Sizing Optimization for Industrial Applications

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1. Abstract
The current work presents the use of sizing optimization for large scale industrial applications with multiphysics phenomena. Presented are some examples which include either structural-acoustic or thermal-structural coupling. Moreover, these incorporate advanced simulation features such as contact modelling and efficient equation solvers dedicated to handle such large models.
This is achieved using the optimization system SIMULIA Tosca Structure as a direct add-on module for Abaqus. This module targets the thickness layout of the different structural sheet components for optimizing the static or dynamic responses of the structure computed using the users' existing Abaqus workflows.
Traditional design responses such as static stiffness, mass, internal and reaction forces and modal eigenfrequencies can be selected for both the objective function and constraints allowing the optimization of typical engineering setups where the shell thicknesses are the primary design variables.
Sizing optimization is a powerful tool for efficient structural design, being already employed across several industries to systematically achieve structure configurations with competitive performance and reduced design times.
The potential of this technology is here illustrated using some large applications from different industries, including a full automotive model from the transportation and mobility sector and a jacket offshore structure for wind turbines from the renewable energy sector. These represent some typical engineering sizing optimization setups of multiphysics and multidisciplinary problems like fully coupled acoustic structural interaction for NVH (Noise, Vibration and Harshness) design or thermo-structural coupling for designing high temperature components. Furthermore, the work demonstrates how sizing optimization benefits from advanced finite element modelling capabilities such as the support for contact inside or outside the design elements and approaches to handle large models with increased numerical efficiency, for instance the automatic multi-level substructuring eigenfrequency solver (AMS) introduced in Abaqus.
2. Keywords: Industrial applications, sizing optimization, structural optimization software

3. Introduction
Structural optimization has shown to be a powerful automatic tool to fulfil the growing industry requirement for efficient resource usage [1, 2, 3]. Frequently, after defining the overall layout of sheet metal structures, there is a need to find the optimal thickness distribution that meets the functional requirements. While trial and error modifications represent a tedious and slow process, the use of sizing optimization tools represents a systematic procedure to automatically obtain optimized sheet thicknesses.
The optimization system SIMULIA Tosca Structure [4] integrates optimization technologies in practical engineering environments as an add-on module easily integrated into the existing Abaqus [5] workflows as shown in Figure 1. After creating the finite element (FE) model of the structure, an optimization task can be defined by selecting the objective function to be minimized or maximized, the respective constraints and elements defined as design elements. All setups and definitions can be done in Abaqus CAE environment for pre-processing. Afterwards, the optimization task is completed by an iterative procedure where the model is automatically updated and modified using a robust non-linear constrained optimizer [8] based on sensitivities derived using the semi-analytical adjoint method [1]. Both the FE equilibrium and adjoint equations are solved by the Abaqus solver. At the end of the optimization job, the final model with optimized thicknesses is readily available for the typical CAE post-processing.
The current paper presents the advantages and possibilities that such a tool is able to bring to engineering design tasks addressing several typical industrial optimization setups. The first example addresses the maximization of the lowest eigenfrequencies for an offshore wind turbine jacket structure. The second example addresses a coupled structural-acoustic problem where the sound pressure generated by a structural excitation is minimized at a certain location. This mimics applications in Noise, Vibration and Harshness (NVH) related to the driving user comfort. To conclude, a full automotive model will be subjected to a mass minimization task with stiffness constraints, underlining the capabilities and ease of use of these optimization tools.

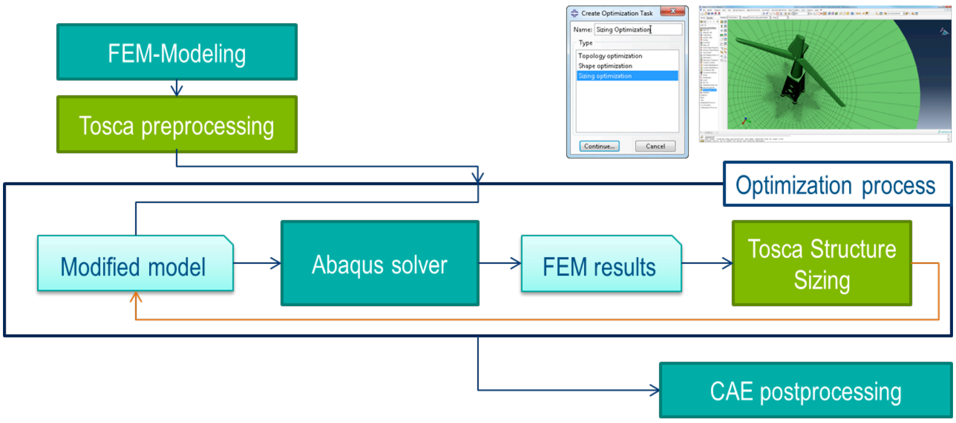

Figure 1 – Tosca Sizing process workflow

4. Offshore wind turbine jacket structure

The first example will consider the modal dynamic behaviour of a jacket structure that supports a 5MW offshore wind turbine, presented in Figure 2a. We optimize the thickness configuration of the truss jacket structure for maximizing its lowest eigenfrequencies. The structure is shown in Figure 2b, modelled with S4R (4 node reduced integration) 3D shell elements. The remaining components of the wind turbine are modelled using continuum elements, membrane elements and rigid bodies, accounting for their correct inertial distribution and the foundations are represented by four piles which are fixed in the ground. The total finite element model has 155300 elements and 612807 degrees of freedom (DOF) and the eigenfrequency analysis is performed using Abaqus AMS (Automatic Multi-level Substructuring) eigenvalue solver [5]. This allows for significant overall analysis runtime reduction in large-scale simulations [6]. The present case requires the evaluation of the first 10 modes and the analyses are performed using 24 CPU cores. The total CPU time is reduced by 60% when compared to the traditional Lanczos eigenvalue solver. This reduction is severely increased when the number of requested modes is increased: 75% for 50 modes and 90% for 100 modes considering the current example. This reveals a good scalability when considering larger models.

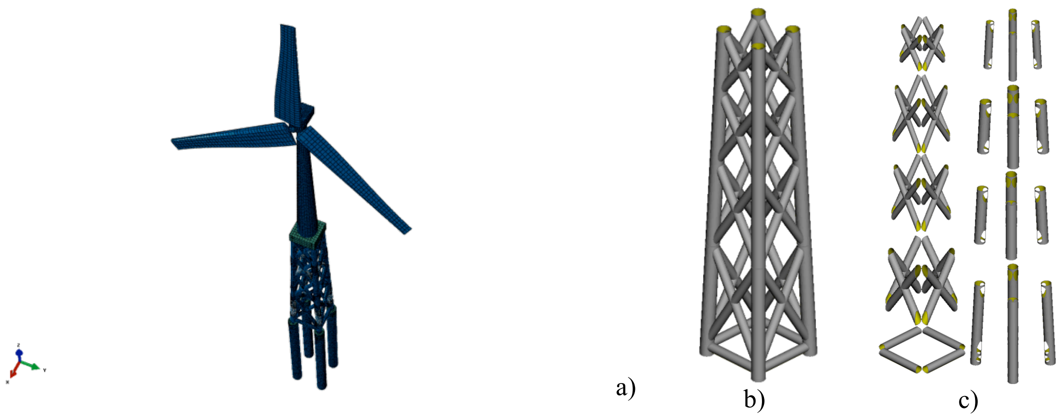

Figure 2 – Offshore wind turbine: a) FEM model, b) shell jacket structure to be optimized and c) defined groups for clustering the thicknesses

The objective of the current optimization task is to maximize the lowest eigenfrequencies of the structure with a mass constraint. The objective function (ϕ) is defined using the Kreisselmaier-Steinhauser [4] formulation described in equation (1) and will consider the first 4 ($J = 4$) eigenmodes:

$$\phi = -\frac{1}{k}\ln\sum_{j=1}^{J} e^{-kf_j}\,, \qquad k = \frac{30}{f_{min}} \tag{1}$$

The design domain is meshed by 83020 shell elements modelling the truss structure. Each of the respective shell thicknesses represents one design variable – also called free sizing. As a consequence, the thickness of the shell

elements varies freely. This will tell how to subdivide the optimized into various constant thicknesses profiles. Alternatively, these elements can be clustered into groups that have the same thickness, therefore reproducible by assembling different parts together. In the present case we consider 9 independent groups represented in Figure 2c, thus reducing the 83020 thickness design variables to 9.
The free optimization result is shown in Figure 3a and the respective optimization iterative process in Figure 4, showing an increase in the objective function of 41% without increasing the weight of the structure. For the clustered optimization the results are presented in Figure 3b and Figure 5. The additional constraint of grouping the design variables still allows an increase for the objective function of 15% for the same initial mass.

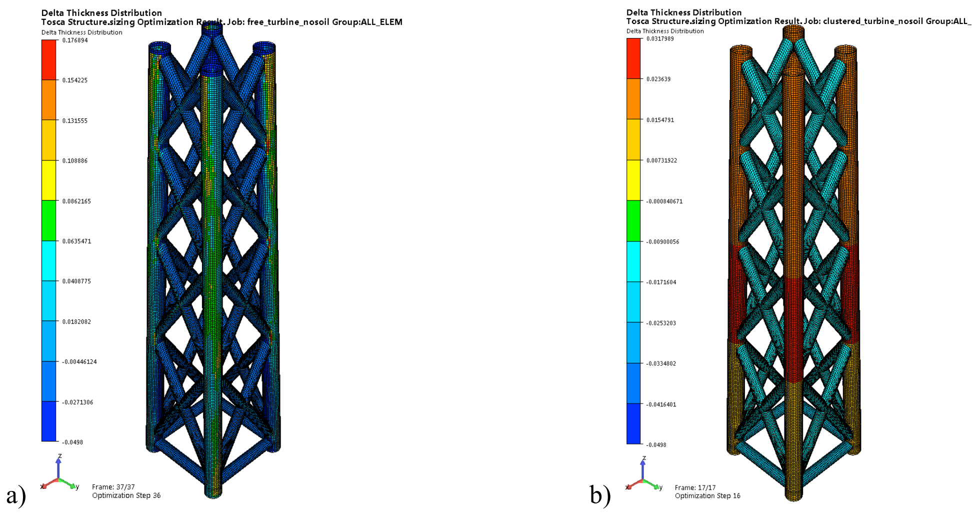

Figure 3 – Change of thickness of the optimized structure: a) with free sizing and b) with clustered sizing

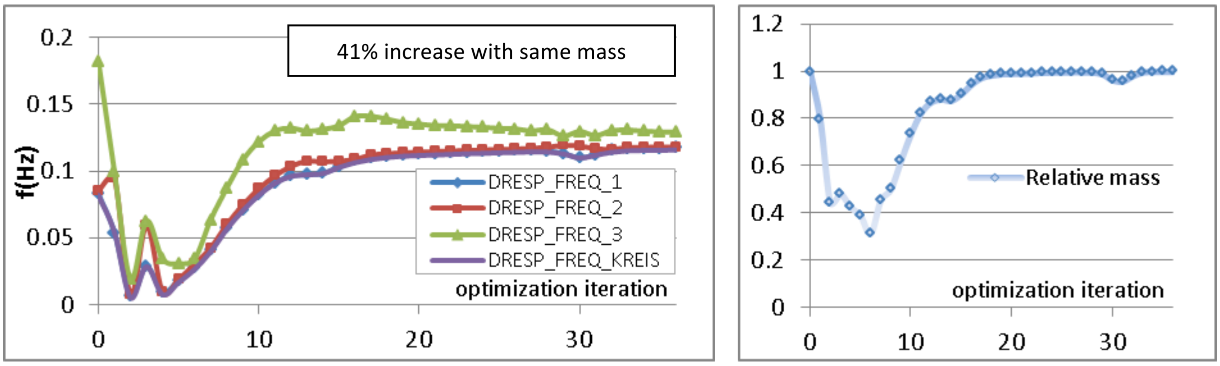

Figure 4 – Optimization iteration history for the free sizing optimization for the objective function, lowest eigenfrequencies and normalized mass

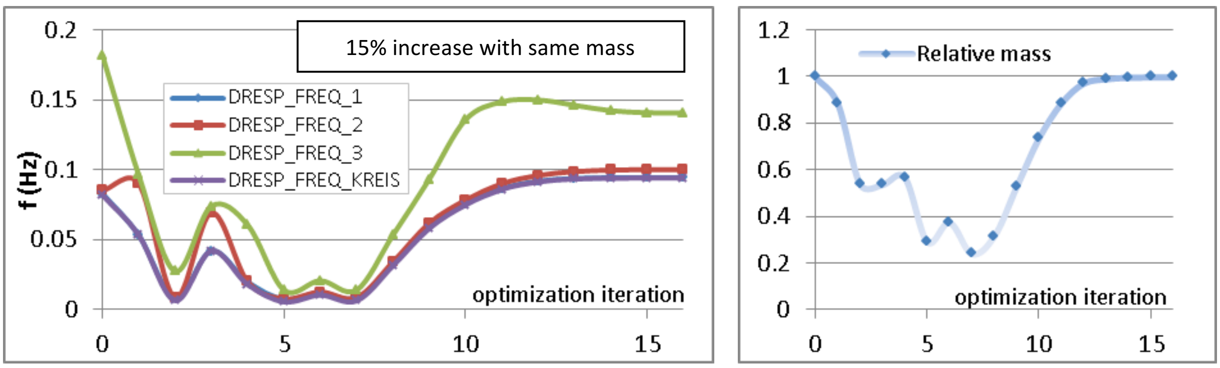

Figure 5 – Optimization iteration history for the clustered sizing optimization

5. Fully coupled structural-acoustic sizing optimization

The current application reveals the optimization of a structural shell plate coupled with an air cavity in order to minimize the pressure measured at a nodal location inside the acoustic domain when the structural component is subjected to a harmonic loading as illustrated in Figure 6a. The plate is simply supported at its boundaries and loaded at its central point.

The governing finite element equilibrium equation for a structure with structural and acoustic domains subjected to harmonic loading and assuming a steady-state time-harmonic response can be described by equation (2) [7]:

$$-\Omega^2 \begin{bmatrix} M_s & 0 \\ A^T & M_a \end{bmatrix} \begin{Bmatrix} u_A \\ p_A \end{Bmatrix} + i\Omega \begin{bmatrix} C_s & 0 \\ 0 & C_a \end{bmatrix} \begin{Bmatrix} u_A \\ p_A \end{Bmatrix} + \begin{bmatrix} K_s & -A \\ 0 & K_a \end{bmatrix} \begin{Bmatrix} u_A \\ p_A \end{Bmatrix} = \begin{Bmatrix} P_s \\ P_a \end{Bmatrix} \tag{2}$$

where Ω is the excitation frequency of the applied load and resulting response. $\boldsymbol{M}$, $\boldsymbol{C}$ and $\boldsymbol{K}$ are the mass, damping and stiffness matrices. $\boldsymbol{P}$ is the load amplitude for the structural or acoustic domain, according to the respective suffix s and a. The interaction between both domains is quantified by the coupling matrix $\boldsymbol{A}$ and the response amplitudes given by $\boldsymbol{u}_A$ and $\boldsymbol{p}_A$ for the structural and acoustic degrees of freedom, respectively.

The objective is to minimize the resulting pressure amplitude at the centre of the cavity considering an exciting frequency from 500 to 1000 Hz while keeping its weight below the initial value. In order to consider the frequency response across the defined spectrum with N discrete excitation frequencies, we introduce the Q-mean norm formulation for the objective function as explained in [7] and defined in equation (3):

$$\min_{\{t\}} \left(\frac{1}{N} \sum_{n=1}^{N} \left(p_A(\Omega_n) \right)^Q \right)^{1/Q} \tag{3}$$

subject to: equilibrium – represented by equation (2) and mass constraint – $m(\{t\}) \leq m_{init}$

where $\boldsymbol{p}_A$ represents the amplitude of the pressure at the node(s) of interest, $\boldsymbol{m}$ the total mass of the structure and $\boldsymbol{m}_{init}$ its initial value and $\{\boldsymbol{t}\}$ the vector of the thickness design variables. $\boldsymbol{Q}$ is set to 6 as it reveals to be numerically stable and to cause only a small error when compared to the min-max formulation [7].

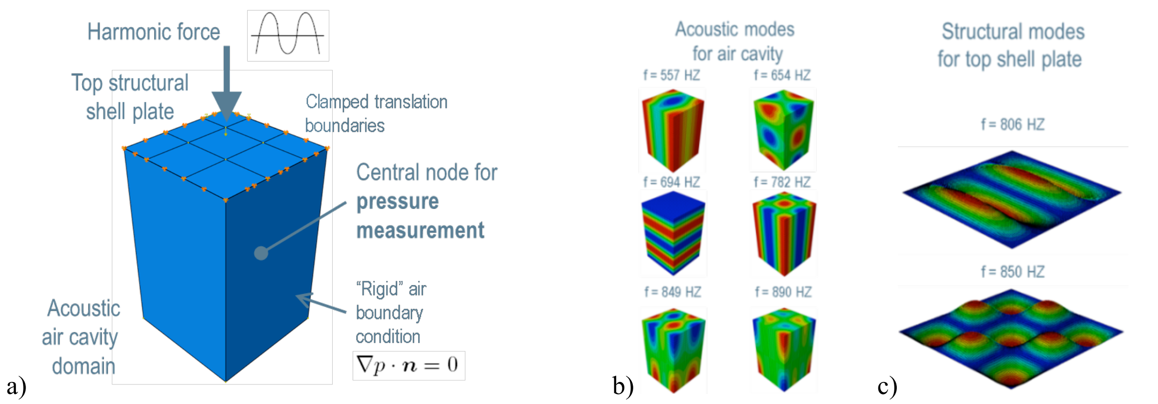

Figure 6 – a) Coupled structural-acoustic model, b) acoustic modes for air cavity and c) shell plate initial modes

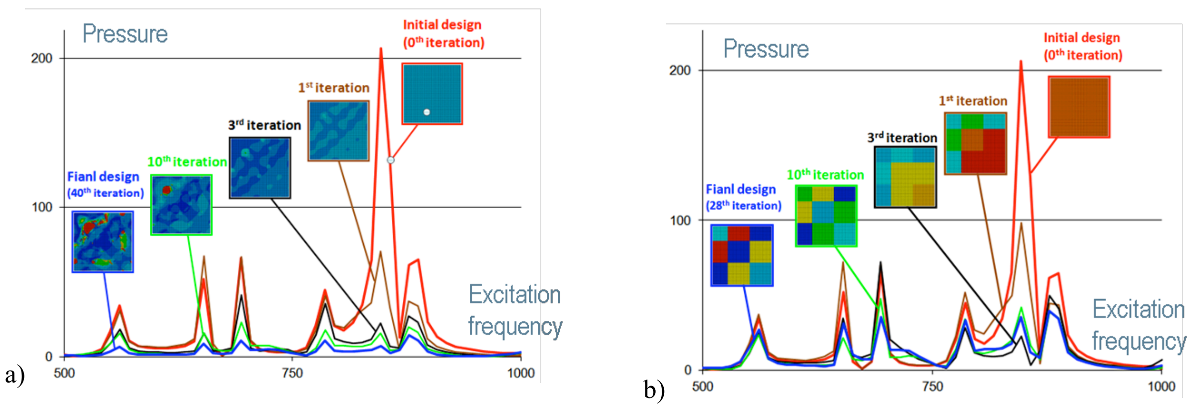

Figure 7 – Optimization iteration history of the pressure frequency response amplitude at the centre of the air cavity for the a) free sizing and b) clustered sizing optimization, respectively

The initial structure modes and eigenfrequencies are represented in Figure 6b and Figure 6c for the uncoupled domains.
The results for free and clustered sizing optimizations are shown in Figure **7**a and Figure **7**b, respectively, where we can observe a significant reduction of nodal acoustic pressure during the optimization iterations.

6. Automotive sizing optimization
The current application of sizing optimization considers a full automotive model available as an Abaqus example model [5], shown in Figure 8. The vehicle is modelled considering 331578 S4 (4 node shell elements with full integration) and 17443 S3R (3 node shell elements having reduced integration) shell elements, totalling approximately 2 million of degrees of freedom.
The objective of the optimization is to minimize the mass of the structure subject to stiffness constraints. Both free and clustered optimization will be considered. The clustered thickness optimization groups the thicknesses into 198 section groups color-coded in Figure 8.

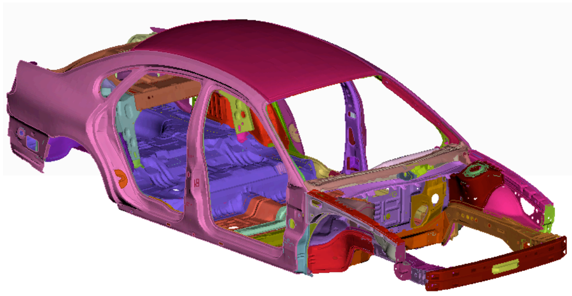

Figure 8 – Automotive finite element model considered for sizing optimization

We will require that the all bending, torsional and axial stiffness remain above the initial values. The structure is submitted to 3 representative different load cases defined in Figure 9 being clamped at the rear and loaded at the front wheel knuckles. The resulting displacement at the load locations is used to represent the stiffness of the car in these scenarios and combined as represented in equation (4) where the first subscript of the displacement u represents its orientation and the second the nodal location. In order to compute the sensitivities of these displacement constraints, the respective adjoint equations are solved for each required DOF and iteration using Abaqus solver.

Bending stiffness: $\frac{u_{z,2} + u_{z,1}}{2} \leq u_{bending}^{target}$ Torsional stiffness: $\frac{u_{z,2} - u_{z,1}}{2} \leq u_{torsional}^{target}$ Axial stiffness: $\frac{u_{x,2} + u_{x,1}}{2} \leq u_{axial}^{target}$ (4)

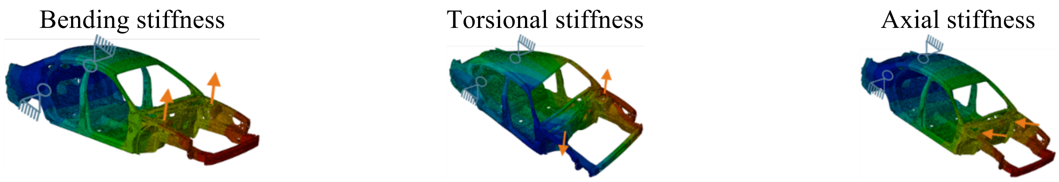

Figure 9 – Illustration of the stiffness constraints for the mass minimization optimization

The shell thicknesses are constrained to vary between -20% and 20% of the initial value and the optimization convergence curves are presented in Figure 10 and Figure 11 for the free and clustered options, respectively. In both cases, a significant mass reduction of 19% and 15% can be achieved with the same axial and bending initial stiffness measures and even with a significant increase in torsional stiffness.

7. Conclusion
The use of sizing optimization with SIMULIA Tosca Structure is capable of bringing major improvements to the design of shell structures. As here demonstrated with several applications we were able to maximize the structural eigenfrequencies or to reduce the acoustic pressure without increasing the weight of the structures. Additionally, we have also minimized the structural mass while also keeping or improving its stiffness. Being able to easily integrate these optimizations into existing workflows, it is a valuable tool for the design of structural shell components.

At the conference, additional examples will be presented illustrating the use of sizing optimization to applications that involve thermo-structural coupling and include contact modelling.

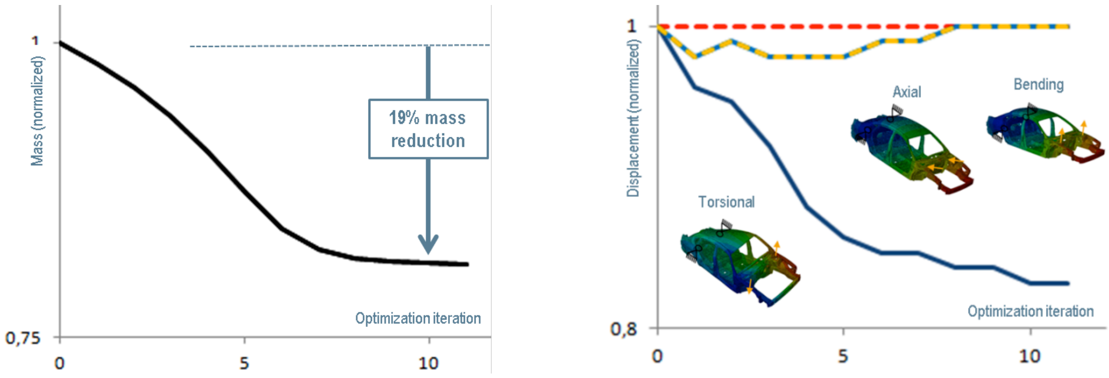

Figure 10 – Mass and stiffness constraints optimization iteration history for the free sizing optimization process

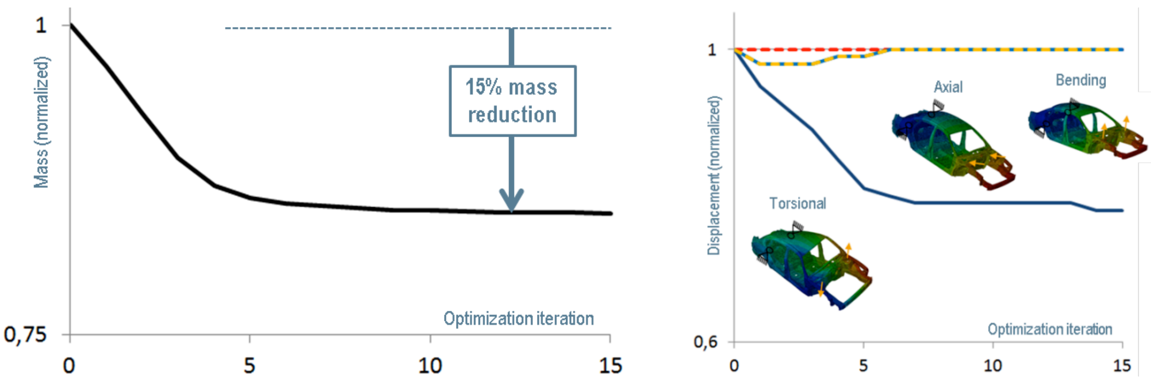

Figure 11 – Mass and stiffness constraints optimization iteration history for the clustered sizing optimization process

8. Acknowledgements

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