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Automotive Chassis Topology Optimization: a Comparison Between Spider and Coupé Designs

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Abstract—**Weight reduction is a major issue for carmaker companies due the need to comply with the emission regulations without reducing the vehicle safety. A classic trial-and-error approach to design in the automotive industry is becoming inadequate and new means are needed to enhance the design process. A major improvement on the end product can be achieved by adopting suitable optimization techniques from the early design stage. In the present paper the problem of automotive chassis design in view of weight reduction is tackled by means of topology optimization. The design methodology proposed is applied twice: at first addressing a chassis for spider vehicles, then for coupé vehicles. The two chassis, together with some intermediate result are discussed and compared. The methodology has been proven to be successful in finding innovative and efficient layouts for automotive chassis.**

Index Terms—**chassis design, topology optimization, multidisciplinary optimization, weight reduction, structural dynamics**

I. Introduction

WEIGHT reduction has become a primary concern in automotive industry. In fact, safety standards and emission regulations impose conflicting performance targets that need to be satisfied at the same time. While the respect of the safety standards pushes the automotive design process towards heavy weight solutions, environmental issues and handling call for a resolute vehicle weight reduction.

Over the last twenty years the average vehicle weight has steadily risen due to the improvements in safety and the growth in number of the vehicle features [1]. This brought to the increase in the aluminum content of vehicles with the aim of restraining their weight, and also to a growing interest towards composite materials, even though their application is still limited to parts of some high performance prototype vehicle for cost reasons.

Apart from the quest for better materials, remarkable weight saves can be also obtained by adopting a new approach in design involving optimization techniques. Optimization is a powerful tool for systematic design in mechanics; it can lead to sensible improvements that could not be achieved with a simple trial-and-error approach.

In order to apply these techniques a suitable parameterization of the investigated problem together with the definition of the objectives and the targets which are sought are needed. The optimization algorithm iteratively generates new samples which are tested through simulation.

Topology optimization [2] is a non-traditional optimization technique, particularly suitable for solving structural mechanics problems at an early design stage using finite elements analyses. It aims at finding the optimum material distribution within the domain given by a finite elements mesh. In a different way than more traditional algorithms, it has the peculiarity that it can change the topology of the object by virtually digging holes in the domain at locations where the algorithm, from local gradient computations, thinks it is less needed. This is made possible by adopting a parameterization based on a fictitious element-by-element material density. Various ways for formulating such an optimization problem exist: according to the SIMP method, which is the most simple and well-known, the material density can vary with continuity between void and full density. The stiffness of the material varies nonlinearly with the density following the formula

$$E = E_0 \left(\frac{\rho}{\rho_0} \right)^p$$

where E is the stiffness of the material, ρ the density, p is called penalty factor, and the subscript 0 stands for the properties of the full density material. Of course an intermediate density material has no physical meaning but accepting its presence is an helpful stratagem that allows the optimization problem to be solved much more easily. Setting a penalty factor larger than unity makes the intermediate density material unfavourable due to its large density-to-stiffness ratio, pushing the optimization process towards “black-and-white” solutions. For more informations on topology optimization we refer the reader to the rich specialized literature over the topic (*e.g.* [3]).

Applications of topology optimization to mechanical problems are numerous in literature, also with regard to the automotive field in particular. Most of these applications are focused on the optimization of small components, where the numerical modeling of the components is relatively simple, and a small number of loading conditions are investigated. In these cases the complexity of the problem is limited, due to the fact that a small number of optimization constrains need to be satisfied, and convergence is likely to be achieved quickly.

For instance, in [4] the optimization of hard drives suspensions is addressed by mean of topology and topography optimizations. The aim of the study is to maximize the torsion, bending, and sway modal frequencies. [5] deals with the topology optimization of an automotive suspension aiming at the minimization of its compliance for a given volume under three different static loading conditions.

In [6] instead, an application of topology and size optimizations of a body-in-white structure is presented, thus involving a larger and more complex domain than it is usually encountered in topology optimization applications in literature. In fact, the domain is given by all the volume that lies below the vehicle styling surface with the exclusion of the room needed for elements like the passenger compartment, the engine, the wheel, the boot, the doors,

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the windshields, and so on. The vehicle skeleton obtained this way undergoes the optimization process under global stiffness, local stiffness, and crashworthiness constraints.

In this paper a methodology for automotive chassis design in view of weight reduction based on topology optimization is applied for creating two types of chassis layouts. The first type regards the design of chassis suitable for a rear-engine spider vehicle, the other for a rear-engine coupé vehicle. The setup of the two optimizations are discussed and the outcoming solutions compared. The optimization process applied is part of a more elaborated methodology already presented by the authors in [7]. This methodology was based on three optimization algorithms (topology, topometry, and size optimizations) to be applied in cascade for refining step by step the chassis layout.

II. Set up of the optimization process

A typical optimization problem can be written in the form

$$\begin{aligned} &\underset{\mathbf{x}\in\mathcal{D}}{\text{minimize}} && f(\mathbf{x}) \\ &\text{subject to} && \mathbf{c}(\mathbf{x}) \geq 0 \end{aligned}$$

where $\mathbf{x}$ is the vector of the optimization parameters, $\mathcal{D}$ the design space or domain, f the objective function, $\mathbf{c}$ the set of optimization constraints.

In topology optimization, as already discussed, the choice of the parameters is given by a fictitious density for each element of a finite elements domain. Thus, In order to set up the topology optimization process, the following still needs to be defined:

1) the domain of the optimization (*i.e.* the finite element mesh of the object to be optimized, suitably subdivided into designable and non-designable areas),
2) the objective of the optimization,
3) the performance targets the chassis must satisfy (given in the form of optimization constraints), together with the loading conditions needed for computing the targets.

The definition of the objective of the optimization is quite straightforward: since the goal of the study is the weight reduction of cars we aim at the minimization of the chassis mass. On the other hand, the definition of the domain and the performance targets are more sensitive issues and will be briefly discussed in the following.

A. Domain of the optimization

The domains of the optimization are shown in Fig. 1. The two domains are the same except for the addition of a solid roof model for the coupé case. They are thought for leaving a large freedom in choosing the optimum layout to the optimization process. In fact, the massive solid blocks in Fig. 1 leave room just for the passenger compartment, the engine, and the gearbox. Room for other components (such as trunk, fuel tank, and so on) is less critical and can be accounted for *a posteriori* when interpreting the topology optimization results into an actual chassis. The chassis mesh is made of 1.97 million four-nodes tetra elements, while the roof mesh is made of 1.97 hundred thousand eight-nodes hexa elements.

The only constraints in the choice of the domain are mainly given by the provisional vehicle dimensions:

1) the wheelbase and track are fixed,
2) the suspensions layout is fixed and the suspensions non-designable,
3) the suspension, seats, engine, and gearbox joints are fixed and non-designable areas,
4) all the rest of the finite element mesh (including the roof for the coupé case) is designable,
5) the whole structure is made of aluminum.

B. Performance targets

An automotive chassis must comply with both handling and safety performance standards. In the present study these structural requirements are set according to Ferrari SpA internal regulations. Their fulfilment is ensured by imposing optimization constraints in terms of admissible nodal displacements, or modal responses. These quantities are measured from finite elements simulations. In particular, the performance requirements used in this work regard:

1) the global bending stiffness of the chassis structure: an elevated structural stiffness, both bending and torsional, ensures a better handling vehicle performance. For evaluating the bending stiffness the four wheel centres are constrained in the finite elements model and the sills are loaded vertically. It is required that the displacement in the loaded area remains below a given threshold.
2) the global torsional stiffness: three wheel centres are constrained and the fourth wheel centre is loaded vertically. It is required that the displacement of the loaded wheel centre remains below a given threshold.
3) the crashworthiness in case of front crash: for being granted the homologation approval, vehicles must show a good performance in case of several crash conditions (*e.g.* frontal, rear, lateral crash, and so on) according to the various safety standards such as Euro NCAP or US NCAP. In the present work the frontal crash alone is taken into consideration. However, due to the inherent restrictions of the topology optimization software employed, it is not possible to include dynamic and non-linear load cases into the optimization process. Thus, the crash analysis had to be simplified into a static linear loading condition. The front surface of the domain is constrained while concentrated loads along the rear-to-front direction are applied to the locations where the main vehicle hanging masses are connected to the chassis structure: wheel centres, seats, engine, and gearbox joints. It is required that the passenger compartment undergo limited deformations under these loads. In particular, the displacements are monitored at several locations along the seats and engine joints, the A-pillar, the pedal area, the flame-shield, the dashboard under the windshield. An additional constraint is added for monitoring the overall compliance of the structure, since it has been seen that elevated compliances can cause difficulties to the finite elements analyses. Since a proper regulation for such a loading condition is missing, the threshold values for the allowable deformations are set equal to the displacements found by applying the same loading condition to a reference chassis model. A check will be needed *a posteriori* to

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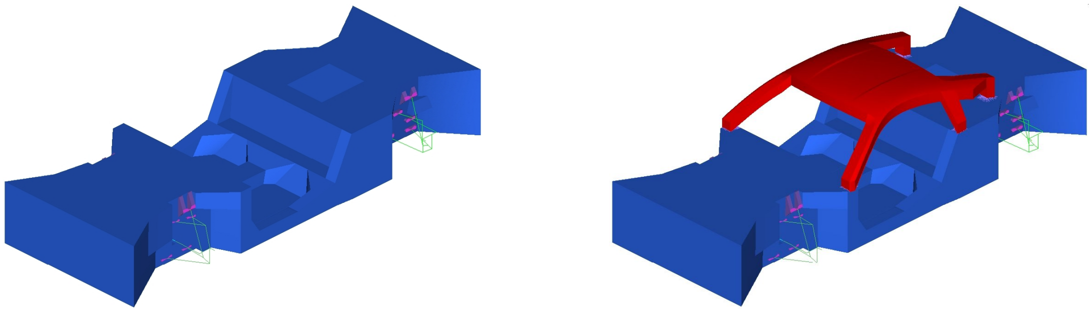

(a) (b)

Fig. 1. Topology optimization domains: (a) spider vehicle chassis, (b) coupé vehicle chassis.

ensure that in case of dynamic non-linear simulations the behaviour of the vehicle in the event of crash respects the performance standards.

4) the modal analysis: the natural frequencies inherent to global structural modes are directly related to the structural stiffness. For this reason, another way for enforcing an elevated stiffness and a good handling performance, is by ensuring elevated natural frequencies. It is required that the first bending and torsional natural mode frequencies remain above a given threshold.
5) the local stiffness at the suspension, engine, and gearbox joints: the sills are constrained while selected points are loaded along the coordinate axes. In particular the load are applied to the front and rear wheel centres along the three coordinate axes, and to the engine and gearbox joints along the vertical directions. It is required that the displacement of the loaded points remains below a given threshold. Since a proper regulation for these eight loading cases is missing, the threshold values are set equal to the displacements which are found by applying the same loading condition to a real Ferrari chassis chosen as reference model. Ferrari internal regulations would actually require an inertance analysis over a wide range of frequencies for local stiffness evaluation at each joint in the structure, but the high CPU requirements that such an analysis would involve makes of it a non viable option to be implemented in a topology optimization task.

The numerical values of the optimization constraints differed between the spider and the coupé cases since a coupé layout, due to the presence of the additional roof structure, is supposed to be stiffer. The chassis domain is symmetrycal along the spanwise direction, and so is the mesh. It is required that the topology optimization process mantains the symmetry in the solution.

III. Results

The model set up and the topology optimizations were performed using the software suite *Altair HyperMesh 10* and its embedded optimization tool *OptiStruct*.

The optimizations on the two models were performed by increasing step by step the complexity of the process, *i.e.* adding the optimization constraints one at a time, and repeating the optimization. This made it possible to check the consistency of the solutions, and have a better understanding of the results. Six test cases are taken into consideration and their relative set up is resumed in Fig. 2 and Tab. I which show the trend of the chassis mass as optimization constraints are added, and the active optimization constraints at the end of the process.

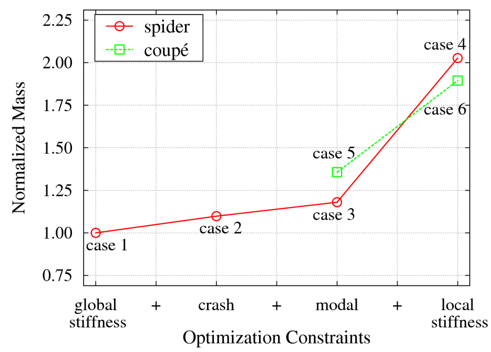

Fig. 2. Trend of the chassis mass as constraints are added to the topology optimization process; the mass is normalized over the mass of case 1. The masses for cases 5 and 6 include the mass of the roof.

TABLE I
Active constraints after topology optimization:
✓ = active, ✗ = inactive, – = not applicable.

Active Optimization Constraints	Cases					
	Spider				Coupé	
	1	2	3	4	5	6
Global bending stiffness	✓	✗	✓	✓	✓	✓
Global torsion stiffness	✓	✓	✓	✓	✓	✗
Crash seat joints displacement	–	✗	✗	✗	✓	✗
Crash engine joints displacement	–	✓	✓	✗	✓	✓
Crash A-pillar displacement	–	✓	✓	✓	✓	✓
Crash pedal displacement	–	✗	✓	✗	✗	✗
Crash flame shield displacement	–	✓	✓	✗	✗	✗
Crash dashboard joints displacement	–	✗	✗	✗	✗	✓
Crash compliance	–	✓	✓	✓	✓	✓
First natural mode	–	–	✓	✓	✓	✓
Local front wheel stiffness along x	–	–	–	✓	–	✓
Local front wheel stiffness along y	–	–	–	✓	–	✓
Local front wheel stiffness along z	–	–	–	✓	–	✓
Local rear wheel stiffness along x	–	–	–	✓	–	✓
Local rear wheel stiffness along y	–	–	–	✓	–	✓
Local rear wheel stiffness along z	–	–	–	✗	–	✗
Local engine joint stiffness along z	–	–	–	✓	–	✓
Local gearbox joint stiffness along z	–	–	–	✓	–	✗
Total	$\frac{2}{2}$	$\frac{5}{9}$	$\frac{8}{10}$	$\frac{12}{18}$	$\frac{7}{10}$	$\frac{12}{18}$

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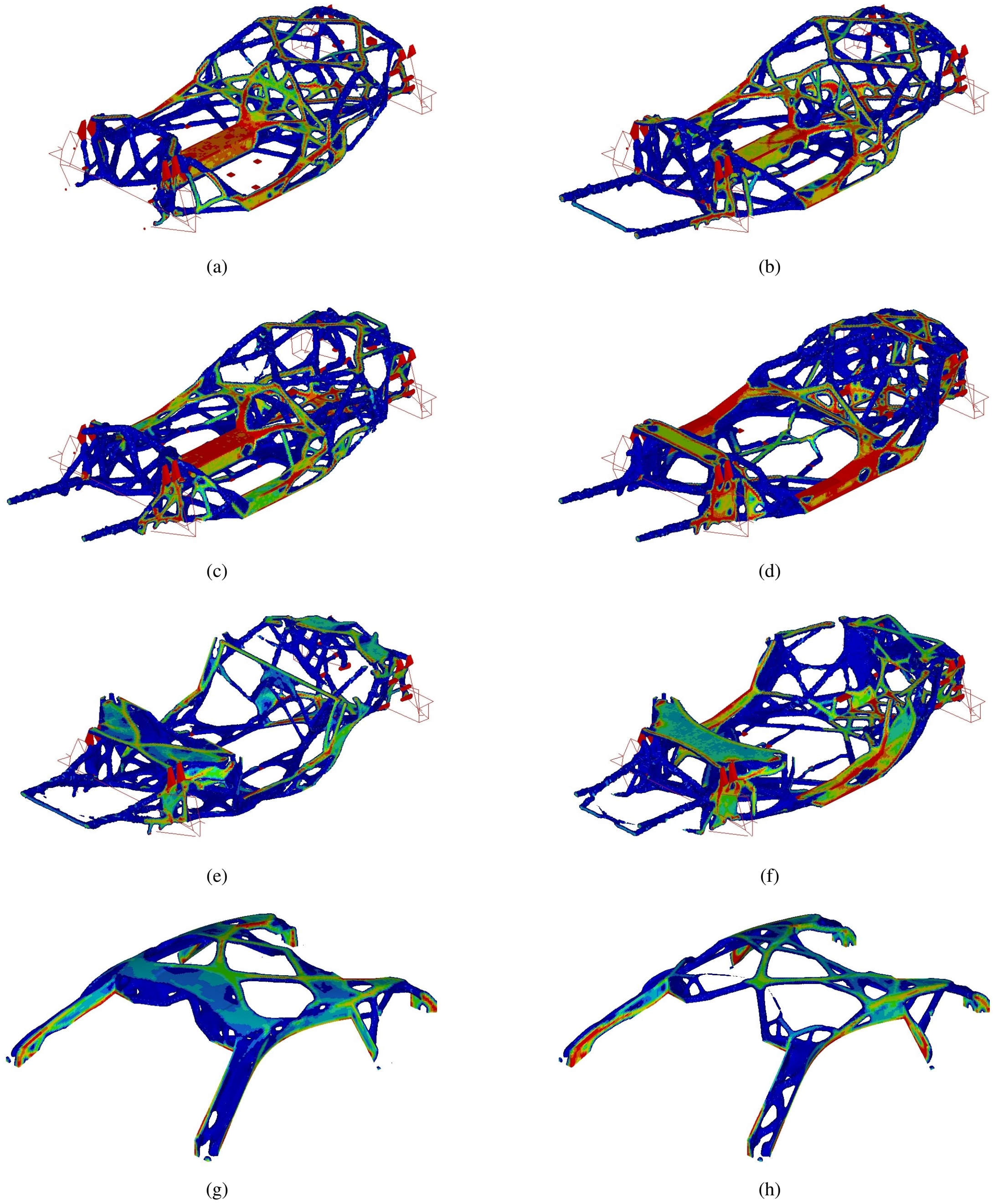

Fig. 3. Topology optimization results: figures (a) to (f) refer to cases 1 to 6 respectively; (g) and (h) are the roofs obtained for cases 5 and 6. The different colours stand for different material relative densities with blue being 10 % and red 100 %. Elements with relative density lower than 10 % are not shown.

It can be seen that despite the growing number of constraints from case 1 to case 6, a relatively large fraction of them is still active at the end of the optimization processes (73 % on average). This also negatively affects the convergence of the optimizations which is not always obtained straightforwardly when the problem is over-constrained. Important constraints, which are always found to be active, regard the A-pillar displacement and the compliance of the structure in the event of crash, and the first natural mode. Most critical are also the local stiffness constraints since most of them are always active, and they heavily affect the final mass of the topologically optimized chassis.

Figure 3 shows the results of the six optimizations performed. Case number 1 (Fig. 3(a)) involve global stiffness constraints alone. As a consequence, the seats, engine, and gearbox joints, as well as some suspension joints, not being loaded, are not connected to the rest of the structure. The loads between the front and the rear are transferred both through the sills and the central tunnel.

In case number 2 (Fig. 3(b)) due to the addition of the crash loading conditions, bumpers in the front and connections to all the junctions appear, this allows the load across the central tunnel to be slightly lower.

The inclusion of the modal constraint (Fig 3(c)) brings to a slightly simpler structure and to a significant reinforcement of the bumpers support area.

Case number 4 (Fig. 3(d)) gives rise to a much more complex and different chassis layout. The whole structure

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is thickened and the number of beams on the rear part is much larger. It must be considered that loads along the lengthwise and the vertical directions in the suspension areas were already included also while evaluating the global structural stiffness and the frontal crashworthiness. Thus, it was expected that the most critical loading conditions during the assessment of the local stiffness of the suspension joints would have been those along the spanwise direction. In fact, in Fig. 3(d) a large spanwise structure connecting the front suspension joints appears on the upper part of the domain, and the transversal beams linking the suspension joints area to the central tunnel on the lower part of the domain are doubled in size. The tunnel itself disappears and is substituted by a web of beams going from the front end of the former tunnel up to half the way along the sills. Such a sensible change in the layout and such an increase in weight for what it should have been a "local" stiffness issue must be considered carefully. In fact, it can be argued that such a behaviour is mainly due to the loading condition chosen for the evaluation of the local stiffness which involves the clamping of the sills, so that the lines of force must go through the sills, thus making the central tunnel superfluous. However, this is not a loading condition to which a vehicle is actually subject while on the road. Other loading conditions have been tried in substitution of the inertance analysis, but all of them are prone to similar issues. On the other hand, it is out of doubt that the previous structure (Fig. 3(c)) lacks reinforcements along the spanwise direction, which are properly included in the solution proposed in Fig. 3(d).

The cases 5 and 6 (Figs. 3(e) and 3(f) for the chassis layouts, and Figs. 3(g) and 3(h) for the roof layouts) are coupé versions of the cases 3 and 4 respectively. In these cases, the beams are much thinner compared to their spider versions, as a consequence of the fact that a portion of the load is carried upwards through the roof of the vehicle. This also allows the final structure to be much simpler (a lower number of beams is found in the solutions), and the central tunnel to disappear completely. Despite the fact that in case 5 (Fig. 3(e)) local stiffness load cases are not included, spanwise structures appears both on the front and on the rear of the chassis. These structures converge towards the areas in which the roof is connected to the rest of the chassis and serve to transfer the loads between the chassis and the roof. The spanwise structures are still present and even reinforced in case 6 (Fig. 3(f)) due to the local stiffness loads. It is interesting to notice how the bumpers in both the copué versions are more distant from the ground level, are not parallel to each other but slightly converging, and their two support beams are less slanted and meet nearer to the car front. These things suggest that the lines of force in the event of front crash are going to remain higher on the level ground and be pushed towards the sides of the car in order to go through the roof structure.

The two roof layouts (Figs. 3(g) and 3(h)) are similar in that apart from having two sets of beams running all the way across the left and the right side of the car, they have an elongated "X" shape on the top with some reinforcements to the rear. The structure in Fig. 3(h) is thinner and has a reticular front doorpost as a result of having assumed a relatively thick topology domain around the doorpost area.

IV. Conclusions

The methodology adopted for automotive chassis design by means of topology optimization has been proved to be effective, being able to trace an optimum draft design for the chassis by itself, by simply starting from a massive design space. It must be considered that the problem set up phase must be addressed carefully since the outcome of the optimization can be significantly affected by the choice of the optimization objective, constraints, and simulations means. This makes no exception, since every optimization problem is always deeply and intrinsically dependent upon these aspects.

The methodology has been applied in order to design chassis for both spider and coupé vehicles. Different layouts has been found by topology optimization and have been compared and discussed in the present work. The results are consistent with the loading conditions applied and the performance targets imposed which are derived from Ferrari SpA internal regulations. The comparison made, allowed a better understanding of the way the chassis works and on the way in which the lines of force are transported across the structure.

Addressing a coupé vehicle, it was found that a suitable exploitation of the presence of the roof from a structural point of view allows a remarkable stiffnening of the chassis, which is always a welcomed characteristic in view of improving the vehicle handling. The other way round, for given stiffness targets, significant savings in terms of structural weight can be achieved. Thus, a joint approach to design, able to address the structural role of the roof and of the chassis at the same time, aided by collaborative optimization techniques, is recommended, and can lead to innovative and more efficient solutions.

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