A Systematic Study on Topology Optimization of Crash Loaded Structures using LS-TaSC™

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1 Introduction

Using topology optimization methods, new structural concepts can be generated. These methods are efficient in the field of structural design, taking into account linear structural properties and linear static loading conditions. Usually the mean compliance is considered, subject to a mass constraint. Therefore, the design space is divided into small volumetric elements (so-called voxels) and the algorithm decides based on an analytical sensitivity for every voxel, is there material or not. After this optimization, the engineer has a good proposal and the possibility for the interpretation and the generation of a CAD model.

1.1 Topology optimization of crash loaded structures – state of the art

When crash load cases have to be considered, the special characteristics of the highly nonlinear dynamic crash problems have to be taken into account. Large deformations and rigid-body motions occur during a crash incident. The used material laws are mostly nonlinear because the kinetic energy is transformed to plastic deformation. For the correct prediction of the material behaviour, strain rate dependencies and complex failure criterions have to be considered. The majority of the forces is transmitted via contact. Because of missing sensitivity information, pure mathematical methods are not able to consider all these necessary nonlinearities in topology optimization of crash loaded structures. Various approaches exist to overcome these difficulties. The three following methods represent the current state of the art.

Ortmann and Schumacher [1] developed a graph and heuristic based topology optimization, which use finite shell elements instead of a voxel mesh. A complex CAD activity is necessary and is realized by mathematical graphs. The related optimization software involves several rules based on engineer expert knowledge and can consider different crash relevant objective and constraint functions.

The equivalent static loads method, Park [2], is a voxel-based topology optimization method, which uses static methods for the optimization. After a dynamic simulation, the nodal forces that would lead to an equivalent displacement with purely elastic material behaviour are determined. With this load, the displacements are minimized in a linear-static topology optimization. By means of multiple load cases during the static optimization, different time steps of the dynamic simulation can be optimized simultaneously. A static optimization is followed by a dynamic simulation until the densities converge.

With the hybrid cellular automaton method (Patel et al. [3]), the internal energy density of a structure gets homogeneously distributed by heuristics derived from biological growth rules. The cellular automaton describes a discretization of the space with voxels. The state of a voxel, in this case its density, is controlled from iteration to iteration by transition and neighbourhood rules.

1.2 Content of this contribution

Based on the hybrid cellular automaton method, LSTC developed the software LS-TaSC™ [4]. This software runs with LS-DYNA® and is used for the studies in this paper. With two demonstrative applications the influence of scaled loads, intermediate densities and conflicting constraints are discussed. As no analytical sensitivities are available in explicit finite element simulations, LS-TaSC™ uses the internal energy density as sensitivity, which makes it an efficient optimization strategy for stiffness applications. This paper deals with the question, what is the behaviour of the voxel approach for topology optimization depending on crash relevant objectives in LS-TaSC™?

The first provided example is an academic one. A maximum stiffness as well as a crashworthiness design is developed. In the latter case, the multipoint method of LS-TaSC™ is used to cope with the acceleration of an impactor as objective function. The second example has a high applicability. The optimal topology of a vehicle rocker under pole impact is sought. Following the topology optimization with LS-TaSC™, the design proposal is realized in a shell model, whose wall thicknesses are optimized.

2 Material model

In the presented applications a nonlinear material model of type 24 is used with piecewise linear isotropic hardening but without damage mechanism. The particular characteristic values of the EN AW-6005 T6 aluminium alloy for extrusion profiles are a yield strength of 240 MPa, a density of 2.7×10^{-6} kg/mm^3 and a young's modulus of 70000 MPa. As the topology optimization method of LS-TaSC™ uses intermediate densities, the scaled stress-strain-curves are visualized in Fig. 1. The artificial element density of the *i*-th element is designated with x_i, which equals the ratio between the scaled density and the original material density.

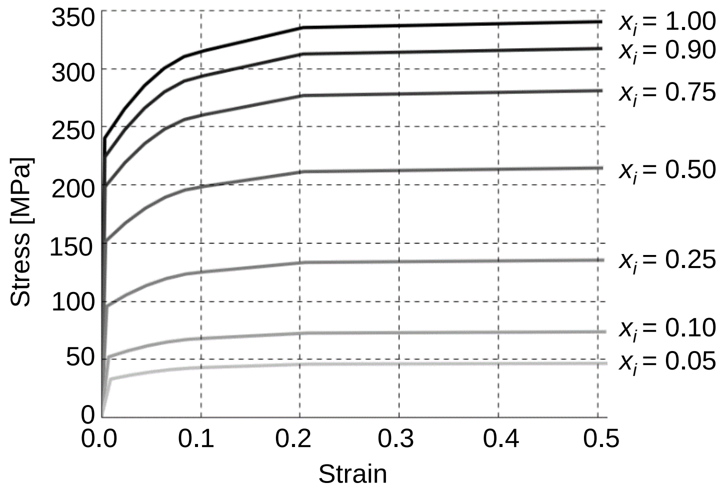

Fig.1: Scaled stress-strain-curves for intermediate element densities

At low densities the yield strain increases and the hardening tends to zero. That means, after reaching the yield stress, the elements stretch with nearly no resistance. By default, elements with an artificial element density $x_i < 0.05$ are not considered in the simulation to improve the numerical stability. By using a filter for the internal energy densities, once deleted elements can be regenerated if the adjacent elements are highly loaded.

3 Application 1 – "cantilever impacted by rigid sphere"

The first regarded example is a cantilever, fixed on the left side and impacted by a sphere of mass m = 0.1098 kg with a velocity v_0 = 10 m/s as shown in Fig. 2. The cantilever has an extrusion depth of 5 mm in *z*-direction.

The intrusion of the sphere is calculated using the Finite-Element-Method in LS-DYNA® with an explicit time integration. The cantilever model is meshed with regular hexahedrons with an element edge length of 1.0 mm and the solid element formulation 2 (fully integrated solid), whereas the sphere is meshed with tetrahedral volume elements and consists of a rigid material. A non-design-space with a width of 1 mm is defined at the clamping side and at the bottom side, where the sphere hits the cantilever. Apart from that, the whole body is defined as design space. Extrusion direction is the *z*-direction.

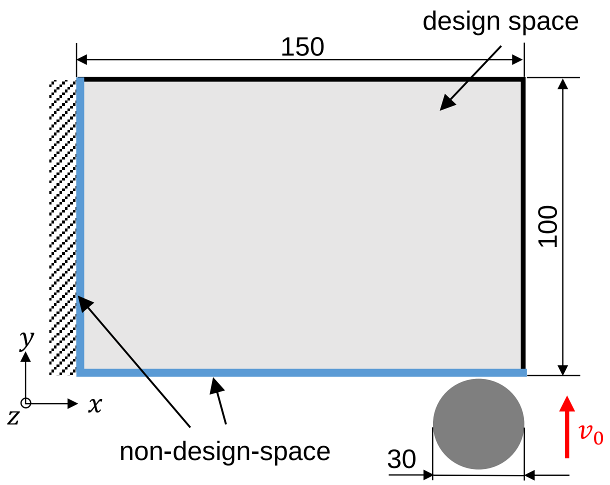

Fig.2: Application 1 – cantilever impacted by rigid sphere

3.1 Optimization task A – stiffness design

The first optimization task is the minimization of the intrusion of the sphere in *y*-direction. This will result in a stiff structure. The mass fraction of the design space has to be 10 %.

3.1.1 Topology optimization with LS-TaSC™

The default LS-TaSC™ filter radius is used, which means that all neighbouring elements, that share at least one node, will be used to filter the internal energy densities. The optimization ends, when the mass redistribution falls below 0.002 or when the maximum number of 100 iterations is reached.

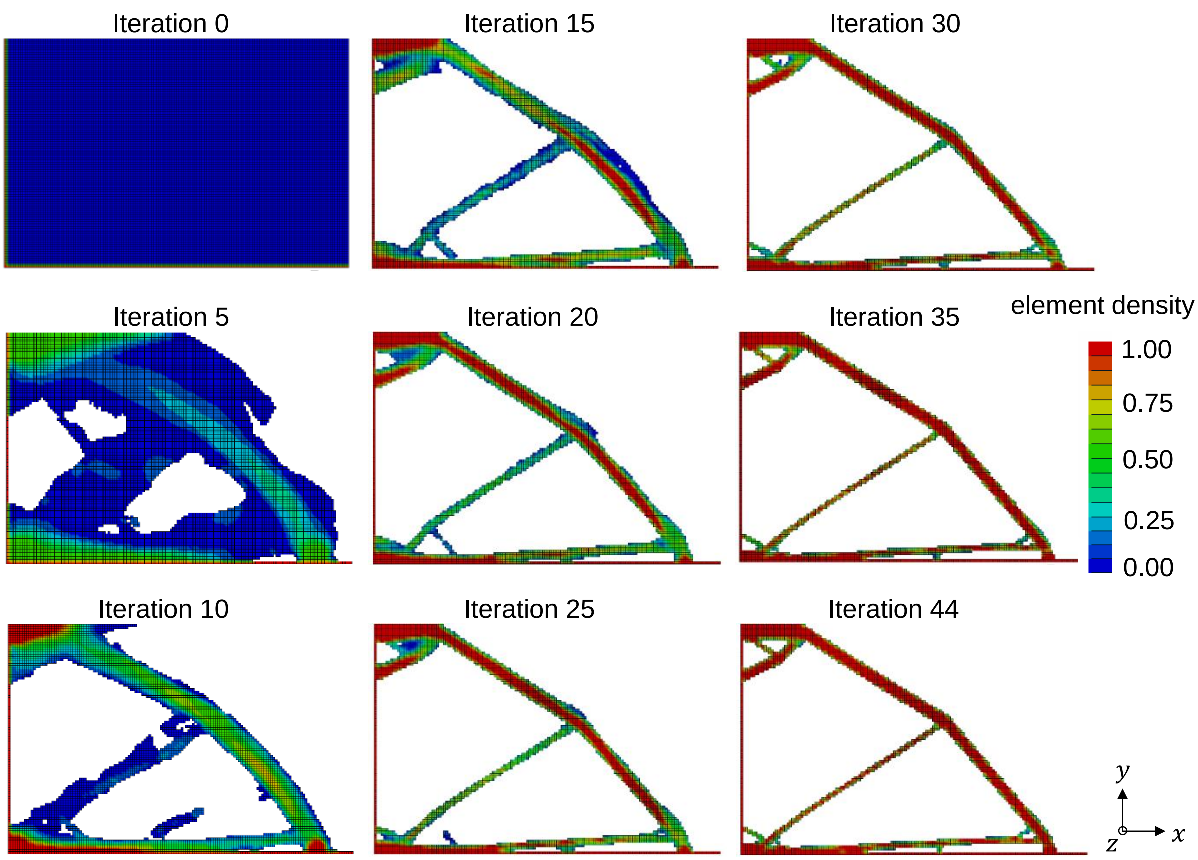

Fig.3: Optimization history of cantilever task A: stiffness design

The optimization converges after 44 iterations, while the mass is constant over the iterations. The history is shown in Fig. 3. The result is close to a black and white design, where only few elements with intermediate densities appear. Only some shorter walls with intermediate densities have developed. The structure resembles a framework structure like a linear elastic design proposal.

3.1.2 Structural behaviour of the hexahedral model

As shown in Fig. 4b, the walls under compression and tension work perfectly together as a framework. The location, where the sphere hits the frame, plasticizes under load. So the wall under tension works after the plasticizing and therefore it is not exactly horizontal. The resulting displacement of the sphere in *y*-direction of 3.74 mm is mainly driven by the plasticizing on the left side of the lower non-design-space at the location of the highest bending moment, cf. Fig. 4a.

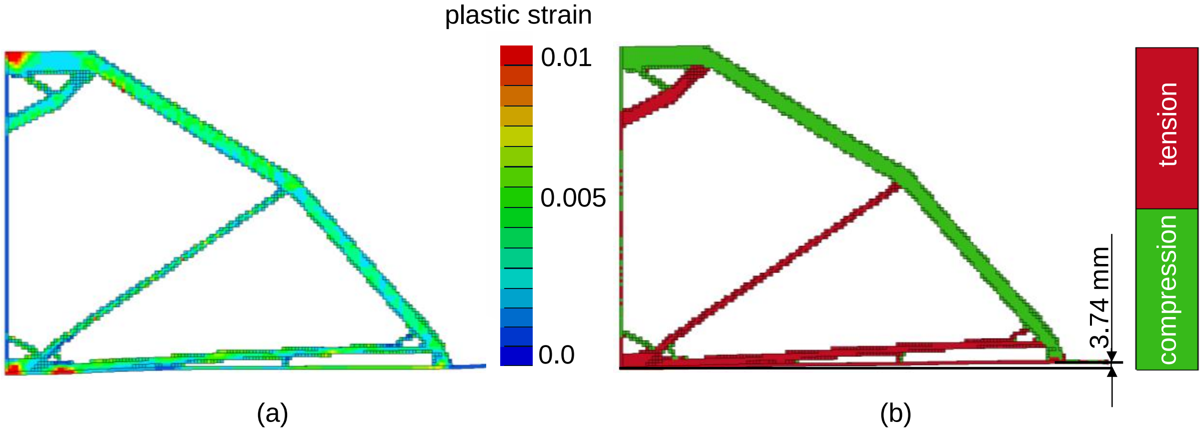

Fig.4: Structural behaviour of the hexahedral model of the stiffness optimization

3.1.3 Shell interpretation and sizing

As the result of the topology optimization with hexahedrons in crash loaded structures can only be a design proposal, a thickness optimization of a shell interpretation with the same mass of 23.3 g and the same objective function is performed. The 18 design variables are derived from the sections of the walls. The lower bound for the thickness is set to 0.2 except for the variable t_1 (cf. Fig. 5a), which has a minimum thickness of 1.0 mm, because it is part of the non-design-space. As there appear no additional mass accumulations along the non-design-space, only the thickness t_1 is a design variable. For the global optimization, an evolutionary algorithm and for the following local optimization, a gradient based method is used. The resulting thickness distribution, presented in Fig. 5a, is found in good accordance to the hexahedral model. With this optimization, a maximum *y*-displacement of 2.26 mm is reached.

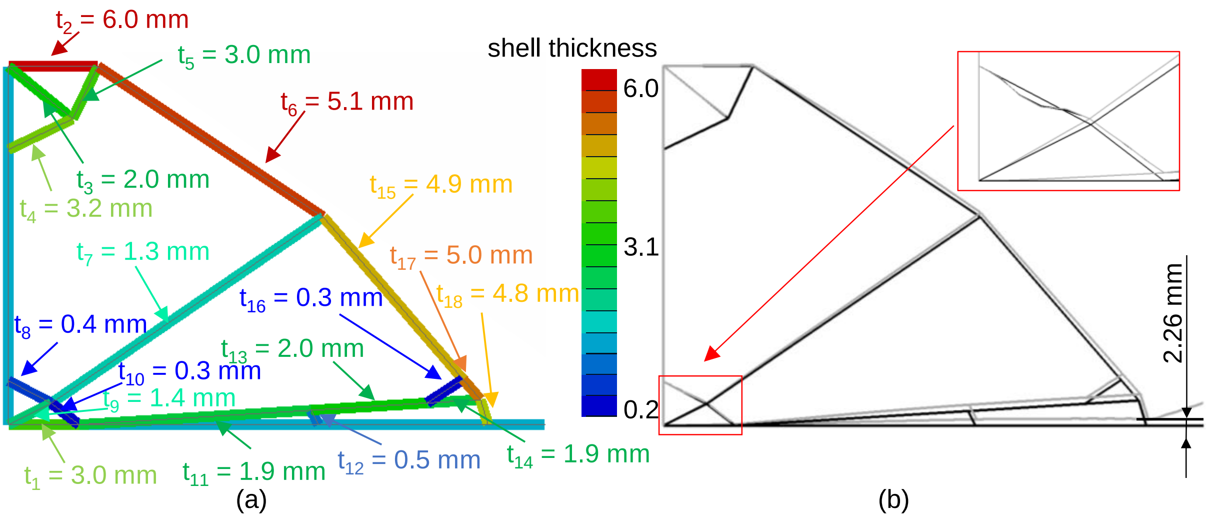

Fig.5: Optimized thickness distribution of the shell interpretation of the stiffness optimization (a), comparative deformation view (b)

The structural behaviour is slightly different with the optimized shell interpretation (cf. Fig. 5b). The deformation comes from the buckling of wall number 8 but not from the plastic strain at the clamping area.

3.1.4 Scaled loads

As the mechanical behaviour of the resultant design proposal is similar to the one of an elastic load case except for plasticizing, a further investigation is motivated, whether a scaled load would lead to a different topology. Therefore the velocity of the sphere is increased to 20, 30 and 40 m/s.

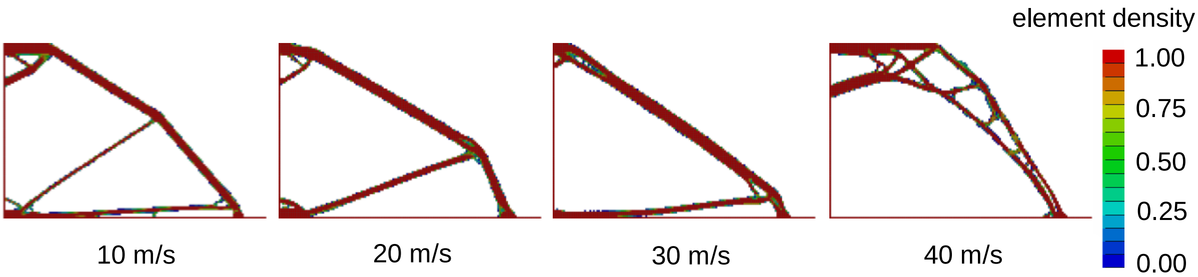

Fig.6: Optimal topologies derived with higher velocities

The resultant topologies are shown in Fig. 6. A cross checking of the topologies with the other velocities shows, that the topologies indeed work best for their deriving velocity, cf. Tab. 1. Moreover, the structure of the 40 m/s-load case is the only one, which has the structural integrity for this load.

		maximum displacement sphere in *y*-direction [mm]			
		optimized topology for velocity [m/s]			
		10	20	30	40
v_0 [m/s]	10	3.738	3.840	3.784	4.908
	20	10.774	7.346	7.757	11.419
	30	-	28.834	13.836	24.804
	40	-	-	-	39.953
mass [g]		23.276	23.278	23.276	23.272

„-“: infinite displacement (structure does not stop the motion of the sphere)

Table 1: Displacement table for topologies with scaled loads

3.2 Optimization task B – minimum acceleration design

The idea for minimizing the acceleration of the sphere comes from the occupant protection in cars. Accelerations, or decelerations respectively, are often used to categorize the load on the occupant in a crashing car. The load case is the same as for optimization task A, but the objective function is now the acceleration of the sphere. The new task is to minimize the maximal acceleration subject to a maximum displacement of 50 mm in *y*-direction.

3.2.1 Topology optimization with LS-TaSC™

The reduction of the acceleration will be achieved with an adjustment of the mass fraction. In LS-TaSC™ this is implemented by the multipoint method. The mass fraction becomes a global design variable. With two additional simulations in each iteration, the mass adjustment is determined. This is the only way to deal with arbitrary objective functions. This application uses the standard constrained optimization method. The initial mass fraction is 30 %. Again the default filter radius is used and the optimization ends, when the mass redistribution falls below 0.002 or when the maximal number of 100 iterations is reached.

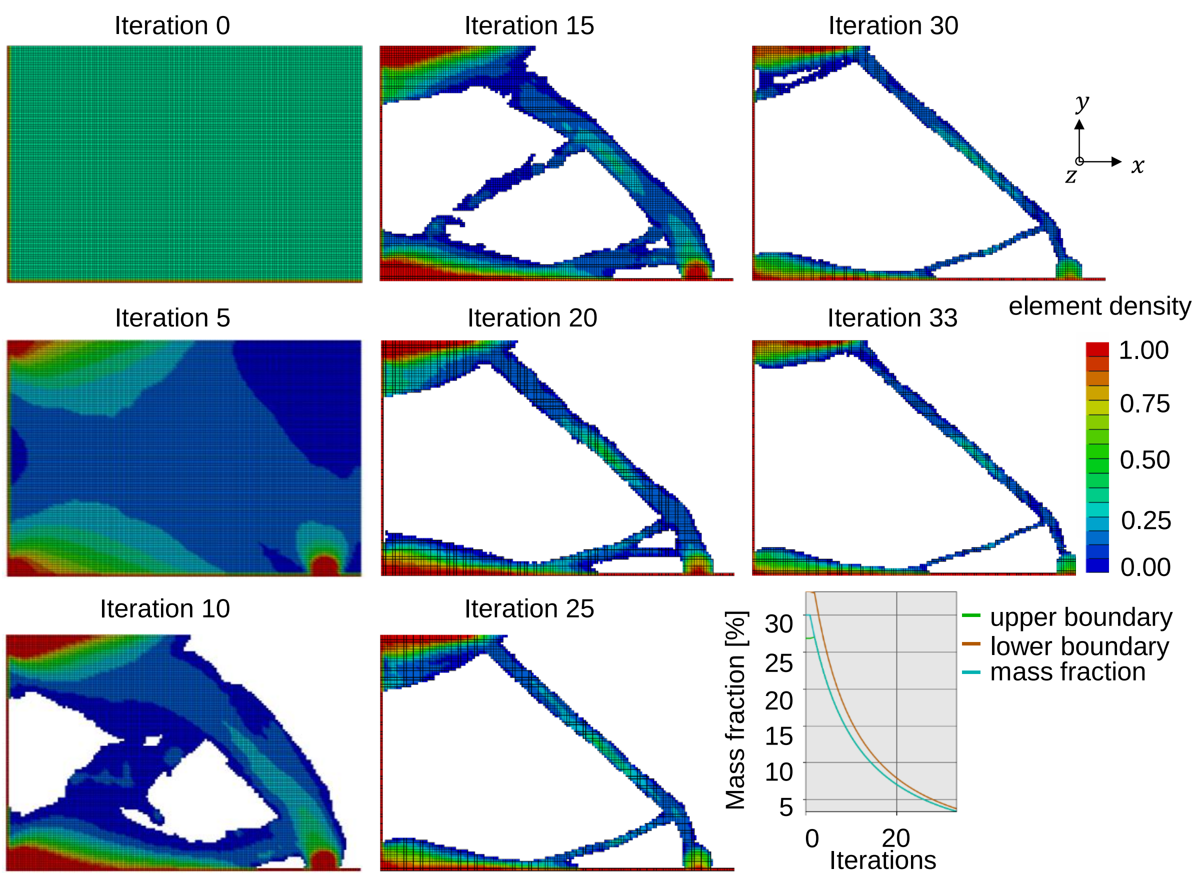

Fig.7: Optimization history of cantilever task B: minimum acceleration design

The optimization procedure converges after 33 iterations, with 100 FE-Simulations underlying. The optimization is driven by the high internal energy density at the clamping side and in the impact zone, where the elements become massive (cf. Fig. 7). The thinner walls instead have intermediate densities. The global optimization reduces the mass fraction until a further reduction would delete the walls and the structural integrity gets lost. The mass fraction falls far below the maximum mass fraction of 30 %. Nevertheless, the displacement constraint is far from being violated. The maximum displacement of the sphere in *y*-direction is 11.94 mm. The acceleration of the sphere in the final iteration is 10074 m/s².

3.2.2 Structural behaviour of the hexahedral model

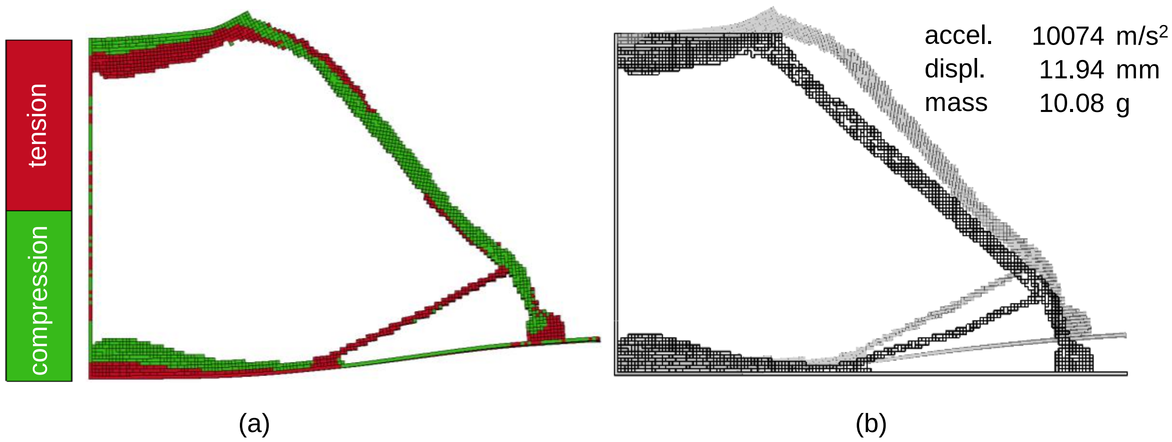

Fig.8: Structural behaviour of the hexahedral model of the acceleration optimization (a), comparative deformation view (b)

The acceleration of the sphere comes with the bending of the massive walls at the clamping side and the slight buckling of the intermediate density bar. From the acceleration-displacement-curve in Fig. 9, the driving force for the mass reduction can be discovered: The impact force of the sphere in the first contact. But as the impact zone remains highly loaded and therefore massive, the first acceleration peak remains the highest and dominant for the optimization.

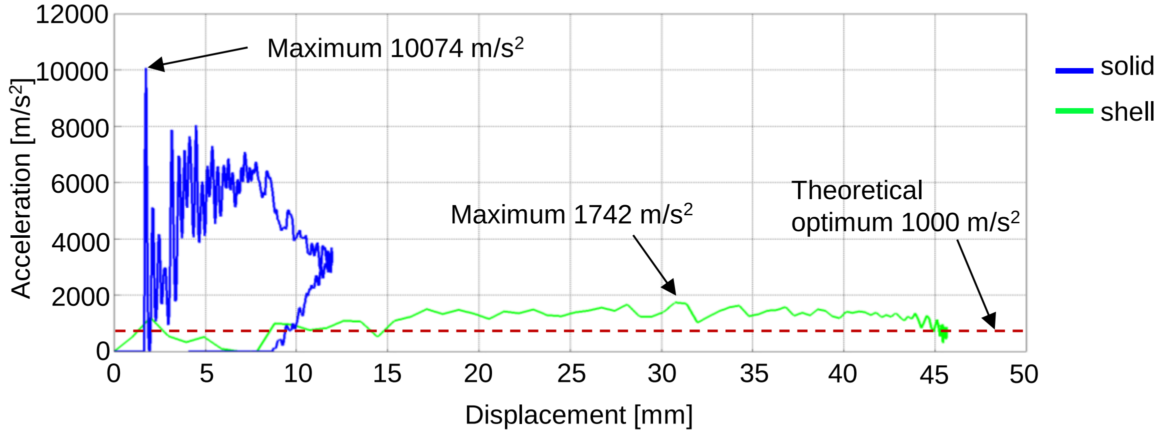

Fig.9: Acceleration-displacement-curve of the sphere

3.2.3 Shell interpretation and sizing

For the bending walls at the clamping side, two walls are drawn in to enable the compression-tension behaviour similar to the hexahedral model, cf. Fig. 10a. As constraints for the thickness optimization, a lower bound of 0.2 mm and an upper bound of 6.0 mm were defined. Again, the walls of the former non-design-space have a lower bound of 1.0 mm. The deduced shell structure changes its appearance significant, cf. Fig. 10b. The mass of the shell structure with 17.51 g is even far below the maximum mass of 63.11 g that corresponds to a mass fraction of 30 %. As it is a minimum acceleration design and not a minimum mass design, the walls in the upper left corner appear much too thick, but there is no motivation for the optimization algorithm for a local mass reduction. At the impact zone, the vertical wall has only minimum thickness in order to reduce the impact force on the sphere. The end of the horizontal non-design-space with maximum thickness is a piece of mass, which gets accelerated. The other maximum thickness walls ensure the structural integrity, so that the sphere does not exceed the displacement constraint. The minimum thickness wall in the compression zone of the hexahedral model has no benefit and would be eliminated if allowed.

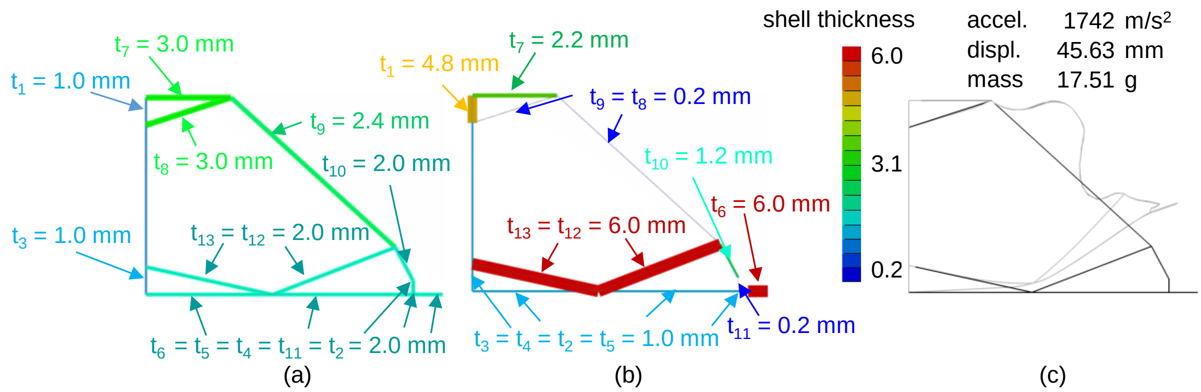

Fig.10: Shell interpretation of the optimized voxel structure (a), thickness optimization result (b) and comparative deformation view of the optimized thickness model (c)

The resulting acceleration-displacement-curve is also shown in Fig. 9. The theoretical optimum is derived from the energy conservation law for an ideal plastic impact, where the cantilever would convert the whole kinetic energy of the sphere into deformation.

4 Application 2 – "rocker beam in pole impact load case"

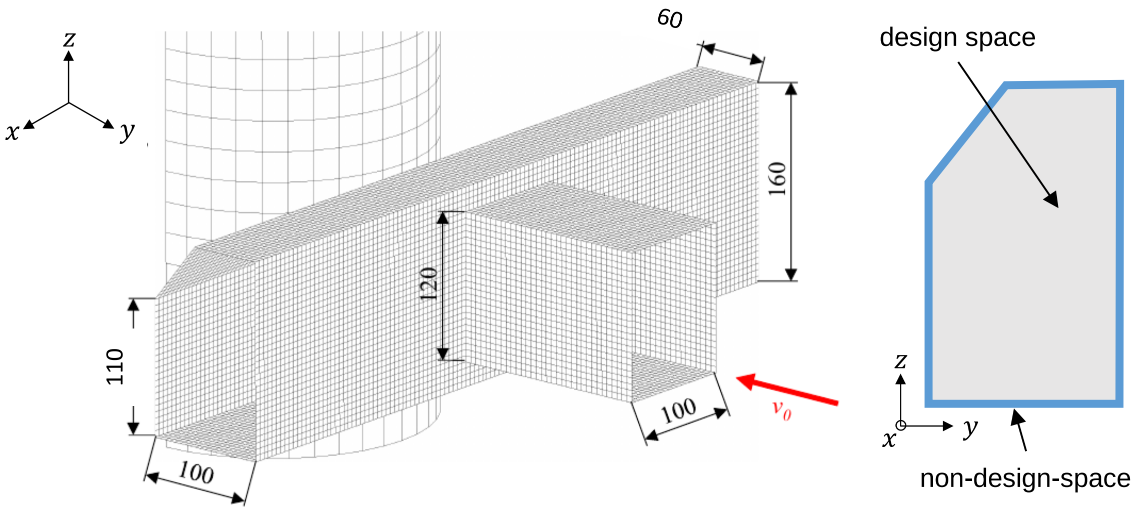

Fig.11: Application 2 – rocker beam in pole impact load case

The second presented application is a segment of an aluminium rocker beam of a vehicle. The load case is based on the EURO-NCAP pole impact. It is connected to a part of the seat cross member and moves with an initial velocity of 29 km/h against a rigid pole as shown in Fig. 11. A rigid wall of mass 85 kg with the same velocity is connected to the end of the cross member to increase the impact energy. The length of the rocker beam is 600 mm. Boundary conditions ensure, that this extract behaves like included in a full vehicle crash. Therefore the seat cross member is guided in y-direction and the 110 mm cutting edges of the rocker are restricted in z-direction. The intrusion denotes the maximum value of the indentation in the rocker beam. This is determined by the maximum y-displacement of the rigid wall, which is positioned at the end of the seat cross member.

4.1.1 Topology optimization with LS-TaSC™

The goal of the optimization is the minimization of the maximal occurring contact force of the rigid wall. Through this optimization objective an even force-time- respectively force-displacement-curve is desired, since the structure is forced to convert the kinetic energy of the load into deformation, but the maximum force should be as small as possible.

The mass constraint is 10 % mass fraction as upper bound. In order to ensure structural integrity, a displacement constraint for the moving of the rigid wall is defined. The movement in *y*-direction is restricted to a maximum of 70 mm. Non-design-space is the surrounding of the rocker with a width of 1.6 mm.

As the extrusion plane is now not rectangular anymore, the finite elements are irregular hexahedrons. The average element edge length in the extrusion plane, that is the *y*-*z*-plane, is 3 mm. In the extrusion direction, the elements are stretched by factor 4, to keep the numerical effort within a limit. The element type is now -1, which is an efficient formulation for elements with poor aspect ratio.

The filter radius is again default, in this case, this results in a four times broader neighbourhood in the extrusion plane than in the extrusion direction. The convergence criterion and the multipoint options remain the same as for the application 1B.

Convergence is reached after 31 iterations with 94 FE-simulations in total. Over the course of the optimization process (cf. Fig. 12), the reintroduction of elements can be discovered. Between iteration 5 and 10, the middle wall loses the connection but grows to the front again. The design proposal has a slightly reclined middle wall with intermediated densities and reinforcements in the impact area and the support zone to the cross member.

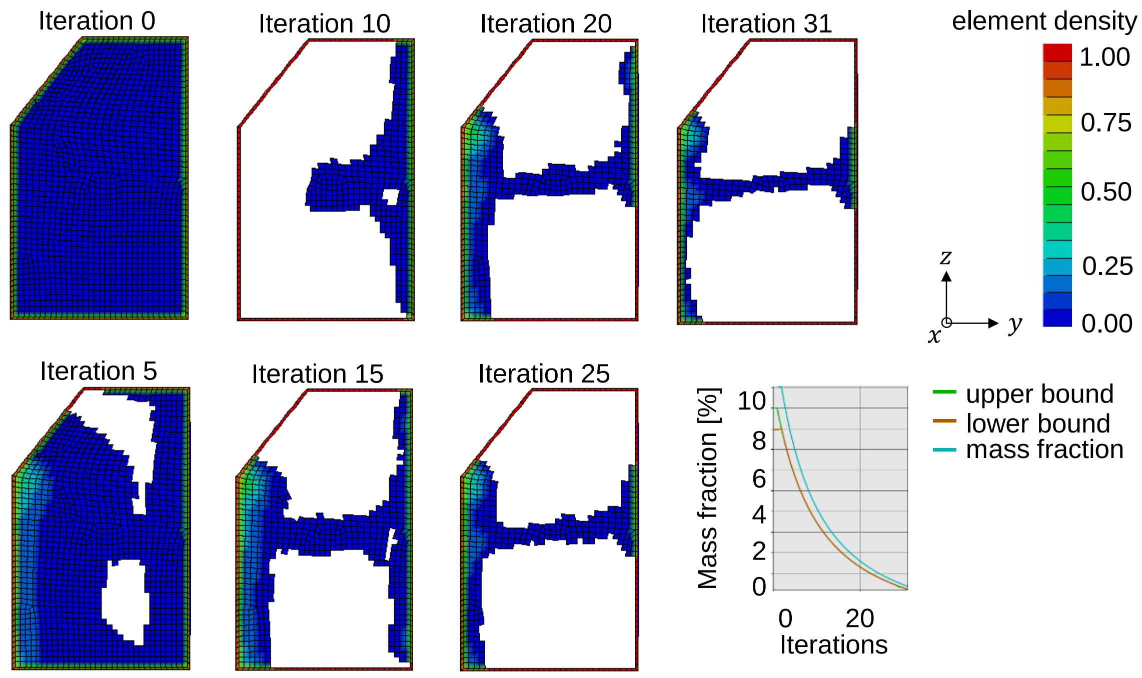

Fig.12: Optimization history of application 2: minimum reaction force

Again, the mass is far below the mass constraint and also the displacement of the rigid wall with 53.19 mm does not reach its bound. The maximal reaction force is 75.58 kN. In Fig. 13, this force is compared with the theoretical value for an even force-displacement-curve of 40.34 kN with the same mass of 1.55 kg for the rocker beam.

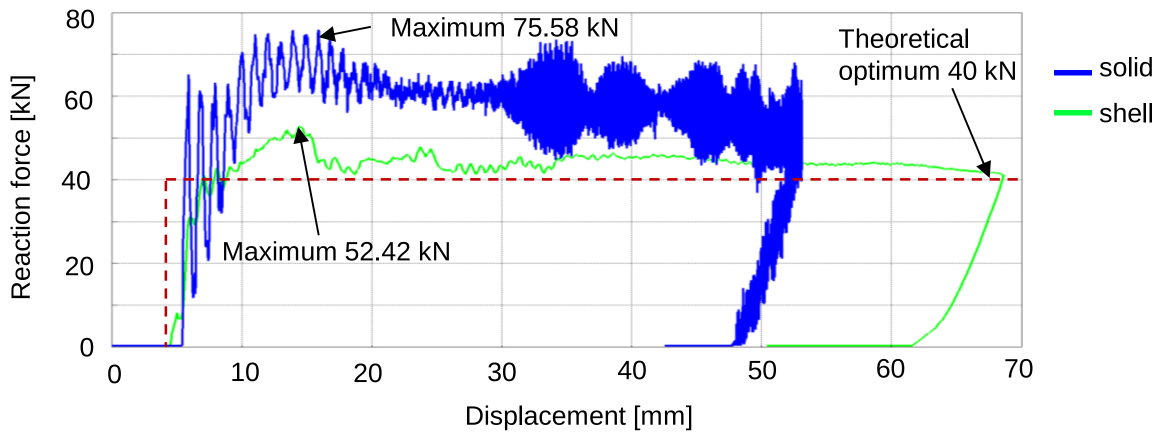

Fig.13: Force-displacement-curve of the rigid wall

4.1.2 Structural behaviour of the hexahedral model

The buckling mode of the reclined middle wall takes the kinetic energy of the model, cf. Fig. 14b. It has no direct connection to the cross member in moving direction. That leads to a three-point-bending in the wall, which is connected to the cross member. The massive impact zone and the upper and lower elongations prevent the rocker from higher intrusions, because the vertices are stiffened. The force-displacement-curve in Fig. 13 shows a quite rectangular course. In this application, the maximum force has a delay, because the force is measured at the end of the cross member, where the force impact arrives when the walls of the rocker beam are working.

4.1.3 Shell interpretation and sizing

The lower boundaries for the shell model are set to 1.6 mm for the walls, which fall together with the non-design-space. For the middle wall the lower bound is 0.2 mm. All design variables have the same upper bound of 6.0 mm. As shown in Fig. 14c, the middle wall is thickened up and the impact zone gets thinner. The buckling mode of the middle wall changed the side, see Fig. 14d. By construction the shell model cannot be as stiff in the vertices as the hexahedral model is. This makes it easier for the shell model to use the whole intrusion space. The force-displacement-curve is also drawn in Fig. 13. In this application, the dominating force peak comes with the same structural behaviour as in the hexahedral model.

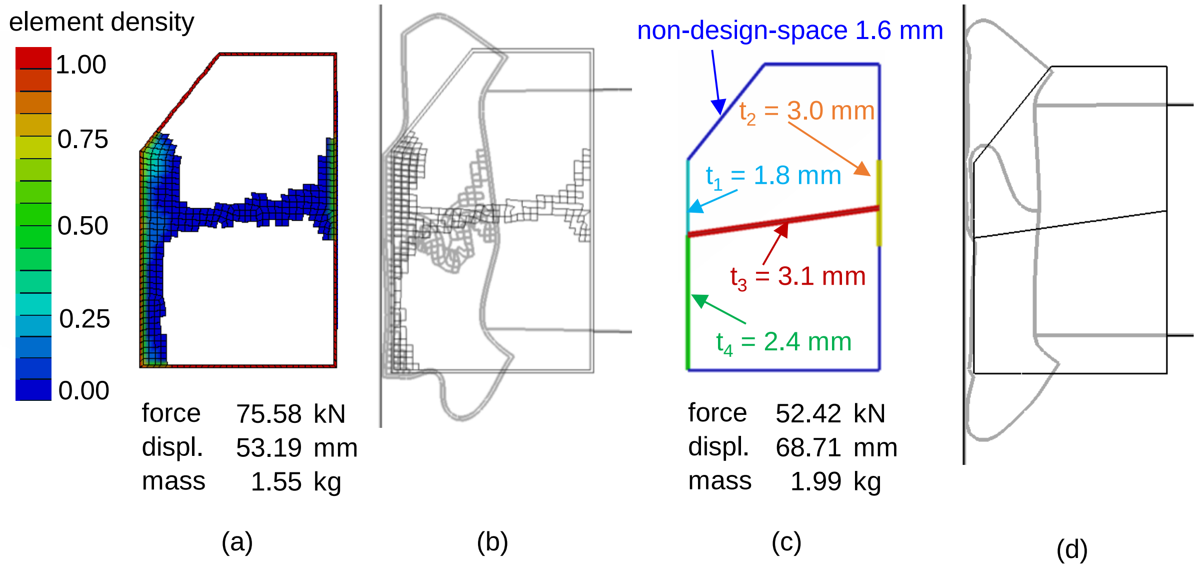

Fig.14: Design proposal from LS-TaSC™ for the rocker optimization (a), comparative deformation view of the hexahedral model (b), optimized shell model with wall thicknesses (c) and comparative deformation view of the shell model (d)

5 Summary

It has been demonstrated, that the topology optimization method implemented in LS-TaSC™ is able to deal with crash loaded structures. For stiffness applications, the method derives reliable results. When objective functions as accelerations or reaction forces are required, the multipoint method is able to adapt the mass fraction depending on the optimization goal. But the quality of the result highly depends on the load case.

The main difference between the optimization problems of application 1 and 2 are different load directions. For the rocker, the load path goes directly from the impact point to the moving mass at the cross member end. In the cantilever, the load path is not straight and there are high bending moments at the clamping sides. During the optimization of the rocker, a load path is determined and weakened with the mass reduction. In the same way, an engineer would insert a bulkhead inside a welded shell rocker. The impact and support zone are more balanced with the load path. In contrast to the cantilever application, where the moments at the clamped side and the compact impact zone are overrated compared to the low density wall under compression.

The big advantage of LS-TaSC™ is the applicability for stiffness applications. It is able to deal with crash loaded structures, irregular meshes and manufacturing constraints. The multipoint method is one way to deal with arbitrary objectives. For users, who have set up topology optimizations yet, it is easy to use the application. Even the number of iterations is acceptable. But the high number of voxels in crash simulations leads to a huge numerical effort for each iteration. In large displacement applications, the intermediate densities lead to numerical instabilities.

Current state of the art voxel based topology optimization methods are not yet able to optimize crash relevant objectives considering crash relevant constraint functions. In order to extend the applicability of the voxel based methods, sensitivity information are necessary. One idea is the numerical calculation of the so-called topological derivatives [5], where the sensitivities of the objective function are derived from a meta model.

6 Literature

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