New optimization strategies for crash design

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Abstract:

This contribution deals with optimization strategies for the crash design. After a short introduction of the mathematical background, the optimization algorithms, and current software systems, special approaches are presented. The first important strategy is the handling of the scattering of the influence variables. A corresponding optimization example is discussed. In order to come to totally new and better designs, effective possibilities for geometry model changings are necessary. Here, the methods for shape optimization are discussed. For the difficult part of topology optimization of crash structures two approaches are presented. The first method is to add or replace parts of the structure using a model library. The second method is based on introducing openings resp. holes in the structure. This paper gives an overview of applications of the optimization strategies and gives an outlook for the next years.

Keywords:

optimization strategies, crash optimization, robust design, shape optimization, topology optimization

1 Using optimization algorithms and systems

Important in any case is the accurate definition of the optimization problem. Necessary is the consideration of the following questions:

- Which design variables have to be considered?
- What are the objectives and constraint functions?
- Which simulation sequences are necessary?

The mathematical formulation reads as follows:

minimize $f(\mathbf{x})$ objective function, (1)

so that the following constraints are fulfilled:

$g_j(\mathbf{x}) \leq 0$ $j = 1,m_g$ inequality constraints

$h_k(\mathbf{x}) = 0$ $k = 1,m_h$ equality constraints

$x_i^l \leq x_i \leq x_i^u$ $i = 1,n$ side constraints, upper and lower bounds),

This formulation is valid for all optimization problems, because max $f(\mathbf{x})$ = -min $f(\mathbf{x})$ = min $-f(\mathbf{x})$.

Figure 1 classifies possible design variables. The simplest definitions of design variables are the dimensions. The most difficult design variables describe the principal idea of the design. The most helpful structural optimization of industrial products in the last 10 years was the topology optimization for finding principal concepts for static or quasi-static load-cases [1,2].

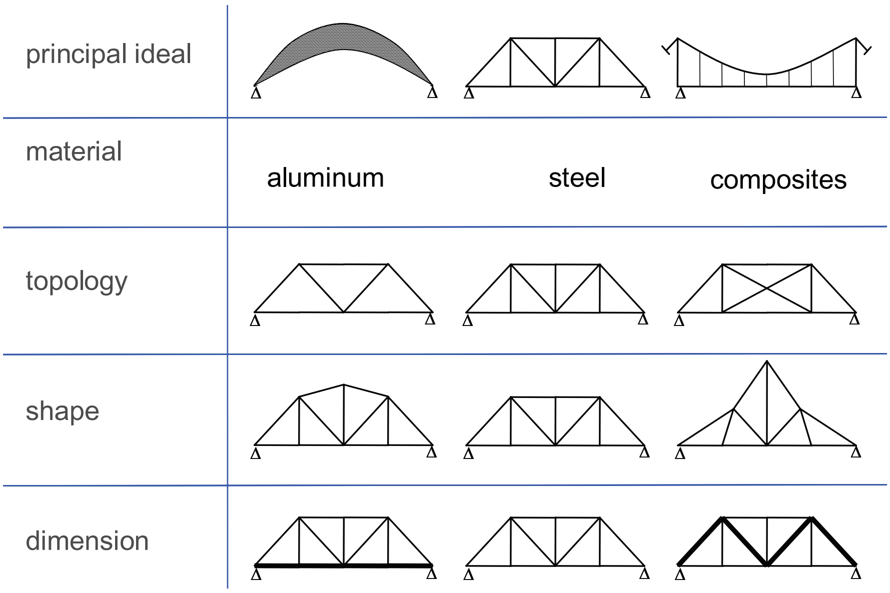

Fig. 1: Classification of possible design variables

The following typical crash development requirements have to consider as constraints in the optimization loop:

- Special acceleration values like the head injury criterion HIC value,
- high energy absorption with fold buckling,
- special force levels of crash elements in order to save neighbourhood parts,
- smooth force-displacement curve,
- smooth acceleration-time curve,
- special force paths of frontal and side crashes,
- high stiffness of special parts of the system, e.g. parts in a main force paths in the passenger area,
- low stiffness of special parts, e.g. at positions of the head contact of a pedestrian,
- special vehicle safety criteria, e.g. no leakage of the petrol system.

The use of mean compliance [1] or other functions coming from linear static approaches are not useful for crash optimization applications.

An efficient use of optimization needs knowledge of the principal behavior of the optimization algorithms. The first thing is to know, that many optimization algorithms find only the nearest local minimum. Only for functions with only one minimum we can guarantee to find of the global minimum. This is the case for convex functions. The finding of sufficiency conditions for the optimal point is not so easy. The formulation of conditions for a global optimum of arbitrary problems is not possible, until now. Beside the large range of mathematical optimization algorithms other optimization possibilities exist, e.g. the method of evolutionary algorithm [3] which uses the evolution in the biological sense: mutation, selection and recombination (participation of former generations). The algorithm runs as follows:

1. Definition of the parent designs (initial design),
2. Calculation of these designs,
3. Generation of a new population with mutation,
4. Calculation of these designs,
5. Selection and recombination of the best designs,
6. Check the convergence criteria: if there is no fulfilling, go to step 3,
7. Optimal design.

The advantage of these algorithms is the higher possibility for finding the global optimum, especially in non-smooth applications, e.g. coming from vehicle crash design.
The use of response surfaces (RSM) based on Design of Experiment (DoE) lists is interesting for problems with a small number of design variables (lower than 10). The simplest case is the polynomial approach:

$$\tilde{g}(\mathbf{x}) = c_1 + c_2 \frac{x_1 - x_1'}{x_1''} + c_3 \frac{x_2 - x_2'}{x_2''} + c_4 \left(\frac{x_1 - x_1'}{x_1''}\right)^2 + c_5 \left(\frac{x_2 - x_2'}{x_2''}\right)^2 + c_6 \left(\frac{x_1 - x_1'}{x_1''}\right)\left(\frac{x_2 - x_2'}{x_2''}\right) + \ldots \quad (2)$$

with $x_i' = 1/2 \cdot (x_{i,min} + x_{i,max})$ and $x_i'' = 1/2 \cdot (x_{i,max} - x_{i,min})$. The quality of the RSM can be checked by several criteria [1].

Software systems have an architecture shown in figure 2. The analyses model can consist a simulation sequence with a number of different CAE software (CAD system, Preprocessor, FE software, Postprocessor, …). Helps for selection and roll-out of a commercial system in the company can be found in [1,2].

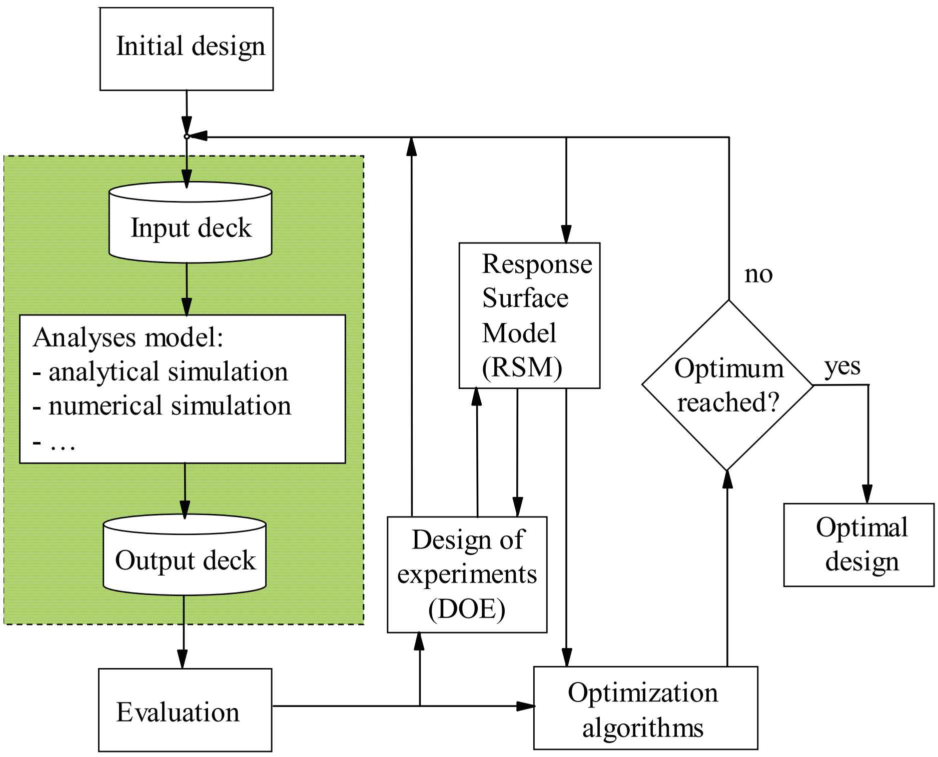

Fig. 2: Architecture of general purpose software

2 Necessary optimization strategies for crash development

By involving crash simulations in the optimization loop, we have to handle the following problems:

- Normally, there is a non-smooth structural behaviour,
- there not exists enough material data (e.g. polymer material, composite…),
- the material data and all other influence variables scatters,
- there are mesh-dependent results,
- there are physical bifurcations,
- there are simulation bifurcations,
- the input deck is often optimized for a special design point.

The considering of scatterings of the influence variables (figure 3) can be handle during the optimization by involving a robust analysis in the loop, but it is typical very computer time consuming [4]. The example in figure 4 shows a function with two minima. By consideration of the scattering we see, that the right minimum is more robust than the left minimum.

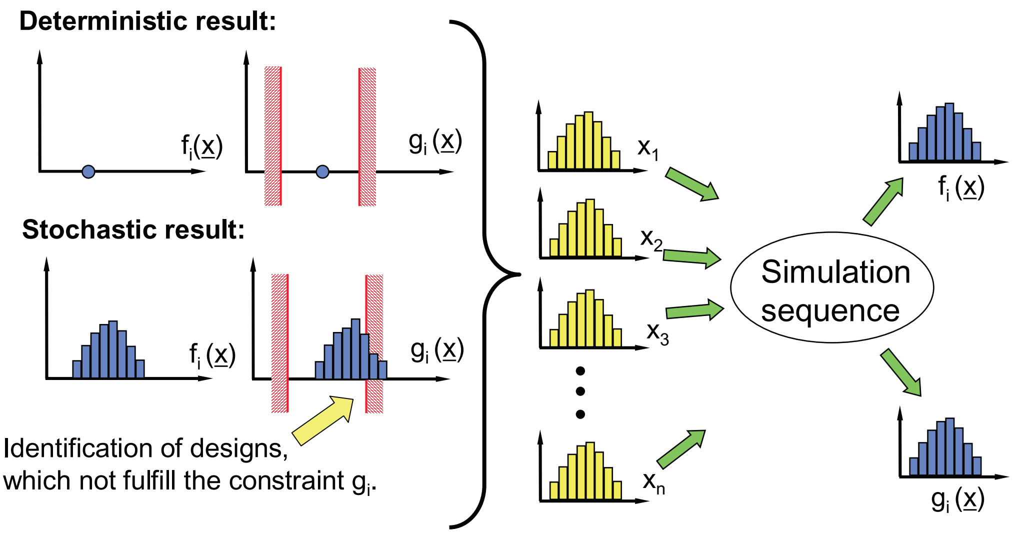

Fig. 3: Consideration of scatterings

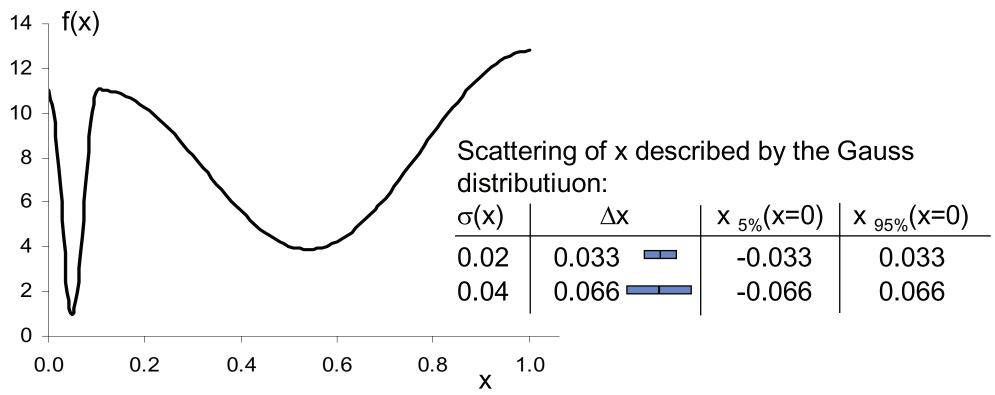

Fig. 4: One-dimensional example for robust design

One idea to handle some scatterings is the use of quantiles [5]. E.g., a 5%-Quantil is the value at which 5% of all samples are less. In order to find a robust design (optimization based on robust analyses) in a limited time, the following principal steps are necessary:

1. Selection of all interesting influence variables with their expectation values and their variants.
2. Carry out a Design-of-Experiment list (DoE) considering the selected variables.
3. Generate a Response Surface Model (RSM) based on these results.
4. Time-saving calculation of the quantiles of structural responses based on the RSM for all

points of the DoE list in step 2.
5. Generate a RSM (similar to step 3) with all results from step 4.
6. Optimize the structure based on the RSM.
7. Verification of the optimization results and if necessary, using of more local analyses for a more local optimization in the area of the optimal.

One practical example for showing the influence of scatterings is the optimization of the four wall thickness' of a frame side rail component model of an aluminum vehicle. The main load-case for the frame side rail is the frontal crash (figure 5).

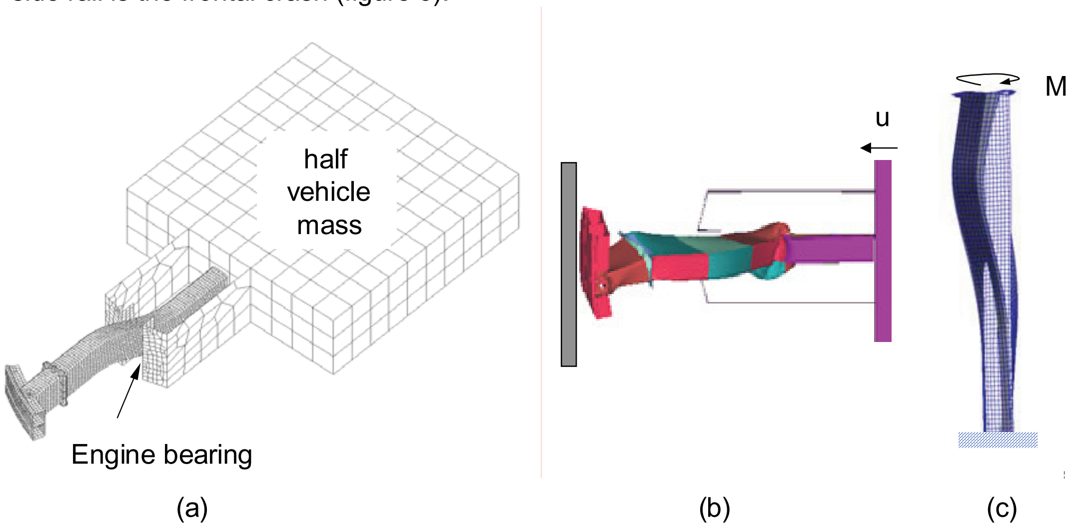

Fig. 5: Thickness optimization example: FE model for the frontal crash simulation of a frame side rail (a), Structural responses of the frame side rail in crash (b) and in static load-case (c)

For reasons of the driving dynamics and the strength analysis an additional simulation with MSC/NASTRAN® is necessary. In the structural optimization, both simulations with different input decks for the same structure are taken into account simultaneously. Objective of the optimization is the maximization of the 5%-Quantile of the internal crash energy ($f_{5\%}$) during a front crash. The constraints are formulated as follows:

- The 95%-Quantile of the deformation displacement during the frontal crash ($g1_{95\%}$) has to be lower than 300 mm,
- The 95%-Quantile of the mass of the frame side rail ($g2_{95\%}$) has to be lower than 2.5 kg,
- The 95%-Quantile of the static deformation because of a torsion load-case ($g3_{95\%}$) has to be lower than 2.4 mm.

With help of a DoE list, simulations for 41 combinations of multiple wall thickness' between 2 mm and 6 mm are carried out. Based on these simulations we generate 2^{th}-degree polynomial RSM. The scattering of the multiple wall thickness' is 0.06 mm. The results of different optimization loops are summarized in the following table. All optimization results are fulfilling the constraints g1, g2 and g3:

Optimization results	X_1 [mm]	X_2 [mm]	X_3 [mm]	X_4 [mm]	f [Nmm]
Opt. 1: without considering the scattering	5.0	2.0	5.5	2.0	1.049e7
Opt. 2: with considering the scattering of f, g1, g2, g3	4.9	2.0	4.4	2.0	0.817e7
Opt. 3: with considering the scattering of f, g1, g3	5.2	2.0	5.3	2.0	0.934e7

3 Shape optimization

In the last years, a lot of shape optimization applications of crash relevant components are shown, e.g. [6,7]. The simulation sequences for these applications are more complex as for wall thickness optimization problems. At least, we need a morphing tool like HYPERMORPH® or MSC/SOFY®, which change the finite element model directly. For a small changing of the shape these tools work very well, but normally the topology of the finite element mesh is fix and large shape variations lead to bad finite elements. A possibility to overcome that is the use of an adaptive mesh tool.
A more systematic approach is the involving of a CAD system. Based on the new parameter set of the CAD model, a new finite element model is generated and the structure is calculated. Here we need very good interfaces, because all steps have to be done automatically for every iteration of the optimization loop.
An interesting possibility is the use of SFE CONCEPT® [8,9], which is developed with the well-founded CAE background. The SFE CONCEPT tool offers a specially adapted solution to this problem. The design is performed in a purely declarative manner using abstract high order elements to describe a structure. Typical elements include, for instance, surface domains, cross sections, beams and even complex joints. All these elements and their interaction constitute the complete and consistent surface description of the structure. The description is fully parametric and the model can be estimated at every design state using the integrated finite-element-generator [10, 11, 12]. An example for a shape change in a SFE CONCEPT model and in the generated FE mesh is shown in fig. 6.
Design variables can be defined in SFE CONCEPT using the graphical user interface just by interactive recording the modification of the model due to parameter changes [8]. Lower and upper bounds of the design variables can be set with graphical control. Export of all the information about the design variables can be done directly to different optimizer tools, the complete optimization loop is shown in fig. 7.

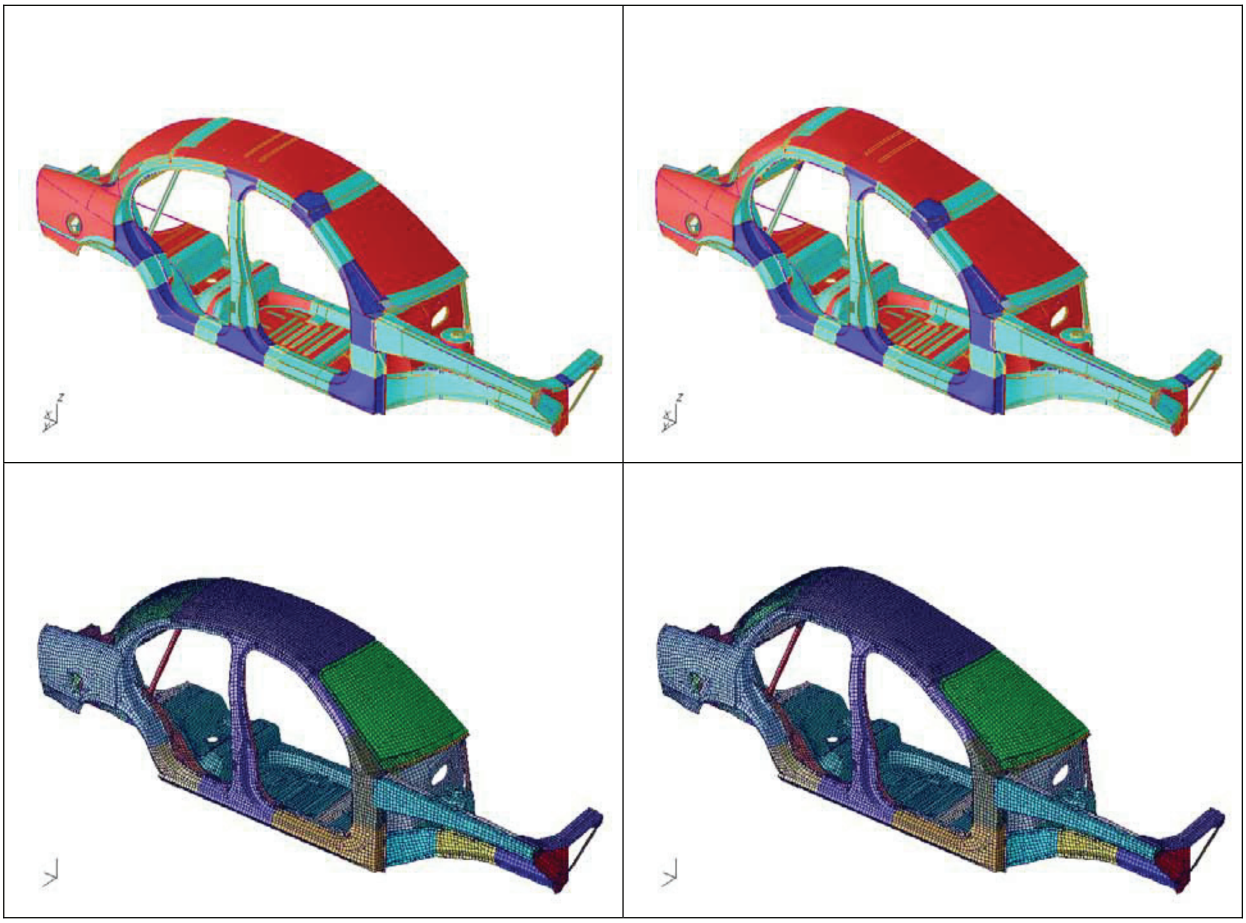

Fig. 6: Example for a shape change in SFE CONCEPT model and in generated FE mesh

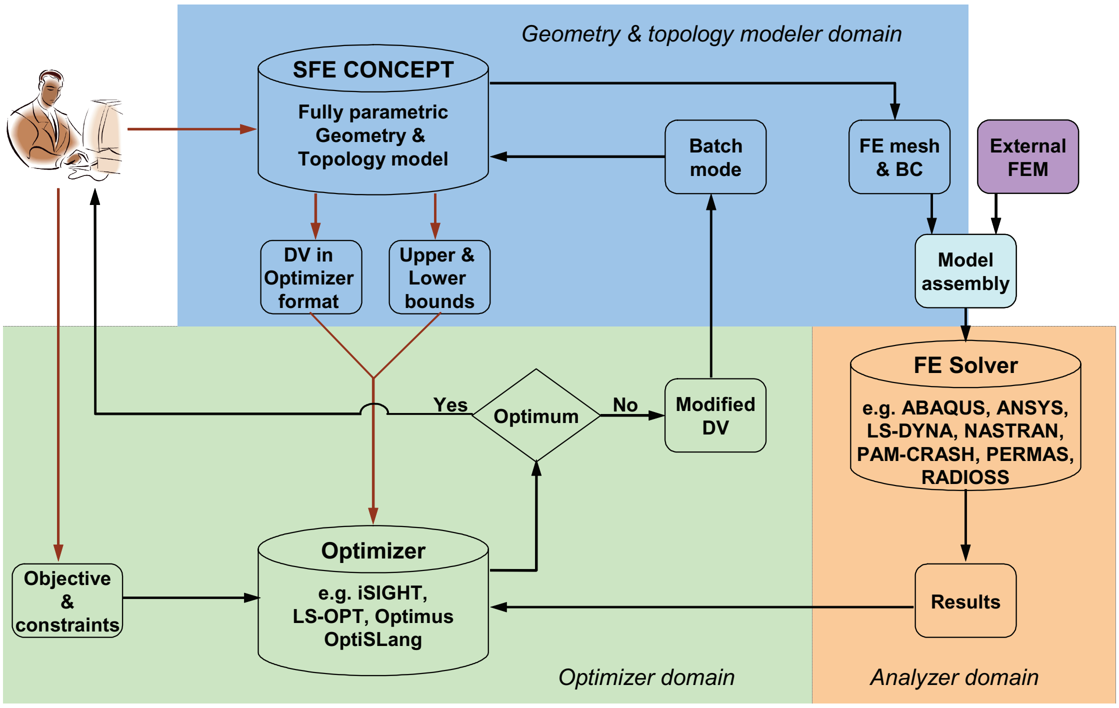

Fig. 7: Flowchart for shape optimization using SFE CONCEPT

4 Topology optimization

4.1 Combinatory optimization

Topology changes can be fulfilled by adding, deleting or replacing surfaces or parts of surfaces in a structure. In optimization it is essential to have only topology changes that make sense, because the design room for all topology changes is too large to search for an optimum. So the parts to add or replace should be chosen by engineering experience.
SFE CONCEPT provides a model library within all kinds of objects – from simple parts to complete models – can be organized. Parts can be added or replaced easily due to the use of smart connections. Fig. 8 shows an example for assembling parts to receive different model variants. This could also be done in an optimization loop. Additionally, shape optimization of the actual topology could be done to estimate the topologies after shape changes.

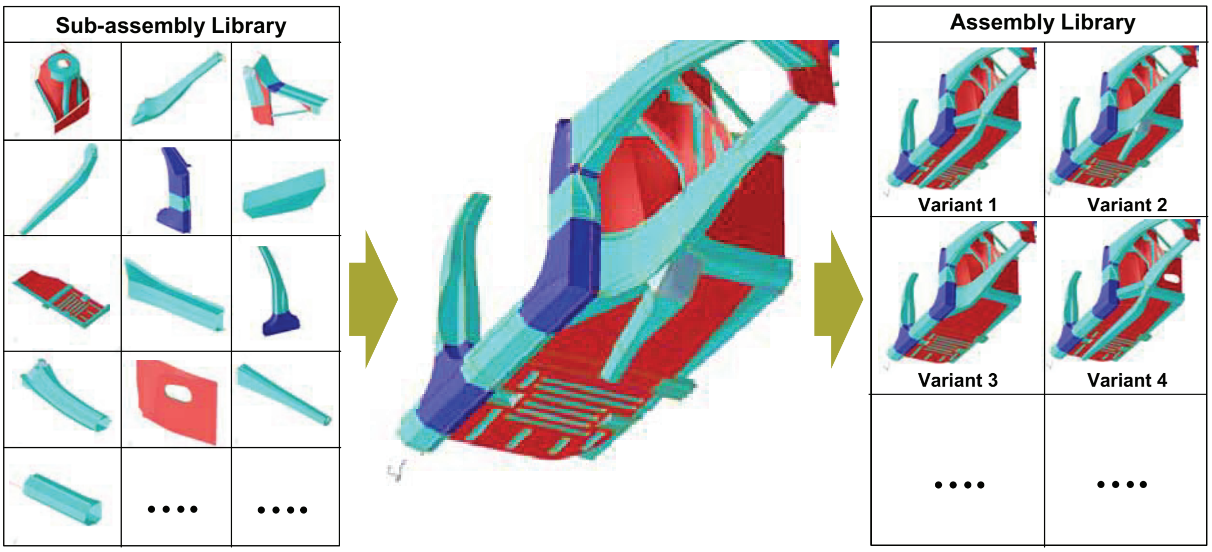

Fig. 8: Model assembly using SFE CONCEPT library

For this library based topology optimization approach, it is not possible to calculate all combination of the different variants of the different sub-assemblies. We need special strategies for finding the changes with the best chance of improvement. One idea is the identification and improvement of the parts with the worst buckling behavior [13].

4.2 Positioning of openings

The basic approach here is the bubble method [14,15] with the following solution strategy for the simultaneous optimization of shape and topology:

Step 1: For a given topology domain, a shape optimization is carried out considering all relevant objective and constraint functions. After the shape optimization, the structure of the component cannot be improved any further in this topology class.

Step 2: By inserting a opening resp. a hole (change of the topology class), we try to achieve improved results. We require the coordinates of the optimal position of the new hole. The positioning is carried out via a positioning criterion, which is to be determined analytically for special objective and constraint functions, or numerically for general cases.

Step 3: After the positioning, a shape optimization (equal to step 1) is carried out in order to find the optimal shape of the new bubble and the other variable boundaries. Go to step 2.

This generates an iterative process, and the optimization terminates when the positioning is performed at a variable boundary and if the approach function shall not be refined any further.
Figure 9 shows the optimization loop of a static loaded cantilever disc. The objective function is the mean compliance described by the elastic energy U. The constraint function is the weight of the structure defined by the half filled topology domain. The material is aluminium. The disc is fixed at the left boundary and has a load on the right (topology domain 150 mm x 100 mm, thickness 3 mm).

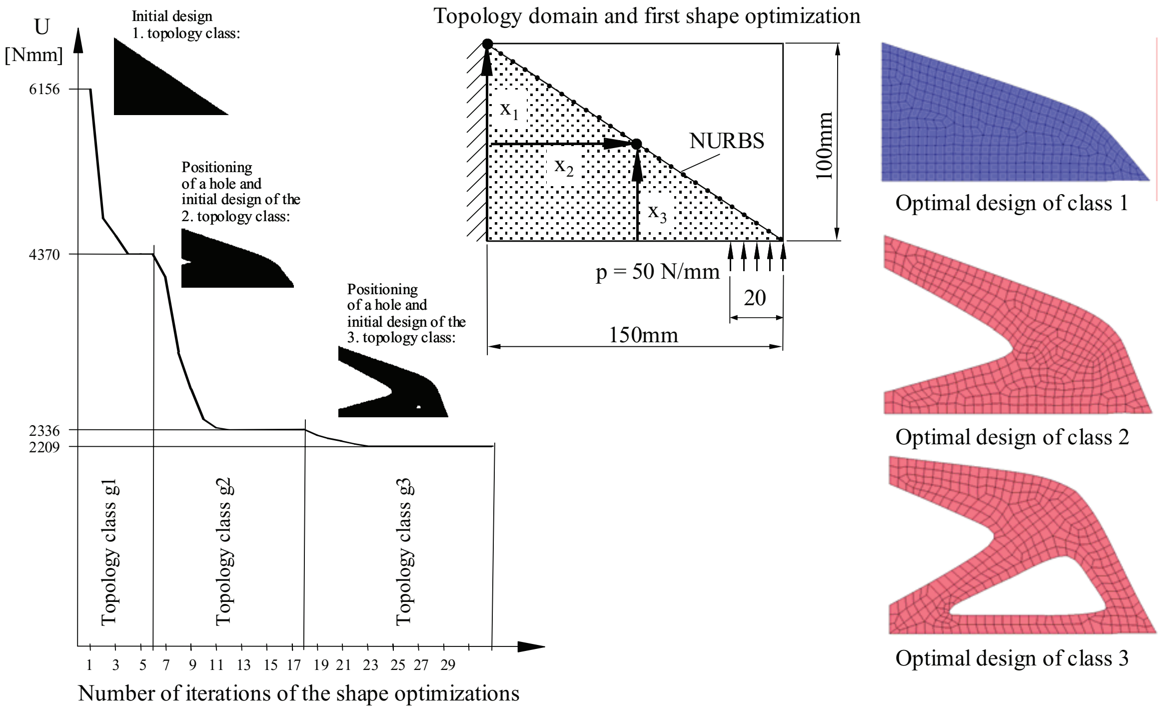

Fig. 9: Topology and shape optimization considering the mean compliance as objective function

In crash application, the use of mean compliance or other functions coming from linear static approaches are not useful. This is shown by carrying out crash simulations of the optimized structures. Now, we have a load coming from a 20 mm x 20 mm x 20 mm cube, which has a theoretically mass of 20 kg and a vertical upward speed of 1,389 m/s. The material of the disc is defined with a piecewise linear plasticity approach (Young´s modulus= 72000 MPa, stress [N/mm^2]/plastic strain [-]: 24./0., 37./0.04, 50./0.12, 100./1.e10). The out of plane deformation is not blocked, but the bending stiffness of the disc is increased by factor 1000. The results are shown figure 10 and in the following table:

Topology class	max. y-displ.	time of max. displ.	HIC_{15}	max.. eff. strain
g1_t = 3 mm	21.7 mm	28.9 ms	3.54	0.1414
g2_t = 3 mm	19.4 mm	26.5 ms	4.68	0.3512
g3_t = 3 mm	17.4 mm	22.0 ms	7.21	0.3060
g2_t = 1,6 mm	31.0 mm	46.0 ms	1.83	0.8250
g3_t = 1,5 mm	27.9 mm	38.0 ms	1.98	0.8390

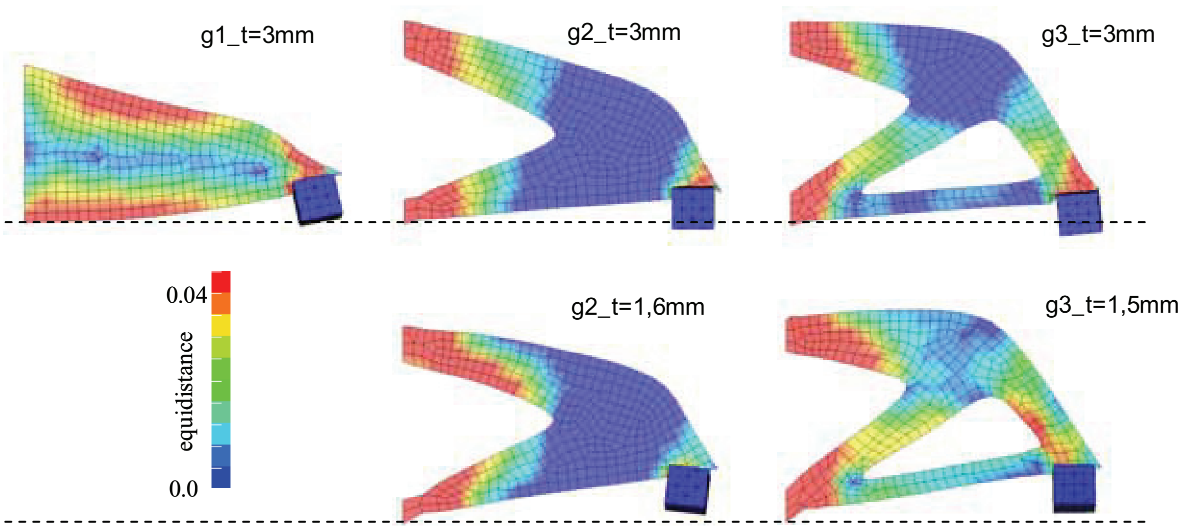

Fig. 10: Crash analysis of the "mean compliance" optimal structures (fringe: effective strains)

The best structure (g3_t = 3 mm) in the focus of mean compliance is the worst structure in the focus of the HIC value. Another point of view is to compare the structure topologies with the same values of mean compliance. Therefore, the thickness of g2 and g3 is reduced. For this material behaviour, and maybe only for this material behaviour, g2_t = 1,6 mm and g3_t = 1,5 mm comes to better results for the HIC value. This is also not a right way for topology optimization of crashworthiness structures. At least, we have to consider the right nonlinear material behaviour.
We see, for crash topology optimization we have to involve the correct objective functions and constraints. One possibility is the evaluation of the following characteristic function depends on the stress states $\sigma_{1(n)},\sigma_{2(n)}$ of the single time-steps in the component [15]:

$$\Phi_{1A}(\sigma_1,\sigma_2)=\sum_{n=1}^{n_{max}}\frac{1}{2E}\left[(\sigma_{1(n)}+\sigma_{2(n)})^2+2(\sigma_{1(n)}-\sigma_{2(n)})^2\right] \quad , \qquad (3)$$

which is a sum over all time-steps. This function has to be evaluated for each point of the structure. For increasing the stiffness C, a new hole is positioned at that point of the structure, where the characteristic function attains a minimum. Thus, the positioning vector **r**(x,y) is determined:

$$\mathbf{r}(x,y)_{opt\_C\_max}=\mathbf{r}(x,y)\left[\Phi_{1A}(\sigma_1,\sigma_2)_{min}\right] \quad . \qquad (4)$$

For the minimization of the acceleration and the HIC-values, it is necessary to have a low stiffness C of special parts of a crashworthiness structure. Here, we use the same characteristic function as we used for the high stiffness, but now the bubble is positioned at that point of the structure, where the characteristic function attains a maximum:

$$\mathbf{r}(x,y)_{opt\_C\_min}=\mathbf{r}(x,y)\left[\Phi_{1A}(\sigma_1,\sigma_2)_{max}\right] \quad . \qquad (5)$$

The first example deals with the hole positioning in the cantilever disc example shown in figure 9. The optimal hole position for the maximization of the stiffness C is found by the equations (3) and (4) and for the minimization of the stiffness C by the equations (3) and (5). Considering the computer time, the question is: Is it possible to carry out optimal hole positions based on information only from a short time interval after the beginning of the crash simulation? The answer is: The results are dependent on the considered crash time. Only the crash calculation time of 30 ms calculate the whole crash process until the mass have no velocity in the y-direction. The results are highlighted in figure 11.

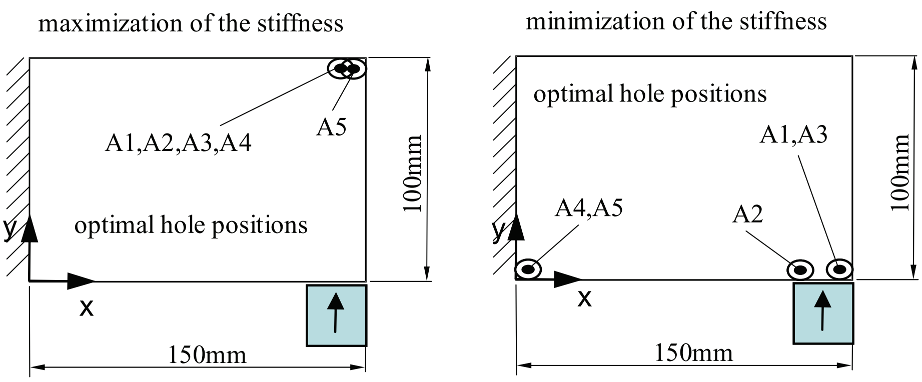

Fig. 11: Optimal hole positions in the cantilever disc solved with the characteristic function in dependence on the crash calculation time

The hole positioning for the maximization of the stiffness is robust. The minimization depends on the considered crash time.

In real life, the holes have not an infinitesimal size. For comparison, finite circular holes with a diameter of 30 mm are modelled in the disc. In order to generate response surfaces with the terms x, y, x^2, xy, y^2, x^3, x^2y and xy^2, for different hole positions HIC_{15} values and mass deformation distance of the cube are calculated (figure 12).

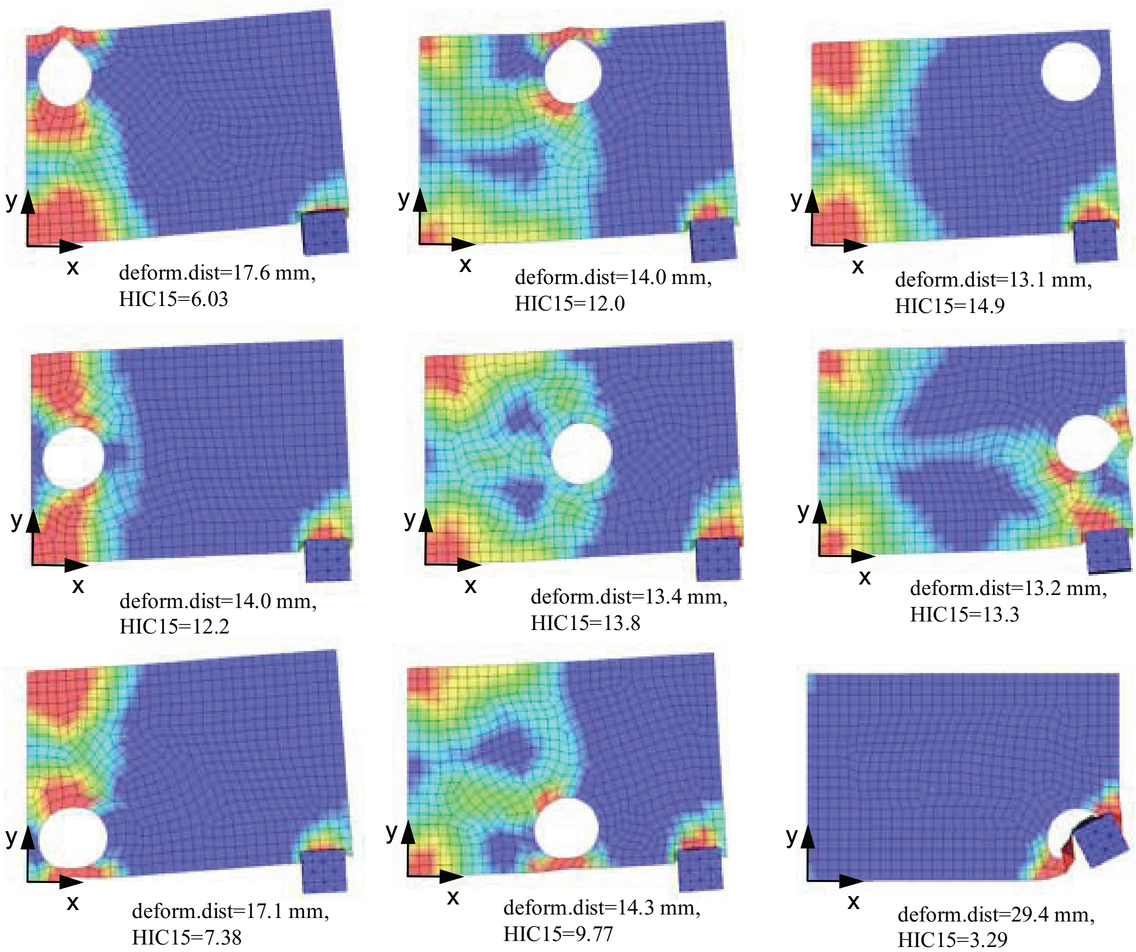

Fig. 12: Calculation of the crash behaviour for different hole positions

Here, the whole crash time until the mass have no y-velocity is calculated. The response surfaces for the HIC_{15} value and the deformation distance are shown in figure 13. These are the objective functions. For these two objective functions, the Pareto optimized boundary is calculated with 5 sub-optimizations using the constraint oriented approach is calculated. The first results of the analytical approach are nearby the numerical approach. The results suggest, that the analytical hole positioning approach do not need the calculation of the whole process.

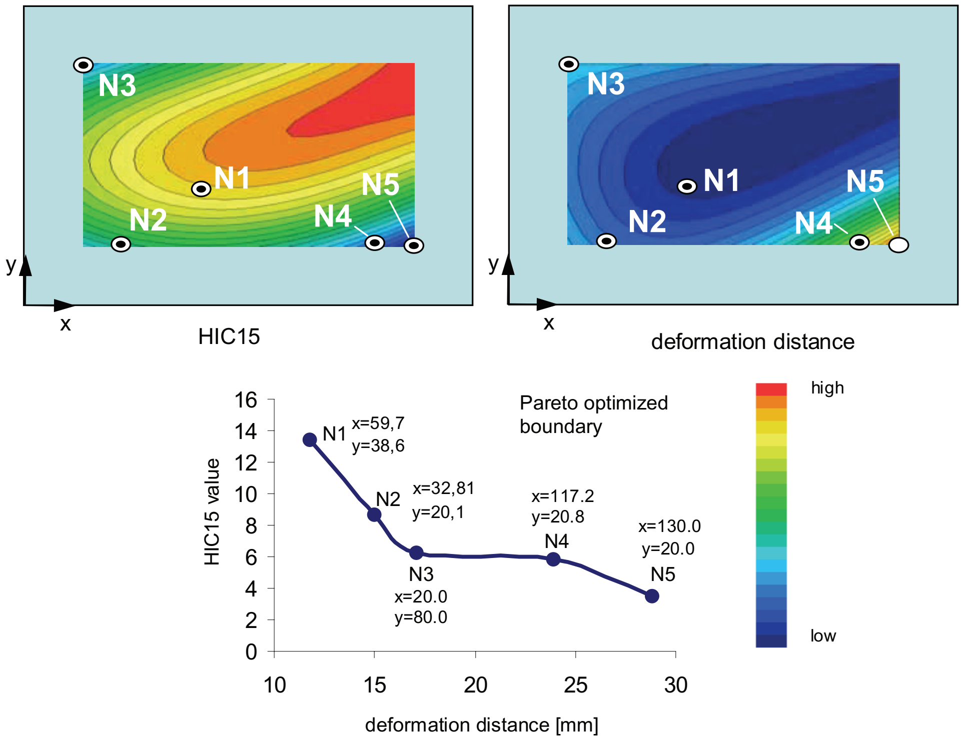

Fig. 13: Response Surfaces and Pareto optimized boundary for circular holes in the cantilever disc

The next example is the same disc discussed before, the difference is position and velocity direction of the 20 kg cube. The optimal hole positions are shown in figure 14. Here, the analytical and numerical approach give the same results for the optimal hole position.

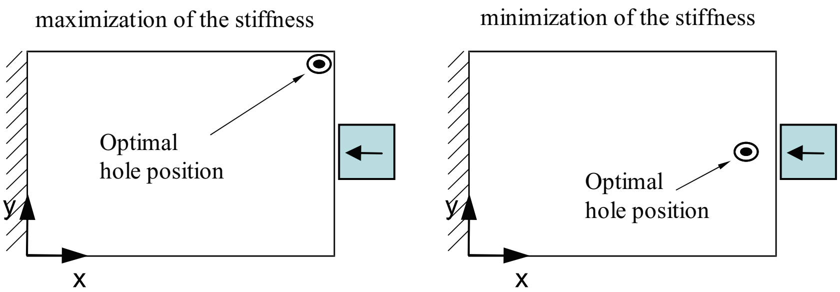

Fig. 14: Optimal hole positions in the direct loaded disc

A quarter of a crush tube with square cross section (70 mm x 70 mm, steel, length 320 mm, wall thickness 1.2 mm) is modelled because of symmetry [16]. The nodes on top of the tube are assigned extra mass (¼ model: 100 kg) and given an initial velocity in the y-direction of -5.646 mm/s. The nodes on the bottom of the tube are fixed in y-direction. Figure 15 shows the crash process and the optimal positions of the holes for maximal stiffness max_C and minimal stiffness min_C considering the first 0.2 ms of the crash process. For verification, finite holes with a size of 20 mm are modelled at the optimal hole positions. The maximal acceleration are nearby the beginning of the crash process: Reference structure without holes: 290 mm/s^2, max_C-structure with a small hole at the corner: 290 mm/s^2, min_C-structure with a small hole at the middle: 240 mm/s^2.

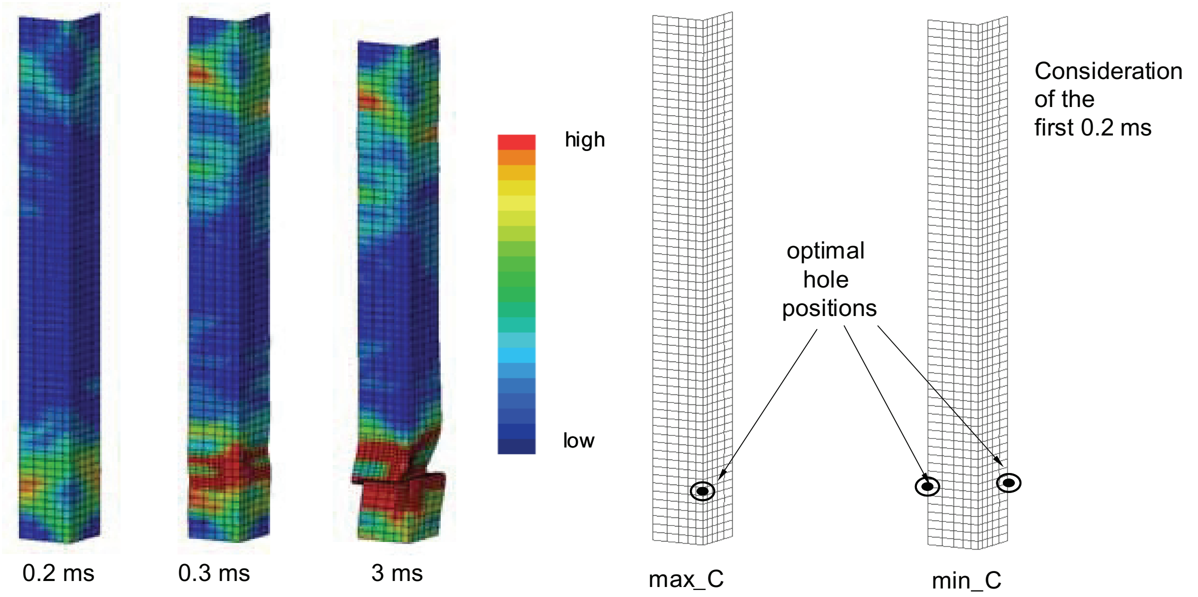

Fig. 15: Optimal hole positions in the square tube

5 Conclusion

The discussed optimization strategies are specialized for the crash design of structures. Currently, the goal for using mathematical optimization tools in the crash development process is not to make human engineering input dispensable. Mathematical optimization tools are able to give trends for the design of new crash structures. It is necessary to consider the scattering of all influence variables in shape and topology optimization.

6 Literature

[1] Schumacher, A.: Optimierung mechanischer Strukturen – Grundlagen und industrielle Anwendungen, Springer, Berlin Heidelberg, 2005
[2] Schumacher, A.: “Mathematical optimization in the development process of mechanical structures - Overview and rollout investigations –“, Proc. of the NAFEMS Seminar “Optimization in the Structural Mechanics”, Wiesbaden, April 27-28, 2005
[3] Rechenberg, I.: Evolutionsstrategie: Optimierung technischer Systeme nach Prinzipien der biologischen Evolution. Frommann-Holzboog, Stuttgart, 1973
[4] N.N.: Proceeding of “Weimarer Optimierungs- und Stochastiktage 1.0“, Dec. 2.-3.2004 CD-ROM, Dynardo GmbH
[5] Schumacher, A.: “Time-saving Robust Design Considering Vehicle Crash Performance”, in: I.C. Parmee, P. Hajela: Optimization in Industry, Springer-Verlag, London, 2002
[6] Quadbeck, M.: “Formoptimierung mit parametrisierten Flächenmodellen und LS-OPT“, 4. LS-DYNA Anwenderforum, organized by DYNAmore, Bamberg 2005
[7] Wang, H., Müllerschön, H., Mehrens, T.: “ Shape Optimization of a Crashbox using HyperMorph and LS-OPT“, 4. LS-DYNA Anwenderforum, organized by DYNAmore, Bamberg 2005
[8] SFE GmbH, SFE CONCEPT^{TM} V3 Reference manual, 2005
[9] Stuhec,U.; Zhou,S.;Shahidi,B.;Zimmer,H.;Prabhuwaingankar,M.: CAE Concept Design-a Key Enabler in Virtual Product Development: 11th International Conference on Human-Computer Interaction, Las Vegas, Nevada USA, July 22-27,2005
[10] Hoppe, A.; Zimmer,H.; Widmann,U.; Papke,L.; et al: Multidisziplinäre Optimierung parametrischer Fahrzeugkomponenten: Entwicklungen im Karosseriebau, Hamburg, 4.- 5.Mai 2004: VDI Berichte 1833
[11] Heiserer,D.; Zimmer,H.; Schäfer,M.; Holzheuer,Ch.;Kondziella,R.: Formoptimierung in der frühen Phase der Karosserieentwicklung: Berechnung und Simulation im Fahrzeugbau, Würzburg, 29.-30.September 2004: VDI Berichte 1846

[12] Automobilentwurf durch Simulation und Optimierung; auto>opt Abschlussbericht [www.auto-opt.de] Juni 2005 ; gefördert vom BMBF Kz: 01 IR B01; Herausgeber: Fraunhofer-Institut für Algorithmen und Wissenschaftliches Rechnen SCAI
[13] Olschinka, C.: „Studie zur Topologieoptimierung von Fahrzeugstrukturen hinsichtlich der Crashsicherheit“, Diploma Thesis University of Applied Science Hamburg, 2005
[14] Eschenauer, H., Kobelev, V., Schumacher, A.: “Bubble method for topology and shape optimization of structures”, J. of Structural Optimization 8, 42-51, 1994
[15] Schumacher, A.: “Parameter-based topology optimization for crashworthiness structures”, CD-ROM-Proc. of the WCSMO-6 conference, Rio de Janeiro, Brazil, May 30-Juni 03, 2005
[16] Reid, J.D.: LS-DYNA Example Manual, LSTC, 2005