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Original Research Article

Application and optimization of damping pad to a body-in-white of a vehicle for improved road noise, vibration and harshness performance

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ABSTRACT

Road noise is expected to become even more important in the vehicle product development cycle due to electrification and challenging lightweight/emission targets. In this study, a topology optimization algorithm is applied to determine the damping pad layout on the roof and floor panels of a Body-in-White (BIW), being the dominant contributors on road noise, vibration and harshness (NVH) performance of an automotive. Optimization algorithm yields the prescribed % of the surface area of these panels where the damping pad should be distributed set by the automotive Original Equipment Manufacturers (OEMs). The objective function is the minimization of the overall acceleration of these panels for the frequencies up to 200 Hz, while the weight of the BIW is considered as the optimization constraint. The results of the optimization are compared with the road NVH performance of panels with full damping and no damping. The optimization results indicate that by using 25% of the damping pad on the roof and floor panels improve the vibration performance especially in the frequency range of 80 Hz to 150 Hz significantly compared to bare BIW panels. Besides, the performance of the 25% damping is almost same as the application of full damping pad for frequencies between 90 Hz to 110 Hz. The results show that the methodology is able to address the trade-offs between road NVH and weight targets effectively.

Keywords: Damping Pad, Body-in-White (BIW), Topology Optimization, Passenger Cars, Frequency Response Analysis, Finite Element Modeling (FEM)

1. Introduction

Electric and hybrid electric vehicles have become more common in the recent years due to stringent requirements on emissions and customer expectation from the fuel economy. It has been forecast that the sales for electric vehicles will increase considerably by 2020 [1]. In addition to being more environmentally friendly, electric vehicles are also quieter than vehicles equipped with internal combustion engines. This is attributed to the "silent" characteristics of electric powertrains and, thus, road NVH will become more dominant for vehicle NVH problems with advancement of this technology [2]. Road NVH is a challenging development area for automotive OEMs since it encompasses a wide range. The problem becomes even more challenging as it may conflict with other design requirements such as lightweight. More specifically, by using materials such as composites, sound package and thicker panels, the road NVH can be improved significantly. However, this is not desired due to its implication on the development cost and lightweight requirements. Considering the challenging targets of automotive industry in 2020s, we can only think that the road NVH development will be one of the core development areas in the automotive industry [3].

Road NVH is a very complex phenomenon related to many subsystems and components of a vehicle including suspension, elastomer properties, chassis components, sound package, and vehicle body structure. In addition, vehicle speed and road-tire interaction plays an important role on the road NVH characteristics of a vehicle [4]. The product development cycle is usually based on virtual prototyping and hardware testing. Even though, there is a vast body of literature on the test-based methods [5], the recent trend is to reduce hardware testing and the cost/time associated with it.

A wide variety of Computer Aided Engineering (CAE) tools and analytical methods are available as commerical or open-source codes for the NVH development. Multi-body dynamics software are used to determine the loads during customer use; while the finite element (FE) software is generally employed to study modal alignment, body sensitivity and airborne performance. Similarly, computational fluid dynamics simulations are performed to study wind noise, and statistical energy analysis for the calculation of high frequency vibrations and acoustics. A detailed review of these software can be found in [6]. There is a vast body of literature on the application of these tools to resolve road NVH related problems especially in the recent years. For example, a multi-body dynamics model in Automatic Dynamic Analysis of Mechanical Systems (ADAMS) software is used to determine the stiffness of the suspension bushings for static and dynamic loading for a pick-up truck [7]. Xu et al. developed a new method using the frequency response functions (FRFs) using finite element models (FEM) in order to determine the road noise, which was verified experimentally [8]. A sound package development study was performed using a FEM for calculating the attachment point sensitivity for low frequencies and a statistical energy analysis for the high frequencies [9]. A full FEM of a vehicle is used to study the effect of road conditions and speed on road NVH by determining the panel and modal participation factors from the results of design sensitivity [2]. A multi-disciplinary approach is taken for the calculation of dynamic stiffness of elastomer elements, determined from the force spectrum at the suspension attachment point up to 100 Hz, and finally a finite element model is used to determine the structure-borne noise inside the cabin for Road NVH [10].

The advancement of the aforementioned tools led to the development of multi-disciplinary optimization codes and software to address the trade-offs in the product design cycle. For example, a multidisciplinary design optimization (MDO) method using meta-models is applied to vehicle structures to address the trade-offs between different vehicle attributes [11]. A shape optimization for the center floor panel as the major contributor on the Road NVH in terms of sound pressure level inside the cabin is performed using genetic algorithms for the low frequency range. The methodology is demonstrated on a sport utility vehicle (SUV) and experimentally verified [12]. A topology optimization algorithm is used for the design of automotive joints with beam-like cross sections according to the manufacturing methods such as welding and stamping under BIW body static stiffness and attachment point dynamic stiffness [13]. A sensitivity-based optimization algorithm is employed to optimize the gauges of body components made of aluminum under static and modal analysis by reducing 9% of the overall weight [14].

Topology optimization is an optimization algorithm in order to determine the optimum material distribution in a structure given the objective function and constraints [15]. There have been many applications of this algorithm to automotive applications. In one of the recent studies, Sun et al. [16] applied topology optimization algorithm to a vehicle door to derive the optimum tailor-welded blank design. Tuncer and Sendur [17] applied a frequency based topology algorithm to improve the sound quality of a vehicle door by distributing the damping pads on the door outer panel. Even though the application of the topology optimization algorithms to structural engineering applications is common, there is a gap on its application to determine to optimal damping material layout on the BIW of a vehicle body on important vehicle attributes such as road NVH in the literature. The objective of this paper is to determine the optimum layout of damping distribution of vehicle body panels for the low frequency range. OEMs are expected to benefit from the methodology to finalize their BIW design and make the body shop related planning in their assembly plant. For that purpose, two main contributors are chosen: vehicle roof and floor panels for their known contribution to the structure-borne vibration characteristics. Since the structure-borne vibration is related to the frequencies up to 200 Hz [12], a frequency based objective function in terms of the panel accelerations under customer use is chosen. The weight of the damping package is constrained in order to meet OEMs lightweight targets.

The remainder of the paper is organized as follows: firstly, the details of the finite element model of BIW, damping pad model and road NVH simulation details are presented in Section 2. Then, the topology optimization problem is described in Section 3. Section 4 is reserved for the detailed discussion of the results. Finally, the paper is concluded with conclusions in Section 5.

2. Materials and Methods

In this section, details of the finite element model of the BIW (Section 2.1), damping pad (Section 2.2) and the details of the road NVH simulation (Section 2.3) are described.

2.1. BIW finite element model

The BIW of the vehicle body constitutes the base for the analysis model. The forces from the road are represented for forces applied at front and rear suspension towers. The methodology is demonstrated on a passenger car. For that purpose, the FEM of a 2010 Toyota Yaris (shown in Figure 1.a), developed by Center for Collision Safety Analysis (CCSA) [18], is used as a test case. Important aspects of the simulation model are summarized below:

- BIW is the metal sheet structure of the body assembled in the assembly plant. The components, such as bumper beams, that are attached by bolts are also in the finite element model (Figure 1.b).
- The windscreen is added to BIW model as it adds structural rigidity and plays an important role on the static and modal characteristics of a vehicle body.
- Major panels are modeled by two types of finite elements: i) CQUAD4 element, which is a quadrilateral plate element connection, and ii) CTRIA3 element, which is a triangular plate element in Nastran [19].
- Connections are important on the static and dynamics response of the vehicle structure. Most common types of connections in the form of spot-welds are modeled in the FEM of BIW. For that purpose, RBE3-HEXA-RBE3 elements are used (Figure 1.c). RBE3 element is an interpolation constraint element in Nastran to connect the spot-welds to the panels that are welded to each other, while the HEXA element in Nastran represents a six-sided solid element connection. The modeling approach is standard in the automotive industry. For more detailed information on modeling, the reader is referred to [20, 21].
- A similar modeling approach as the spot-weld modeling is followed to model the glue-bondings (such as bonding of windscreen and roof bows). The bonding is modeled as a continuous RBE3-HEXA-RBE3 elements between the windscreen and cowl panel for the bonding of windscreen (Figure 1. d) and between roof and roof bows (Figure 1.e).
- Chassis attachment points are modelled with a rigid element (RBE2) elements (Figure 1.f). Since the suspension attachment points are where the forces from road are transferred to the vehicle body for road NVH assessment, modeling of these locations is critical.
- Attachment holes are modelled with quad elements with an extra circle outside the rigid element. This makes the modeling of such stress concentration locations more accurate.
- Bolt joints are modeled with RBE2-CBAR-RBE2 elements. This type of modeling takes into the flexility of these connections in the vehicle structure. RBE2 element represents

rigid body element, while CBAR element is the beam element in Nastran.

The model is verified against a set of finite element quality criteria given in Table 1. For more information, the reader is referred to the definition of finite element quality criteria [19]. This type of model is generally deemed sufficient to study the low frequency noise and vibration investigations in the literature [20].

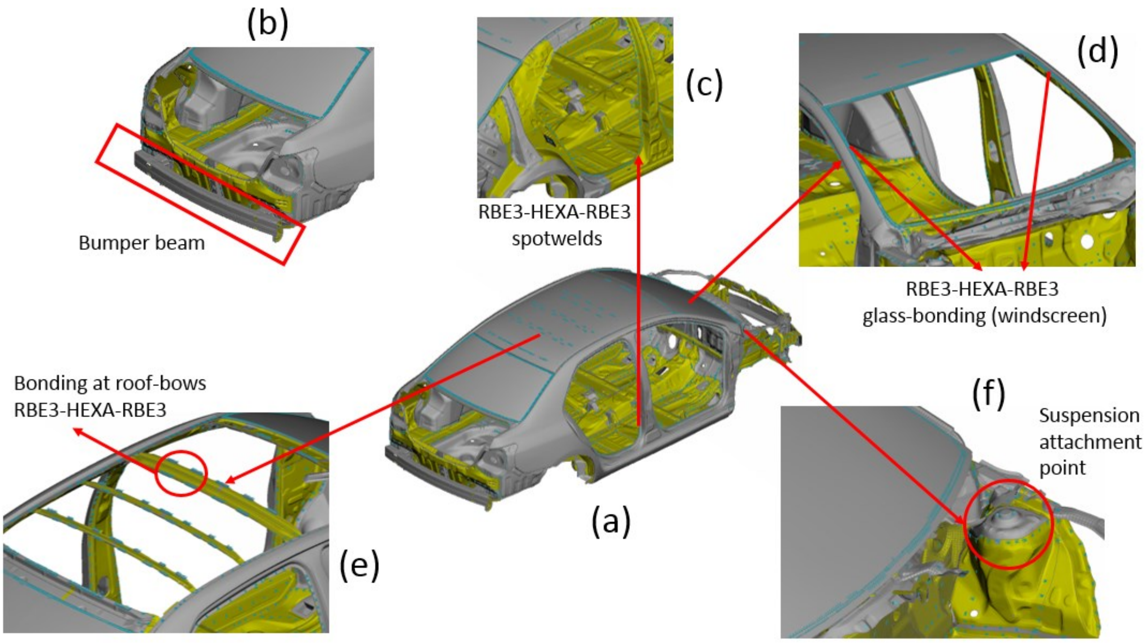

Figure 1. Finite element model of Toyota Yaris a) BIW, b) bumper beam model, c) RBE3-HEXA-RBE3 type spotweld model, d) glass-bonding model, e) bonding between roof and roof bows and f) modeling of front suspension attachment point

Table 1. Quality criteria of the FEM

Quality criteria	Value
Aspect ratio (max.)	3
Skewness (max)	45°
Warpage (max)	10°
Min. angle for quads	45°
Max. angle for quads	135°
Min. angle for trias	30°
Max. angle for trias	120°
Jacobian	0.7
Min. element length	3 mm
Max. element length	12 mm

2.2. Damping pad model

The damping pad is simply modeled as shell elements which are created by copying the finite elements of the panels they are attached to by an offset which is the half of the thickness of the damping pad. The damping pad is characterized by its density, Young's modulus and damping loss factor. Young's modulus is more effective in terms of the overall stiffness of the panel. Density changes the mass of the damping pad and overall mass of the panel, therefore, it has an effect on the modal characteristics. The thickness of the damping pad is chosen as 2 mm. The damping pad models for the roof and floor panels are shown in Figure 2.a and Figure 2.b, respectively. Finally, the damping loss factor is related to the structural damping of the panel. The parameters of the damping pad model are shown in Table 2.

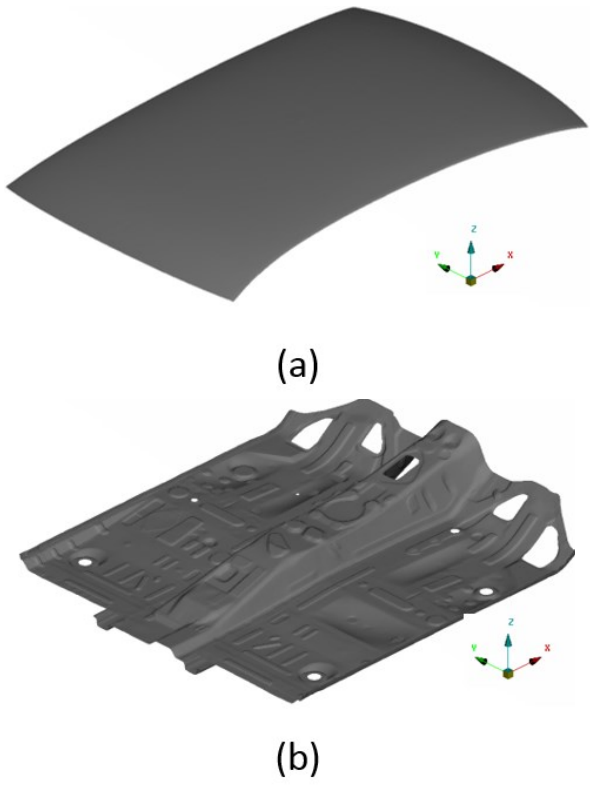

Figure 2. Damping models a) roof panel, b) floor panel

Table 2. Material properties of the damping pad [21]

Properties	Value
Density	2 g/cm^3
Young's Modulus	4000 MPa
Poison's Ratio	0.3
Damping Loss	0.3
Thickness	2 mm

2.3. Road NVH assessment

Road NVH performance is determined by performing a frequency response analysis (FRA) in Nastran software. For that purpose, the BIW of the vehicle is excited by a harmonic function at four suspension attachment points as shown in Figure 3. The frequency of the excitation is from 0 to 200 Hz with 1 Hz increment. In the literature, it is known that structure-borne frequency is mostly related to up to 200 Hz [12].

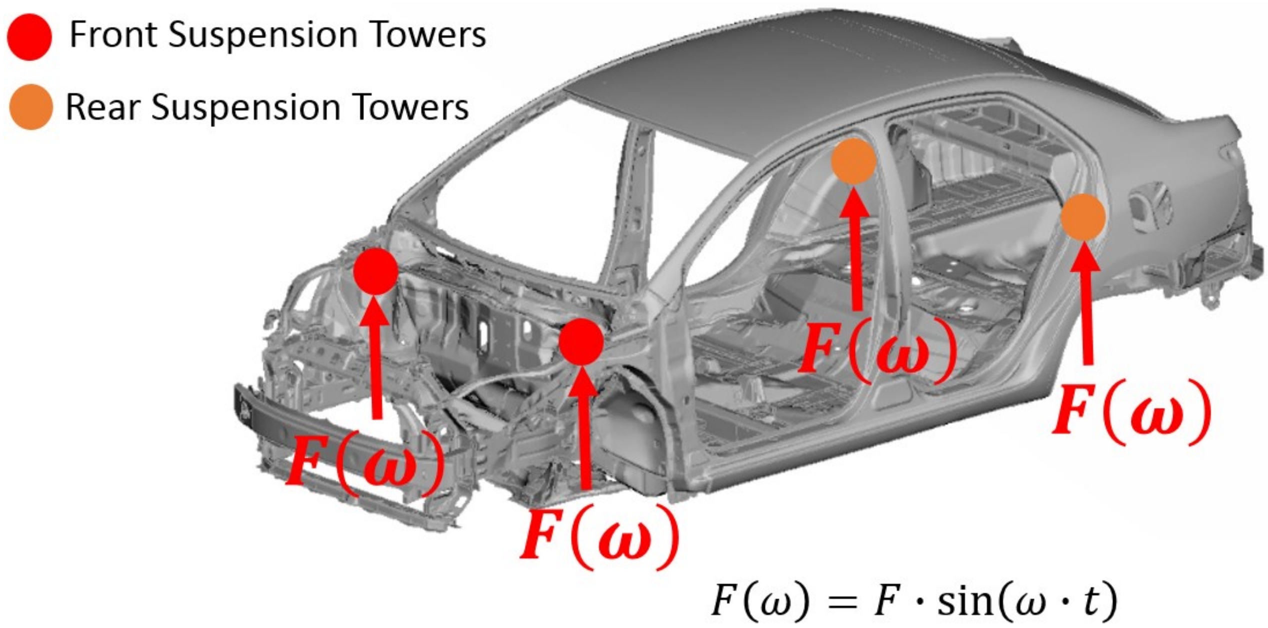

Figure 3. Excitation at front and rear suspension towers

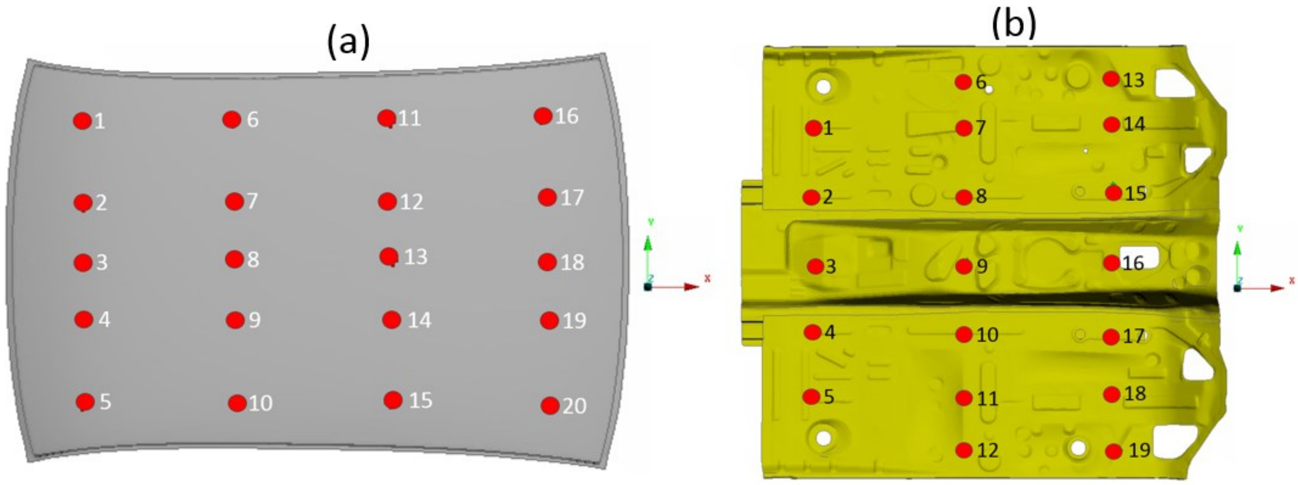

Figure 4. Measurement points a) roof panel, b) floor panel

The outputs from the FRA are the acceleration (or displacement/velocity) of the points of interest on the mechanical system. Floor and roof panels are considered as the main contributor of the Road NVH [12]. The acceleration of these panels on the points shown in Figure 4 are calculated as a measure of the vibration of these panels. The floor FEM model of the roof and floor panels contain many nodes, and calculating the acceleration of all the nodes as a measure of the vibration level of these panels is computationally expensive. Therefore, evenly distributed number of points on these panels is used for the calculation of the accelerations. This approach enables the efficient determination of the overall vibration from these panels in an average sense. For this purpose, 20 points are determined on the roof

panel (shown in Figure 4.a) and 19 points are used on the floor panel (shown in Figure 4.b).
The vibration of a panel, which is proportional to the sound radiation, is calculated according to Equation (1) by using the acceleration spectrum up to 200 Hz of the points for the panels. Since the accelerations can be positive and negative over the frequency range of interest, the acceleration is squared and integrated first to account for the direction. According to this expression, the frequency response for each point is calculated for point 1 to point N using the finite element model. Then, the magnitude of the acceleration for each point is squared and integrated over the frequency of interest (continuous summation). Once all the integrals are calculated separately for point 1 to point N, they are summed (discrete summation). Finally, V_{panel} is calculated by taking the square root of the summation. The reader is referred to [22] for more detailed explanation of the mean square calculation. Lower values of V_{panel} is desired for better NVH performance of the vehicle cabin. Therefore, minimization of this quantity will be considered as the objective function in the topology optimization problem.
where V_{panel} is the overall vibration metric, ω_{min} and ω_{max} are the lower and upper frequency limits for the low-frequency range, respectively. In this study since the low frequency is taken up to 200 Hz, ω_{min} is set as 0 Hz. and ω_{max} is 200 Hz. In this equation, i represents the number of points on the panel where the acceleration is calculated (i is 20 for roof panel, and i is 19 for the floor panel). N is the total number of points on the roof and floor panels, which is 39 for this case study? The calculation of Equation (1) is performed in ANSA software.
Modal frequency response analysis is performed in Nastran to calculate the acceleration of the panel measurement points. For that purpose, a modal analysis up to 500 Hz (more than twice of the frequency range of interest, which is 200 Hz for low frequency) is performed as the initial step of the FRA.

$$V_{panel} = \sqrt{\int_{\omega_{min}}^{\omega_{max}} [acc_1(\omega)]^2 \cdot d\omega + \int_{\omega_{min}}^{\omega_{max}} [acc_2(\omega)]^2 \cdot d\omega + \cdots + \int_{\omega_{min}}^{\omega_{max}} [acc_N(\omega)]^2 \cdot d\omega} = \sum_{i=1}^{N} \sqrt{[acc_i(\omega)]^2 \cdot d\omega} \quad (1)$$

3. Topology Optimization to Application of BIW

The objective of the topology optimization is to determine the best locations where the damping material is to be applied. Therefore, previously defined panel vibration, V_{panel}, is used. The topology optimization problem is defined in Equation (2). The objective function is the minimization of the overall acceleration of the body structure panel where damping pad is applied. The constraint is to limit the volume of the damping pad determined by the OEMs. This ratio is, in general, related to several factors in the product development cycle such as the cost and weight increase due to the addition of the damping pads. Even though, the volume ratio is predetermined as 0.25 in this case study, the topology methodology proposed is general and can be exercised for different volume ratios. This means that the optimizer will determine the optimum location of the 25% of the full pad (and remove 75% of the full damping pad) to minimize the objective function given in Equation (2).

$$Find: \rho$$

$$Minimize: V_{panel} = \sum_{i=1}^{N} \sqrt{\int_{\omega_{min}}^{\omega_{max}} [acc_i(\omega)]^2 \cdot d\omega} \quad (2)$$

$$such\ that\ V_f \le 0.25$$

$$0 < \rho \le 1$$

where ρ is the topology variable, which is constrained between 0 and 1. The values closer to 1 indicate the critical areas on the panel, while lower values mean they are least needed. The output of the topology optimization can be, optionally, color-coded in ANSA software for visualization purposes to show the critical locations for the application of damping pad; see Figure 6 as an example. Finally, the pre-determined volume (%) of the material can be applied on the floor and roof panels in order to minimize the panel vibrations. Readers are referred to [15] for more detailed information about the topology optimization methodology.

4. Results

4.1. Effect of full damping application on road NVH metrics

Before proceeding with the topology optimization, the response for the addition of full damping pad is compared with the panel with no pad. Therefore, a frequency response analysis is performed for these two cases. Figure 5 shows the summation of the acceleration of the 39 points up to a frequency range of 200 Hz. The results show that the amplitudes are reduced significantly by using the full damping pad on the roof and floor panels especially in the frequency range of 80 – 150 Hz. This frequency is related to error states such as boom noise. Therefore, the full damping coverage of both roof and floor panels is expected to improve overall vehicle NVH performance. However, this results in some weight increase in the BIW design. More specifically, the weight is increased by 14.6 kg.

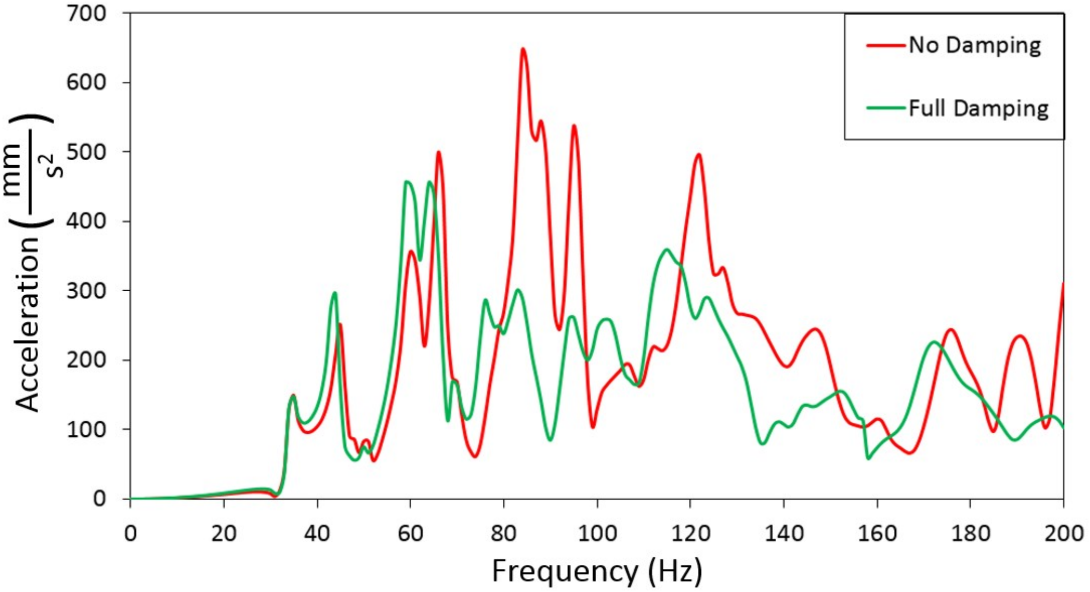

Figure 5. Overall vibration from roof and floor panels with no damping and full coverage of damping on both panels

4.2. Topology optimization results

There are two stopping criteria for the optimization algorithm to terminate: 1) maximum number of iterations is reached, or 2) the % change in the objective function between two consecutive iterations is less than a predetermined value. In this case study, the maximum number of iterations is set to 50 to make sure that there is no error in the finite element model and set-up of the optimization problem. The pre-determined % change of the objective function is set as 0.1%. Topology optimization is performed on a Dell Precision T5810 computer with Intel® Xeon® CPU E5-1607 running of 3.10 GHz sampling and 32GB RAM. The optimization algorithm converged in 5 iterations. The topology optimization cycles for the roof and floor panels are shown for each design iteration in Figure 6. The results show that the vibration of the roof and floor panel according to Eq. 1 is reduced by 1522 $mm \cdot \sqrt{Hz}/s^2$ (for iteration 1), 2487 $mm \cdot \sqrt{Hz}/s^2$ (for iteration 2), 3465 $mm \cdot \sqrt{Hz}/s^2$ (for iteration 3), 4312 $mm \cdot \sqrt{Hz}/s^2$ (for iteration 4), and 4990 $mm \cdot \sqrt{Hz}/s^2$ (for iteration 5) compared to the no-damping case. The finite element model of the optimum damping layout by keeping the 25% of the full damping pad (as shown in Figure 2) is shown in Figure 7.

The FRA for the BIW with this optimum damping is re-run and the results of the FRF analysis are compared for the BIW with no damping, BIW with 100% damping and BIW with 25% damping. The FRFs for three cases (no damping, full damping and 25% damping) are plotted in Figure 8. The results show that the optimized damping pad layout is quite effective. The visual comparison of the FRFs indicate that the optimum damping reduces the undesired acceleration peaks compared to no damping case effectively in the frequency range of 80 Hz to 150 Hz. It is also concluded that the performance of the optimum damping coverage is almost same as the full damping application in the frequency range of 90 Hz to 110 Hz. The weight increase with the full application of the damping pad is also reduced by 75%, which is 10.95 kg, since the topology optimization distributed the 25% of the damping pad at the critical locations on the roof and floor panels.

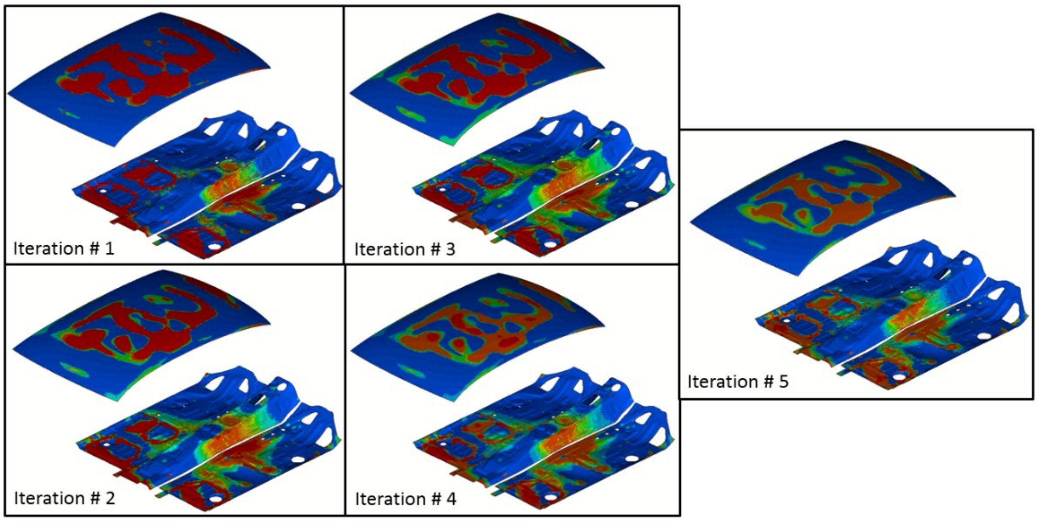

Figure 6. Topology cycles for roof an floor panels of BIW

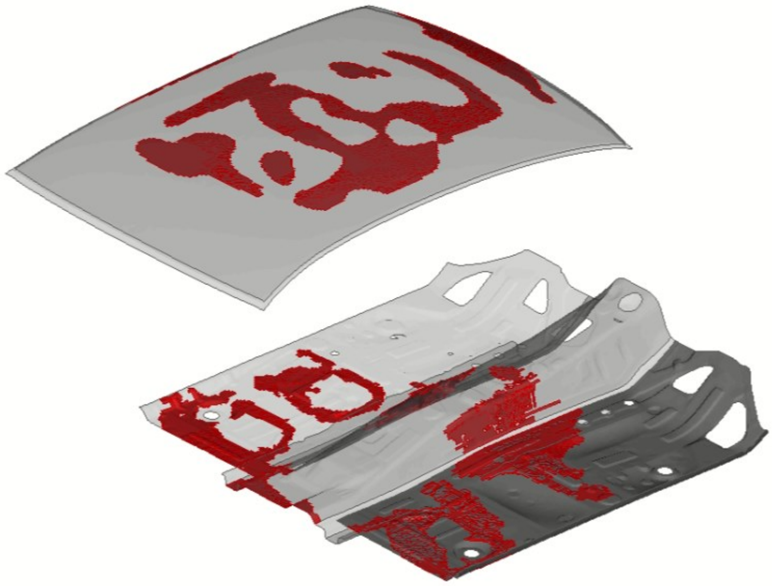

Figure 7. Finite element model with 25% damping on the roof and floor panels

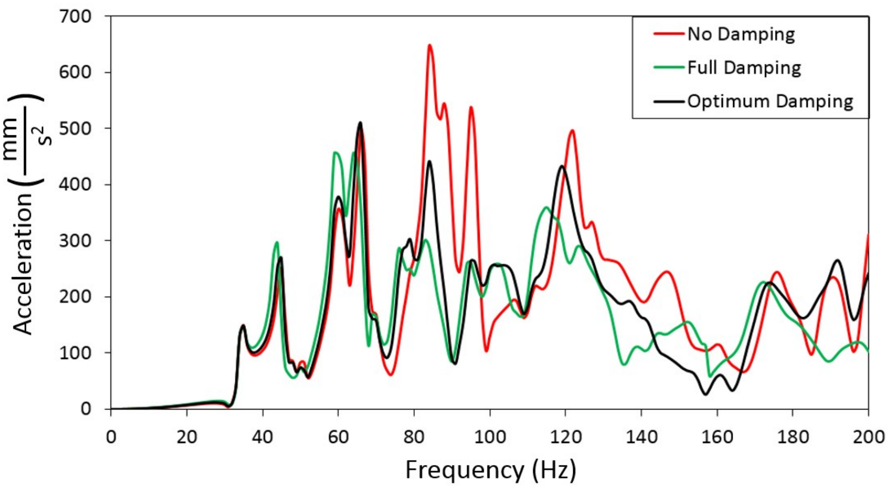

Figure 8. Overall vibration from roof and floor panels with no damping, full damping and optimum damping on both panels

5. Conclusions

A topology based optimization methodology is applied to determine the optimum layout of the damping pad distribution on the roof and floor panels of a BIW. The study concludes that the integration of optimization algorithms with simulation tools will be a great contributor to meet future's challenging weight and emission targets. Key findings from this study are summarized below:

- By fully covering the sheet metal on the aforementioned BIW panels, the vibration performance is increased significantly. More specifically, the overall acceleration of the roof and floor panels are reduced significantly in the frequency range of 80 Hz to 150 Hz compared to bare BIW panels. However, the vehicle weight is increased by 14.6 kg by the application of damping pad on both panels.
- The optimum distribution of the damping pad by only using the 25% of the full damping layout is obtained by the application of a topology optimization study. The comparison of the optimum damping coverage with the bare panels and full damping coverage indicate the effectiveness of the optimized damping pad. More specifically, the overall acceleration levels in the 80-150 Hz is reduced significantly compared to the frequency response of the bare panels. Besides, a very comparable performance is achieved with the optimum damping coverage as the full damping application in the frequency range of 90 Hz to 110 Hz. The performance improvement with the optimum damping coverage increases the vehicle weight by only 3.65 kg, which is 25% of the full damping coverage.

The transformation of the topologies obtained from the topology optimization to a more manufacturable design so that OEMs can apply these in their assembly plant is an important research topic, which is left as future work. Multidisciplinary optimization for powertrain NVH, road NVH and other vehicle attributes is also acknowledged as a future work.

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