Topography optimization-driven design of added rib architecture system for enhanced free vibration response of thin-wall plastic components used in the automotive industry

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Abstract
A relentless paradigm switch in the automotive industry is being witnessed, where traditional combustion engines are progressively replacing with electric counterparts. In order to compensate for heavy electric engine technology, plastic components used in the interior parts of electric vehicles seems to be a reasonable strategy aligned with the lightweight trend as the automotive industry’s top priority. Extremely thin plastic components (ultra-lightweight) attached to ribs-based architectures have been identified as an adequate solution offering a good balance between lightweight and structural/vibrational response of the overall composition. Still, they may yield detrimental features regarding thermal-induced deformations (i.e., warpage and shrinkage) associated with the typical mold injection process for their manufacturing. This paper uses topology optimization to determine the precise location of the added rib architecture system for enhanced vibration response of the overall plastic component (thin original plastic part and ribs architecture). Following a constant mass criterion, the topology optimization-driven design of the additional material is replaced with a more convenient ribs-based architecture component, showing a reasonable similarity from a vibration standpoint. The component in its initial state (without some ribs-based architecture) shows a vibration response of 16.69 Hz. Once the topology optimization is applied, a significant improvement is observed since a value of 26.05 Hz is obtained. At a postprocessing stage, the design is analyzed and subjected to typical static loading cases to verify its stiffness properties, getting a significant displacement reduction from 7.68 to 3.51 mm. Finally, the design implementation of a warpage and shrinkage analysis confirmed that the final design was not adversely affected according to standard considerations.

Keywords Design · Topology optimization · Modal analysis · Warpage · Shrinkage · Manufacturing

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1 Introduction

The industry in general, especially the automotive industry, is at the forefront in searching for new ways to reduce the environmental impact of production processes. Automotive companies are targeting ECO-Innovations, opening a wider field beyond engine and powertrain technology. [1]. On the other hand, the automotive industry is constantly faced with a constant attempt to balance excellent performance, economy, comfort, and reduction of polluting gases. These conflicting requirements are the fundamental impetus for the challenging advances that today’s engineering needs to address [2].

Lightweighting has been identified as a top priority within the automotive industry. Recent reports by consultants in the automotive sector reveal that by 2030, automotive manufacturers will have to increase the percentage of lightweight components in vehicles from 30 to 70% in order to compensate for heavy electric powertrains and engine technology. In that sense, using plastic components in the interior parts of electric vehicles seems to be a good strategy in line with the identified lightweight trend. Lightweighting is beneficial in electric vehicle (EVs) design as it allows a higher range for the same cost or a smaller battery and motor with constant capacity and performance [3]. In addition, automakers devote significant resources to developing electric, autonomous, and hybrid vehicles. Therefore, automotive lightweight remains a fundamental strategy from the early design stages [4]. Obviously, a possible strategy aligned with the recent lightweight trend for increased autonomy of electric vehicles would be to decrease the thickness and therefore decrease the mass of these plastic components. However, this might yield undesired collateral effects. Specifically, the vibration response of these components can be adversely affected, which can lead to highly undesired consequences such as resonance or amplified forced vibrational responses. In addition, the stiffness properties of these plastic parts can also be compromised/deteriorated if the thickness is decreased.

In the automotive industry, plastic materials are widely used due to their wide versatility in appearance and manufacturing in the face of different and complicated geometries. However, the processes become complex, expensive, and challenging because of complex geometries and premium parts appearances. Therefore, the most significant advantage of injection molding in interior components for electric vehicle OEMs is the simplest: weight [5]. As the most critical and widespread procedure of polymer processing for finished plastic parts [6, 7], injection molding can address several challenging competitive demands: Plastic parts are often lightweight and offer excellent design freedom for the designer of the product at a low cost [1].

In order to compensate for the detrimental features that extreme thickness reduction of these components entails from both vibrational and stiffness standpoints, it is customary to attach a ribs-based architecture to the original (thin) plastic component (see Fig. 1 for an illustration of this strategy). The addition of this ribs-based architecture clearly helps to improve the deteriorated vibrational and stiffness response of the original thin-wall plastic component. Unfortunately, the introduction of such architecture may introduce some undesired drawbacks. Specifically, the indiscriminate use of ribs spreading all over one side of the original thin plastic component entails thickness variations during the injection molding process that characterizes the manufacturing of these components. This, in turn, reflects in increased both warpage and shrinkage of the final manufactured product, namely, thermal-induced deformations occurring due to the temperature variation during the injection process. These two phenomena contribute negatively to the vehicle’s whole assembly process and the perceived appearance quality.

The aforementioned benefits and downsides of introducing ribs-based architectures seem to indicate that this material aggregation needs to be done according to some criterion, never in an indiscriminate manner. It would be tempting to let an experienced designer fix such a criterion. However, the level of sophistication of the problem at hand prevents this as a reliable strategy. Specifically, the vibration response requires solving a generalized eigenvalue problem of the design component, from which the eigenfrequencies can be inferred. In addition, the stiffness response involves the solution of a linear elastic problem, typically through Finite Element analysis. Finally, the warpage and shrinkage in the final product can also be inferred from a Finite Element analysis that models the complex injection process and then considers the thermal fluctuations to determine both warpage and shrinkage. Therefore, an efficient strategy aiming at establishing a design criterion for the optimal
position of the ribs-based architecture needs to be driven by efficient mathematical algorithms. In that sense, topology optimization approach seems to be a wise choice that can help designers.

Since the pioneering work of Bendsoe and KiKuchi [8], the field of structural topology optimization has been extraordinarily prolific. These methods can be broadly classified into [9]: density-based methods, with the Solid Isotropic Material with Penalization (SIMP) as their maximum representative [10], level-set methods [11, 12], phase-field methods [13], topological derivative methods [14] and evolutionary methods [15]. The prolific nature of this field does not restrict to the wide range of methods available and just mentioned, but to the tremendous range of applications where they have been successfully implemented. Thus, there exist countless works in the field of structural topology optimization oriented toward the maximization of structural stiffness [10, 12], or to a reduction of the stress field [13], or even with the aim of improving the natural vibration frequencies of a given structural system [16]. Furthermore, in other works where topology optimization methods have been efficiently applied with the aim to design auxetic materials [17] and other types of metamaterials, namely, materials with unusual macroscopic properties, where physical intuition is generally not sufficient in order to conceive their non-conventional final designs. Additionally, in the last years, there has been a considerable progress in the design of multifunctional materials [18, 19], materials which can be actuated through different physical principles, including thermal, electrical or magnetic stimuli, to name but a few, and that can be used as actuators and energy harvesters.

In this paper, topology optimization methods are considered to enhance natural frequencies (i.e., free vibration) of a thin-wall plastic component used in the automotive industry. Topology optimization in context-free vibration (and even forced vibration) is a very consolidated field of research. There exist many works where the mathematical bases associated with this problem are well established. The work in [20] presented a method to optimize the layout of supports in order to maximize the structural natural frequency by means of the SIMP method. Furthermore, Jensen and Pedersen [21] focused on maximizing the separation of two adjacent eigenfrequencies in structural with two material components. Luo et al. [22] presented a multi-objective topology optimization method for designing aerodynamic structures by means of the SIMP method, where compliance and eigenvalues were integrated as part of the objective function. Maximization of natural frequencies (free vibration) or improvement of the vibrational response in the case of forced vibration has also been applied for the topology optimization-driven design of tall buildings subjected to seismic excitation [16].

Topology optimization in the context of manufacturing is becoming more predominant. There is a countless number of papers on the subject. For instance, Yasin et al. [23] advocates for SIMP-based topology optimization for the design of a ribs system attached to a plastic cover protector with the aim of reducing warpage associated with its mold injection manufacturing process. Gao et al. [24] address the topology optimization-driven design of rib stiffeners for increased bending stiffeners of thin-wall components used in the aeronautics and aerospace sectors. Furthermore, the authors in [25] devised a new smart topology optimization algorithm for support-free designs, showing promising results from the manufacturing standpoint.

In this context, topology optimization (TO) is the technology that focuses on developing optimized structures, considering expected loads, available design spaces, materials, and cost. If TO is implemented in the early stages of component design, designs with minimum mass and maximum stiffness will be obtained. Currently, the largest field of application of TO is in the aircraft industry because an optimized component’s manufacturing process is complex, expensive, and low volumes manufactured. The optimized part at present is usually done through additive manufacturing; this process currently has the limitation of being costly and only handling low volumes, the aircraft industry being one of the few that can afford this type of technology and adapts to the production volumes. On the other hand, the automotive sector requires low-cost, high-volume manufacturing processes such as injection molding.

Studying TO and identifying the benefits of this type of technology, i.e., mass reduction and stiffness increase, is where the conventional injection mold manufacturing process is maintained. Although currently, no research work applies TO in thin-wall plastic components for the automotive industry, this research work begins to determine how to apply TO in thin-wall plastic components used in the automotive industry, which is precisely what all OEMs currently have as their objective: reduction of step to increase the autonomy of the EVs, less consumption of polymers for the circular economy purposes, achieve the desired performance in plastic components and not increase costs for the manufacturing processes thus EVs become in an option for the clients.

The layout of the paper is as follows: Sect. 2 describes the basics regarding free vibration analysis and density-based topology optimization. Section 3.1 introduces the strategy devised in order to increase the eigenfrequencies or natural frequencies of the thin-wall plastic component used in the automotive industry and considered in this paper, where the topology optimization algorithm dictates the distribution of an additional material which needs to be added on top the initial plastic component. The results in terms of new eigenfrequencies of the new component are shown in this section, corroborating a significant improvement in this metric. Moreover, the replacement of the added material...
2.1 Free vibration analysis

Let $\mathcal{B}^h \in \mathbb{R}^3$ be the Finite Element discretization of a three-dimensional continuum $\mathcal{B}$ (see Fig. 2). The latter can potentially represent a thin-wall plastic component within an automobile, such as the one depicted in Fig. 2. An important aspect in many engineering sectors, including the automotive industry, has to do with improving the dynamic response of the final products that they fabricate. This can be effectively achieved by trying to increase the natural frequencies (or eigenfrequencies) of their product of interest.

Vibrational analysis of engineering systems is typically carried out with the help of a Finite Element software, where $\mathcal{B}^h$ is split into a finite number of elements (see Fig. 2). For the particular case of linear tetrahedral discretizations, each of these individual elements is comprised of four nodes (see also Fig. 2). Let $N_{\text{nodes}}$ represent the total number of nodes of the associated discretization. In that case, the body $\mathcal{B}^h$ is characterized by $N_c = 3 \times N_{\text{nodes}}$ possible number of natural frequencies $\nu_j$, namely $j = \{1, 2, \ldots, N_c\}$. Each natural frequency $\nu_j$ is associated with a vibration mode (or eigenmode) $\phi_j \in \mathbb{R}^{N_c}$. From a mathematical standpoint, the problem of finding the natural frequencies of a discrete system $\mathcal{B}^h$ as that in Fig. 2 can be stated as

$$\begin{align*}
\left[K - \left(2\pi \nu_j\right)^2 M\right]\phi_j &= 0; \quad j = \{1, 2, \ldots, N_c\}; \\
\phi_i \cdot M\phi_j &= \delta_{ij},
\end{align*}$$

(1)

with $\nu_1 \leq \nu_2 \leq \cdots \leq \nu_{N_c}$ and where $\delta_{ij}$ represents the $ij^\text{th}$ component of the Kronecker delta tensor. Equation (1) represents a generalized eigenvalue problem where $K$ and $M$ are the stiffness and mass matrices of the discrete system $\mathcal{B}^h$, respectively. Furthermore, each eigenfrequency $\nu_j$, (more specifically the scalar $(2\pi \nu_j)^2$) and its associated vibration mode $\phi_j$, represent an eigenvalue and eigenvector of the generalized eigenvalue problem in (1). Finally, the second equation in (1) represents the standard orthonormality condition of the eigenvectors.

Clearly, the solution of the eigenvalue problem in Eq. (1) for all its possible $N_c$ eigenvalues and eigenvectors can entail a cumbersome task, extremely demanding from the

1. The stems from the three-dimensional nature of the solid $\mathcal{B}$.
2. The eigenvalue problem in (1) can be formulated in terms of the eigenfrequencies $\nu_j$ (with units of Hz) or in terms of the angular eigenfrequencies $\omega_j$ (with units of radians per second), both being related through the following relationship, $\omega_j = 2\pi \nu_j$. 

Fig. 2 a On the left-hand side is depicted the thin-wall plastic component ($\mathcal{B}$) used in the automotive industry; b on the right-hand side is presented the Finite Element discretization of the plastic component ($\mathcal{B}^h$)
computational standpoint, normally when considering very fine discretizations, leading to large values of \( N_v \). Instead, it is common practice to focus on a small set of eigenvalues and eigenvectors \( \tilde{N}_v \), with \( \tilde{N}_v < < N_v \). In that case, the generalized eigenvalue problem in (1) is replaced with its computationally affordable counterpart as

\[
\begin{cases}
(K - (2\pi \nu)^2 M) \phi_j = 0; & j = \{1, 2, \ldots, \tilde{N}_v\}; \\
\phi_i \cdot M \phi_j = \delta_{ij},
\end{cases}
\]

with \( v_1 \leq v_2 \leq \cdots \leq v_{\tilde{N}_v} \), and where the only difference between (2) and (1) resides in the substitution of \( N_v \) with \( \tilde{N}_v \). With regards to the number of eigenfrequencies and eigenmodes that must be considered, the answer is not immediate. Instead, this is normally dictated by the expertise of the structural engineer responsible for the vibration analysis of the continuum \( B^h \).

### 2.2 Density-based topology optimization in the context of free vibration

In density-based topology optimization, a pseudo-density field \( \theta : B^h \rightarrow [0, 1] \) is introduced, where \( \theta = 0 \) represents void regions (no material) and \( \theta = 1 \), solid regions, namely, regions where the presence of material is favored. In general, an iterative gradient algorithm such as the Method of Moving Asymptotes [26], the optimality criterion [10] or similar is used in order to determine in what regions of \( B^h \) the presence of solid (\( \theta = 1 \)) is to be fostered, always driven by the objective of maximizing or minimizing a relevant objective function. In the context of free vibration, \( J(\theta) \) can be defined as a weighted sum of the \( \tilde{N}_v \) eigenfrequencies, which are intended to be maximized, namely

\[
J(\theta) = \sum_{j=1}^{\tilde{N}_v} f_j v_j(\theta),
\]

where \( f_j \) represent appropriate weights, which can modulate the relative importance of each eigenfrequency \( v_j \) within the objective function. The objective function \( J(\theta) \) in (3) seeks the maximization of the first \( \tilde{N}_v \) eigenfrequencies of the engineering system \( B^h \). However, it is customary to formulate optimization problems as a minimization problem rather than as maximization problem. A possible way to transform \( J(\theta) \) into a suitable objective function whose minimization yields the maximization of \( v_j, j = \{1, \cdots, \tilde{N}_v\} \) is by redefining \( J(\theta) \) as follows

\[
J(\theta) = \sum_{j=1}^{\tilde{N}_v} \frac{f_j}{v_j(\theta)}.
\]

In this work, the weights \( f_j \) have been taken as \( f_j = 1 \), \( \forall j = \{1, \cdots, \tilde{N}_v\} \). In addition, the topology optimization problem considers a volume constraint, which restricts the amount of solid material (i.e., \( \rho = 1 \)) that can be used within \( B^h \).

With the aim of reducing the presence of intermediate densities, a material interpolation scheme is required in order to obtain a distinct solid-void topology, characterized with either \( \theta = 1 \) or \( \theta = 0 \), respectively. In order to achieve this, the relevant material parameters featured in the definition of the stiffness matrix \( K \) and the mass matrix \( M \) in the generalized eigenvalue problem in (2) need to be related to the pseudo-density field \( \theta \). In particular, the elasticity tensor within every element \( e \) of \( B^h \), denoted as \( E_e \), is the relevant material property featuring within the stiffness matrix \( K \). Furthermore, the true density (not to be confused with the pseudo-density field \( \theta \) used for optimization purposes) within every element \( e \) of \( B^h \), denoted as \( \rho_e \), is the material property featuring within the mass matrix \( M \). In this work, the SIMP material interpolation scheme is adopted, which yields the following material interpolation schemes for both \( E_e \) and \( \rho_e \), namely

\[
\begin{align*}
E_e & = \left( \epsilon (1 - \theta_e^p) + \theta_e^p \right) E_0; \\
\rho_e & = \left( \epsilon (1 - \theta_e^p) + \theta_e^p \right) \rho_0,
\end{align*}
\]

where \( \theta_e \) represents the value of the pseudo-density field at a given element \( e \) of \( B^h \). Furthermore, \( p \) and \( q \) are penalizing exponents which in this paper take the values \( p = 3 \) and \( q = 1 \). Crucially, \( E_0 \) and \( \rho_0 \) represent the elasticity tensor and the true density of the solid (intact) material. It is customary to permit a residual value for both \( E_e \) and \( \rho_e \) when the element density vanishes, i.e., \( \theta_e = 0 \). For that, a small threshold \( \epsilon \) is normally adopted of value \( \epsilon \approx 10^{-6} \).

With these ingredients in mind, the optimization problem of interest can be finally stated in a compact manner as

\[
\min_{\theta} J(\theta); \quad \begin{cases}
K(\theta) - (2\pi \nu(\theta))^2 M(\theta) = 0; \\
\phi_i(\theta) \cdot M(\theta) \phi_j(\theta) = \delta_{ij}; \\
\int_B \theta \, dV - aV \leq 0; \\
0 \leq \theta_e \leq 1; \forall e \in B
\end{cases}
\]

**Remark 1** In order to prevent results with checkerboard patterns in the pseudo-density field \( \theta \), it is customary to make use of filtering techniques [27, 28]. For instance, the Helmholtz’ or Partial Differential Equations (PDE) filter, yields the filtered pseudo-density field \( \tilde{\theta} \) from the solution of the following boundary value problem...
\[ \hat{\theta} - R^2 \nabla^2 \hat{\theta} = \theta; \quad \text{on } E^b; \]
\[ \nabla \hat{\theta} \cdot n = 0; \quad \text{on } \partial E^b, \quad (7) \]

where \( R \) represents the filter radius and \( \nabla^2 (\bullet) \), the Laplacian operator. In addition, \( \partial E^b \) represents the boundary of \( E^b \).

With the aim of further reducing the appearance of intermediate densities, it is customary to make use of standard projection techniques such as the smoothed Heaviside function proposed in [29], namely
\[ \hat{\theta} = \frac{\tanh(\beta \eta) + \tanh(\beta(\hat{\theta} - \eta))}{\tanh(\beta \eta) + \tanh(\beta(1 - \eta))}, \quad (8) \]

where \( \hat{\theta} \) is known as the physical pseudo-density field, \( \beta \) and \( \eta \) are parameters carefully selected and updated throughout the optimization [29].

### 2.3 Topology optimization algorithm

Here, a gradient-based algorithm is used to carry out the topology optimization process. For that, it uses a gradient-based algorithm implemented in the software Altair OptiStruct\textsuperscript{TM} [30]. For the problem at hand, the gradient-based topology optimization algorithm advocated for needs the evaluation of the derivatives of the eigenfrequencies \( \nu_j \) with respect to each pseudo-density field, namely \( \frac{\partial \nu_j}{\partial \theta_e} \), with \( e = \{1, 2, \ldots, N_{\text{elem}}\} \), being \( N_{\text{elem}} \) the number of Finite Elements of the underlying discretization \( E^b \). This can be found in [21], where \( \frac{\partial \nu_j}{\partial \theta_e} \), with \( e = \{1, 2, \ldots, N_{\text{elem}}\} \) adopts the following expression\(^3\)
\[ \frac{\partial E_e}{\partial \theta_e} = (1 - \epsilon) \rho_{e}^{\epsilon - 1} E_{\theta_e}; \quad \frac{\partial \rho_{e}}{\partial \theta_e} = (1 - \epsilon) \rho_{e}^{\epsilon - 1} \rho_0. \quad (9) \]

\[ \frac{\partial \nu_j}{\partial \theta_e} = \frac{1}{2 \nu_j} \phi_j \cdot \left( \frac{\partial K}{\partial \theta_e} - (2 \pi \nu_j)^2 \frac{\partial M}{\partial \theta_e} \right) \phi_j. \quad (10) \]

This eventually permits computing the sensitivity of the objective function \( \mathcal{J}(\theta) \) in (4) with respect to the pseudo-density field \( \theta \) by simple differentiation, yielding
\[ \frac{\partial \mathcal{J}(\theta)}{\partial \theta_e} = \sum_{j=1}^{N_{\text{elem}}} \frac{\partial J}{\partial \nu_j} \frac{\partial \nu_j}{\partial \theta_e} \]
\[ = - \sum_{j=1}^{N_{\text{elem}}} \frac{1}{8 \pi^2 \nu_j^2} \phi_j \cdot \left( \frac{\partial K}{\partial \theta_e} - (2 \pi \nu_j)^2 \frac{\partial M}{\partial \theta_e} \right) \phi_j. \quad (11) \]

Based on the vector of sensitivities \( \frac{\partial \mathcal{J}(\theta)}{\partial \theta_e} = \left[ \frac{\partial \mathcal{J}(\theta)}{\partial \theta_1} \ldots \frac{\partial \mathcal{J}(\theta)}{\partial \theta_e} \right]^T \), the algorithm determines the new value of each elemental pseudo-density field \( \theta_e \) in the topology iteration \( k + 1 \) as a function of the elemental pseudo-density field \( \theta_e \) at iteration \( k \) according to
\[ \theta_{e|k+1} = \theta_{e|k} + t \frac{\partial \mathcal{J}(\theta)}{\partial \theta_e}. \quad (12) \]

where the step length \( t \) is chosen internally by the algorithm with a two-fold aim: to guarantee a decrease in the objective function \( \mathcal{J}(\theta) \) while permitting an affordable computational cost, as a small value of \( t \) entails an excessive number of topology optimization iterations. The main ingredients necessary for the computational implementation of the topology optimization algorithm described can be seen in Algorithm 1, presented in a pseudo-code format.

**Algorithm 1** Flowchart of the gradient-based optimization algorithm used

- **Set pseudo-density variables** \( \theta \) complying with volume constraint \( \theta_e = \alpha \).

- **Get filtered pseudo-density** \( \hat{\theta} \) from (7).

- **Get physical pseudo-density** \( \hat{\theta} \) from (8).

**While** user defined criterion

- **Solve** Generalized Eigenvalue Problem in (2).
  - Get eigenfrequencies \( \nu_j \) and eigenmodes \( \phi_j \).
- **Compute** sensitivity \( \frac{\partial \mathcal{J}(\theta)}{\partial \theta_e} \) in (11)
- **Determine** step length \( t \) in (12) and update \( \theta_e|k+1 \) compliant with the volume constraint.
- **Evaluate objective function** \( \mathcal{J} \) in (4) and its relative change with respect to previous iterations to determine termination.
- **Get filtered pseudo-density** \( \hat{\theta} \) from (7).
- **Get physical pseudo-density** \( \hat{\theta} \) from (8).

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\(^3\) Since the pseudo-density dependent fields \( E_e \) and \( \rho_e \) featuring in both \( K(\theta) \) and \( M(\theta) \), depend on the elemental pseudo-density field \( \theta_e \) according to (5), the terms \( \frac{\partial K}{\partial \theta_e} \) and \( \frac{\partial M}{\partial \theta_e} \), featuring in (10) require the computation of \( \frac{\partial E_e}{\partial \theta_e} \) and \( \frac{\partial \rho_e}{\partial \theta_e} \), namely
3 Results

3.1 Topological optimization applied to an automotive plastic component

Although topology optimization approach has been described throughout Sect. 2, structural integrity and safety considerations prevent to implement of dramatic topological modifications in the already extremely thin geometry of the plastic component $B$ (see Fig. 3a), and hence, question the application of topology optimization directly upon $B$. Clearly, the application of topology optimization (imposing a given volume fraction constraint) over $B$ would lead to an intricated truss-like design (see Fig. 3), which is undesired from the safety and structural integrity standpoints. On the contrary, shape optimization does seem like a reasonable approach in this particular case, where the thickness of the original engineering part (Fig. 3a) and the collocation of control points parameterizing its geometry could be conveniently varied with the aim of improving the objective function of interest.

However, both strategies, as mentioned earlier, are not pursued in this paper (neither topology optimization directly upon $B$ or shape optimization of $B$). Instead, a more convenient engineering solution is sought. This is illustrated in Fig. 4. In this figure, it is possible to observe, from two different visual angles, that on the side which is not visible to the final user, an additional volume of a constant thickness (4 mm) has been added (gray volume). This entails that the new volume $B$ is comprised of two parts, the original and non-optimizable blue volume in Fig. 4, denoted as $B_{NO}$, and the additional gray volume in Fig. 4, where topology optimization can be applied, denoted as $B_O$, with $B = B_{NO} \cup B_O$. From the discretization standpoint, the structural coupling between both volumes, which share the area where the intersection between $B_O$ and $B_{NO}$ occurs, namely in $B_{NO} \cap B_O$, has been carried out through the consideration of a contact type element which ties the displacements of the nodes of $B_{NO}$ with those of $B_O$ over the surface $B_{NO} \cap B_O$. This strategy entails a certain degree of flexibility as it permits the consideration of non-matching Finite Element meshes for both $B_O$ and $B_{NO}$ on $B_{NO} \cap B_O$.

Once both non-optimizable and optimizable volumes $B_O$ and $B_{NO}$ are discretized and properly tied (connected) as previously discussed, a volume fraction constraint over the optimizable volume $B_O$ needs to be prescribed. In this work, a volume fraction $\alpha = 0.2$ has been considered, which restricts the total amount of the optimizable volume $B_O$ which can be used to a 20% of its original volume.

Finally, from the Finite Element standpoint, the superposition of both volumes yields two Finite Element meshes (tied through their surface intersection) with the information shown in Table 1.

With regards to the eigenvalue problem in the optimization problem (6), for the case $\theta = 1$ (solid plastic component), the material parameters featuring in the stiffness matrix $K(\rho)$ and the mass matrix $M(\rho)$ are the Young’s modulus $E$, the Poisson’s ratio $\nu$, and the true density of the material $\rho$, which can be found in Table 2.

3.2 Results from topology optimization analysis

Before topology optimization is applied on the optimizable region $B_O$, the first six eigenfrequencies of the original part $B_{NO}$ without considering any addition of material on top (namely now $B_O$) are computed and shown in Table 3.

The strategy advocated for is that described in Sect. 3.1, where topology optimization is carried exclusively on the additional volume $B_O$, placed on top of the non-optimizable region $B_{NO}$ on the side, which is not visible to the final user. Fixing a target volume constraint of 20%, the gradient-based algorithm (see Algorithm 1) yields an optimal solution characterized by the material distribution shown in Fig. 5. From Table 4, the six first eigenfrequencies corresponding with the component in Fig. 5 (non-optimizable region $B_{NO}$ and the

Fig. 3 a On the left-hand side, the application of TO onto the plastic part is presented, and b on the right-hand side, a design exhibiting undesirable geometrical features such as holes and thin bars is shown.
The final design of $B_{O}$ are shown. Crucially, this results entail that the value of the objective function in Eq. (4), decreases from 0.175 (for the case where no addition is considered in last row of Table 3) to 0.0936 (see last row in Table 4), yielding a $(0.0936 - 0.175)/0.175 = 46\%$ reduction with respect to the initial plastic component without any volume addition $B_{O}$.

### 3.3 Design of ribs-based architecture

The topology optimization-driven design shown in Fig. 5, although efficient from the vibration standpoint, is not conventional in the automotive industry. From manufacturing point of view, this design is not feasible, because it can adversely affect the warpage and shrinkage response of the component. Furthermore, this design entails difficulties with regard to the mold manufacturing process. Instead, it is customary to place a ribs architecture on the non-visible plastic part of $B_{NO}$, a solution which facilitates manufacturability in an injection model. In fact, this is a commonly used solution in injection molding due to the great theoretical and practical knowledge about the architecture of ribs and it helps to control both warpage and shrinkage. This is precisely the aim of the present section, which is illustrated in Fig. 6. From this figure it can be seen that the optimized domain $B_{O}$ (highlighted in yellow in Fig. 6), is replaced with a ribs-based architecture which is topologically similar to its yellow counterpart (see Fig. 6). The width of the squared grid of ribs and its height are then conveniently modified with the aim of obtaining a similar mass to that of its yellow counterpart. Crucially, the material properties of both models (the yellow solid optimal region in Fig. 6) and those of its ribs-based surrogate are the same (i.e., elastic properties and true density).

Although a purely mass-driven criterion has been adopted in order to replace the yellow topology optimization design with its ribs-based surrogate, it is clear that the frequency response of both models (yellow design and ribs-based surrogate) must be relatively similar. Otherwise, the ribs-based architecture would not represent a suitable approximation of the yellow design. Therefore, once the geometrical features of the ribs-based architecture have been obtained according to the mass criterion described in the previous paragraph, a free vibration analysis must be carried out over the latter, where its first six eigenfrequencies must be computed. Table 5 shows them. It can be observed that a relatively good agreement between the eigenfrequencies of both models is obtained, where a maximum discrepancy of $32\%$ is observed in the second eigenfrequency. For all the six eigenvalues, the ribs-based surrogate has frequency values which are lower

| $\nu_1$ | 16.6965 |
| $\nu_2$ | 22.1170 |
| $\nu_3$ | 38.9320 |
| $\nu_4$ | 56.2351 |
| $\nu_5$ | 61.1155 |
| $\nu_6$ | 95.4906 |
| $\sum_{j=1}^{6} \frac{1}{\nu_j}$ | 0.175 |

---

**Table 1** Finite Element information for the final Finite Element mesh comprises the discretization of both optimizable and non-optimizable volumes in Fig. 4.

| Finite Element information for $B_{O} \cup B_{NO}$ |
|-----------------------------------------------|
| Total number of elements (excluding contact)  | 1446354 |
| Total number of nodes                        | 355097  |
| Total number of contact elements (tie)       | 50310   |
| Total number of displacement degrees of freedom | 355097×3 |

**Table 2** Material parameters featured in the eigenvalue problem in (6) for $\theta = 1$.

| Material parameters featuring in eigenvalue problem in (6) for $\theta = 1$ |
|-------------------------------|
| $E$ (MPa) $\nu$ $\rho$ (kg/m$^3$) |
| 2.29 0.347 1050 |

**Table 3** Six first eigenfrequencies (in Hz) for the original part $B_{NO}$ without adding the optimizable part $B_{O}$. The last row includes the metric introduced to quantify the vibrational response in the objective function in Eq. (4).

**Table 4** Illustration of the topology optimization strategy advocated for, where the union of a non-optimizable (blue) part $B_{NO}$ and an optimizable (gray) part $B_{NO}$ is considered (the latter attached to $B_{NO}$ in the blind area for the final user). Here, two different views of the superposition of both volumes are shown.
than those of its topology optimization-driven counterpart in Table 4.

Crucially, comparing the last row in Table 5 with that in Table 4, it can be seen that the relative difference in the overall metric that includes the first six eigenfrequencies is $(0.1165 − 0.0936)/0.1165 = 24.65\%$. Comparing the ribs-based surrogate with the frequency response of the single part (no additional optimizable part is considered, see Table 3), the overall metric represents a reduction of $(0.1165 − 0.175)/0.175 = 33.42\%$ with respect to the latter. Figure 7 shows the six natural frequencies for the three cases considered, namely, when no additional volume is considered (blue line), for the topology optimization-driven design (red line), and for the ribs-based surrogate (yellow line). Ideally, the yellow line should be as close as possible to the red line. However, the tendency shown, where the yellow line is closer to the red one rather than to the blue one, except for the second eigenfrequency, indicates that the surrogate model represents a reasonably good approximation of the design obtained in the topology optimization stage.

The eigenfrequency response between the designs obtained at the topology optimization stage and that obtained after its replacement with the ribs-based architecture are reasonably similar. However, if a closer approximation of the ribs-based architecture to that yielded by the initial topology optimization design stage is desired, it is possible to change the ribs configuration and increase its total mass, in a feedback-type process.

### Table 4

| Frequency (in units of Hz) information for TO-based design in Fig. 5 |
|-------------|
| $v_1$       | 30.7923 |
| $v_2$       | 47.9436 |
| $v_3$       | 71.8369 |
| $v_4$       | 92.8920 |
| $v_5$       | 108.7890|
| $v_6$       | 155.8657|
| $\sum_{j=1}^{6} \frac{1}{v_j}$ | 0.0936 |

#### 3.4 Flowchart for design/Finite Element analysis assisted manufacturing process

Prior to their manufacturing, engineering components as that designed on this paper, go through a verification process that extends beyond the conceptual stages described throughout Sects. 3.1–3.3. This is illustrated in Fig. 8, where the conceptual stages (first three boxes in the figure) match each subsection from Sect. 3.1 to 3.3. Typically, additional Finite Element analysis is carried out in order to validate the proposed design from a different engineering metric, not just that considered in the topology optimization-driven design stage, where only vibrational aspects have been considered. For instance, it is customary to perform additional linear elastic analysis for standardized static loading cases. Furthermore, although it is described more in detail through Sect. 3.6, the manufacturing of the engineering component $\mathcal{B} = \mathcal{B}_O \cup \mathcal{B}_{NO}$ is done through an injection molding process where thermal variations may yield undesired deformations, named as molding defects, once the plastic component has been cooled in a controlled manner to reach room temperature. These further analyses in Stage 5 (see list of items below) are the object of the following two subsections and are also conveniently illustrated in the flowchart of Fig. 8.

- **Stage 1.** This stage is part of the pre-processing in the optimization software environment, Altair OptiStruct™ [31]. Here, it is considered to elaborate on the original geometry of the piece by adding material to the back part of the piece. Additionally, the initial and boundary conditions are assigned for the solution of the problem.
- **Stage 2.** This stage corresponds to processing or solving the problem according to the boundary conditions assigned in the previous step.
- **Stage 3.** Here, it deals with postprocessing the results obtained with the conditions assigned to the problem. In this step, the ribs are adapted according to the resulting geometry of the optimization problem.
**Fig. 6** Diagram of the topology optimization showcasing the replacement of the continuum design (yellow) by a squared grid architecture. The obtained volume through TO is transformed into a squared grid that is added to the automotive part analyzed.

**Table 5** Six first eigenfrequencies (in Hz) for the component considering both non-optimizable part $B_{NO}$ and the ribs-based surrogate for $B_{SR}$. Last row includes the metric introduced to quantify the vibrational response in the objective function in Eq. (4).

| Frequency (in units of Hz) for surrogate-based model in Fig. 6 | $\frac{v_j \times 100}{v_j^*}$ | Metric $\%$ |
|---|---|---|
| $v_1$ | 26.0533 | $\frac{v_1}{v_1^*} \times 100$ | -15.38% |
| $v_2$ | 32.4875 | $\frac{v_2}{v_2^*} \times 100$ | -32.25% |
| $v_3$ | 60.1380 | $\frac{v_3}{v_3^*} \times 100$ | -16.28% |
| $v_4$ | 79.0446 | $\frac{v_4}{v_4^*} \times 100$ | -14.90% |
| $v_5$ | 94.0964 | $\frac{v_5}{v_5^*} \times 100$ | -13.5% |
| $v_6$ | 133.8680 | $\frac{v_6}{v_6^*} \times 100$ | -14.10% |
| $\sum_{j=1}^{6} \frac{1}{v_j}$ | 0.1165 | | |

**Fig. 7** Six first eigenfrequencies for three cases considered, namely, when no additional volume is considered (blue line), for the topology optimization-driven design (red line), and the ribs-based surrogate (yellow line).
Stage 4. The results in stages 2 and 3 are validated through the mechanical behavior and modal analysis of the part. Finally, if the results are not in agreement, the design is returned to stage 1. Consequently, imposing new initial conditions on the problem again.

Stage 5. This step consists of taking the component previously validated in the modal analysis (whose density of the optimized material corresponds to the number of ribs) and performing an analysis of its mechanical behavior, subjecting the piece to loads of \( F = [0 \ -50 \ 0] \) (N). The final analysis corresponds to the filling of the piece in the mold, thus evaluating that the deformation and contraction of the piece are within the allowed limits.

### 3.5 Validation of advantage of the topology optimization over a design on structural stiffness

Topology optimization (TO) over the optimizable (gray) part \( B_O \) in Fig. 4 is intended to have an impact on the free vibration response of the overall structural component \( B = B_O \cup B_{NO} \). This is due to the underlying nature of the objective function \( J(\theta) \) in Eq. (4), which directly aims at maximizing a finite number of eigenfrequencies of the combined component \( B \). Despite the specific vibration-based nature inherent in the objective function, it is possible to check, at a postprocessing level, if the final design of \( B_O \) brings additional benefits not necessarily related
to vibration. This work intends to quantify the gain in 

**stiffness** introduced by the surrogate ribs-based design in Fig. 6. In order to do that, a system of three forces is considered, each one of value $F = [0 \ -50 \ 0]$ (N), acting on the overall component (see Fig. 9), which is subjected to zero displacement boundary conditions on the nine holes (see Fig. 9).

When subjected to these boundary conditions, a standard linear elastic analysis was carried out (see Remark 2) of the ribs-based surrogate model in Fig. 6 when subjected to these boundary conditions and the monitored contour plot of the Euclidean norm of the displacement field can be seen in Fig. 10a. In addition, the same linear elastic analysis is carried out when the added optimizable volume $B_O$ is not present. The latter result can be observed in Fig. 11b. Crucially, the maximum displacement (which occurs at similar regions in both figures) yields a reduction from 7 mm to 4 mm for the ribs-based surrogate, yielding a 57% reduction of the maximum displacement with respect to the case excluding the optimizable domain (no ribs).

**Remark 2** The linear elastic analysis carried out in order to compare both models (ribs-based surrogate in Fig. 10a and that devoid from any reinforcement at all in Fig. 10b) corresponds with the solution of the following boundary value problem

$$
\sum_{j=1}^{3} \frac{\partial \sigma_{ij}(\varepsilon(u))}{\partial x_j} = 0; \quad i = 1, 2, 3; \quad \text{in } B
$$

$$
u = u^*; \quad \text{on } \Gamma_D
$$

$$\sigma(\varepsilon(u))n = t; \quad \text{on } \Gamma_N
$$

where $\nu$ represents the displacement field and $\varepsilon(u) = \frac{1}{2} \left( \frac{\partial u}{\partial x} + \frac{\partial u}{\partial x}^T \right)$. For the case of linear elasticity considered, $\sigma$ is obtained in terms of the elasticity tensor $E$ through the linear relationship

$$
\sigma_0 = \sum_{k,l=1}^{3} E_{ijkl} \varepsilon(u)_{kl};
$$

$$E_{ijkl} = \frac{E}{2(1 + \mu)} (\delta_{ik} \delta_{jl} + \delta_{il} \delta_{jk}) + \frac{E}{(1 + \nu)(1 - 2\nu)} \delta_{ij} \delta_{kl};$$

where $E$ and $\nu$ represent the Young’s modulus and Poisson’s ratio (see Table 2). Furthermore, $\Gamma_D$ and $\Gamma_N$ represent the part of the boundary of $B$ where both Dirichlet (displacement) boundary conditions and Neumann (tractions or forces) boundary conditions are prescribed. The weak form
of above Eq. (13) yields to the well-known principle of virtual work, which can be recast as

$$\int_{B} \sigma(\epsilon(u)) : \epsilon(v) \, dv - \int_{\Gamma_{W}} t \cdot v \, da = 0,$$

where $v = 0$ in $\Gamma_{D}$ represents the test functions. Finite Element discretization of (15) yields the well-known linear system of equations in terms of the nodal displacements $U$ of the underlying Finite Element mesh, i.e.,

$$KU = F.$$  

### 3.6 Validation of advantage of the TO over injection molding process (warpage and shrinkage)

During the injection molding process, defects such as warpage and shrinkage are common quality problems for plastic molding products that affect their performance. A common solution is to strengthen the plastic part by adjusting both the type and number of ribs to avoid excessive increase in stiffness, then this study aims to evaluate molding defects by using numerical simulation of the filling of a plastic part whose attached ribs have been previously established by topology optimization in Sect. 3.3.

The manufacturing of the automotive component finally designed in Fig. 6 is carried out through an injection molding process, where the solidification of an initially plasticized polymer flowing through a mold will lead to the final plastic component (see Fig. 12a). For the specific case of the plastic component considered in this paper, the used mold has one cavity and three injection points represented by the three yellow conical shapes, as seen in Fig. 12b.

The filling process occurring through the injection inlets and the subsequent cooling down process can be mathematically modeled by means of a system of conservation equations comprising the continuity equation (conservation of mass), the conservation of linear momentum equation, and the conservation of energy equation. These can be stated as

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \upsilon) = 0;$$

$$\rho \frac{\partial \upsilon}{\partial t} + \rho(\nabla \upsilon)\upsilon = -\nabla p + \nabla \cdot \tau + \rho g;$$

$$\rho c_p \frac{\partial T}{\partial t} + \rho c_p \nabla T \cdot \upsilon = \nabla (\kappa \nabla T) + \eta \dot{\gamma}^2,$$

which must hold on $B \times [0, T]$, where \( \{ \rho, \upsilon, p, T \} \) refer to the density, velocity, pressure and temperature of the injected flowing polymer. Furthermore, $g$ is the gravity, and $\{c_p, \kappa, \eta \}$ are the specific heat constant at constant pressure, the heat conductivity and the viscosity of the fluid, which can depend on a non-linear fashion with respect to the strain rate $\dot{\gamma}$ respectively. The expression for the latter and the remaining terms in (17) which remain to be defined are
Usually, an additional constitutive equation is required relating the pressure field \( p \) to the temperature and density fields. However, the software used in this paper (Moldflow\textsuperscript{TM} [32]) makes some geometrical assumptions that simplify considerably above equations (the reader is referred to [33]). The boundary value problem in (17) (appropriate boundary and initial conditions need to be imposed, guaranteeing uniqueness of solutions) is simulated in MoldFlow\textsuperscript{TM} [32], reaching a quasi-static solution at time \( t \).

\[
\tau = 2\eta(\dot{\gamma})D; \quad D = \frac{1}{2}(\nabla v + (\nabla v)^T); \quad \dot{\gamma} = \sqrt{2\tau(D^2)}.
\]

The injection molding process entails the injection of a flowing plasticized polymer through the cavity of a mold. Inside the cavity, the fluid will experience a cooling down process until solidification, reaching its required dimensions of the solidified plastic component at room temperature. However, during this process, residual stresses can develop due to the high pressures reached, the change in temperature, and the polymer chain relaxation, yielding a contraction and deformation of the final solidified plastic component. In order to quantify this temperature-induced deformation once the component has cooled down outside the mold, an additional numerical simulation was carried out a posteriori (once a solution for (17) has been

---

**Fig. 12** a Sketch of a typical mold where an owing polymer is injected through the inlet cavity; b True mold for the plastic component considered in this paper, created in the commercial software Moldflow\textsuperscript{TM} [32]

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**Fig. 13** The two cases considered for both warpage and shrinkage analysis: a ribs architecture spanning all over the non-optimizable domain; b design of ribs architecture yield from topology optimization analysis in Sect. 3.3
obtained). The Duhamel-Neumann constitutive law [23, 34] is considered with that purpose, where the stress field $\sigma$ in the cooled down component is defined according to

$$
\sigma_{ij} = E_{ijkl}(\varepsilon(u))_{kl} - (2\mu + 3\lambda)a \Delta T \delta_{ij};
$$

$$
E_{ijkl} = \mu(\delta_{ik} \delta_{jl} + \delta_{jl} \delta_{ik}) + \lambda \delta_{ij} \delta_{kl},
$$

(19)

with $a$ the coefficient of thermal expansion of the solidified component and $\mu$ and $\lambda$ defined in Remark 2. In addition, $\Delta T$ represents the thermal variation experienced by the material in the cooling down process.

The thermal-induced deformation and contraction are two crucial factors determining the quality of the final product that went through the injection molding process. Mathematically, the deformation/warpage or more precisely, the displacement is characterized by the solution of a linear elastic problem similar to that on Remark 2 where no static forces are applied but where the thermal part of the stress field $\sigma$ in Eq. (19) is responsible for the deformation. The second factor identified, namely contraction, is also referred to as shrinkage and will be monitored for each Finite Element after the thermal-induced linear elastic analysis described has been carried out.

In quantifying warpage and shrinkage, a comparison of these two defects was carried out between the design
obtained in Fig. 6 and the case where an indiscriminate use of ribs is considered. Both cases are displayed in Fig. 13.

With regards to thermal-induced warpage, Fig. 14 shows the contour plot distribution of the Euclidean norm of the displacement field for both cases displayed in Fig. 13. Clearly, the maximum displacement field value is obtained for the case where the ribs architecture extends all over the non-optimizable domain. In particular, a warpage defect reduction from 6.4 mm to 4 mm is observed in the latter case, proving that although the ribs-based architecture has been designed with an underlying vibrational aim, a collateral benefit has been obtained too from the warpage standpoint.

Finally, with regards to thermal-induced shrinkage also associated with the cooling down process, Fig. 15 shows the contour plot distribution of the shrinkage field for both cases displayed in Fig. 13. In this case, there is no remarkable difference between the contour plot distribution for both configurations in Fig. 13, as an approximate value of 5% contraction is observed for both configurations in Fig. 15. Therefore, it can be concluded that the ribs-based design obtained in Sect. 3.3 does not entail any significant benefit from the shrinkage standpoint concerning the configuration in Fig. 13a. However, it does imply an undesired increase in shrinkage either, which is a positive aspect.

4 Conclusions

This paper demonstrates the applicability of topology optimization methods on thin-wall plastic components used in the automotive industry; from the results of this research can draw the following conclusions:

1. Applying TO correctly on the component determines the feasible areas for ribs-based architecture to increase the values of the component’s frequency. For the free vibration analysis of the component in its initial state without ribs-based architecture carried out in Altair OptiStructTM [31], a value of 16.69 Hz is obtained. Once the component is added to the ribs-based architecture following the areas indicated by the TO, an increase to 26.05 Hz is obtained for the first natural frequency of the entire plastic component.

2. Although not contemplated within the topology optimization stage, the final ribs-based design has introduced additional benefits and enhanced eigenfrequency response. A standard linear elastic analysis of the ribs-based surrogate model and the thin plastic component (devoid of ribs) could be calculated as a 57% reduction in the maximum value of the Euclidean norm of the displacement of the first concerning the latter. It is worth noticing that this is generally used as a metric for structural stiffness, as displacement reduction is intimately related to compliance, which is the inverse of stiffness.

3. Additionally, the thermal analysis associated with the injection molding process of the final design was carried out using the commercial software MoldFlowTM [32]. As a result, it was possible to determine the thermal-induced effects, such as warping and shrinkage, on the final plastic component design. This allowed corroborating an improvement in terms of warpage for the case where ribs are introduced in an indiscriminate fashion when going from a value of 6.76 to 4.93 mm. From the shrinkage standpoint, a similar shrinkage response was obtained to the indiscriminate ribs-system approach, although within the manufacturing recommendations.

A comprehensive guideline for the design of thin plastic components in the automotive industry has been presented in this paper, covering the initial topology optimization-driven design stages and further verification analysis stages encompassing stiffness assessment and warpage/volumetric shrinkage evaluation after the mold injection process. Using TO from the design stages generates great economic bents, time savings, component quality, and ease of manufacturing (injection mold and part) because it is the design stage where rework costs less and innovations like these can be applied for the benefit of the components.

For future work, this research opens the possibility for additional works where, in addition to vibration, stiffness and warpage/shrinkage could be included in a multi-objective type approach within the topology optimization design stage, thus, allowing the topology optimization algorithm to yield designs where a balance between the different objectives is achieved, rather than verifying some of them at postprocessing stages. However, this study is limited to CAE analysis of vibration response, static loading cases, and warpage/volumetric shrinkage, and manufacturing feedback is not considered for the design of ribs architecture configuration. Therefore, also as future work, it is recommended to do this type of analysis with different patterns of ribs, different configurations of height and width, different materials, and different thicknesses of the base components; from these new analyses, results will most likely be obtained where the component improves its stiffness performance with less mass, following what is shown in this research work.

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