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Optimization Methodology for Automotive Chassis Design by Truss Frame: A Preliminary Investigation Using the Lattice Approach

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Abstract. The present paper investigates the application of optimization methodologies to vehicle chassis in view of an integrated and transdisciplinary vehicle design. A detailed sizing optimization in cascade to Topology Optimization (TO) for the design of automotive chassis is investigated. This approach is also known as lattice optimization. The TO is employed to find a coarse optimum chassis lay-out under linear elastic conditions. The second stage of the methodology converts into a truss frame the edges of the FE cells, including only the elements that remained above a certain density threshold after TO. The diameter of each truss is then optimized in view of chassis weight reduction while meeting a set of given design requirements, such as maximum stress and minimum size. Various tubular frame architectures for lightweight solutions are considered complying with different sets of constraints over different design spaces. Finally, the balance between the computational cost and the feasibility of the lattice solution is discussed in comparison to TO.

Keywords. Finite element modelling, Lattice optimization, Multidisciplinary optimization, Automotive chassis design, Truss structure optimization

Introduction

Weight saving is a crucial target for car manufacturers, since it allows vehicles to achieve better structural performance while containing fuel consumption. However, making a lighter chassis generally means to reduce its stiffness and crashworthiness, therefore a suitable compromise has to be found. Balancing between these two aspects, over the years many different types of chassis structures have been adopted in the automotive industry, ranging from backbone to spaceframe chassis [1]. Optimization techniques have been introduced into the design process, in order to find the lightest car framework that fulfils the manufacturer's requirements and the vehicle type approval. In fact, the whole industrial project leading to the design of a new vehicle is extremely vast and transdisciplinary, and precise sets of requirements are defined for many different aspects. Thus, in the recent years the chassis design is necessarily developed within an integrated framework. Concerning the chassis, the main areas of interest

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relate to geometrical and structural constraints [2]. The former are due to the space needed by the many vehicle components and by the passenger compartment for comfort reasons, and also the aesthetic plays an important role. The latter are due to safety, handling, fuel consumption, and ergonomic reasons.

In 2011, Cavazzuti et al. [3] were the first to apply topology optimization to a whole car body with a view to weight saving. Nowadays, TO represents a valid and a commonplace work tool applied during the design of vehicle components for structural performance [4-5]. In 2017, Barbieri et al. [4] presented the methodology for the design of a motorcycle piston in steel manufactured via an Additive Manufacturing process. However during the TO process, a periodic truss microstructure of the piston had not been integrated, and the TO results had been re-interpreted manually. The integration of the manufacturing constraints into the TO had been presented in Mantovani et al. [5], for a complex component like an automotive dashboard in order to obtain results gradually feasible. Optimization, however, comes with a high computational effort, so that a single run can last days or even weeks to be carried out. In order to make the early stages of the automotive design process faster and more flexible, a reduction of this computational cost becomes necessary. This can be achieved by thinking of a car chassis as a truss structure and then optimizing this specific type of vehicle layout. Gauchia et al. (2010) [6] used this approach to optimize a bus chassis, starting from an initial framework and appropriately sizing its members by means of genetic algorithms to improve the structure torsional stiffness while containing its weight.

Since truss optimization has a wide range of applications beside automotive, many different numerical methodologies can be found in the literature. In 2007, Achtziger [7] proposed a deterministic method to simultaneously optimize both the cross-sectional areas and the position of the joints for in plane truss structures. Some years later, Sonmez (2011) [8] presented his stochastic method inspired by bees social behaviour for fixed 2D and 3D geometry beams sizing. In 2014, Zegard and Paulino [9] introduced the Ground structure Analysis and Design (GRAND) method for highly interconnected truss structures. This method "provides an easy-to-use implementation for the optimization of least-weight trusses embedded in any domain geometry". The following year they extended this method from two- to three-dimensional domains [10].

In this paper the application of size optimization in cascade to topology optimization for designing a tubular automotive chassis [1].

. This approach named in the following *lattice optimization* is able to automatically change the solid design domain into a truss-like domain and optimize it.

The present paper investigates whether the results obtained in this way are compatible with the industrial product development process, in the context of computational cost reduction in favour of design versatility.

Topology, size and lattice optimization

In order to fully understand the topics presented in this paper, a brief introduction about some of the main structural optimization strategies, in particular topology and size optimizations, is necessary. Thereafter, it will be shown how lattice optimization operates as implemented in *Altair Optistruct 2017*.

Topology optimization intends to find the optimal material distribution for a structure to minimize an objective function (such as the mass) while complying with given performance constraints for certain load cases. In this paper, such material distribution is obtained through the SIMP method (Solid Isotropic Material with Penalization): the optimization domain is discretized in 3D elements and each one of them is assigned a relative density ρ_r, defined as

$$\rho_r = \frac{\rho_{FE}}{\rho_{mat}} \tag{1}$$

where ρ_{FE} is the element density and ρ_{mat} is the density of the design domain material. The relative density can vary with continuity between 0 and 1 and indicates how much a finite element is "permeated" with material: the value 0 represents the void, while the value 1 represents a solid element. A penalty factor is applied to the optimization process to force the solution toward a discrete topology, in other words to penalize the elements with intermediate relative density. A factor equal to 1 produces no penalty while higher values cause greater disadvantage in terms of decrease of elements stiffness (*i.e.* Young modulus). The relative densities are the design variables of the TO, namely the quantities that can be varied and that determine the value of the objective function to be minimized. Once the design domain is discretized, the load cases of interest are applied and the respective performance targets are set in the form of optimization constraints. More extended bibliographical information on TO is available in [11].

Size optimization is a refinement process which operates on an already defined topology. It aims at minimizing the objective function by finding the optimal dimension for each element of the structure. Typical design variables are, for instance, the thickness of 2D shell elements or the radius of 1D beam elements. Similarly to topology optimization, load cases with respective optimization constraints must be applied to the design domain.

Lattice optimization combines the two previous strategies, as follow:

1. At first, a TO is carried out after which the 3D elements of the design domain are substituted by 1D beam elements created along their edges; the beams initial size depends on the relative density of the solid element they replace. The 1D elements are tapered and have circular section: each of them is characterized by two radii, one for each end, so that every grid point can have concurring beams with equal cross-section. However, the maximum section radius is limited by the initial 3D element size, thus larger solid elements can generate thicker (and longer) beams.
2. In the second phase a size optimization of the beam elements takes place resulting in the final optimized truss structure.

1. Model set up

1.1. Design space

The first step of the model set up consists in the definition of its design space, that represents the domain of the optimization problem, where the optimal material distribution of the chassis is pursued. The vehicle design space takes into account:

- the volume occupied by the vehicle main parts, such as the engine, the gearbox, the steering system, the passenger compartment, *etc.*
- the wheelbase and the track of the vehicle
- the suspensions layout and their joints position, as well as the location of other non-design elements
- the external dimensions and shape of the vehicle

The design space FE model is directly derived from the one developed by Cavazzuti *et al.* (2011), [1]. It consists of solid tetrahedral elements of the first order (4 nodes each) with one Gauss integration point. The suspensions, which are non-design domain, are modelled as 1D beam elements interconnected by elastic 0D joints.

In this paper, we take into consideration two variants of the same design space which differ in the presence of non-design front crash absorbers, modelled as 1D beam elements with given cross-section and material. The element count is also different: 914 for the variant with the crash absorbers and 962 for the other one. The former is shown in Figure 1; the non-design absorbers, embedded in the solid mesh, are depicted as red bars. As we will see in the following, an additional load case representing the front crash is applied to this version.

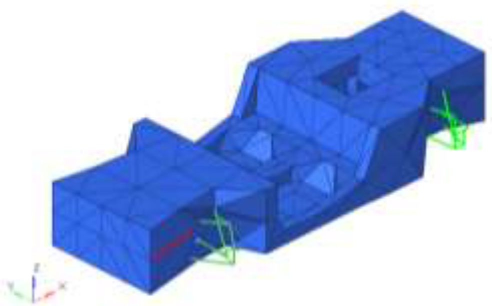

Figure 1. Discretized design space.

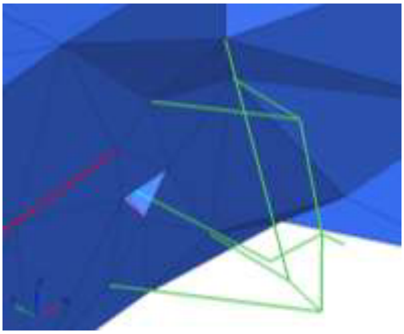

Figure 2. Particular of suspension joints.

Due to lattice optimization characteristics, the design space geometry is required to be as simple as possible, in order to obtain as large elements as possible. In this way the tapered beams radii might vary in a wider range of allowable values during the size optimization. As a consequence of this assumption the suspensions, seats, engine and gearbox mounts are modelled as single-node connections, rather than multi-node connections, as doing so would raise the number of nodes and thus decrease the size of the elements. Specifically, as shown in Figure 2, the suspensions spherical joints are obtained via RBE2 elements between two coincident nodes.

Such a coarse mesh is not expected to give accurate topology results, but it is necessary to generate the initial truss structure.

1.2. Load cases and constraints

The two optimizations (topology and size) are set with the same objective and constraints. The objective is the minimization of the chassis mass while the constraints are given in terms of maximum allowable displacements or compliance for a given set of load cases. These concern:

- Global vertical bending stiffness: the wheel centres in the FE model are constrained and the sills loaded with a vertical force F at the wheelbase middle point (Figure 3). The vertical displacement of the nodes of the sills around the loaded area is required to remain below a certain threshold Δz, so that the target bending stiffness

$$k_f = \frac{F}{\Delta z} \left[\frac{\mathrm{N}}{\mathrm{m}}\right] \tag{2}$$

  is achieved.
- Global torsional stiffness: one of the front wheel centres is loaded with a vertical force while the remaining are constrained (Figure 4). The vertical displacement of the loaded point is required to remain below a certain threshold, so that the target torsional stiffness

$$k_t = \frac{M}{\Delta \alpha} \left[\frac{\mathrm{Nm}}{\mathrm{rad}}\right] \tag{3}$$

  is achieved, where M is the moment applied to the chassis and $\Delta\alpha$ is the maximum allowable angular displacement between the front and the rear wheel centres sections. It is noted that since this load case is not symmetrical to the longitudinal middle plane of the vehicle and the size optimization does not feature symmetry conditions, two specular load cases are necessary.
- Local stiffness of the suspensions joints (Figure 5): the sills are clamped and each wheel centre is loaded with a force F with components in the x, y, and z directions (four load cases altogether). It is required that the displacement of the wheel centres remains below a certain threshold Δs.
- Crashworthiness (only for the variant featuring the front crash absorbers): the inertial forces acting on the chassis in the event of a front impact are modelled with equivalent static forces applied to the seats, engine and gearbox mounts and to the wheel centres. The constrainment is applied to a row of elements in the front of the design space, including the extremities of the crash absorbers (Figure 6). It is required that the displacement of several points of the passenger compartment remains below a certain threshold and that the structure's compliance does not exceed a given value. The displacements are collected at the seats, the pedal area, the door hinges, the engine mounts, the flame shield, and the cowl. Being this load case the outcome of a linearization, the capability of the chassis to fulfil all the crash requirements must be checked *a posteriori*, after the optimization process is completed.

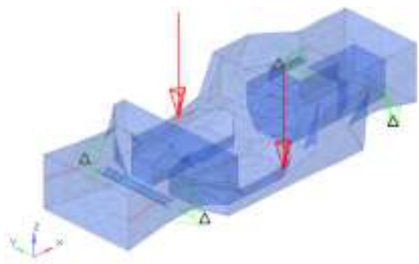

Figure 3. Global bending stiffness.

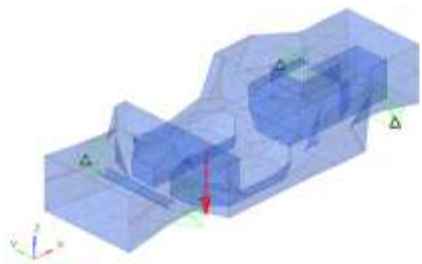

Figure 4. Global torsional stiffness.

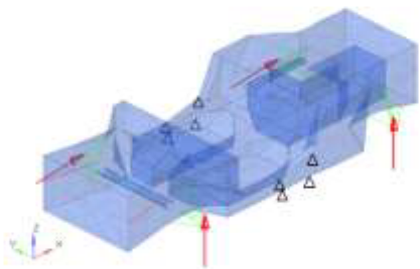

Figure 5. Local stiffnesses.

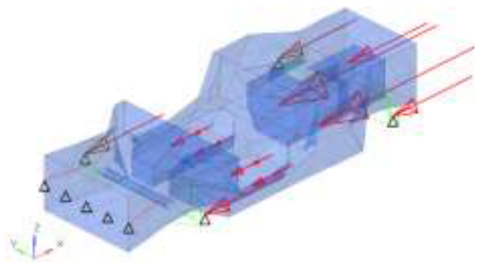

Figure 6. Crashworthiness (linearized front crash).

1.3. Optimization parameters

Lattice optimization is performed on the model shown in Figure 1. The solver used for the optimization is *Altair OptiStruct 2017*, while the model was set up with *Altair HyperMesh 2017*.

All the parameters for both the optimization phases are set at the same time. For the topology optimization the most significant parameter is the penalty factor, which can be chosen among three possible values: 1, 1.25, and 1.8. Since a truss structure is required, there is not need to force the relative densities towards discrete values, thus, the penalty factor is set to 1. In other words no penalty is given to elements having intermediate density. In this particular application, the material distribution found through the first phase is of limited interest since the mesh is too coarse to be able to give a sensible outcome. However, depending on the relative density assigned to each element it is possible to decide whether to delete the element, to replace it with a lattice structure or to keep it solid. A very low threshold was set to delete elements characterized by very low density, while no solid element was pursued.

The design variables of the second phase (size optimization) are the newly generated tapered beams radii. The maximum value they can take depends on the size of the element from which they derive, while the minimum can be set in terms of minimum radius or minimum radius to length ratio. Strictly correlated to these parameters is the clean filter: if active, at the end of the optimization the beams featuring radii below the threshold are deleted. Even though this functionality may prove useful for obtaining a feasible structure, the drawback is that the limit has to be set *a priori* and is not embedded in the calculus. This is a flaw which can make this optimization process inefficient, since it could take many attempts to set the right clean filter's threshold. If this parameter is set too high then necessary elements are deleted, while if it is set too low then too many unnecessary beams are kept, making the structure uselessly complex and heavy.

The size optimization also features an additional constraint, namely the maximum stress value for the tapered beams. This constraint is particularly significant when combined with load cases featuring non-scalable loads and constraints, such as the linearized front crash for this model. On the other hand, attention must be paid when global stiffness based load cases are present, since here the focus is not on the local stresses themselves. The maximum stress constraint has a strong influence on the optimized beams section and, combined with the clean filter, can be exploited in order to avoid too thin a framework.

2. Results

The optimizations were carried out in parallel, on an Intel i7 1.73 GHz 4-core CPU with 4GB of RAM. The results were obtained with very little computational effort, with an overall run time not longer than 10 minutes. For this reason, the computational costs are not a real issue as they would be with an ordinary topology optimization approach. Figure 7 derives from the first variant of the model, which does not feature the linearized front crash load case nor the crash absorbers. This is evident, since no beam was kept in the front part of the chassis, except for the ones connected to the suspension joints. It can be noted that the beams of the sills are robust (considering their cross-section area and inertia) compared to the others, since here the dominant loads are torsion and global vertical bending.

Figure 7. First model variant results (maximum admissible stress 5 MPa).

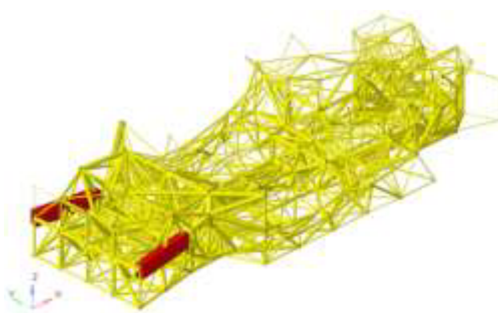

Figure 8. Second model variant results (maximum admissible stress 20 MPa).

The maximum admissible beam stress is set to 5 MPa: otherwise, the stiffness constraints alone causes the beams to be too thin and thus to be deleted by the clean filter. Such a strict stress constrain is thus necessary to see the consequent structure, but also makes it oversized for the stiffness target. Therefore, this result is of little practical interest, but it shows one of the major flaws of this method: since the clean filter is not part of each iteration of the optimization, fundamental beams can be deleted and the remaining ones are no further optimized considering this deletion.

When the second variant of the model is considered, the resulting truss appears significantly more robust, due to the front crash load case, as can be seen in Figure 8.

The non-design crash absorbers are depicted in red and with their actual cross section. Despite their presence, the front part of the chassis is characterized by an apparently unnecessary massive beam structure. This is due to the maximum allowable beam size, strictly dependent on the dimensions of the solid FE from which it derives. In fact, during the size optimization, when the solver cannot further increase an element radius, it thickens the adjacent ones in order to fulfil the stiffness requirements. Another limit of this method then emerges: the restriction to the radii obtainable values strongly limits the structure performance, forcing the solver to give inefficient solutions.

Compared to the previous chassis, it can be noted that the sills are thinner while the central tunnel is much thicker. This is due to the addition of the front crash load case, which require the structure to hinder the compressive equivalent static loads applied to the engine and gearbox mounts.

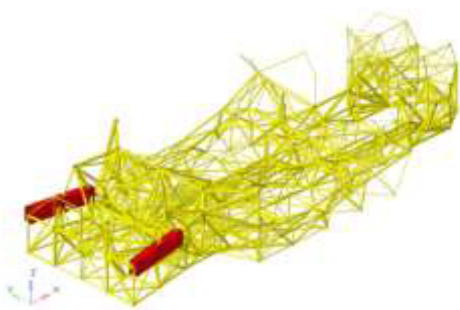

Figure 9. Second model variant results (maximum admissible stress 40 MPa).

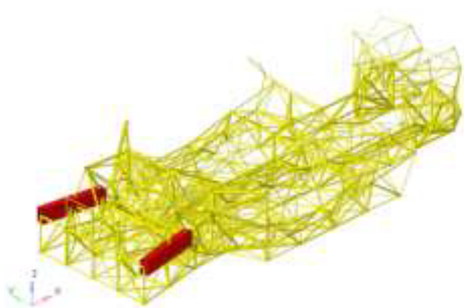

Figure 10. Second model variant results (maximum admisible stress 50 MPa).

The results shown in Figure 8 are obtained setting the stress constraint value to 20 MPa, in order to avoid the deletion of too many elements by means of the clean filter. Higher values of maximum allowable stress cause slight violation of some constraints. Figure 9 and Figure 10 show the results for the same model variant with maximum stress value set to 40 MPa and 50 MPa, respectively. This constraint combined with the clean filter causes the beams to be overall thinner and fewer. As anticipated above, in these two last cases the requirements in terms of global bending stiffness and local stiffness of the rear shock towers are not fulfilled.

In all the resulting chassis a slight asymmetry can be observed: this may be due to the computational noise on the radii values that are very close to the threshold set by the clean filter.

3. Conclusions

Lattice optimization is a promising technique primarily meant for periodic additive-manufactured microstructures design. Here, it has been applied to the design of an automotive chassis in order to test and to stress its capability in an uncommon context.

The mesh has been kept very coarse in order to limit the number of trusses favouring the feasibility of the final solution, taking hint from the idea of tubular chassis. The goal is to compare lattice to more consolidated approaches like topology optimization. Unfortunately, the results of lattice left almost no room for interpretation, whereas this is a distinguishing feature of topology.

Even though the lattice analysis required a very short amount of CPU time to be completed compared to topology, the results show that this methodology still needs to be refined in order to be used in an integrated vehicle design process. The biggest flaw of this optimization strategy is that its outcome is strongly affected by the initial mesh and the design domain, making it impossible to draw close to a feasible solution in case of such a complex structure.

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