Original Article

Dynamic size optimization approach to support railway carbody lightweight design process

Alessio Cascino ⓘ, Enrico Meli ⓘ and Andrea Rindi

Abstract

The transition to a globalised electrical power supply pushes rolling stock manufacturers to find structural solutions for reducing the power required by the vehicles. Within a market where details can make the difference in economic terms, the concept of design optimization is becoming established. However, current industrial procedure for the evaluation of carbody structural strength, in static field and according to EN 12663-1:2015 standard, does not include any optimization process. More in general, optimization processes, and especially dynamic optimization, are not widely used for designing of railway vehicles. In this framework, the present paper proposes a new dynamic optimization approach to support the design of railway vehicle carbodies subjected to static loads. Proposed methodology aims to minimize the mass of the metallic structure, working on the thicknesses of the optimized components, maintaining the baseline geometry. The constraint function was imposed on the first natural vibration frequency of the system. The optimization strategy is based on a dynamic size optimization process in conjunction with modal analysis techniques, applied on the single carbody shell. The procedure has involved the roof assembly of a single tram vehicle body. The proposed approach is resulted numerically efficient in terms of calculation times. Encouraging results have been achieved in terms of mass saving and mechanical behaviour of the carbody shell. Optimized components were 21% lighter than the original, that corresponds to 3% if evaluated on the total mass of the single carbody metallic structure. The methodology turns out to be a useful tool for supporting designers to reduce the mass of the carbody structures, reducing overdimensioning condition that could affect the carbody structure.

Introduction

Nowadays, due to environmental pollution caused by CO_2-emissions, a global transition to electricity is taking place within railway industry.[1,2] Lightweight design of railway vehicles allows to reduce the power required during their operation conditions. In addition, the lower is the mass of the car body, the lower is the mass of other components such as bogies and traction systems. Then, the possibility of reducing mass is confirmed as a crucial aspect in the future development of railway vehicles, not only for the carbody structures. Structural optimization processes do not yet have a well-defined role in the design procedure for railway vehicles carbodies, especially when dynamic optimization constraints are involved (including natural frequencies and vibration modes of the system). Available literature illustrates that not many studies about optimization processes, applied on railway vehicles carbodies design, have been carried out. This aspect is even more accentuated for tram vehicles, object of the present work. More in general, topological optimization and structural optimization processes, represent effective techniques for designing and improvement of several classes of components. Control arms for automotive applications have been optimized to

reduce mass.[3,4] Structural optimization design of an automotive vehicle body-in-white and topology optimization for high performance automotive chassis design, have been carried out in.[5,6] Topology optimization approaches, combined with additive manufacturing techniques, represent an effective design procedure for turbomachinery components, as illustrated in.[7–11] Innovative geometries have been proposed, thanks to the possibility of changing the distribution of material in space. This concept represents a key point when designers need to choose the most suitable optimization approach for their benchmark. Referring to railway field, the importance of sealing the carbody due to the aerodynamic conditions of the vehicle,[12] could preclude the adoption of topology optimization which could open several holes in design volume. The present activity had the objective of maintaining the baseline geometry of the carbody, optimizing the thicknesses of the reference extrude

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profiles, respecting vehicle dynamic constraints in terms of frequency. Other research activities, in railway sector, have illustrated different design approaches. For instance, size optimization and a material selection method have been combined to test an electrical multiple unit (EMU) carbody.[13] An optimized hybrid body has been obtained with a mass saving of 29%. Topology optimization technique has been used in the lightweight design of a railway carbody, including crashworthiness performance within the methodology.[14] A multidisciplinary design optimization method of a lightweight carbody for fatigue life prediction was presented in,[15] where a rigid-flexible multibody system dynamics model of vehicle and finite element model have been developed to simulate the coupled interactions. Ribs, used in the body of a railway vehicle, have been innovated in shapes through an optimization process which included topology, shape and size optimization,[16] with the objective to reduce vehicle mass. A railway anchor bracket has been redesigned taking advantage of a topology optimization process and introducing composite materials.[17] Some studies were also performed for mass-reduction of rolling stock carbodies using innovative materials like Magnesium alloys, as reported in.[18,19] With the aim of achieving an acceptable running comfort condition, the decoupling of the carbody from bogie instability motion, as well as from the suspension frequencies,[20] must be considered. In,[21] the effect of bogie hunting on the carbody elastic vibrations, has been presented. The topic of suppressing the carbody flexible vibration has been deeply analyzed in,[22–24] considering the major effect of the mounted equipment.

Object of the present research was the carbody shell of a tramway vehicle. From a structural point of view, rolling stock carbodies are composed by three main assemblies: roof assembly, side-wall assembly and floor assembly, as shown in Figure 2(b). They have very different structural behaviours and perform totally different tasks. Nowadays, modern tramway vehicles have a low floor. This feature facilitates entry and exit by car even for people with disabilities. However, this important feature reduces the space under the carbody, barely enough for running gear (wheelsets, engines, transmission, brake system, etc.), forcing designers to use the roof for the positioning of many components. Because of this, it is important to understand that the roof is increasingly stressed and it must be considered a principal structural component.

In this article, authors present a design approach for rolling stock carbodies including a dynamic size optimization process. This procedure can be applied to the general design process in conjunction with EN 12663-1:2015 standard,[20] with the aim to design a lighter carbody complying with the structural requirements of the standard. The optimization process is applied on the single carbody without change its baseline geometry, acting on the thicknesses of the extruded profiles that make up the structure. It is combined with modal analysis technique in free-free conditions. Object of the present activity was the roof assembly. The proposed strategy allowed not only the reduction of the system mass, but also to respect dynamic constraints in terms of natural frequencies. The natural frequencies of vibration and mode shapes of the carbody are functions of the structural properties of a system. They are

affected by the stiffness and the mass of the structure such as the simplest cantilever beam. For this reason, the stiffness of the carbody structure is one of the main parameters that designers need to consider. The procedure respects this fundamental concept, researching the minimum value of mass that allows to maintain the required performance in terms of frequency of vibration and, indirectly, of stiffness.

Methodology

In this chapter, the methodology proposed by the authors to reduce the carbody mass, using a dynamic size optimization process, is briefly described. Starting from the project specifications and thanks to the experience of the designers, finite element models of the single carbody structure and the complete vehicle were constructed. Different steps of the procedure will be analyzed more in details in the following chapters. Figure 1 illustrates the flowchart of the methodology, that can be summarized as follows:

1. First, the complete finite element model of the vehicle must be tested according to the reference standard, EN 12663-1:2015. Load conditions are always applied on the whole vehicle and not that of the single carbody. First test on the structure is generally not a problem. In railway field, the
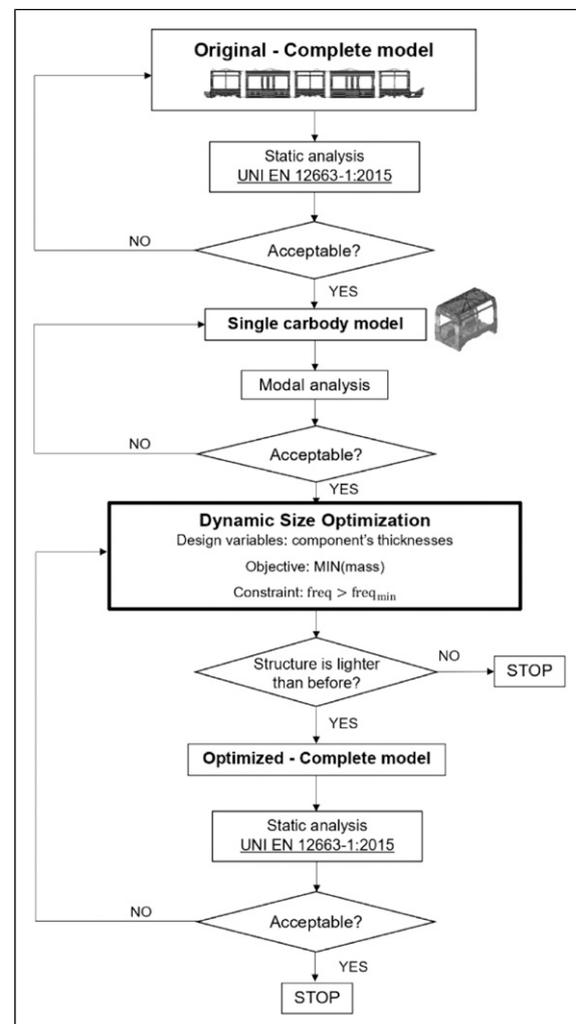

Figure 1. Flow chart of the proposed design methodology.

building of a new vehicle platform starts always from similar project that have been already completed by the company. Then, the experience of the designers become important. The preliminary design is often subjected to an overdimensioning condition: too large thicknesses make the structure very heavy, with no real advantage in terms of strength, which results already sufficient. An acceptable condition of the baseline model, in terms of strength, is fundamental to proceed with the second step of the procedure. It would not have sense to apply an optimization process on a structure which does not support the loads acting on its original configuration.

2. Second, starting from the finite element model of the vehicle, each carbody must be isolated with the objective to study its modal behaviour through a modal analysis in free-free conditions. The parameter of interest is the first natural frequency of the structure. Mainly, the lowest natural mode is the vertical bending mode, but the class of the modes could depend on the properties of the system analyzed: twist and compression deformation modes are also important in carbody design. To avoid resonances, as mentioned in,[20] a common practical design rule is to keep the first natural frequency of the carbody as high as possible. Focusing on the first vertical bending mode, discussed and observed during the present activity, it offers the main contribution to the vertical vibration in the middle part of the carbody. Rarely, these frequencies may be below 10 Hz, which is close to the vibration frequencies to which the human body is sensitive. Passenger comfort is always a priority. This step allows the designer to know the modal behaviour of the structure, in terms of frequencies and mode shapes, useful to define the constraint condition for the optimization step. If the first frequency of vibration would be too low, it means that the carbody needs to be redesigned with the aim of increasing this fundamental parameter.

3. Third step consists in the effective optimization process. It could act on very different parameters. Because of that, only a thorough comprehension of the problem allows to define the most correct optimization method useful in railway sector. The definition of these conditions and therefore of the entire optimization procedure, represents the crucial point of this activity. The optimization process is applied in combination with the modal analysis of the single carbody, realized in free-free conditions. It works on the thicknesses of the extrude profiles selected (design variables), changing their values within a predetermined range, without changing their baseline geometry. The objective of the analysis is to find the minimum of the mass function. The constraint condition is imposed on the minimum value of the first natural vibration frequency of the carbody, that must be higher than a predefined threshold value. This value represents the minimum,

permissible value, to respect the dynamic behaviour of the tested carbody. Alternatively, without further information about the dynamic of the vehicle, the designer could choose the value of the first frequency of vibration obtained from the first modal analysis, provided that it was greater than an acceptable value. This value should never be lower than 8 Hz. If the optimized carbody would not result lighter than before, it would mean that the structural material has already reached its best performance considering the geometry configuration of the carbody.

4. Once the optimization process on the single carbody is positively concluded, a lighter structure with the required first natural vibration frequency has been obtained. Then, the original complete model of the vehicle must be updated with new thickness values of the optimized components. These values, for each carbody, must be introduced in the complete finite element model of the vehicle, obtaining the final version of it. In order to conclude the procedure, the mechanical performance of the optimized structures must be tested according to EN 12663-1:2015 standard.

Tram vehicle model description

The tram platform was composed by five carbodies and three bogies. It was monodirectional and it ran in an urban area. Figure 2(a) illustrates the finite element model of the complete vehicle with carbody reference names. Two classes of carbody can be distinguished: suspended or not suspended. The two suspended carbodies (RA, RB) are not connected to a bogie and they are longitudinally longer than the other ones. Considering EN 12663-1: 2015 standard, this vehicle was part of P-V category, dedicated to passenger vehicles and in particular to tramway vehicles. This first classification defines the different load conditions with which to test the system.

The structure was totally made with aluminium alloys that could be different according to the component or the assembly considered. Table 1 illustrates some mechanical properties of these according to EN 1999-1-1:2014 standard.[25] As shown in Figure 2(b), each carbody was composed by three main parts: roof assembly, frame assembly and upright assembly. Each of them, was made with the union of aluminium extruded profiles through welding process and connected to others assemblies using riveting. Two cabines were joined to the end carbodies using bolting. This vehicle was monodirectional and the two cabins had different characteristics. One important note must be highlighted about the presence of a coupler in the front cabin named "CAB_MA": this tram platform would never be coupled with other trams during its service life and then it did not have a coupler. However, the possibility of several failures and consequent stops must be considered. Within the urban line several trams made by different manufacturers were already moving. Then the need for one solution which allowed to connect trams with different features was necessary and it was designed: it
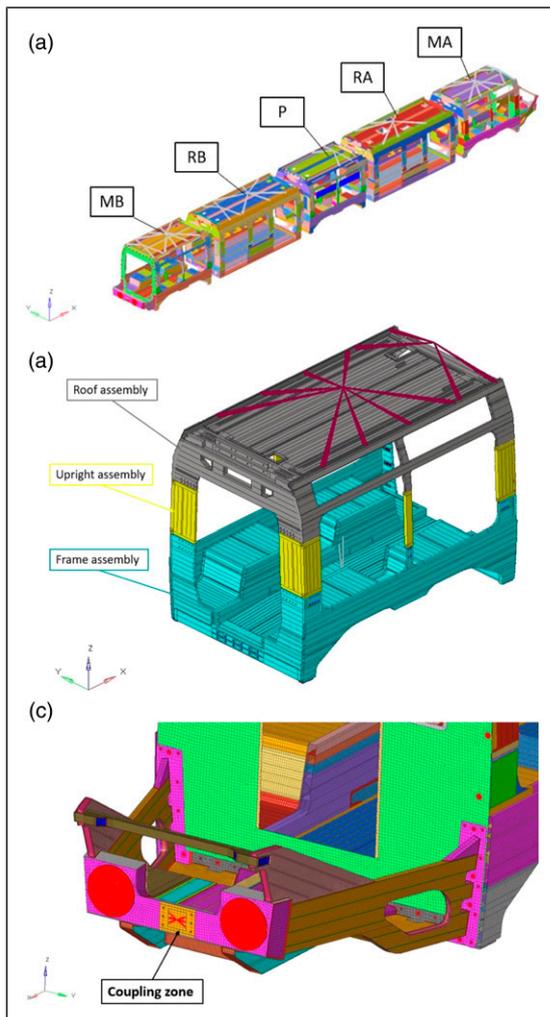
Figure 2. (a) Finite element model of the complete tram vehicle, (b) Main assemblies of the single carbody, (c) Front mask coupling zone.

consisted of a component mounted in the frontal area of the CAB_MA which made the coupling between two different vehicles possible. It was called "coupling mask". Despite the development of this component was not an objective of the present work, it was very important considering the loads applied for testing the carbody structure. The load that should be applied in the couplers zone has not been considered, while a new load was applied in the front mask zone. This aspect proved to be very important because it was not suggested by the reference standard but was a choice of the designers. Figure 2(c) illustrates the coupling zone in the frontal area of the CAB_MA. Material properties of Table 1 are referred to thickness of aluminium extruded profiles lower than 5 mm.

Dynamic size optimization of the suspended carbody roof

FE analysis of the tram platform

As described above, the reference standard required to test the complete model of the vehicle. The finite element model was composed almost entirely of shell

elements at first order. Table 2(a) summarizes the characteristics of it, including some specific information about dimensions of both types of carbody. The model had 25 contact interfaces. With the objective to perform a linear analysis, all were FREEZE type. Material model has been supposed linear elastic and isotropic, with card image MAT1. The activity has been carried out using HyperMesh 2017 and Optistruct solver. Table 2(b) shows the reference masses, as indicated in the reference standard, UNI EN 15663:2019.[26] Tram platform has been tested under different loading conditions, in agreement with the reference standard EN 12663-1: 2015; Table 2(c) summarizes them.

Constraint conditions were always represented by an isostatic configuration: the carbody was supported vertically at the secondary suspension, laterally at the side pads and longitudinally at the posterior buffers of the cabin. Simulation results have illustrated wide margin of strength. The structure has shown low maximum values of stress, compared to the reference limit of the materials (Table 1). This outcome depended on the design of the carbody, in terms of geometry and thicknesses of the extruded profiles. Reduced margins, instead, should suggest that the material has almost reached its limit performance. Then, with the goal of achieving an interesting mass saving from the subsequent optimization approach, either you change the material, or you vary the geometry of the system. In order to evaluate the effectiveness of the procedure, static stress and inflection, have been considered both for the complete vehicle and the single carbody. Permissible values have been respected. The comparison between the performance of the original structure and the optimized one, will be present in next paragraphs.

Modal analysis of the single carbody

As introduced above, modal behaviour of the single carbody is always evaluated with the aim of monitoring its first elastic frequency of vibration in free-free conditions. The decoupling between carbody and bogie motion, represents one of the most important conditions that must be ensure, with the aim of achieving running stability and comfort for passengers. Modal analysis has been carried out on all the carbodies. They have showed a first bending mode that involved the upper panel of the roof assembly and first frequencies equal or higher than 10 Hz. In order to validate the proposed procedure, only the most critical carbody has been tested using the optimization process. Next pictures and results will be referred to only one carbody, named with the abbreviation "RA". It was the most critical due to its longitudinal length, the several holes on the roof assembly and the heaviest equipment positioned over it. In addition to the mass of the aluminium structure, masses of the major equipment positioned on the carbody have been considered, with the objective to simulate a condition as real as possible. The comparison, in terms of frequencies, is illustrated in Table 3(a). Figure 3 shows the finite element model of RA carbody.

Table I. Mechanical properties of the cited aluminium alloys.

Material	Density	Young's modulus	Poisson's ratio	Proof strength (0.2%)	Ultimate tensile strength	Proof strength (0.2%)	Ultimate tensile strength	Minimum elongation
				Base material		Welded material		
	[kg/m³]	[N/mm²]	[—]	[N/mm²]		[N/mm²]		[%]
EN AW 6005 T6	2700	70000	0.30	215	255	115	165	8
EN AW 6106 T6	2700	70000	0.30	200	250	95	160	8

Table 2. (a) Characteristics of the vehicle model, (b) Reference masses, (c) Loading conditions according to EN 12663-1:2015.

(a) Complete model	
Number of nodes	6,119,166
Number of element	5,599,665
Suspended carbody	
Width [mm]	2390
Length [mm]	5400
Height [mm]	2780
Not-suspended carbody	
Width [mm]	2390
Length [mm]	3720
Height [mm]	2780

(b) Carbody mass [kg]	
C0	Tare condition
C2	C0 + seated passengers + standing passengers (4 pass. For square metre)
C4	C0 + seated passengers + standing passengers (6 pass. For square metre)

(c) Loading conditions	Load calculation formula
Max vertical load	Vertical load = 1.3*g* C4
Compressive load on the buffers in C0 condition	Vertical load = C0 * g Longitudinal compressive load on the anterior buffers = 200000 N
Compressive load on the buffers in C4 condition	Vertical load = C4 * g Longitudinal compressive load on the anterior buffers = 200000 N
Tensile load on the drawbar in C0 condition	Vertical load = C0 * g Longitudinal tensile load on the drawbar = 55000 N
Compressive load on the drawbar in C0 condition	Vertical load = C0 * g Longitudinal compressive load on the drawbar = 100000 N

Dynamic size optimization process

The goal of the size optimization process is to find the minimum value of the thickness that could meet the objective function and the design constraint function of the problem. The formulation of a structural optimization problem is described in,[27] as follows:

$$\text{Find } b \in R^n, z \in R^l \tag{1.1}$$

$$\text{to minimize} f(b, z) \tag{1.2}$$

$$\text{subject to } K(b)z = f \tag{1.3}$$

$$g_j(b, z) \leq 0; j = 1, \ldots, m \tag{1.4}$$

$$b_L \leq b \leq b_U \tag{1.5}$$

From the first line to the last one, $b \in R^n$ represents the design variable vector, while $z \in R^l$ is the vector of nodal displacements. The two exponents n and l indicate the number of design variables and the total number of degrees of freedom of the system, respectively. K is the
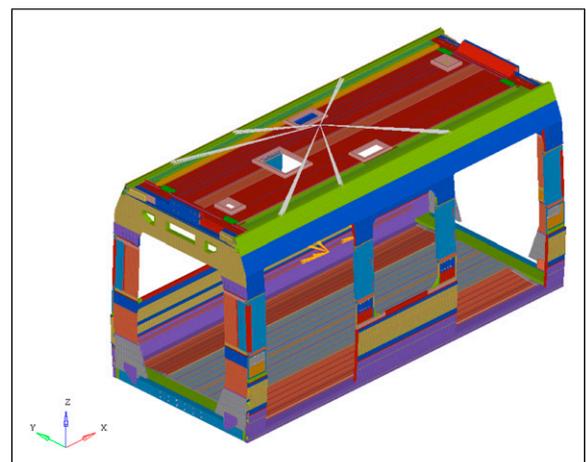

Figure 3. Finite element model of RA carbody.

stiffness matrix and f is the vector of external loads. f represents the objective function of the problem, g_j is the constraint function (number of constraints equal to m) and finally $K(b)z = f$ is the governing equation of linear

Table 3. (a) Design variables changes. (b) Frequencies comparison for the first ten vibration modes of RA carbody.

Component	No.	Original thicknesses		Optimized thicknesses	Thickness range of variation (max - min)	% of thickness reduction
[−]	[−]	[mm]		[mm]	[mm]	[−]
Central extruded profile	1	Skin	3	2.4	3 - 1	20.00
	2	Rib	3	1.3	3 - 1	56.66
Medium extruded profile	3	Skin	3	2.7	3 - 1	10.00
	4	Rib	3	2.5	3 - 1	16.66
Medium-lateral extruded profile	5	Skin	3	2	3 - 1	33.33
	6	Rib	3	1.6	3 - 1	46.66
Lateral extruded profile	7	Skin	4	3	4 - 1	25.00
	8	Rib	2.8	2	2.8 - 1	28.57
Medium extruded profile (rail)	9		4	1.8	4 - 1	55.00

Num mode	Original model	Optimized model	Frequency variation
[−]	[Hz]	[Hz]	[%]
Mode 1	10.54	10.01	−5.03
Mode 2	14.23	14.00	−1.62
Mode 3	14.78	14.54	−1.62
Mode 4	16.30	15.54	−4.66
Mode 5	17.34	16.52	−4.73
Mode 6	18.55	17.67	−4.74
Mode 7	18.58	17.80	−4.20
Mode 8	20.37	20.10	−1.33
Mode 9	22.70	20.60	−9.25
Mode 10	24.71	22.34	−9.59

finite element analysis. At last line, b_L and b_U are the lower bound and the upper bound of the vector b, which define the range of variation of the design variables values.

Design variables have been selected to work on the roof assembly, as explained below. Table 3(a) summarizes their characteristics, before and after the process. The optimized extruded profiles have shown a general reduction in thickness. Figure 4(a) and (b) illustrate the selected components of the roof assembly, subjected to the optimization process, including a detailed cross-section view. The tram platform was recently designed and optimal performance of aluminium extrusion technology could be considered. Minimum values of thickness have been imposed all equal to 1 mm. This choice may allow designer to understand how much the material could still be exploited. Recent technical improvements for extrusion of aluminium extruded profiles make it possible to obtain thinner and thinner thicknesses, near the imposed minimum value. Summarizing the main conditions of the problem: the objective function was represented by the minimum value of the carbody mass, design variables were the thicknesses of the selected components, design constraint function required the first natural frequency of the carbody to be greater than 10 Hz. This value represents the minimum, acceptable frequency of vibration to respect the dynamic behaviour of the present carbody.

Figure 5 shows the comparison between the first vibration mode of RA carbody, before and after the optimization process. In both cases a bending mode was observed. It involved the roof of the vehicle and in particular the optimized components. Focusing on the mass variation, they turned out to be 21% lighter than the original ones, thanks to the reduction of thicknesses. The modal behaviour of the carbody did not change in terms of mode shape, ensuring also the constraint condition required by the problem. In fact, The maximum variation obtained in terms of frequency was about 10% for the ninth and tenth modes, which are less relevant for the dynamic of the vehicle. All other cases showed a variation of 5% or less. Table 3(b) illustrates the complete results obtained by the modal analysis of the original and the optimized configuration of the carbody, considering the first ten vibration modes of the system. Mode shapes showed a very good matching.

Results and discussion

Once updated the original finite element model of the vehicle with the new thickness values, it has been tested again in accordance with EN 12663-1:2015 standard. The mass has decreased: 21% of mass saving has been obtained for the optimized components, that corresponded to a 3% respect to the whole mass of the carbody metallic structure. Mechanical behaviour of RA carbody and that of the vehicle, did not change. From the point of view of its dynamic behaviour, the mode shape of the first mode of vibration has not changed and the minimum frequency of 10 Hz has been ensured. This means that the optimization procedure has achieved the
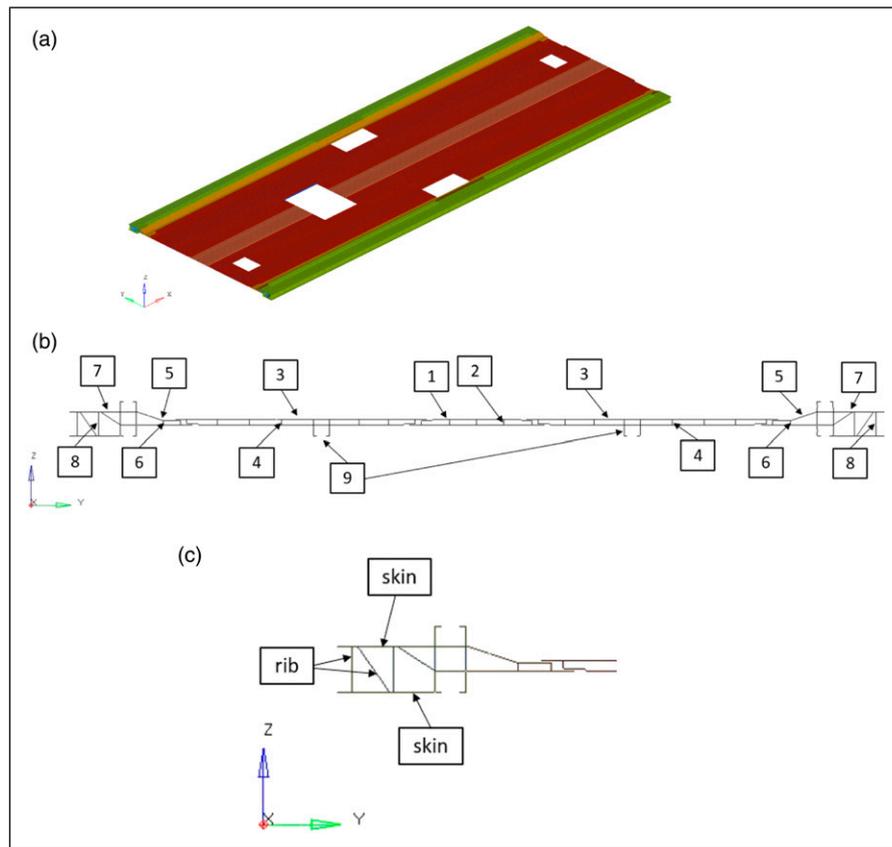
Figure 4. Optimized components of the roof assembly: (a) Isometric view, (b) Cross-section view with references, (c) Detailed view for skin and rib.
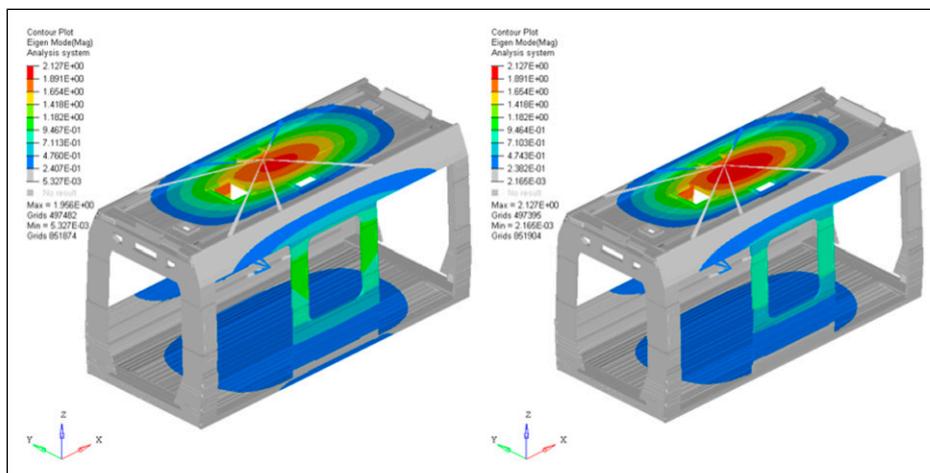

Figure 5. First vibration mode of the original RA carbody (on the left) and the optimized one (on the right).

goal. All the load cases have been applied on the complete vehicle and never on the single carbody, as indicated by the reference standard. The results are referred to the optimization process applied only on the roof assembly. Nowadays, due to the equipment mounted on it, it must be considered as an important structural component. For this reason, the authors in agreement with their industrial partners, decided to start the development of the present optimization procedure from that assembly, considered as a system with extensive room of improvement in terms of mass saving. Table 4 illustrates the FE analysis results of RA carbody,

in terms of max stress value, for each loading condition. Main concentrations of stress have maintained their location, with little changes in intensity. This outcome underlines that the optimization process has not generated any new ones, despite mass reduction. Figure 6(a)–(e) show only the results of "Max vertical load" condition (see Table 2(c)), which has generated the worst response in terms of stress and inflection of the roof assembly. Figure 6(a) illustrates the global displacement distribution of the vehicle. The global behaviour of the tram platform is practically not influenced by the optimization procedure. This aspect is confirmed

by Figure 6(b), that shows the global displacement distribution of the single RA carbody. The value of inflection in the mean longitudinal position of the carbody is the same, but a wider contribution of the roof can be observed. This effect does not represent a problem since the mechanical behaviour of the carbody, in terms of displacement, results unchanged. More in detail, Figure 6(c) highlights that the maximum inflection of the optimized components, which represent the roof panel of the structure effectively, has not increased. This result is promising because it is one of the most representative problems for roofs of railway

vehicles carbodies. Figure 6(d) shows the stress on the complete roof assembly, while Figure 6(e) on the roof panel. Both cases illustrate a small increase of the maximum value of stress. With reference to the upper panel, can be observed a little wider distribution. Consequently, also the utilization coefficient of the material, which is referred to the max stress value, did not have remarkable variations. To conclude, the comparison between stress of the original roof components and the optimized ones, has shown acceptable values. The utilization coefficient of the material must be less than one for a positive verification according to

Table 4. FE analysis results of RA carbody.

| Phisycal measures	Loading conditions	Model configurations	Max values [N/mm^2]	Allowable values [N/mm^2]
Von Mises stress	Max vertical load	Baseline	200	215
		Optimized	202	
	Compressive load on the buffers in C0 condition	Baseline	101	215
		Optimized	108	
	Compressive load on the buffers in C4 condition	Baseline	172	215
		Optimized	182	
	Tensile load on the drawbar in C0 condition	Baseline	74	215
		Optimized	73	
	Compressive load on the drawbar in C0 condition	Baseline	82	215
		Optimized	83	
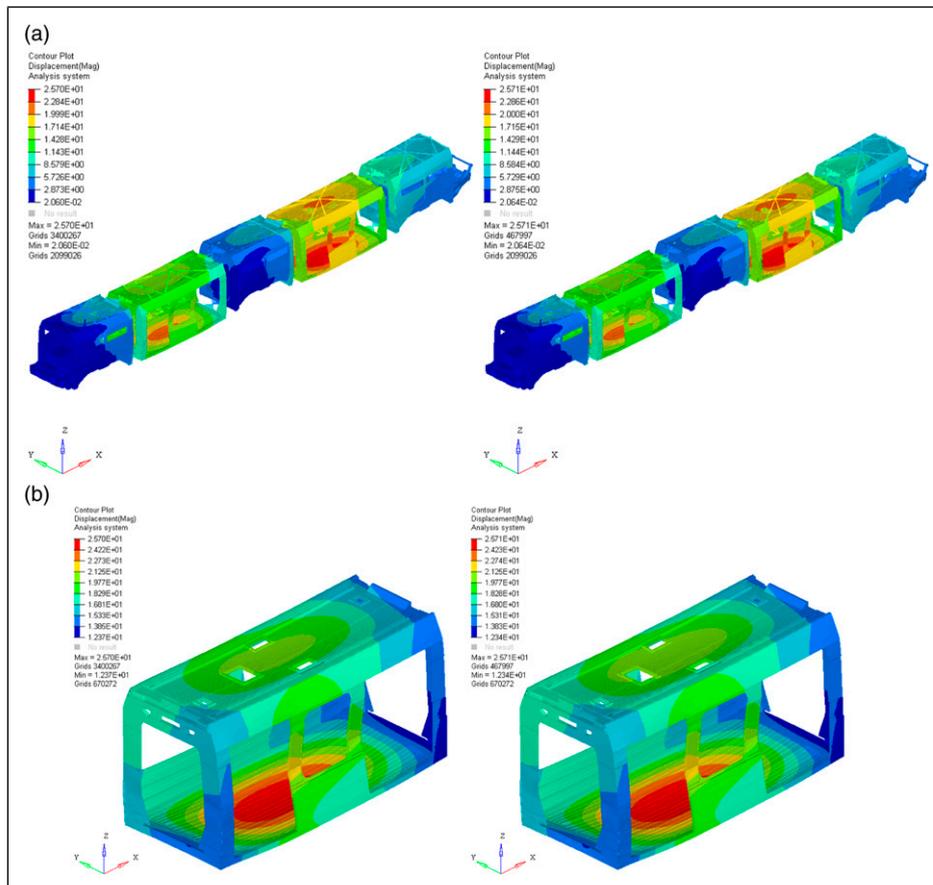
Figure 6. Baseline model on the left, optimized model on the right. (a) Global displacement of the vehicle, (b) Global displacement of RA carbody, (c) Displacement in Z direction (inflection) of the RA carbody roof, (d) Von Mises stress distribution of the RA carbody roof assembly, (e)Von Mises stress distribution of the upper panel of RA carbody roof.
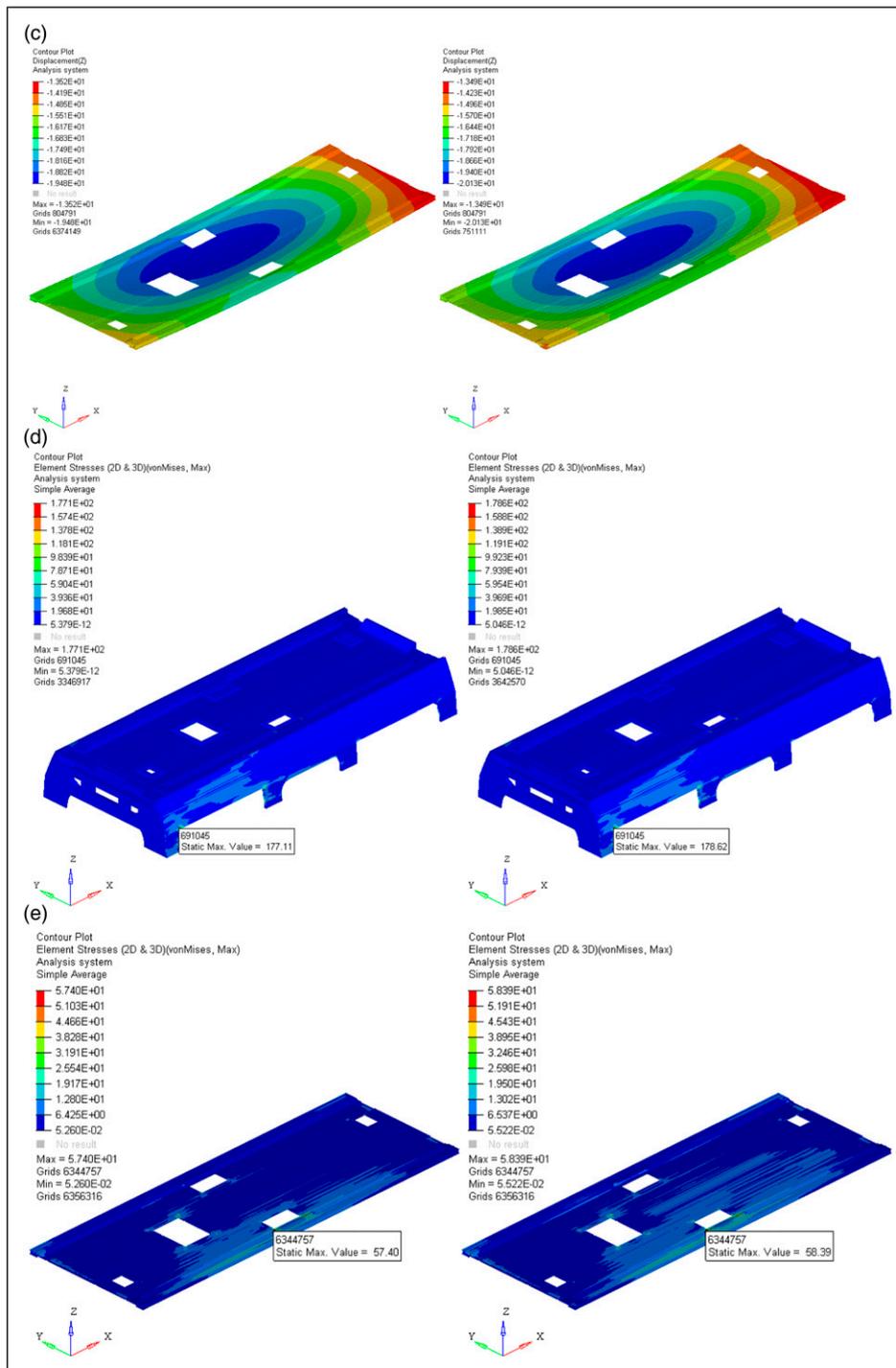
Figure 6. Continued.

EN 12663-1:2015 standard. This condition has been always respected.

Conclusions and future developments

The present activity aimed to develop a structural optimization procedure to support the lightweight design process for railway carbodies and then, environmental sustainability in transportation field. The reference standard EN 12663-1:2015 does not include any structural optimization process, that could help designers to reduce the frequent overdimensioning condition afflicting carbodies shell. The

dynamic size optimization process has the objective to reduce the mass of the carbody shell, ensuring possibly its original mechanical performance and dynamic behaviour. This last condition can be obtained introducing a dynamic design constraint on the first frequency of vibration of the carbody, preserving indirectly the stiffness of the system. The proposed strategy takes advantage of the single carbody model which must be generally created to test its modal behaviour, without making the ordinary procedure more complex. It acts on the thickness of the extruded profiles, defining the minimum value within a predetermined range, that meets the problem conditions. The method turned out to be numerically efficient. It has

required less than 0.5 h for completing the optimization process (that includes also modal analysis), considering a model with 671759 shell elements at first order. The computer used for the calculation process had the following characteristics: Intel(R) Xeon(R) CPU E5-2643 v4 @ 3.40 GHz, RAM 32 GB. In terms of mass reduction, the optimized components were resulted 21% lighter than the original ones, that corresponds to 3% if evaluated on the total mass of the single carbody aluminium structure. In addition, they respected all the conditions in terms of stress and inflection, according to the reference standard. These promising results could be improved, extending the application of the optimization approach to more areas of the single carbody. Despite that, currently, the proposed procedure represents an effective tool for reducing the overdimensioning condition of the roof assembly. Even if these preliminary results are encouraging, much work remains to be done and many future developments are planned. First of all, the possibility to apply the optimization procedure to the whole carbody: this important step could require introducing additional parameters and conditions, due to the different behaviour that can be shown by different assemblies. In addition, new lightweight materials and topology optimization, will be explored with the aim to innovate the roof assembly, reducing mass structure of the single carbody and then that of the vehicle.

Declaration of conflicting interests

The author(s) declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

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