OPTIMIZATION OF SURFACE DAMPING TREATMENTS FOR VIBRATION CONTROL OF MARINE STRUCTURES

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ABSTRACT

Uncontrolled vibration develops into a serious of problems in machinery and structures and damping is the simplest method of limiting the amplitudes. Active damping involves complicated electronic gadgets and is yet to gain popularity. Structural damping is usually very small in metals but can be greatly enhanced by either adding a layer of viscoelastic (VE) material to form what is called as Free Layer Damping (FLD) or sandwiching the viscoelastic layer in between two or more metal layers, an arrangement known as Constrained Layer Damping (CLD). Different beam samples were made comprising of Mild Steel beams as well as CLD and FLD beams varying material thickness. Their response to sinusoidal excitation was measured at different frequencies and the results plotted. These samples were also modeled using ANSYS and analyzed as per Ross, Kerwin and Ungar (RKU) method as suggested by Macioce [2]. The loss factor was calculated for CLD as a function of layer thickness by the Oberst and Schommer [3] approach with the help of RKU equations. From these equations optimum loss factor was determined. It was found that the results as projected by the Finite Element Analysis validate the experimental results. A CLD thickness of 1.5mm increases the loss factor from 0.107 for the base MS beam to a value of 0.123, an increment by 16% for a CLD beam. A marine structure comprising two VE layers of 1mm thickness each sandwiched between two MS layers of thickness 1.5mm and a base plate of 12mm was also taken up as a case study and the results are plotted. In conclusion both CLD and FLD improve damping characteristics significantly and between these two, CLD performs better. Hence these techniques open up an economical and simple method of controlling undesirable vibrations.

1. INTRODUCTION

Vibration of structures, automobiles, aircraft, instruments etc, can have varying effects: minor irritation, discomfort, severe injury and even major breakdown. Detailed studies, too numerous to be discussed here, have been carried out on this aspect over the last century, the most noteworthy being the collapse of the Tacoma Narrows bridge in US due to wind-induced vibrations in 1940[1]. Damping in whatever form, in general tends to control the vibration amplitude due to its ability to convert the mechanical energy into heat. A viscous damper supplies a resisting force which is directly proportional to the velocity. A Coulomb damper on the other hand offers a constant force (independent of the velocity of the body) and is present whenever the two contacting bodies have dry friction. Structural or hysteresis damping is due to friction between the internal planes which slip or slide as the deformation takes place beyond the elastic limit.

Of the above three passive forms, the third one opens up unlimited opportunity for the researcher since different material combinations can be tried out. Since the materials used have both viscosity and elasticity they are appropriately referred to as viscoelastic materials and when they are used for vibration control they are subjected to shear strain or direct strain. A viscoelastic material is one that has considerable energy dissipation capabilities in addition to strain energy storage capability. High strength structural materials have little inherent damping whereas viscoelastic materials, typically elastomers and plastics, have lesser stiffness and greater damping. Combinations of structure and viscoelastic materials can take advantage of the strength of one and the damping of the other. In the simplest arrangement a layer of viscoelastic material is attached to an elastic one and is usually called as FLD. In another arrangement a viscoelastic layer is sandwiched between the elastic layers. This arrangement is known as CLD. Though a scientific application of this is of very recent origin it was used empirically by a violin manufacturer in Venice three hundred years ago and the damping was sufficient to create a rich full tone that made his violins famous [12].

In 1957 Plass studied free-layer damping of circular bars, and also proposed and discussed briefly the related problem of a sandwich plate with a viscoelastic core. He assumed the face sheets to be very thin, and as a result, he accounted for their stiffness only by an equivalent moment of inertia of the sandwich structure as opposed to modeling them
explicitly. Kerwin in 1959 was the first to present a general analysis of viscoelastic material (VEM) constrained by another metal layer, where the loss mechanism is primarily shear in the VEM. His models predicted attenuation of a traveling wave on either a simply supported or infinitely long beam by assigning the beam a complex composite stiffness. An important point is that the target structures were thin-skin beams or plates and the damping treatments consisted of a viscoelastic material.


2. Description of the Equipment

PC based data acquisition and analysis system of type 3560, is used for noise and vibration analysis. It provides the platform for a range of PC based measurement solution. A pulse, system consists of a PC, pulse software, Windows NT, Microsoft office, multiprocessor DSP board(s), an interface, and data acquisition front and hardware. An FFT analyzer generates a FRF takes the input signal and the output signal and produces the FRF spectra. The vibration transducer is attached to the machine to convert the mechanical vibration into an electrical signal. An accelerometer of type-4384 having the acceleration range 20 m/s$^2$ to 1000 km/s$^2$ and frequency ranges 0 Hz to 60 kHz is used for measuring and analyzing machinery vibration. Force transducer of type-8200 in the force range 1000N tensile to 5000N compressive is used for Measurement of Frequency Response Functions. Vibration exciter of type-4809 is used.

The beam is clamped to simulate a fixed-end boundary condition. The accelerometer is positioned on the beam. Sinusoidal excitation is provided by the exciter. The transducers are positioned approximately 1mm away from the beam. The response of the beam is measured by the accelerometer and is processed through signal conditioning equipment, and the resulting response is stored in a computer for later analysis. From the response spectrum the frequencies and damping values of the various modes of vibration of the beam are measured. The experimental set up is shown in Fig.1

3. Experimentation

The frequency responses of MS, FLD and CLD beams were measured for different thicknesses of viscoelastic materials and the analysis is done by Ross, Kerwin and Ungar method. This analysis has been developed for a three layer system and is used to handle both experimental and shear types of damping. As per this analysis, flexural rigidity of a three layer system is given by

$$EI = \begin{cases} 
E_1 \frac{H_1}{12} + E_2 \frac{H_2}{12} + E_3 \frac{H_3}{12} - E_2 \frac{H_2}{12} \left(\frac{H_{31} - D}{1 + g}\right) + E_4 H_4 D^2 + E_2 H_2 \left(H_{21} - D\right)^2 \\
+ E_4 H_4 \left(H_{31} - D\right)^2 - \frac{E_2 H_2}{2} \left(H_{21} - D\right) + E_3 H_3 \left(H_{31} - D\right)
\end{cases}$$

$$\frac{H_{31} - D}{1 + g}$$

where

$$D = \frac{E_2 H_2 \left(H_{21} - H_{31} / 2\right) + g \left(E_2 H_2 H_{21} + E_3 H_3 H_{31}\right)}{E_1 H_1 + E_2 H_2 / 2 + g \left(E_1 H_1 + E_2 H_2 + E_3 H_3\right)}$$

$$H_{31} = \frac{\left(H_1 + H_3\right)}{2} + H_2 \quad H_{21} = \frac{\left(H_1 + H_2\right)}{2} \quad g = \frac{G_2}{E_3 H_3 H_2 p^2}$$
Where ‘$E$’ is the Young’s modulus, ‘$G$’ is the Shear modulus, ‘$I$’ is the second moment of area, ‘$H$’ is the thickness, ‘$p$’ wave number. Subscript 1 refers to the base structure, subscript 2 to the damping layer, and 3 to the constraining layer. No subscript refers to the composite system. ‘$D$’ is the distance from the neutral axis of the three layer system to that of the original beam.

To incorporate the damping terms into RKU equations above, it is only necessary to replace each modulus term with a complex modulus, as given below.

$g = g(1+i\eta_2)$, $E_2 = E_2(1+i\eta_2)$, $E_3 = E_3(1+i\eta_3)$, $E_1 = E_1(1+i\eta_1)$, and $E = E(1+i\eta)$

4. ANALYTICAL PROCEDURE BY USING FEA

An Oberst beam is used in the finite element analysis. Oberst beam is a cantilever beam (base beam together with one or two layers of other materials). As the base beam is made of a rigid and lightly damped material (steel, aluminum), the most critical aspect of this method is to properly excite the beam without adding weight or damping. The measurement of the vibration response of the beam is usually made using an accelerometer. As per the ASTM E75 standard the dimensions of the test beam are taken as

- Length = 250mm, Width = 12mm, Thickness = 3mm
- Thickness of damping material is 1mm

The beam used for Finite Element Analysis is shown in Fig.2. RKU analysis has been carried out to find the optimum constrained layer thickness for 3mm base plates. Constrained layer thickness is varied from 0.5 to 2.5mm and the corresponding loss factors are found out.

5. FEM ANALYSIS

As the experimental procedure is laborious an attempt is made initially to design the cantilever beam with and without damping to reduce the excessive vibrations at resonance conditions by using FEM. The MS, FLD and CLD beams has been modeled in ANSYS. Modal analysis has been carried out to find out the natural frequencies of the beams to validate with experimental natural frequencies by taking two elements of shell 63 and shell 99. After the results obtained from the FEM analysis for MS, FLD, and CLD beams are plotted in Figs.3, 4, 5, 6, 7 respectively

6. RESULTS AND DISCUSSION

In this paper, the structure for the present study is a foundation used to support the marine structure. This foundation is subjected to vibration, and this paper aims to reduce the effects of vibrations by adding damping. Vibration reduction is carried out by surface damping treatment using CLD and FLD configurations. RKU analysis has been used to predict the damping of the structure for CLD.

As the beam is excited over a frequency range and at a certain frequency, resonance condition is encountered. That frequency can be treated as natural frequency of the cantilever beam, because large amplitude is observed when the excitation frequency and the natural frequency of the beam are coinciding with each other. Damping ratio is calculated using the half power bandwidth method. The response spectrums of the MS, FLD and CLD beams are shown in Figs 3,4,5,6 and 7 and it is observed that treatment of MS beam with CLD and FLD had reduced the vibration response levels i.e. 10 to 20% reduction in amplitude.

For the beam structure, CLD is effective than FLD at frequencies up to 4KHz, which falls in the range of application of most machinery of marine application, since major contribution of machinery and auxiliaries is dominant in the 10Hz to 4KHz band. For most machinery involving marine application, the excitation and operating frequencies fall in the 10Hz – 4 KHz band. The damping treatment for vibration suppression is therefore required to perform well in this range. The effect of damping treatment on frequencies beyond this range is neglected as they are not relevant for the application.

From the Figs 8 and 9, it can be observed that treatment of MS beam with CLD and FLD has reduced the vibration response levels. However, it is seen that CLD is giving better performance as compared to FLD. CLD has a limitation in its degraded performance on vertical surfaces, and FLD is found to be useful for vertical surfaces. When weight is a constrained, FLD can be useful.

The experimental results of natural frequencies are compared with the analytical values are shown in tables 1,2,3,4 and 5 and found that the variations in the results are up to 10%. Thus the results obtained are comparable and validating the experiment. Since FEM is an approximate mathematical technique, and since the FE approach assumes homogeneity in material or boundary conditions, which vary from those in real-life structures, the results are subject to deviation from experimentally measured values. However further detailed analysis, which is beyond the scope of this paper, would bring down the deviation between experimental and FE values.
Fig 10 shows the variation of loss factor with constraining layer thickness. The loss factor value increases up to constraining layer thickness of 1.5mm and then the value decreases. Thus the highest loss factor which is achieved with 1.5mm constraining layer thickness is taken as the optimized thickness. Thus the highest loss factor which is achieved with 1.5mm constrained layer thickness. If the weight penalty is constrained, the CLD treatment can be applied partially over the base layer, in this case the damping can be introduced at some modes. If many modes are to be damped, it is almost certain that full coverage treatment will be needed. In this case full coverage treatment is modeled based on finite element analysis to optimize damping treatments [5]. The developed method uses modal strain-energy information of bare structural panels to identify flexible regions, which in turn facilitates optimization of damping treatments with respect to location and size.

7. CASE STUDY

CLD treatment was applied on the base frame of the marine equipment Fig 11 and a flat plate below S&V mounts connecting to foundation. The viscoelastic layer is joined to the plate of the vibrating surface by means of an adhesive, and this layer is further constrained by a metallic layer. The results indicate an appreciable reduction of 8-10dB, underlining the utility of CLD as a feasible practical solution for vibration transmission control. In the frequency range of 1000-1500Hz the reduction in amplitude after CLD and in the frequency range of 10-80Hz there is an increase in amplitude. For most machinery involving marine application, the excitation and operating frequencies raise in the 10Hz – 100Hz band. This is due to the amount of modal energy (due to that mode’s contribution) being absorbed by application of damping material. So the effect of increase in amplitude in this range is neglected as they are not relevant for the application.

8. CONCLUSIONS

The following conclusions are made:

- Constrained layer damping treatment is effective over a wide frequency range.
- RKU analysis is best available method for predicting the response when damping is present. RKU analysis methodology has been validated for a number of studies on beams and plates of varying configuration and an empirical relationship obtained. This analysis methodology is a measure to efficient compensation and provides quick and valid results for damping treatments. This analysis is widely used for design of damping treatment. Fully covered CLD treatment gives reduction in vibrations in wide frequency range, but where weight penalty is a constraint, partial coverage can be resorted to.
- Application to a marine structure indicated good reduction in transmitted vibration, underlining the utility of CLD.
Fig. 3: Response spectrum of mild steel beam

Fig. 4: Response spectrum of FLD beam (material 1)

Fig. 5: Response spectrum of FLD beam (material 2)

Fig. 6: Response spectrum of CLD beam (material 1)

Fig. 7: Response spectrum of CLD beam (material 2)

Fig. 8: Comparison of experimental responses of MS, FLD and CLD beam (material 1)

Fig. 9: Comparison of experimental responses of MS, FLD and CLD beam (material 2)

Fig. 10: Loss factor vs. Constraining Layer Thickness
### Table. 1: Comparison of Experimental and Analytical Natural Frequencies of MS beam

<table>
<thead>
<tr>
<th>Experimental Freq (Hz)</th>
<th>Damping Ratio (%)</th>
<th>Analytical Freq (Hz)</th>
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<tbody>
<tr>
<td>43</td>
<td>12.1</td>
<td>39.36</td>
</tr>
<tr>
<td>200</td>
<td>11.91</td>
<td>156.62</td>
</tr>
<tr>
<td>1092</td>
<td>7.07</td>
<td>995.71</td>
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<tr>
<td>4792</td>
<td>2.01</td>
<td>4698</td>
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</table>

### Table. 2: Comparison of Experimental and Analytical Natural Frequencies of FLD beam (material 1)

<table>
<thead>
<tr>
<th>Experimental Freq (Hz)</th>
<th>Damping Ratio (%)</th>
<th>Analytical Freq (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>24</td>
<td>14.05</td>
<td>37.99</td>
</tr>
<tr>
<td>996</td>
<td>20.55</td>
<td>921.68</td>
</tr>
<tr>
<td>1472</td>
<td>6.94</td>
<td>1392.5</td>
</tr>
<tr>
<td>1996</td>
<td>1.74</td>
<td>2149.4</td>
</tr>
</tbody>
</table>

### Table. 3: Comparison of Experimental and Analytical Natural Frequencies of FLD beam (material 2)

<table>
<thead>
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<th>Experimental Freq (Hz)</th>
<th>Damping Ratio (%)</th>
<th>Analytical Freq (Hz)</th>
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</thead>
<tbody>
<tr>
<td>32</td>
<td>12.56</td>
<td>37.62</td>
</tr>
<tr>
<td>480</td>
<td>19.13</td>
<td>685.98</td>
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<tr>
<td>960</td>
<td>6.2</td>
<td>924.98</td>
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<td>1612</td>
<td>5.69</td>
<td>1383.3</td>
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### Table. 4: Comparison of Experimental and Analytical Natural Frequencies of CLD beam (material 1)

<table>
<thead>
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<th>Experimental Freq (Hz)</th>
<th>Damping Ratio (%)</th>
<th>Analytical Freq (Hz)</th>
</tr>
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<tr>
<td>76</td>
<td>23.9</td>
<td>82.64</td>
</tr>
<tr>
<td>260</td>
<td>26.9</td>
<td>150.95</td>
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<tr>
<td>572</td>
<td>3.41</td>
<td>516.58</td>
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<td>1020</td>
<td>8.28</td>
<td>935.83</td>
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### Table. 5: Comparison of Experimental and Analytical Natural Frequencies of CLD beam (material 2)

<table>
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<th>Experimental Freq (Hz)</th>
<th>Damping Ratio (%)</th>
<th>Analytical Freq (Hz)</th>
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<tbody>
<tr>
<td>79</td>
<td>17.1</td>
<td>82.86</td>
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<td>232</td>
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<td>937.98</td>
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<td>1484</td>
<td>6.24</td>
<td>1444</td>
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![Fig11: Comparison of amplitude before and after CLD](image)
REFERENCES


