approach helps to identify load paths and corresponding structures to minimize injury risks and maximize energy absorption in dedicated areas by principle package considerations. As shown in the recently finalized thesis [40], this approach is as well able to handle multi-load cases scenarios (front, side and read impacts), a result is given in Figure 13.

It should be noted that the approach here is based on linear finite element results. Hence in areas with strong plastic deformations (material non-linearity) and in general large deformations (geometrical non-linearity) the modelling might represent the force levels and load paths approximately but will not represent local buckling etc (some global contact conditions can be integrated into this approach if they are considered by a special set of ESL; local contacts are more difficult). Hence if the local topology of a structure, e.g. the cross-section, needs to be determined, this method will not help to identify the best topology. Here truly non-linear and more local approaches should be used. For example, if the topology of the bumper beam needs to be determined with respect to the performance in a high-speed crash, the authors believe that their method is inappropriate. The objective of homogeneous energy density based on compliance or stiffness does not correspond to local buckling with the formation of plastic hinge lines. The question of identifying load paths through the global structure is nevertheless in a certain way independent of how the force-deformation curve is generated and linked to local buckling modes. Important for this approach are mainly the force level and the transfer of forces through the structure. The work presented here is embedded in further research activities like realizing a chain of optimization steps by coupling sequentially topology and shape optimization ([43,44,45]) to ameliorate the design by more detailed geometry adaptation. Additional literature on crash topology optimizations can be found in [46-53].

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