Passive constrained layer damping patches have been extensively used for vibration and acoustics applications due to their capability to dissipate vibrational energy into heat. In this study, we perform an optimization study of damping pads on a real 3D structure. For this purpose, finite element models (FEM) of the sheet metal structure and damping pads are created. Correlation of the finite element model with experimental data is performed both on modal analysis and frequency response functions. In order to determine the optimum locations of the damping pads, the body panel is divided into nine regions. Damping pad is applied on each region one by one and the frequency response functions are compared with the panel without any damping pad. Root Mean Square (RMS) based damping index is calculated between these configurations and bare panel, and each region is ranked from highest to lowest based on the damping index. Two optimum configurations are evaluated and resulting frequency response functions are also compared with the full damping pad. Finally, the thickness of the damping pad is varied to see its effect on the frequency response functions.

Keywords: damping, damping loss factor, modal analysis, frequency response, optimization

1. Introduction

Increasing pressure on reducing the cost and weight of the products force companies to well optimize their engineering solutions for noise, vibration and harshness problems. The most common countermeasures applied in the industry to improve vibration performance are: i) making the structures stiffer by adding reinforcements, ii) adding damping material on critical parts of structures, iii) insulating the structures with insulating treatments such as foams and felt [1]. All or combinations of these solutions exist in many engineering applications.

Passive constrained layer damping patches have been employed in many acoustics and vibration applications [2]. The primary function of damping treatment on a vibrating panel is to reduce vibration levels or radiated sound power. Such patches use a viscoelastic core and dissipate vibrational energy by deforming in compression and tension under surface vibration in bending [3].

While companies have been targeting sheet metal panels as one of their prime methods to reduce weight via reducing their thickness, applying holes, or using lighter materials, to achieve their lightweight goals, these actions are usually in conflict with vibration requirements. Therefore, they have
been investing on the measurement of the damping characteristics of the damping pads and are looking ways to increase the damping properties. Various methods are available in the literature to measure the damping characteristics of the damping pads most commonly used one being the Oberst method recommended by ASME [4] and SAE [5]. Even though, the measurement of the damping characteristics is straightforward, the determination of where to apply these pads on engineering structures is more difficult. Mostly, companies use their experience from previous products to “cary over” the damping pads or use excessive amount of damping pads to make sure that they face with no major vibration or acoustics problems. However, this sometimes results in over-engineered designs with heavier and costly products.

Dynamic analysis of plates has been studied in detail [6-7]. In addition, full damping coverage of simple plates has been studied previously [8]. Lall et al. found out the damping patch size and location using Rayleigh Ritz method for the determination of partial coverage of a plate [9]. Kung and Singh [10] developed an energy based methodology to calculate the vibration performance of a plate with multiple viscoelastic patches. There have been also studies on the analytical model development for active and passive damping patches. Ritz-Galerkin method has been used by Lam et al. [11] to determine the time response of a structure with active and passive patches. More recent research studies mostly focused on finite element models [12] and control algorithms [13]. In order to improve vibration performance, piezoelectric patch and passive viscoelastic applications have been considered in conjunction [14-15]. Active and passive vibration methods have been integrated and applied to thin rectangular plates by Plattenburg et. al [16]. Analytical models for passive patch, active patch and active-passive patch interaction have been verified with experiments in this study. Even though there has been a vast body of literature on the modeling of passive and active patches and control, studies on optimization of passive patches are quite limited. The goal of this paper is to fill this gap by first validating the simulation models with experimental data and proposing a method for the determination of the location of damping pads.

For this purpose, finite element models (FEM) of the sheet metal structure without and with damping pads are correlated to experimental data from modal testing. A method to determine the application of damping pads is proposed. Organization of this paper is as follows. In section 2, details of the modeling and correlation of a body panel with and without the damping pads are presented on both modal response and frequency response functions. Sensitivity and optimization of the damping pads are explained in Section 3. Finally, section 4 is reserved for the conclusions.

2. Modal and Frequency Response Correlation of a Panel

The FEM of the sheet metal structure is created using 2D elements. The average mesh size of the model is 3 mm. There are approximately 40000 shell elements, 5% of which are tria elements. The sheet metal structure and corresponding FEM model are shown in Fig. 1 and Fig. 2., respectively.

![Sheet metal structure](image1.png)

![Finite element model](image2.png)

Figure 1: Sheet metal structure.
Figure 2: Finite element model.

2.1 Modal Analysis and Testing Correlation

Comparison between simulation and modal testing is made for the first nine modes (upto 150 Hz in modal testing) along with the relative error in Table 2. Mode shapes for the second mode from simulation and modal testing are shown in Fig. 3 and Fig. 4, respectively. The modes at 84.5 Hz and
89.1 Hz were not seen in modal testing due to limited measurement of the acceleration of the structure. The relative error on the rest of the modes ranged from 0.6% to 11.7%.

Table 1: Comparison between simulation and test results on structure modes

<table>
<thead>
<tr>
<th>Mode No</th>
<th>Simulation (Hz.)</th>
<th>Test (Hz.)</th>
<th>Relative Error (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>20.1</td>
<td>18</td>
<td>11.7</td>
</tr>
<tr>
<td>2</td>
<td>67.4</td>
<td>67</td>
<td>0.6</td>
</tr>
<tr>
<td>3</td>
<td>84.5</td>
<td>Not seen</td>
<td>NA</td>
</tr>
<tr>
<td>4</td>
<td>89.1</td>
<td>Not seen</td>
<td>NA</td>
</tr>
<tr>
<td>5</td>
<td>93.2</td>
<td>94</td>
<td>0.9</td>
</tr>
<tr>
<td>6</td>
<td>110.5</td>
<td>107</td>
<td>3.3</td>
</tr>
<tr>
<td>7</td>
<td>128.1</td>
<td>119</td>
<td>7.6</td>
</tr>
<tr>
<td>8</td>
<td>142.3</td>
<td>133</td>
<td>7.0</td>
</tr>
<tr>
<td>9</td>
<td>154.8</td>
<td>149</td>
<td>3.9</td>
</tr>
</tbody>
</table>

2.2 Correlation on Frequency Response Analysis

The accelerances from the simulation and modal testing (repeated for 3 times to see the variation) are compared for eight points, marked as point 100 to 107, with the coordinates shown in Fig. 5.

The results of the comparison are plotted in Fig. 7 and Fig. 8 for points 106 and 107, respectively. The correlation between the simulation and experiment is good except for two frequencies (at 55 Hz and 117 Hz), which are seen in the experiments but not in the simulations.
Effect of mass loading and the location of the excitation on the frequency response is investigated to see their effect on the correlation. For that purpose, the mass of the sensor, which is 30 gr, is added to the finite element model. The results for the mass loading is shown in Fig. 9. It can be shown that the mass of the sensor affects the acceleration results at the peak around 55 Hz. In order to compare effect of excitation location, four more locations within 20 mm of the original excitation location are excited in the finite element model. The results are plotted in Fig. 10. The excitation location affects the results in the frequency range of 25-30 Hz region.

![Figure 9: Effect of mass loading on FRF](image1)

![Figure 10: Effect of location on FRF](image2)

2.3 Modeling and Correlation of Damping Pad

The damping pad application on the sheet metal and its FEM model are shown in Fig. 11 and Fig. 12, respectively. The damping pad is modeled as shell elements and connected node to node to the sheet metal. Material properties of the damping pad in FEM model are given in Table 2.

![Figure 11: Damping pad on sheet panel.](image3)

![Figure 12: FEM of the damping pad](image4)

<table>
<thead>
<tr>
<th>Density</th>
<th>Young’s Modulus</th>
<th>Poison’s Ratio</th>
<th>Damping Loss Factor</th>
</tr>
</thead>
<tbody>
<tr>
<td>2 g/cm³</td>
<td>4000 MPa</td>
<td>0.3</td>
<td>0.3</td>
</tr>
</tbody>
</table>

Simulation results on FRF for point 104 and point 105 are plotted in Fig. 13 and Fig. 14, respectively. The simulation results show the effectiveness of the damping pads.

![Figure 13: FRF results for point 104.](image5)

![Figure 14: FRF results for point 105.](image6)
Sensitivity Studies and Optimization

2.4 Sensitivity Study

The sensitivity of damping pad application of each region is calculated by dividing the panel to nine regions as shown in Fig. 15. The frequency response is calculated for each excitation point with the damping pad applied to each region separately and compared to the panel with the base design for the frequency range of 0-150 Hz.

![Figure 15: Effect of mass loading on the results.](image)

Frequency response function between each input (excitation) and output points has been calculated for the base model and the model with each damping pad on each region separately. The effectiveness of the damping pad in each region is determined according to the damping index given in Eq. (2). Frequency range up to 150 Hz is chosen to determine the effectiveness of the damping pad for structureborne performance. Bigger index values indicate the effectiveness of the application of the damping pad for that region.

\[
\text{Damping Index} = \sqrt{\frac{\sum_{\omega=1}^{150 \text{ Hz}} (acc_b(\omega) - acc_d(\omega))^2}{150 \text{ Hz} - 1 \text{ Hz}}}
\]  

(2)

where \( acc_b(\omega) \) is the acceleration of the bare panel and \( acc_d(\omega) \) is the acceleration of the sheet panel with damping pad.

Damping index is calculated between each excitation point and the other eight points. The damping index is sorted from the maximum to the minimum and the matrix showing the importance of each region between each input and output point is shown in Fig. 16.

![Figure 16: Sensitivity of each region](image)

The results from Fig. 16 show that the application of damping pad on region 4 is quite effective between many input and output points. Similarly, region 9 and region 1 are more effective than the
application of damping pad on other regions. This conclusion could be drawn by counting the number of occurrences in the first three rows (number of occurrences for region 4, region 9 and region 1 are 39, 38 and 27, respectively). Therefore, two optimization packages are generated: i) Application of damping pad on region 4 and region 9 (shown in Fig. 17), ii) Application of damping pad on region 4, region 9 and region 1 (shown in Fig. 18).

![Figure 17: Optimization package 1.](image)

![Figure 18: Optimization package 2.](image)

### 2.5 Optimization of the Damping Pad

The results of the frequency response analysis for the base design and designs which include the optimization packages 1, 2 and full treatment are shown in Fig. 19 (excitation at point 105, acceleration outputs for all 8 points). The results indicate that there is still significant improvement with both optimization packages 1 and 2 compared to the base design.

![Figure 19: Optimization results for excitation at pt. 105.](image)

### 2.6 Effect of Thickness of the Damping Pad

The thickness of the damping pad is varied to see its effect on the frequency response. For this purpose, the thickness of 1 mm and 2 mm are evaluated. The results of the thickness variation study are shown in Fig. 20 and Fig. 21 for optimization package 1 and 2, respectively (excitation at point 105, acceleration for 8 points) compared to the optimized design with a thickness of 3 mm. The results show that the optimization package 1 and 2 have similar accelerance performance when the thickness of the damping pad is 2 mm. However, accelerance results increase when the thickness is reduced to 1 mm. Final decision of the thickness of the damping pads should be based on system level verification.
3. 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. Finite element models of body panels and damping pads are correlated to the modal testing results on body panel modes and frequency response functions. In order to determine the optimum locations of the damping pads, the body panel is divided into 9 regions. Damping pad is applied on each region one by one and the frequency response functions are compared with the panel without any damping pad. Root Mean Square (RMS) is calculated between these configurations and bare panel, and each region is ranked from highest to lowest based on the RMS. These results first suggest that damping pad design plays an important role in minimizing the vibration characteristics of the sheet panels. Using simulation models and experimental techniques could lead to optimum design in terms of functional performance, weight and cost.

Two optimum design proposals are developed using the proposed methodology. Verification of these design proposals is acknowledged as future work. The effect of the damping pad up to 150 Hz
is investigated for mostly structural borne vibrations. However, it is likely that optimization of the damping pad in terms of consideration of higher frequencies could result in different damping pad sensitivities and therefore different optimum design. This is considered as a future work. Body panel could be divided into more regions in order to increase the resolution of the application of the damping pad. Optimization using formal topology optimization methods such as size and shape optimization [17], and topology optimization [18] is acknowledged as future work.

REFERENCES