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4pSA4.   Lightweight distributed vibration absorbers for marine structures

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  When combined with attached motors and rotating machinery, the lightly damped, thick plating required in maritime applications becomes a broadband noise and vibration control problem. A typical solution is to adhere heavy and dense damping materials for dissipation of the plate vibrational energy. In order to attenuate low frequencies, significant mass must be added to the structure. This paper will review the development of two, passive treatments intended to resolve this issue. HG blankets are constructed using small masses embedded into poroelastic material. Together with the inherent stiffness of the poroelastic material, the masses become embedded mass-spring-dampers and their presence is found to notably increase the low frequency transmission loss of the host material. DVAs are compact vibration absorbers that distribute continuous mass and spring elements over the surface while generating ample reactive damping at low frequencies. This paper will overview the concepts and development of adapting DVAs and HG blankets for use on heavy plate structures, their testing for broadband control performance, as well as their versatility for thinner panels. A comparison with a conventional, marine noise control treatment will be considered.
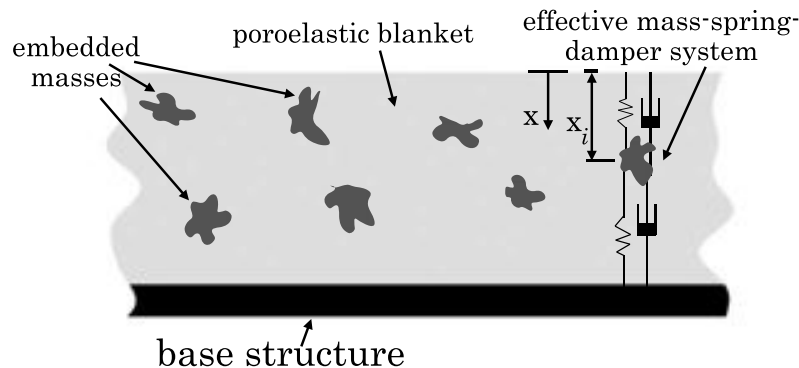
Figure 1: Diagram of poroelastic material with embedded masses, the HG blanket.

1   Introduction

The heavy, isotropic materials used in some maritime vessels can become a source of low frequency noise if any attached machinery vibrates the structural members. A common solution is to outfit the structure with a dense, high-damping material to reduce vibration and, therefore, radiated sound levels. Unfortunately, low frequency attenuation requires a significant increase in mass in order to achieve appreciable attenuation.

Vibration absorbers are long known to be an effective means to decrease the low frequency vibration and noise response of a structure. The classic mass-spring-damper design helps alleviate structural vibration by providing canceling forces in the vicinity of the absorber's tuning frequency and location of attachment. Two such designs have been recently experimented with at Virginia Tech.

The first treatment, known as a heterogeneous [HG] blanket, is composed of a poroelastic material into which a number of masses are embedded. It was shown by Fuller[1] that the masses will interact with the elasticity of the poroelastic material in order to form an array of mass-spring-dampers, or, alternatively tunable vibration absorbers, with a certain range of tuning frequencies, see Figure 1.

Idrisi[2] determined that these tuning frequencies were a function of the foam layer elasticities, the embedded mass itself, the mass depth, the distance between masses and the embedded mass shape. Kidner[3] found that over the tuning frequency range, the embedded masses could increase the insertion loss over the foam material itself by as much as 15 dB. For the structures utilized in this paper, HG blankets typically contribute an additional 5–10% of weight compared to the bare plate (5–10% mass ratio, m.r.).

A recent test illustrated the potential of increasing low frequency sound absorption of standard poroelastic layers[1]. A thin, aluminum panel was mounted in a transmission loss facility at Virginia Tech and poroelastic treatments were then attached to evaluate their performance. Figure 2 shows the panel radiated intensity without treatment (black line), with a 3 inch layer of melamine foam covering the (black dash), with a 2 inch thick sheet
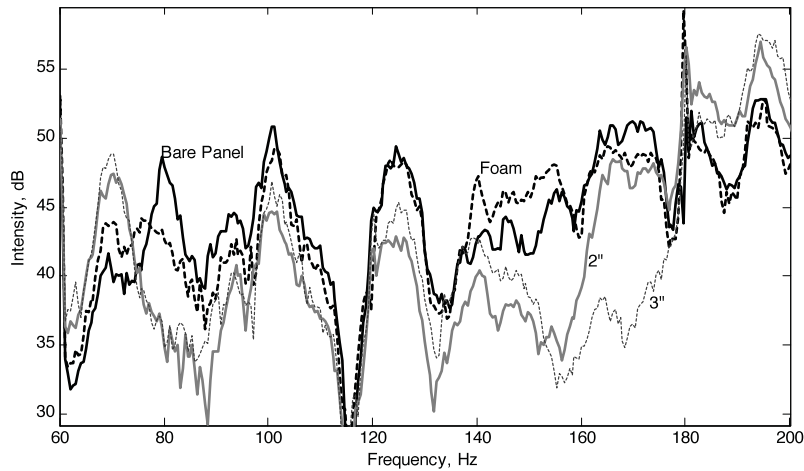
Figure 2: Narrowband sound intensity of thin aluminum panel radiation without treatment (black), with 3 in melamine foam (black dash), with 2 in melamine and embedded masses (gray) and with 3 in melamine and embedded masses (gray dash)[1].

of melamine foam with embedded masses (gray line) and with a 3 inch thick melamine with embedded masses (gray dash).

Figure 2 shows that the added masses noticeably decreased the radiated sound intensity from 60–180 Hz as compared to just the foam layer. Additionally, the masses represented only an additional 6% increase in mass compared to the weight of the aluminum panel and the 2 in. thick HG blanket (gray line) was in fact a lighter treatment than the 3 in. thick melamine foam layer (black dash).

The second treatment, Distributed Vibration Absorbers or DVAs, was originally developed by Fuller and Cambou[4] and uses a design that spreads the contributing mass and spring elements of an absorber over a large area, akin to the evolution from point absorber to distributed absorber in Figure 3. The particular spring design is one in which a material is woven into a sinusoidal shape and constrained on one side by a lightweight base layer and on the other side by the distributed mass element.

Since the mass and spring elements of the vibration absorber are now spread out, the mass per area and spring transverse stiffness per area become the important parameters for tuning frequencies. In addition to the single-degree-of-freedom [SDOF] vibratory motion, the top mass is now capable of vibrating in a variety of frequencies due to its planar dimensions. Coupling effects between the spring layer and top mass, however, are seen to restrain the top mass from vibrating purely like a free plate.

To further increase the global reduction performance, the woven spring layer has here been constructed in a sandwich form with a viscoelastic polymer between the thin woven sheets. With the addition of a viscoelastic sandwich woven layer, DVAs are capable of attenuating mid to high frequencies, well above their SDOF tuning frequencies.

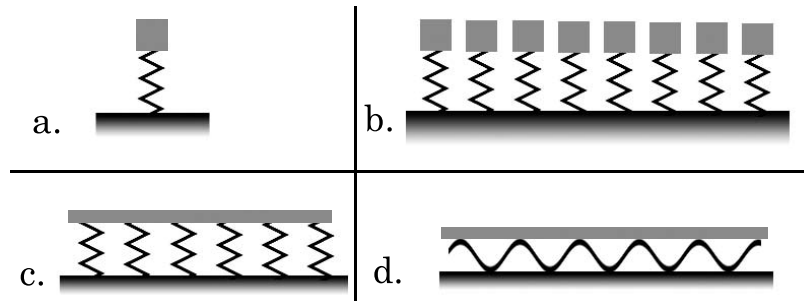The DVA was recently studied Marcotte[5] in depth and has been applied to a variety

Figure 3: Progress from point vibration absorber (a), towards a continuous mass and continuous spring design as achieved in (d).

of thin structures. Most prior work with DVAs focused on using a spring layer composed of a poroelastic material, in a similar manner as with HG blankets[5]. This study, however, retains the original woven design as a means of increasing the absorber damping and therefore increasing the off-resonance vibration absorption capabilities.

Both designs, HG blankets and DVAs, are evaluated analytically and are tested on a large, heavy vibrating plate typical of maritime structures.

2 HG Blankets

2.1 Development

It was shown that the resonant frequency of an embedded mass is inversely proportional to the square root of the distance at which it is placed within a poroelastic layer[3], $f_n \propto 1/\sqrt{x}$, measuring x as shown in Figure 4. A sample transfer function [TF] between the input acceleration at the base of an acoustic foam block and output acceleration of an embedded mass shows behavior similar to a conventional mass-spring-damper, Figure 5. The inset of Figure 5 shows the typical testing arrangement, with an electromagnetic shaker powered with white noise exciting a melamine foam block with mass placed in its interior. It is clear that the embedded mass primarily responds to frequencies at or very near to its resonance frequency.

A model was developed from the work of Allard[6] to determine the sound transmission loss [TL] and insertion loss [IL] through a porous material attached to an infinite plate. Predicting the sound transmission through such a system could allow one to estimate the effectiveness of the additional embedded masses as well as how to efficiently distribute their tuning frequencies.

Allard points out that predicting the transmission loss through a poroelastic media can only be made "in the context of a model where the air and frame move simultaneously", the frame being considered the solid portion of the porous material. By determining the transmission matrix, $[T]$, whose components are provided in Allard[6], which relates the fluid
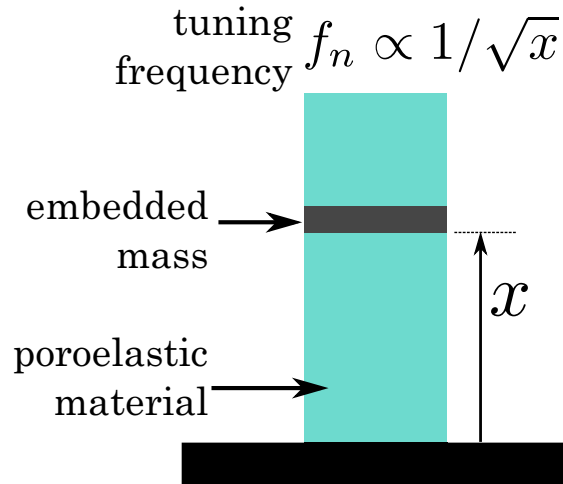
Figure 4: Diagram showing proportionality of HG blanket tuning due to embedded mass distance.
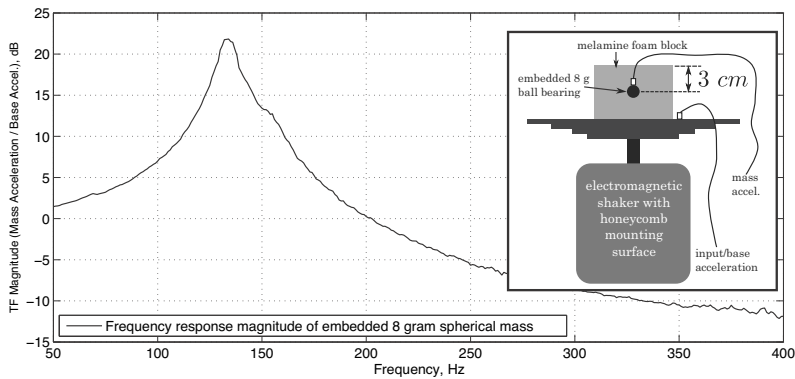

Figure 5: Sample TF magnitude for a 8 gram spherical mass mass embedded into a 5 cm thick melamine foam block. Insert: diagram of experiment with shaker platform exciting the HG sample to resonance with white noise.
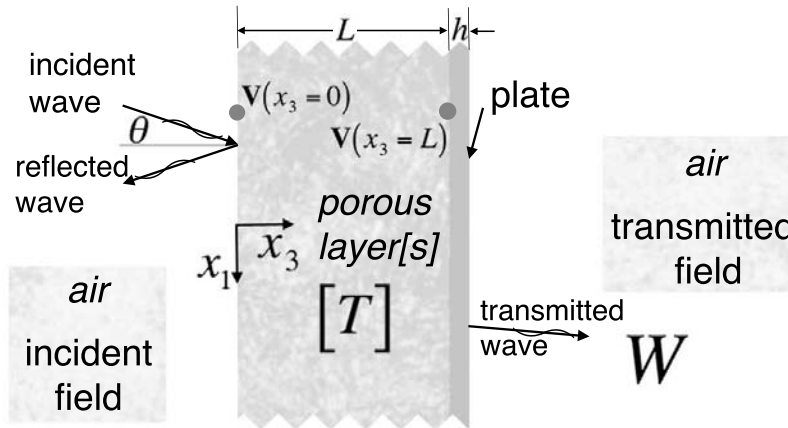
Figure 6: Porous material attached to an infinite plate with incident acoustic field and transmitted field on side of the plate.

and solid stresses and velocities from one region of the poroelastic media to another region, one can find the transmitted stresses and velocities present at the plate resulting from an incident acoustic field. A diagram of the scenario under study is shown in Figure 6. This transmission matrix then relates the stresses and velocities from the incident field, $x_3 = 0$, to those present at the plate, $x_3 = L$.

$$\mathbf{V}\left(x_3 = 0\right) = [T]\,\mathbf{V}\left(x_3 = L\right) \tag{1}$$

where the vector $\mathbf{V}$ is composed of the following quantities:

$$\mathbf{V}\left(x_3\right) = \begin{bmatrix} v_1^s & v_3^s & v_3^f & \sigma_{33}^s & \sigma_{13}^s & \sigma_{33}^f \end{bmatrix}^{\mathrm{T}} \tag{2}$$

where v_i denotes velocity, σ_{ii} denotes stress tensor components, s denotes the solid frame, f denotes the fluid components and $\mathbf{T}$ denotes the transpose. In deriving $\mathbf{V}\left(x_3 = L\right)$, both the infinite panel impedance and the transmission coefficient, W, appear allowing one to then solve for the transmission loss through the poroelastic and plate system. The diffuse field transmission loss can then be calculated from Equation 3.

$$TL = -10\log_{10}\left(2\int_0^{\pi/2} |W\left(\theta\right)|^2 \cos\theta\sin\theta\mathrm{d}\theta\right) \tag{3}$$

Allard found that despite the infinite panel assumption, experimental values of transmission loss for a finite poroelastic and plate system match very closely to the results predicted by the model.

The insertion loss is calculated with similar ease. The insertion loss is the ratio of sound pressure level with and without the plate/poroelastic system in place from some desired observation point or array of points. For primarily dissipative systems, the transmission loss and insertion loss are nearly identical. Full derivation is available in the text[6].

Kidner[7] determined that the presence of embedded masses could be included into this model by assuming the masses to be an additional panel impedance as shown in the modified panel impedance of Equation 4. This simplification can only be met by assuming that the embedded masses are evenly and densely distributed in both space and frequency, allowing their combined impedance to be "spatially averaged", shown in Equation 5. This assumption therefore ignores localized effects of the masses but treats them as a lumped impedance.

$$z'(\omega) = \frac{1}{j\omega}\left[D(1+j\eta)(k\sin\theta)^4 - m\omega^2\right] + \langle Z_a \rangle \tag{4}$$

$$\langle Z_a \rangle = \frac{1}{N}\sum_{n=1}^{N} Z_{a,n} = \frac{1}{N}\sum_{n=1}^{N} j\omega m_{a,n}\frac{\omega_{a,n}^2 + j\omega\omega_{a,n}\eta_{a,n}}{\omega_{a,n}^2 - \omega^2 + j\omega\omega_{a,n}\eta_{a,n}} \tag{5}$$

The panel impedance $z'(\omega)$ is a function of angular frequency ω, plate flexural rigidity D, plate loss factor η, mass area density m and wavenumber $k = \omega/c$, where c is the speed of sound in air. The "spatially averaged" impedance of the embedded masses, $\langle Z_a \rangle$, is a function of the individual mass $m_{a,n}$, the number of masses N, the corresponding tuning frequency $\omega_{a,n}$ and loss factor $\eta_{a,n}$.

By modifying the panel impedance present in $\mathbf{V}(x_3 = L)$, one can then model the sound transmission or insertion loss through a poroelastic material with a large number of mass inclusions, similar to that of an HG blanket.

A numerical model was composed using Allard's derivation and Kidner's approximation of the embedded masses to consider the advantages of embedded masses in a host poroelastic material. The simulated structure is a thin, aluminum panel, 3.1 mm thick and assumes an infinite span, according to Allard's model. Treatments applied to the structure in the simulation include a foam layer, vibration absorbers on their own and, finally, an HG blanket which is composed of the foam layer with embedded masses tuned to equivalent frequencies as were the absorbers (60–140 Hz).

The vibration absorbers correspond to a 2% mass ratio to the structure, totaling 0.12 kg/m^2 as compared to the 6.5 kg/m^2 of the panel, and are modeled as being evenly and randomly distributed over the structure. The foam layer is modeled as being 5 cm thick and the remaining properties come from Allard's text: tortuosity, 1.98; density, 16 kg/m^3; flow resistivity, 65e10^3 N s/m^4; porosity, 0.99; shear modulus, 1.8+j0.18 N/cm^2; Poisson ratio, 0.3; shape factor, 1.8. Figure 7 plots the increase of insertion loss as compared to the bare panel when the panel is treated with: the foam layer (solid line), the vibration absorbers (dash-dot line) and the HG blanket (dashed line) which is composed of the foam layer with masses embedded within.

An interesting feature, evident in Figure 7, are the resonances appearing in the insertion loss for the foam layer and HG blanket at high frequencies. These are produced by the acoustic coupling between the fluid of the poroelastic material and the surrounding fluid, here both air. The foam thickness, density, porosity and loss factor are important parameters in the determination of the spacing and magnitude of these resonances.

Next, one can see the increase of insertion loss that both the HG blanket and vibration absorbers provide within the range of their tuning frequencies, 60–140 Hz. Above this range,
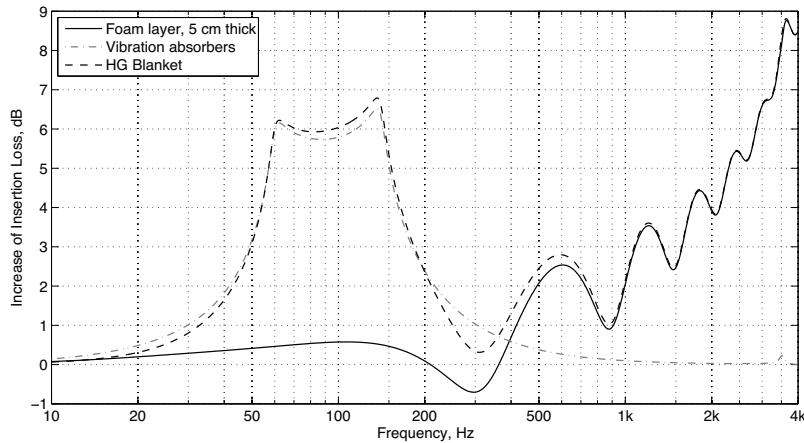
Figure 7: Increase of insertion loss for the thin panel when the treatment is a foam layer (solid line), vibration absorbers tuned from 60–140 Hz (dash-dot line) and an HG blanket (dashed line).

Table 1: Properties of foam layer used in simulation

Thickness $[m]$	Density $[kg \cdot m^{-3}]$
0.07	30

Bulk Modulus $[kPa]$	Flow Resistivity $[kPa \cdot s \cdot m^{-2}]$
79	11.89

however, the vibration absorbers lose the capacity to alter the insertion loss of the structure since at frequencies well above their resonance the absorbers appear merely as a small added mass to the host structure.

Finally, the HG blanket is seen to benefit from both the classical vibration absorber response as well as the host poroelastic material damping effects at high frequencies. The HG blanket overcomes the requirement of adding significant foam layer mass to achieve low frequency attenuation by adding a minor amount of solid, point masses into the foam which act as an array of vibration absorbers.

Next, a model was constructed to emulate an experiment performed earlier in which the usefulness of embedded masses in a foam layer attached to an acoustically-excited, thin, lightly damped plate was evaluated[3]. The foam material was modeled with the properties of Table 1. The panel was of dimensions 1.11 m by 1.11 m with a total mass of 3.4 kg. The HG blanket masses represented an additional 0.4 kg to the structure—11% of the panel mass—and were modeled as having tuning frequencies evenly spread from 60 to 140 Hz to satisfy the modeling assumptions.

Figure 8 presents the model results, comparing the insertion loss of the HG blanket with panel with that of the bare foam layer and panel, plotted in 1/3 octave bands from 63 Hz to
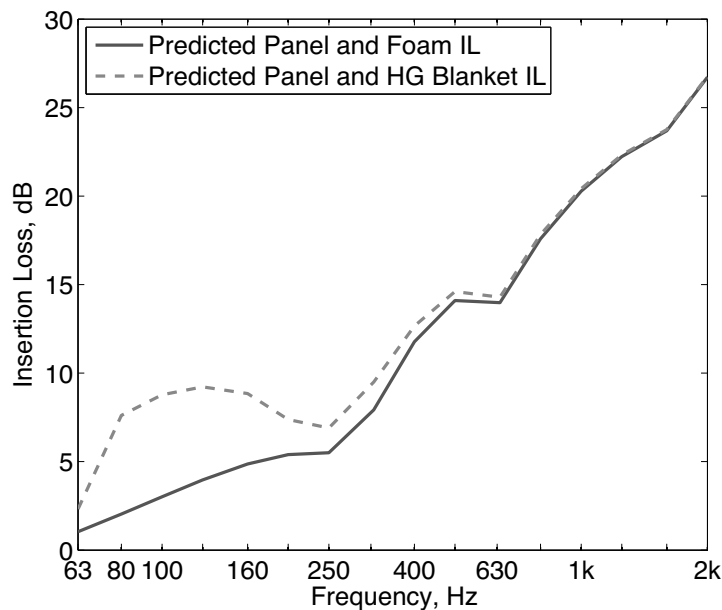
Figure 8: Modeled insertion loss of the foam layer and panel (solid line) and of the HG blanket and panel (dashed line).

2 kHz. The simulation shows a steady increase in insertion loss of the panel and foam layer treatment as one considers higher and higher frequencies. However, the HG blanket provides an increase in IL as compared to just the foam and panel system—an average increase of 4.9 dB from the 80–160 Hz 1/3 octave bands, a maximum of 5.5 dB for the 80 Hz 1/3 octave band. At frequencies well above the tuning frequencies for the masses, the plots converge as expected since the additional, embedded vibration absorbers are not excited in that range.

The test that the above model depicted, described by Kidner[3], utilized 50 spherical, 8 gram masses for the HG blanket, locating them randomly throughout the foam layer. The foam material tested was Willtec foam, as manufactured by Illbruck, with characteristic properties as provided in Table 1. The results of the test are shown in Figure 9 and compare qualitatively well with the trends of Figure 8. The difference in IL between the HG blanket and panel system and the foam and panel system is provided in Table 2.

The simulation, in fact, under-predicted the benefit of the embedded masses to increase the low frequency transmission loss as Table 2 shows the average IL of the experiment to have been greater than that for the model, for the 1/3 octave bands from 80–160 Hz. Figure 9 also reveals the convergence of the HG blanket IL back down to the level of just the foam and panel system at high frequencies. The benefit of the sound control treatment at high frequencies is then limited to the attenuation capabilities of the poroelastic material itself.

An interesting effect appears in the experiment around the 200 Hz 1/3 octave band. The IL of the HG blanket temporarily drops beneath the level produced by just the foam. This
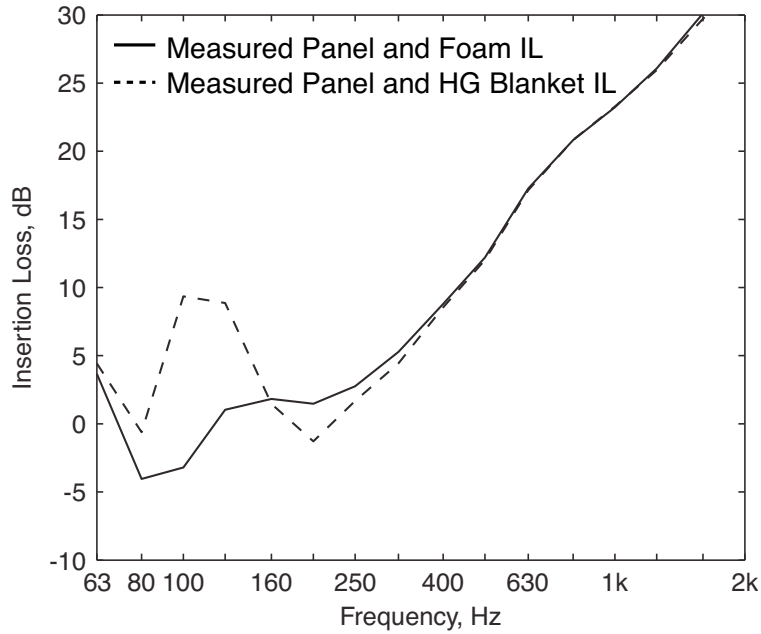
Figure 9: Experimental insertion loss of the foam layer and panel (solid line) and of the HG blanket and panel (dashed line)[3].

Table 2: Change in insertion loss from panel and foam case to panel and HG blanket scenario for given 1/3 octave band

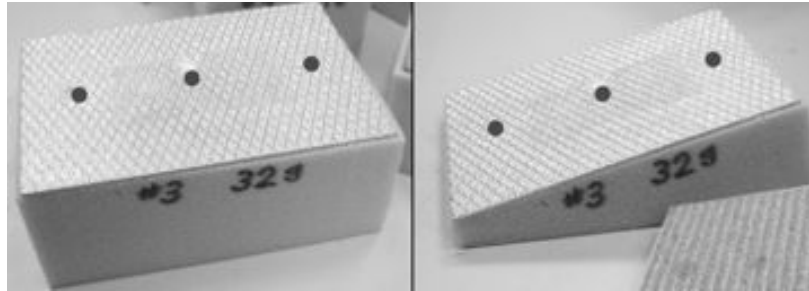|	80 Hz	100 Hz	125 Hz	160 Hz
Model	+5.5 dB	+5.2 dB	+4.9 dB	+4.1 dB
Experiment	+4 dB	+15 dB	+9 dB	0 dB
	Average 80–160 Hz			
Model	+4.9 dB			
Experiment	+7 dB			

Figure 10: HG piece tested with constant foam thickness (left) and varied foam thickness (right) with dots representing accelerometer measuring locations.

effect does not emerge from the simulation and is likely due to a coupling effect between the masses and surrounding foam. As with most multi-body systems, some dynamics may produce in-phase motion, here evidenced by a sudden drop in IL given that the fluid motion and embedded masses are likely oscillating in-phase.

Nevertheless, both the experimental and simulated results show that embedded media within the poroelastic material will increase the low frequency attenuation capability of the host material at frequencies to which the masses were tuned. This encourages one to design an HG blanket which is tuned to the widest frequency range possible for a given target mass ratio desired for the HG blanket treatment.

2.2 Mass Inclusion Optimization

One method to achieve a wide tuning frequency range is to design an HG blanket which utilizes a large proportion of the foam thickness available. Figure 10 shows the difference between a small HG sample with a uniform foam thickness and the same HG sample with a variable foam thickness beneath the mass. In this instance, a continuous mass was used for testing such that mass per area and stiffness per area characteristics play a role in determining the SDOF tuning frequency.

Figure 11 shows the transfer functions for 3 measured points on the top of the samples of Figure 10, where the TF is between the base input acceleration and the sum of the squares of the output accelerations. The case of constant foam thickness below the mass layer shows a single resonance. Interestingly, even though the mass is continuous, the 3 measuring points of the variable thickness sample produce 3 unique SDOF resonance frequencies, suggesting a continuum of resonance frequencies is present, the bandwidth of these frequencies being defined by the variation of mass per area and stiffness per area along the absorber's varied thickness axis. The objective is design is thus to exploit this potential range of "tuning frequencies".
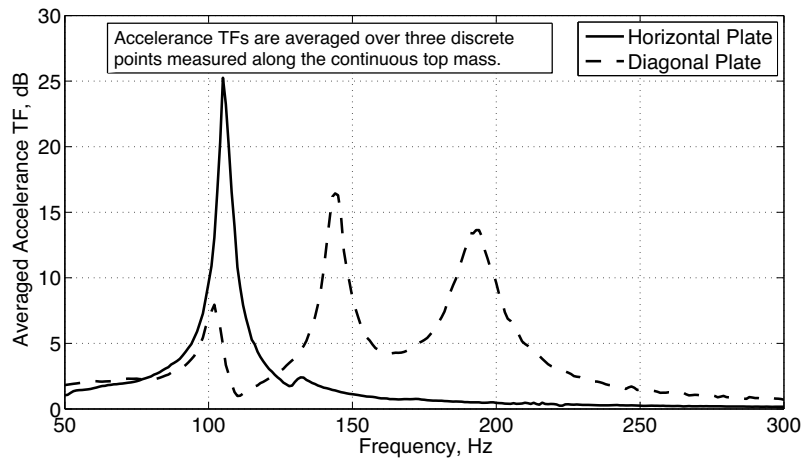
Figure 11: Comparison of TF between constant foam thickness and varied thickness measured at 3 points, as located in Figure 10.

2.3 Testing

Testing was performed at Virginia Tech to evaluate the HG blanket design which incorporated variable thicknesses of foam layer beneath the masses. As marine structures are often characterized as being composed of heavy, thick plates, the treatments were attached to a large structural plate measuring 2 ft by 4 ft by 1/4 inch thick. The plate was suspended atop automobile inner tubes to approximate free suspension. A hole 3/16 inch in diameter was drilled 1/2 inch from a corner of this large plate for attachment of an electromagnetic shaker. The shaker, Ling Dynamic Systems 50 lb shaker, was vertically suspended and attached to the plate at a corner via a short stinger and force transducer, PCB model 208 A03. An array of accelerometers, PCB model 352 A10, were randomly placed on the underside of the plate to measure plate vibration. The shaker excited the plate with white noise from 20–10,000 Hz and transfer functions were found between the input force and the sum of the squares of the acceleration measurements.

The test plate was fully covered with melamine foam 2 inches thick adhered using spray glue. The accelerance TF was then measured as the plate was excited by the shaker. Next, an HG blanket was constructed using variable foam thickness to achieve the greatest bandwidth of tuning frequencies possible for a certain thickness of melamine foam. To aid in this goal, "wedged" foam was used since the variation of thickness is already present in the foam design.

Figure 12 shows a portion of a 2 ft by 2 ft by 2 inch thick HG blanket with a large number and variety of steel and brass washers placed atop the exposed foam faces. This masses were placed randomly and the final HG treatment represented tuning frequencies from 60–250 Hz.

The plate was then fully covered with this HG blanket design, reaching an 8.5% mass ratio [m.r.], also using spray glue for attachment. The accelerance TF was measured as before. A comparison of the plate vibration without treatment (black dash), with melamine
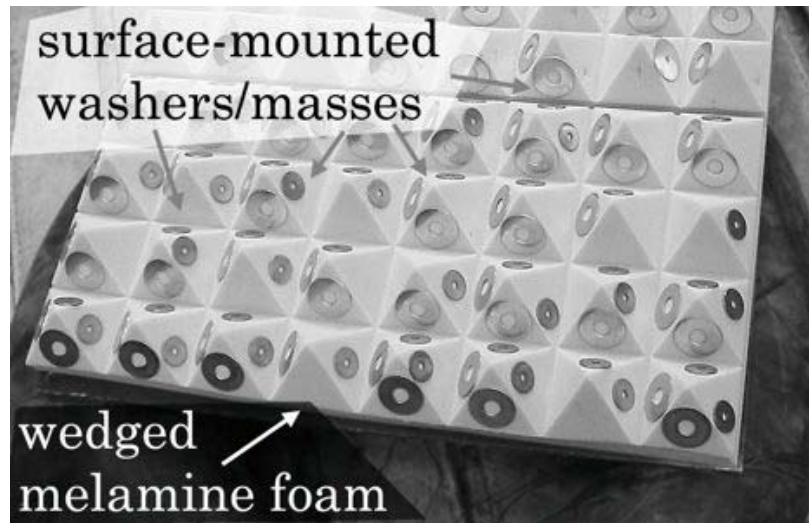
Figure 12: HG blanket constructed from wedged acoustic foam with embedded masses tuned from 60–250 Hz.

foam (gray line) and with HG blanket (black line) is shown in Figure 13.

The melamine foam itself attenuates little to no vibration at frequencies less than 300 Hz. In the range of its tuning frequencies, the HG blanket provides attenuation on the order of 5–7 dB in 1/3 octave bands. The narrowband plot shows that the HG blanket reduces the plate resonances on the order of 5–18 dB. Since the melamine foam and HG blanket were both 2 inches thick, the addition of the masses in the HG blanket are seen to significantly increase low frequency absorption, at a cost of an additional 7.5% of the plate mass (the foam itself totaled 1% of the plate mass). The HG blanket absorber design therefore takes advantage of the inherent elasticity of the porous media to suppress low frequency vibration which the poroelastic is unable to attenuate on its own and widening the tuning frequency range using a varying thickness of foam layer allows for maximum passive attenuation.

3 Distributed Vibration Absorbers

3.1 Development

The original design of the DVA, described in detail by Cambou[4], was intended for a combination of active and passive noise control, using a piezoelectric film as the choice woven layer allowing the distributed active vibration absorber [DAVA] to be electronically controlled. In the current study, however, active elements were removed to pursue a versatile form factor not hindered by control design complexity.

The DVA is composed of four, basic elements (Figure 14): *(i)* a continuous top mass, *(ii)* a continuous woven spring, *(iii)* a base layer and *(iv)* some method for adhering the joints
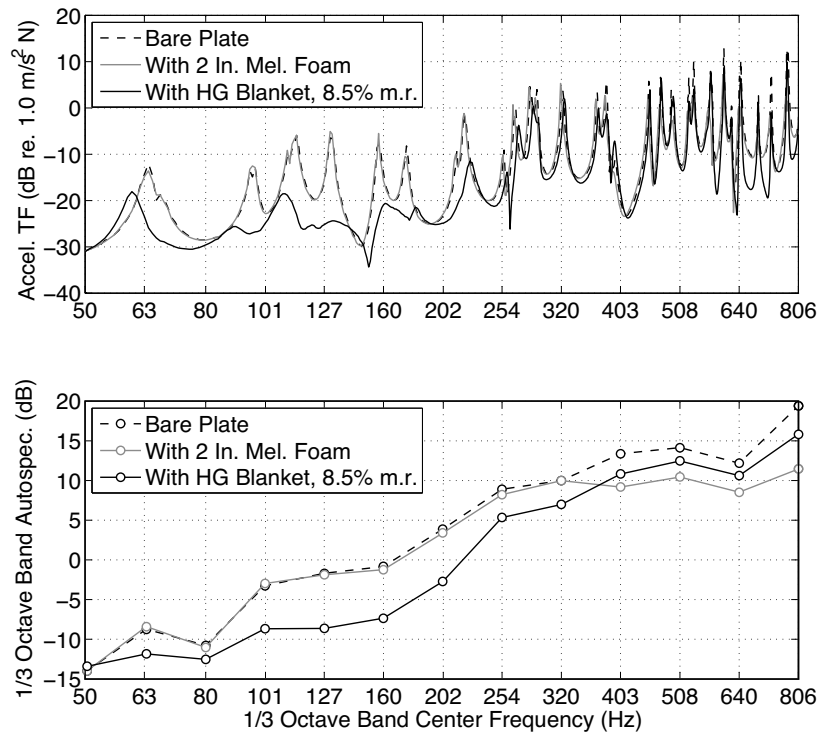
Figure 13: Comparison of plate vibration without treatment (black dash), with melamine foam (gray line) and with HG blanket (black line) in narrowband (top plot) and 1/3 octave bands (bottom plot).
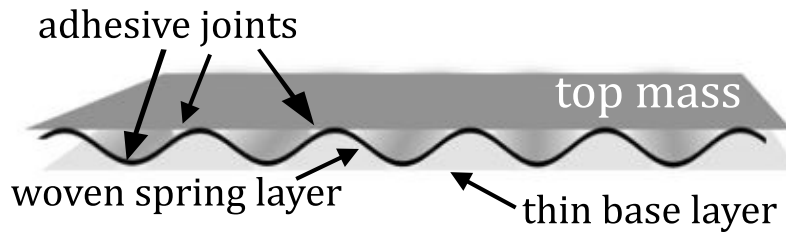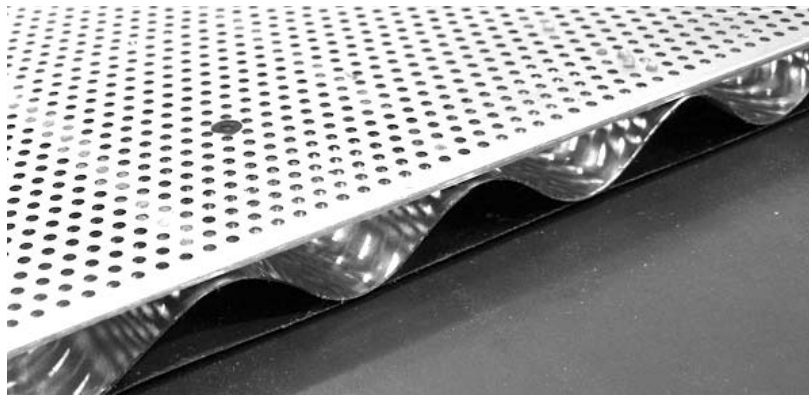
Figure 14: The elements of the DVA.

Figure 15: Sample DVA piece with steel and viscoelastic woven layer.

together. A variety of adhesive, epoxy and metal bonding techniques have been evaluated in this study and the selection of adhering method is ultimately decided based on one's manufacturing restrictions or tuning goals.

Tuning of the DVA can be accomplished by modifying: *(i)* the top mass area density, *(ii)* the woven wavelength in the spring layer, *(iii)* the weave material thickness and *(iv)* by choice of adhesive method. The damping of the DVA is also a product of the adhesive choice, notably a function of how soft or firm the joints become after a bond is made. In this study, damping was added in the form of a viscoelastic damping polymer sandwich within the woven layer.

Figure 15 shows a sample DVA with a perforated steel mass, steel and viscoelastic woven layer and a steel base layer. The viscoelastic layer—placed in between two strips of 0.001 inch thick steel—is too thin to be evident in the photo. The adhesive used to hold the part together was a high-strength epoxy, chosen so as to require as little epoxy as possible since it can contribute to increasing the total mass.

A sample DVA TF is shown in Figure 16 showing, just like for an embedded mass in melamine foam of Figure 5, a clear SDOF resonance. This DVA was identical to that pictured in Figure 15 but did not contain a viscoelastic sandwich woven layer. The woven spring layer was a single, thin metal strip. Thus, this TF is expected to be narrower in
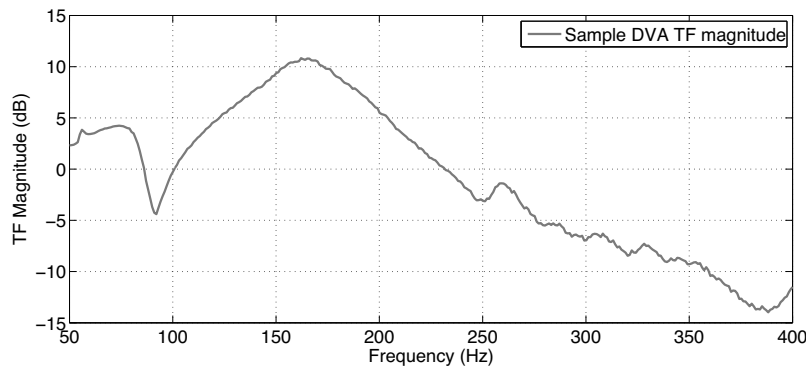
Figure 16: Sample DVA TF for piece tuned to 165 Hz.

bandwidth at resonance than that resulting from the piece with the visco sandwich spring design.

Since the original development, most DVA designs that were studied used acoustic foam as the spring layer. There is an advantage, however, in using the woven spring approach of this study as opposed to a poroelastic "spring" for the DVA design. It is known that damping is an important low-mass method of increasing the absorber's response at frequencies away from resonance[8]. Unfortunately, the damping characteristics of a foam material are inherently fixed, being a characteristic property of the poroelastic material itself. In light of achieving both a tunable absorber design and the possibility of global vibration control with increased damping, the woven spring design has an advantage over using a poroelastic material as the spring layer.

Thus, this project returned to the original woven spring layer design which was deemed more versatile in this respect. The viscoelastic sandwich woven layer could then be tested for its benefit over the standard thin woven layer, particularly in terms of off-resonance vibration absorption. Other locations for the viscoelastic layer have been considered—covering the bottom surface of the mass or covering the top surface of the base layer with the visco, for example—but only the woven layer sandwich design will be discussed here.

3.2   Testing

Given the size of the plate to which these DVAs would be attached, a large number were constructed all using the same top mass dimensions, 6 inches by 12 inches. For full plate coverage, this would require 16 pieces in total. A variety of steel and aluminum sheets of perforated metal were available, representing several area densities. The DVAs were also constructed with several woven layer wavelength dimensions and steel sheet thicknesses. This resulted in DVAs with unique tuning frequencies, ranging from 80–400 Hz. Perforated masses were preferred over solid masses in light of knowledge that air pumping effects can generate high frequency damping resulting from the turbulent flow through the perforations[9]. The perforations also play a role in decreasing the radiation efficiency of the DVAs such that the
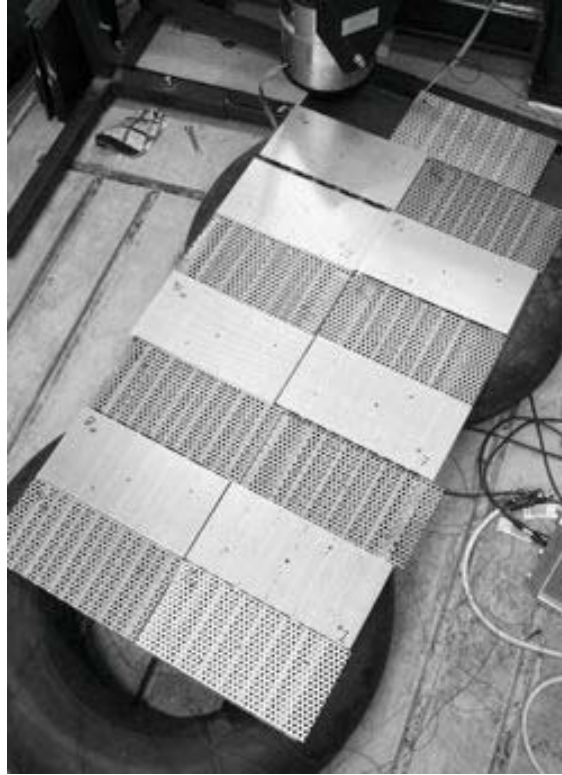
Figure 17: DVA treatment attached to heavy test plate.

vibrating absorbers do not themselves become sources of noise.

To compare the advantage a viscoelastic sandwich woven layer design provides, one batch of DVAs was created with the viscoelastic sandwich layer and one without, the latter using just a single thin woven layer. The DVA treatment using just a thin woven layer had a m.r. of 9.1% while the DVA set with the visco sandwich woven layer had a m.r. of 10%.

No optimization strategy was used for placing the DVAs onto the plate; rather, they were placed wherever there was available room. As with the HG blankets, spray glue was used to secure the DVAs to the plate. Figure 17 shows the DVA treatment that used the visco sandwich layer. It should be noted that only 15 DVA pieces were actually used on the plate for each treatment since the shaker attachment to the plate inhibited access to the plate top surface.

The performance of both DVA treatments was evaluated in a similar manner as was done with the HG blankets—attachment to the test plate, shaking the plate with white noise and taking the accelerance TF with and without treatments present. Figure 18 shows a comparison of the plate vibration with no treatment (black dash), with the DVA treatment of 9.1% m.r. (gray line) and with the DVA treatment of 10% m.r. which used the visco sandwich design (black line).

It is clear that over the range of their tuning frequencies, both DVA treatments reduce the
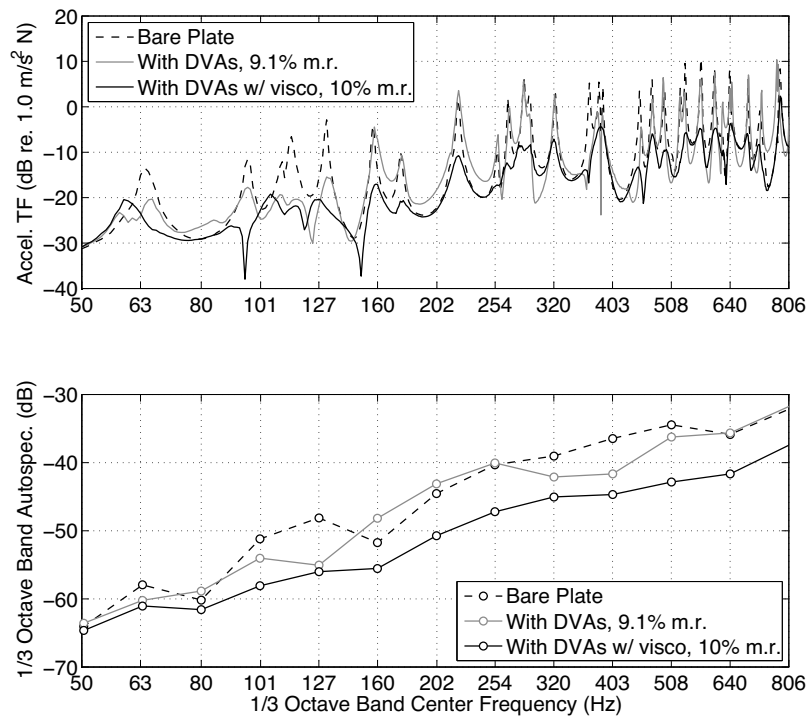
Figure 18: Comparison of plate vibration without treatment (black dash), with DVAs of 9.1% m.r. (gray line) and with DVAs of 10% m.r. using visco (black line) in narrowband (top plot) and 1/3 octave bands (bottom plot).
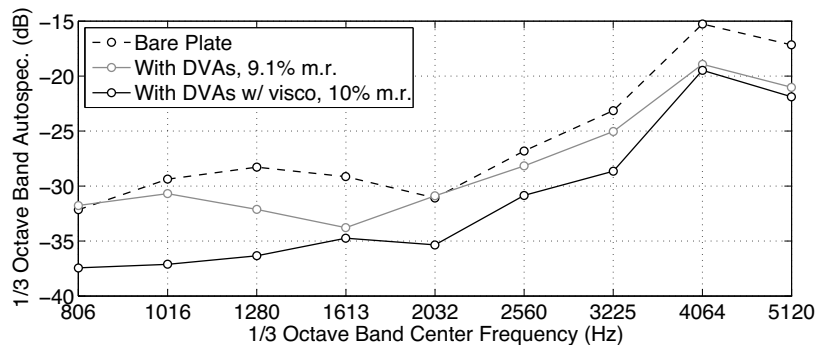
Figure 19: Comparison of plate vibration without treatment (blue dashed), with DVAs of 9.1% m.r. (green solid) and with DVAs of 10% m.r. using visco (red solid) in 1/3 octave bands at high frequencies.

plate's response by as much as 20 dB from resonance magnitudes in narrowband. However, from the 1/3 octave band comparison, the DVAs using the viscoelastic sandwich woven layer provide greater global attenuation, an average of 7 dB from 100–400 Hz. While both DVAs treatments were tuned over the same bandwidth, 80–400 Hz, the treatment which used the visco is seen to outperform the treatment lacking it.

Furthermore, considering higher frequencies, Figure 19 shows that the attenuation of plate vibration is not limited to just the tuning frequency region. The 10% m.r. DVA treatment suppresses vibration by 5 dB or greater at frequencies greater than 1,000 Hz.

In the frequency range plotted in Figure 19, it is seen that the disparity of performance between the DVA treatments decreases in this bandwidth. This suggests that the DVA design itself, and not simply the viscoelastic layer, plays a role in generating global vibration reduction. This high frequency damping is currently hypothesized as being produced by air pumping effects through the perforations and/or through the woven layer cavities.

4 Comparison

To assess how well HG blankets and DVAs perform against a commercially available product a conventional noise control tile treatment was acquired. The "damping tiles" used here are a composite material and are available in standardized sizes. The tile area density was much greater than that of the HG blanket or DVA treatments. When fully covering the plate—as was tested for HG blankets and the DVAs—the damping tile treatment produced a mass ratio of 30%.

The damping tiles were applied to the heavy plate and tested as before for the previous absorber treatments. A comparison of the low frequency performance of plate vibration attenuation is shown in Figure 20. At frequencies less than 250 Hz, all three treatments produce similar amounts of vibration reduction. This is significant if one considers that
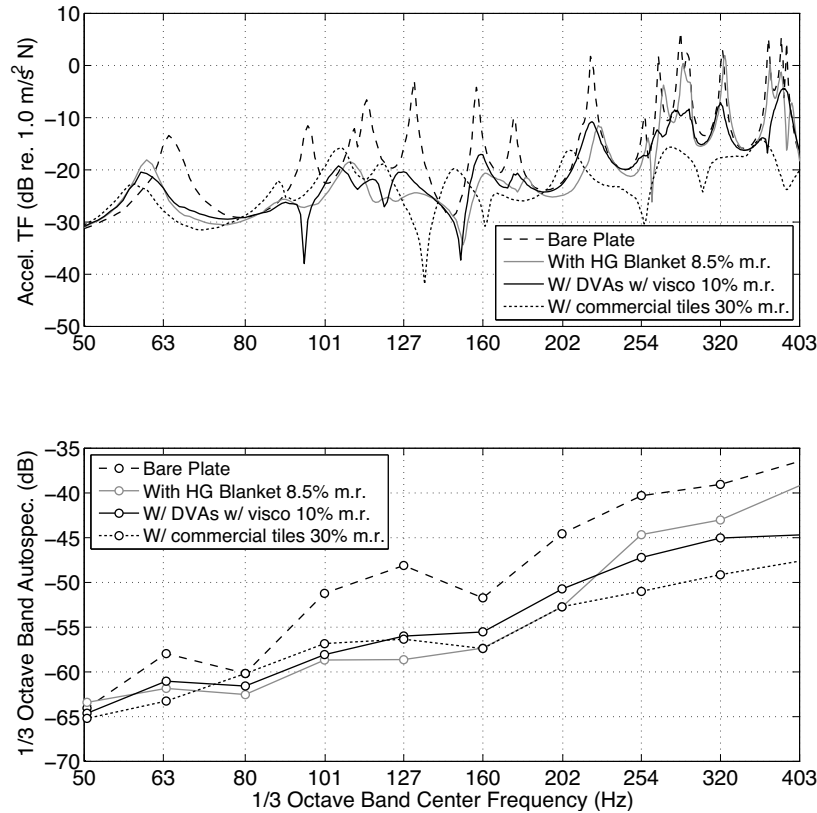
Figure 20: Comparison of plate vibration without treatment (blue dashed), with HG blanket (cyan solid), with DVAs using visco (red solid) and commercial tiles (black solid) in narrowband (top plot) and 1/3 octave bands (bottom plot).

the DVA and HG treatments tested were roughly one-third the mass of the commercial tile treatment. For applications where minimizing weight is a priority, both HG blankets and DVAs provide a viable solution to achieving low frequency noise control.

An additional benefit of both DVA and HG treatments is the ability to easily tune such absorbers to a certain bandwidth of important frequencies. Many commercially available sound absorbing treatments do not provide for this flexibility and would therefore require more mass to be added to the structure to achieve absorption in this low frequency range.

5   Conclusions

Two new vibration absorber designs were considered for their low frequency passive sound absorption. HG blankets were modeled and found to have significant absorption over their tuning frequency range. As a result, HG blankets were constructed with a design that maximized the bandwidth of tuning achievable for a given foam thickness. A lightweight

HG blanket was created and compared against a foam layer of equivalent thickness. The HG blanket was found to decrease plate vibration by 5–7 dB over the range of tuning frequencies, in 1/3 octave bands, but as much as 18 dB off of the plate resonance magnitudes. The HG blanket design significantly increased the vibration absorption of the foam layer at frequencies less than 250 Hz.

DVAs using a continuous spring layer and continuous mass layer were designed and tested in a like manner. To evaluate the benefit of a viscoelastic sandwich in the woven spring layer, one DVA treatment was constructed with this visco and one without. Perforated continuous masses were chosen to increase mid to high frequency damping due to possible turbulence and air pumping effects.

When applied to attenuate plate vibration, DVAs with the viscoelastic spring design averaged 7 dB of vibration reduction from 100–400 Hz and were also capable of more than 5 dB of global attenuation at frequencies above 1,000 Hz. The DVA treatment without the viscoelastic spring still showed significant high frequency attenuation suggesting that the DVA design itself, and not merely the visco, contributed to the global vibration reduction.

A comparison was performed amongst HG blankets, DVAs with the visco layer and a commercial tile treatment. It was found that at frequencies less than 250 Hz, the HG blankets and DVAs performed equally well or better than the damping tiles, even though the DVAs and HG treatments were three times lighter, by mass, than the tiles. DVAs and HG blankets therefore provide significant low frequency vibration absorption for a small cost in additional weight to the structure.

6   Acknowledgments

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