TOPOLOGY OPTIMIZATION OF CONSTRAINED AND UNCONSTRAINED DAMPING LAYERS

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Damping is an effective way of suppressing mechanical vibrations. Passive constrained and unconstrained layer damping patches are often used in the industry as damping solution for vibrations and acoustics problems. In this study, topology optimization methods were applied to the finite element models (FEM) of plates and damping patches in order to determine the critical locations for the frequency range of interest. For that purpose, MSC. Nastran SOL 200 optimization is used for both topology and thickness optimization. The structures are excited at single point and the acceleration of the plate at a predetermined location is considered as output. The objective of the optimization is chosen as the minimization of the acceleration spectrum up to 500 Hz. The optimized damping patch topology is compared to the base plate with full application of damping patch on frequency response functions. The simulation results from topology and thickness optimization show that these methods can effectively improve the frequency response functions.

Keywords: damping, constrained damping, finite element methods, topology optimization

1. Introduction

Plate and shell structures are extensively utilized in many aircraft, household and automotive applications. Improper design of these structures may result in undesired noise, vibration and harshness (NVH) characteristics. In order to improve vibration characteristics, viscoelastic materials (VEM) are used as damping treatments to lower the vibration amplitudes. However, the cost and added weight on the design make the companies to develop methodologies to effectively distribute on such structures especially considering the challenging lightweight targets.

There are mainly two kinds of damping treatments: i) unconstrained damping patch bonded to the plate directly, where the energy dissipation occurs due to longitudinal deformations, ii) constrained damping, where the shear deformation is the main mechanism for the dissipation [1].

Kerwin [2] started the studies on constrained layer damping (CLD), he developed a methodology to determine the loss factor of a plate with a CLD treatment. In the last forty years, constrained and unconstrained layer damping treatments have been studied extensively. Considering the lower cost compared to active applications such as piezoelectric patches, passive application is preferred. In the determination of the effectiveness of damping treatments, finite element modeling has been effectively used. In one of the pioneering studies, Lu et al. [3] developed FEM for sandwich plate problems. Furthermore, Johnson and Kienholz [4] evaluated modal loss factors by using of strain energy for CLD treatments.
A vast body of research is available in the literature on CLD treatment applications and their optimization. An optimal model has been investigated by Lall et al. [5] that minimizing modal loss factor and displacement frequency response function in a plate with full CLD covered. Rikards and Chate [6] operated an optimization study for sandwich plates arrangement with soft core and composite layered face, that are figured out cases along with a sandwich plate FE model by the planning of experiments [7]. Optimal design is cared for damping properties of the bonded viscoelastic material and exterior layers of the plate. The complex eigenvalues approach [8] or energy method measured modal loss factors in studies of the damping properties of sandwich-type structures. In order to optimize the CLD damping on the beam, a genetic algorithm is used by Marcelin et al. [9]. This method is also used for determining optimum layout of CLD patches by Zheng et al. [10,11]. Zheng et al. [12,13] found the CLD pad distribution on plates by topology optimization for maximum modal loss factor [12] and minimal sound radiation [13]. Recently, multi objective optimization has been used for the simultaneous material layout and geometric parameter optimization of viscoelastic damping structures [14].

In this study, detailed FE model of a plate treated with constrained and unconstrained layer damping treatments were developed. The objective of this research is to establish a methodology to place the damping pads (both passive unconstrained and constrained type of damping pads) in a structure and optimize it with respect to performance and weight using formal optimization methods such as topology optimization. The objective function is to minimize the acceleration of the plate up to 500 Hz. The paper is organized as follows. In Section 2, finite element modeling and optimization methodology are described in detail. Topology and thickness optimization results are presented in Section 3. Finally, conclusions are given in Section 4.

2. Methodology

2.1 Modelling of unconstrained layer damping (UCLD)
In this type of the damping treatment, the core material is constrained within a structure element exposed to vibration, commonly metallic elements. Hysteretic characteristics in stress-strain relation is known in the literature [15]. Loss factor is a measure used in the literature to quantify the dissipative characteristics of damping treatments.

In this study, we have considered a rectangular thin plate fully covered by thin layer of VEM. The damping treatment is directly connected to the base layer at each node. The homogeneity assumption for the physical characteristics of both the plate and VEM are considered. The plate’s boundary conditions are taken as free-free condition as in Plattenburg et. al [16]. The FEM of the sheet metal structure is created using 2D elements. The average mesh size of the model is 10 mm. There are approximately 1008 shell elements. In the FEM model as shown in Fig. 1.

2.2 Modelling of passive constrained layer damping (PCLD)
In the UCLD configuration adding a constraining layer over the free surface of viscoelastic layer made up the PCLD design. In consideration of the PCLD, the VEM core layer is sandwiched between two layers, hence the damping in bending mode of vibration becomes apparent typically in view the transverse shear deformation/strain of the constrained viscoelastic layer as a means of dissipating vibration energy [17].

Rectangular plate configuration as a sandwich structure is considered. The viscoelastic layer is bonded between two elastic layers. The finite element model consists of 1512 2D elements as shown in Fig. 2.
2.3 Boundary Conditions and Materials

The dimension of the aluminum plate is taken as $L_x=277$ mm, $L_y=174$ mm. The free-free modal analysis is performed as this is a common testing of the plates to characterize the modal characteristics. For that purpose, commercial finite element method (FEM) code ANSA is used as preprocessor and MSC. Nastran is employed as the solver. The excitation is at $\bar{x}_d = \bar{y}_d = 0.45$ and acceleration measurement location is $\bar{x}_0 = \bar{y}_0 = 0.025$, over-bars denote normalization by plate dimension (i.e. $\bar{x}_0 = x_0/L_x$). The boundary conditions and excitation are shown in Fig. 3.

![Figure 3: The excitation input force and measurement location](image)

Viscoelastic adhesive core in constrained-layer between steel constraining layer as passive constraining treatments and aluminum base plate. The material properties obtained from experiments and used in the following simulations are listed in Table 1 [16].

<table>
<thead>
<tr>
<th>Layer</th>
<th>Material</th>
<th>Young’s Mod.</th>
<th>Density (kg/m$^3$)</th>
<th>Poisson’s ratio</th>
<th>Thickness (mm)</th>
<th>Loss factor</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Steel</td>
<td>203 GPa</td>
<td>7600</td>
<td>0.30</td>
<td>0.38</td>
<td>0.005</td>
</tr>
<tr>
<td>2</td>
<td>Adhesive</td>
<td>6.2 MPa</td>
<td>73</td>
<td>0.40</td>
<td>0.94</td>
<td>1.25</td>
</tr>
<tr>
<td>3</td>
<td>Aluminum</td>
<td>31 GPa</td>
<td>2606</td>
<td>0.33</td>
<td>3.2</td>
<td>0.0013</td>
</tr>
</tbody>
</table>

2.4 Topology Optimization

Topology optimization, allows the introduction of holes or cavities in structures which usually results in great savings in weight or improvement of structural behavior such as stiffness, strength or dynamic response rather than shape optimization. Topology optimization has gotten resuscitated enthusiasm since the presentation of the “homogenization approach to topology optimization” by Bendsøe and Kikuchi [18] however, Michell [19] had pioneering works on minimizing the weight of the structures.

An optimal subset was sought in mathematical terms $\Omega_{\text{mat}} \subset \Omega$. That $\Omega$ is an accessible design area. The density vector $\rho$ that includes elemental densities $\rho_e$ is used as design variable $x$. The density vector can be used for forming local stiffness tensor $E$ as an integer formulation.

$$ E(\rho) = \rho E^0 $$

$$ \rho_e = \begin{cases} 1 & \text{if } e \in \Omega_{\text{mat}} \\ 0 & \text{if } e \in \Omega_{\text{mat}} \setminus \Omega \end{cases} $$
a volume constraint

\[ \int_{\Omega} \rho d\Omega = V ol(\Omega_{mat}) \leq V \]  

(2)

Where \( E^0 \) is represents the material properties of a given isotropic material and \( V \) is the volume of the initial design domain. When \( \rho_e = 0 \) element was considered that element is empty since with \( \rho_e = 1 \) represent a filled element. In order to accommodate for non-discrete values [20] SIMP (Solid Isotropic Material with Penalization) method is commonly used. The density compound as:

\[ E = \rho^p E^0, \quad \rho \in [\rho_{min}, 1], \quad p > 1 \]  

(3)

where \( p \) is the penalty factor with respect to that, mediocre elements will penalize get closer 0 or 1. In this study, the area under the acceleration of each grid point is a measure of vibration. Minimization of this is chosen as the objective of the optimization. Optimization work flow is shown in Fig. 4.

Figure 4: Optimization flowchart

3. OPTIMIZATION

3.1 Topology Optimization Results (PCLD)

The results of the frequency response analysis for the base design and designs which include the optimization for removing 30% and 50% of treatment and full coverage are shown in Fig. 5. and Fig. 6., respectively. The results indicate that there is still significant improvement with all optimization packages compared to the base design. (Area under acceleration response is % improvement).

Figure 5: Optimal Distribution of the PCLD (remove 30%) a. Midlayer b. Upper layer c. FRF results
Table 2: Response improvement of 30% removed

<table>
<thead>
<tr>
<th>Area (Hz.m/s²)</th>
<th>Improvement</th>
<th>Weight (kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Base structure</td>
<td>117.48</td>
<td>N/A</td>
</tr>
<tr>
<td>Full Damping</td>
<td>6.81</td>
<td>94.2%</td>
</tr>
<tr>
<td>Optimum damping</td>
<td>4.92</td>
<td>95.8%</td>
</tr>
</tbody>
</table>

Figure 6: Optimal Distribution of PCLD by removing 50% damping. a. Midlayer b. Upper layer c. FRF results

Table 3: Response improvement of 50% remove (Area under acceleration response is % improvement)

<table>
<thead>
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<th>Area (Hz.m/s²)</th>
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</tr>
<tr>
<td>Optimum damping</td>
<td>4.93</td>
<td>95.80%</td>
</tr>
</tbody>
</table>

3.2 Topology Optimization Results (UCLD)

The optimized design of UCLD corresponding to 70% of the full damping material is shown in Fig. 7. The FRF response for this configuration is shown in Fig. 8. The area under the FRF plot up to 500 Hz is slightly higher than the full damping case. The weight is reduced by 0.01 kg.

Figure 7: Optimization of UCLD (removed 30%)  
Figure 8: FRF results (remove 30%)
<table>
<thead>
<tr>
<th>Area (Hz.m/s²)</th>
<th>Improvement (%)</th>
<th>Weight (kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Base structure</td>
<td>117.48</td>
<td>N/A</td>
</tr>
<tr>
<td>Full Damping</td>
<td>4.71</td>
<td>95.9%</td>
</tr>
<tr>
<td>Optimum damping</td>
<td>5.06</td>
<td>95.7%</td>
</tr>
</tbody>
</table>

Similarly, the optimized design of UCLD corresponding to 50% of the full damping material is shown in Fig. 9. The FRF response for this configuration is shown in Fig. 10. Comparable performance is achieved with this configuration while the weight is reduced by 0.02 kg.

![Image of optimization](image1)

![Image of FRF results](image2)

**Table 5: Comparison of base, full damping and optimum damping by removing 50% of damping**

<table>
<thead>
<tr>
<th>Area (Hz.m/s²)</th>
<th>Improvement (%)</th>
<th>Weight (kg)</th>
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</thead>
<tbody>
<tr>
<td>Base structure</td>
<td>117.48</td>
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<tr>
<td>Full Damping</td>
<td>4.71</td>
<td>95.9%</td>
</tr>
<tr>
<td>Optimum damping</td>
<td>5.11</td>
<td>95.6%</td>
</tr>
</tbody>
</table>

### 3.3 Thickness optimization results

The goal of thickness optimization is to determine the optimum thickness of each layer while reducing the overall weight of the structure. The thickness for PCLD and UCLD is optimized with the same objective function using Nastran SOL 200 solver. The results show that the thickness of the upper layer converged to 0.8 mm from the original thickness of 0.94 mm, and the optimum thickness of the mid-layer is 0.32 mm (originally 0.38 mm). Similarly, the optimum thickness of the UCLD layer is found to be 0.78 mm. The optimization solver converged in 8 iterations for both cases. The area under the FRF curves is comparable with optimized thickness for PCLD and UCLD. With optimized thickness for each layer, 0.1 kg weight reduction is achieved for PCLD compared to 0.15 kg for UCLD.
4. CONCLUSIONS

A design and optimization methodology for the determination of the damping pad location on sheet metal based on the minimization of frequency response functions of the panel was proposed. In order to determine the optimum design and locations of the damping pads, the PCLD and UCLD damping treatment techniques are used. As part of the case study, 30% and 50% of the full damping layers were removed from the original models. The frequency response functions are compared with the bare panel. Furthermore, the thickness of each layer of UCLD and PCLD damping treatments is optimized as a weight reduction opportunity. These results first suggest that damping pad design plays an important role in minimizing the vibration characteristics of the sheet panels. The simulation results suggest that the determination of the optimal layout of damping treatments in terms of functional performance, weight and cost could be achieved by topology optimization. The experimental validation of simulation results is acknowledged as future work.

REFERENCES


