Advanced Passive Treatment of Low Frequency Sound and Vibration

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ABSTRACT

It is well known that standard poroelastic materials and viscoelastic damping materials are ineffective at reducing low frequency sound and vibration. This paper overviews two new treatments developed at Virginia Tech which attempt to address this problem. HG material consists of poroelastic material with embedded multiple small masses. The masses combine with the natural elasticity of the poroelastic material matrix to create multiple embedded vibration absorbers with a range of tune frequencies in the low frequency region. The embedded masses are found to significantly increase the low frequency transmission loss and absorption of the poroelastic material. DVAs are vibration absorbers whose active mass and spring are spread over a large area while still maintaining a viable reactive damping effect at low frequencies. DVAs are found to provide global reduction of low frequency vibration of structures in a compact, lightweight configuration. DVAs are also observed to provide mid to high frequency damping most likely to air squeeze damping effects. The paper will overview the concepts, development and testing of both devices. Applications of the new treatments to realistic structures will be considered.

INTRODUCTION

Low frequency noise and vibration is a particular issue for mechanical equipment such as motors or fans. However, traditional viscoelastic and poroelastic acoustic materials do not adequately absorb low frequencies, typically < 500 Hz. This paper summarizes the development and application of two new absorber treatment designs for the reduction of the vibration of realistic, distributed structural materials and systems. Both of the devices presented act primarily by suppressing the vibration of structures to which they are attached over a large area. If the structures also radiate sound, this, too, will be attenuated.

The first treatment, Distributed Vibration Absorbers or DVAs, was originally developed by Fuller and Cambou (1998) and uses a design that spreads the contributing mass and spring elements over a large area, akin to the evolution from point absorber to distributed absorber in Figure 1. 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.

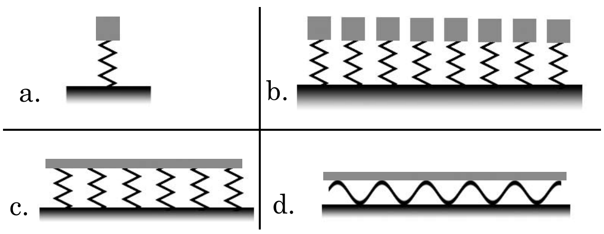


Figure 1: Progress from point vibration absorber (a), towards a continuous mass and continuous spring design as achieved in (d).

Using a variety of mass area densities and by altering the effective spring transverse stiffness by changing the woven layer characteristics, DVAs can be tuned over a large range of frequencies. The DVA was recently studied by Marcotte (2004) in depth and has been applied to a variety of structures, for instance large cylindrical shells (Osman et al. 2001) (Estéve and Johnson 2002).

DVAs have also been developed which utilize a poroelastic material as the spring layer. The continuous top mass is attached to the top of an acoustic foam material. This foam acts like a distributed spring due to the inherent stiffness characteristics. Similarly as with a woven spring layer, this DVA design also allows for tuning based on top mass area density in addition to poroelastic material selection and poroelastic thickness. A diagram of such a DVA is provided in Figure 2.

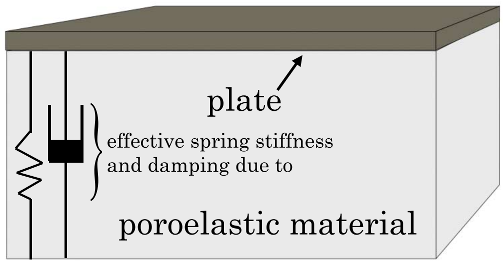


Figure 2: Effective distributed vibration absorber resulting from interaction between continuous top mass and poroelastic material.

The second treatment, known as a heterogeneous [HG] blanket, is composed of a poroelastic material into which a number of masses have been embedded. It was shown by Fuller et al. (2004) 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 3. Unlike a DVA designed with a poroelastic layer, HG blankets can target a range of tuning frequencies.

Idrisi (2008) determined that these tuning frequencies were a complex function of the foam layer properties, the embedded mass itself, the mass depth, the distance between masses and the embedded mass shape. HG blankets using melamine foam of 2 inches in thickness with embedded masses ranging from 3–12 grams will produce a tuning frequency range of about 60–250 Hz. The presence of embedded masses substantially increases the low frequency attenuation capability of the host poroelastic material.

Tests using HG blankets show they are effective at attenuating low frequency radiation in aircraft structures (Fuller et al. 2004) (Idrisi et al. 2009). Kidner et al. (2006) found that over

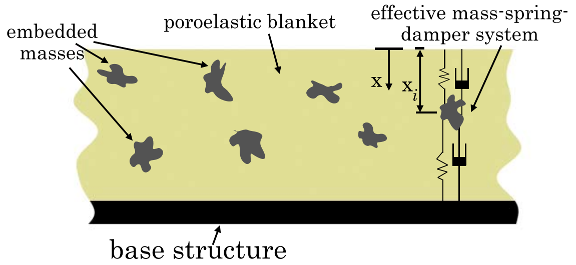


Figure 3: Diagram of poroelastic material with embedded masses, the HG blanket.

the tuning frequency range, the embedded masses could increase the low frequency insertion loss of the foam material by as much as 15 dB when attached to a thin panel.

The development and testing of DVA and HG technology will be reviewed. The application of these treatments to realistic structures will be shown to specifically increase low frequency attenuation of structural vibration and associated sound radiation while contributing minimal additional weight to the structure.

DISTRIBUTED VIBRATION ABSORBERS

Development

The work of Smith et al. (1986) was some of the earliest to discover and analyze the benefits of distributing dynamic absorbers over a continuous structure for increased attenuation. Cambou (1998) found that the distribution of a vibration absorber's mass over the length of a beam more effectively reduced beam vibration than when a single, tuned vibration absorber was applied. Figure 4 shows the two configurations considered while Figure 5 shows the additional reduction of beam vibration resulting from the distributed absorber arrangement.

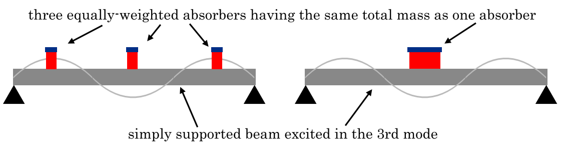
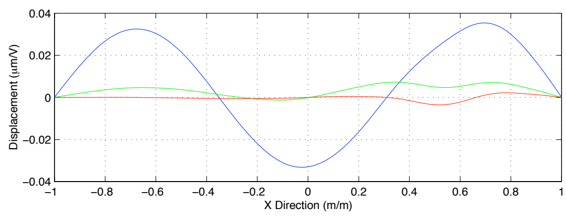


Figure 4: Test of advantage of distributing vibration absorber mass.



Source: Cambou (1998)

Figure 5: Beam displacement without absorber (blue line), with one centralized absorber (green line) and with three distributed absorbers (red line).

The DVA design by Cambou (1998) is composed of four, basic elements (Figure 6): *(i)* a continuous top mass, *(ii)* a continuous woven spring, *(iii)* a base layer and *(iv)* some method for adhering the joints together. While Cambou's design also took advantage of a woven piezoelectric material for the use of actively controlling the DAVA [Distributed Active Vibration Absorber] response, only passive versions will be considered here.

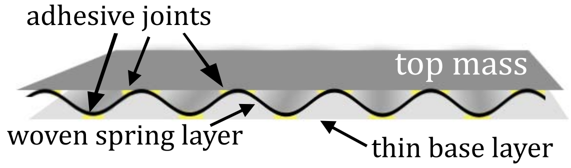
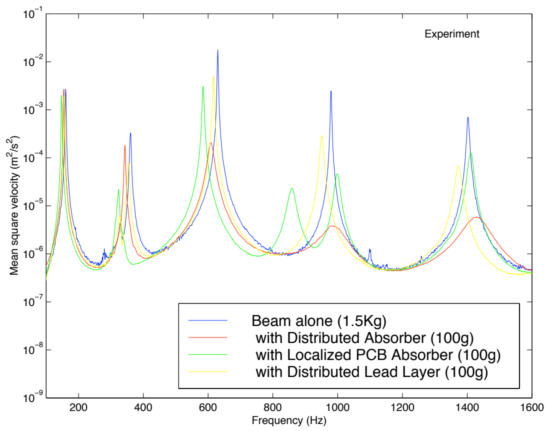


Figure 6: The components of a DVA with woven spring layer.

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. While many materials are options for the spring layer, two commonly used types are thin plastic film and sheets of metal shim.

Replicating the model of Figure 5 in a laboratory setting, Cambou found that the DVA provided greater levels of vibration attenuation than the case of either a point vibration absorber or when the DVA mass was merely glued to the beam surface. The DVA was tuned to 1,000 Hz and Figure 7 shows a dramatic decrease in beam vibration at this frequency while other beam modes are additionally attenuated.



Source: Cambou (1998)

Figure 7: Beam mean square velocity without treatment (blue line), with a DVA (red plot), with a point vibration absorber (green plot) and with the DVA mass simply adhered to the beam (yellow plot).

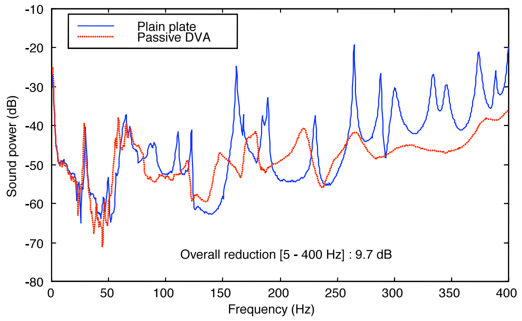
Since the DVA was not tuned to all of the modes that it attenuated in Figure 7, damping within the DVA design must play a role in allowing it to work off-resonance. Fuller et al. (1997) found that damping is a low-mass solution for increasing a vibration absorber's response outside of its tuning frequency. Thus, materials can be selected for the woven layer which increase the loss factor of the spring transverse stiffness. Alternatively, the selection and application of the adhesive material, which holds the DVA together, can be selected to increase damping.

While a woven spring layer provides a means for achieving a distributed spring, the finite number of points of attachment between the base layer and top mass produce a discretized distribution of the reactionary forces. Marcotte (2004) investigated the potential of using a truly continuous spring layer in the form of a poroelastic layer. A diagram of this resulting DVA is shown in Figure 2. This design takes advantage of the tuning properties of the DVA as well as the inherent losses resulting from sound transmission through a poroelastic material.

Marcotte also considered that since the top mass was continuously connected to the spring layer, it would respond more akin to a freely suspended plate than like a single degree-of-freedom system. Indeed, it was found that this configuration produces a DVA where the top layer does not act as a rigid body but rather "as a flexible plate having a bending motion that is coupled with the bending motion of the base plate".

Testing

An application of the DVA design using a poroelastic spring layer was attached to a vibrating plate in a transmission loss facility to evaluate the treatment's global noise reduction capability. Figure 8 shows the resulting reduction in radiated sound power from the DVA treatment. The overall sound power reduction from 5–400 Hz was measured to be on the order of 10 dB with certain plate resonances being reduced by even greater amounts. This result is very promising and applying DVAs to a real structure is the next logical extension to evaluate their performance.



Source: Marcotte (2004)

Figure 8: Plate radiated sound power without treatment (blue plot) and with DVA treatment (red dotted plot).

Source: Osman et al. (2001)

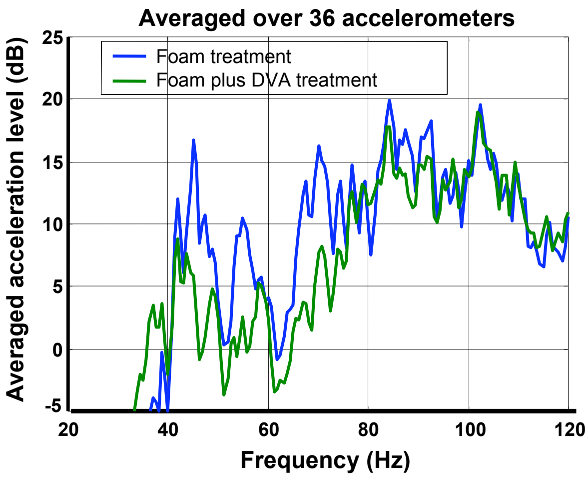
Figure 9: Large payload shroud suspended in laboratory.

Application to a launch vehicle payload shroud

A test was performed to evaluate the attenuation of DVAs attached to a large launch vehicle payload shroud. This structure was a composite, cylindrical shell of 2.8 m in length and 2.46 m in diameter weighing 80 kg (Figure 9). The stiffened end caps each had a mass of 226 kg. The system was excited by exterior acoustic sources, at very high sound pressure levels, in the range of 145 dB.

Tests were conducted with the cylinder having no interior treatment, when the cylinder interior was lined with acoustic foam and when a number of DVAs were added to the foam lining at regular intervals. Osman et al. (2001) found that DVAs using a poroelastic spring layer were capable of reducing the vibration by an additional 5–10 dB over the foam-lined case at frequencies below 150 Hz. Figure 10 shows the resulting vibration levels comparison between the foam lining interior and when the DVAs are added to the foam layer.

It was also found that the DVAs reduced the vibration of frequencies to which the treatment was not tuned—a resonance as low as 54 Hz was attenuated by about 10 dB. Thus, damping in the DVA design leads to off-resonance vibration suppression, as was proposed in Fuller et al. (1997).



Source: Osman et al. (2001)

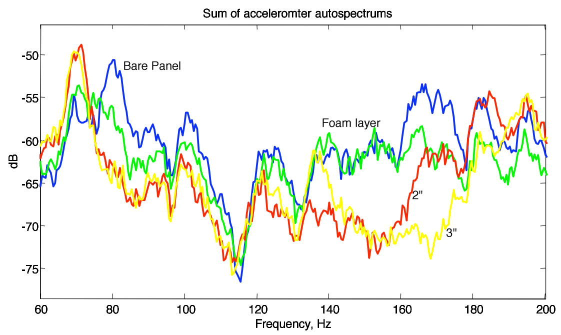
Figure 10: Averaged acceleration level of the cylinder with the foam treatment (blue plot) and with the additional DVAs (green plot).

The foam treatment that was lining the cylinder had a total mass of 15 kg and the DVAs contributed an additional 6 kg. However, the cylindrical shell under study was a massive 532 kg, including the mass of the end caps. Thus, when the DVAs were present with the acoustic foam, this interior vibration absorber treatment contributed just 3.9% additional mass to the structure. Thus, the low frequency losses possible with the additional DVAs show they are a truly lightweight low frequency vibration control solution.

HG BLANKETS

Development

While the transmission loss characteristics of poroelastic media have long been known, recently work has been performed to study methods of increasing the low frequency losses of such materials which are otherwise negligible. The work of Fuller et al. (2004) was some of the earliest to investigate the effects of embedded masses in an acoustic foam layer. Figure 11 shows the results of a test of plate vibration without any treatment and with treatments of acoustic foam 3 inches thick, an HG blanket 2 inches thick and an HG blanket 3 inches thick. The foam layer and HG blankets all utilized the same melamine foam material.



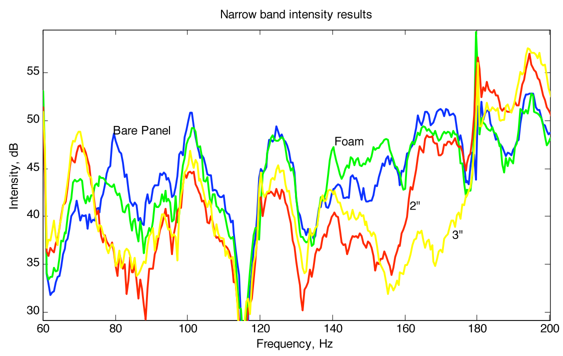
Source: Fuller et al. (2004)

Figure 11: Plate vibration without treatment (blue plot), with a 3 in. layer of melamine foam (green plot), with a 2 in. thick HG blanket (red plot) and with a 3 in. thick HG blanket (yellow plot).

The increase in vibration reduction of the base plate resulting from the additional embedded masses of the HG blanket suggest that the masses interact with the foam layer to generate a matrix of vibration absorbers, akin to Figure 3. It should be noted that the HG blankets contributed just an additional 6% of mass to the plate which is a very lightweight solution to achieving a significant increase in low frequency attenuation.

Figure 12 shows the corresponding reduction in acoustic intensity when these treatments were compared in a transmission loss facility. When the embedded masses were present in the foam layer, the transmission loss increased on the order of 6 dB in one-third octave bands in the frequency range of 60–180 Hz. Of significance, the 2 inch thick HG blanket weighed less than the 3 inch thick melamine foam sheet without masses. However, the 2 inch HG blanket produced a notable increase in both transmission loss and vibration reduction at low frequencies compared with the heavier 3 inch melamine layer.

In an effort to simulate an HG blanket, a model was developed from the work of Allard (1993) to determine the sound transmission loss through a porous material attached to an infinite plate which would serve as a foundation for a complete HG blanket model. Predicting the sound transmission through such a system could allow one to estimate the effectiveness of the additional



Source: Fuller et al. (2004)

Figure 12: Plate radiated sound intensity without treatment (blue plot), with a 3 in. layer of melamine foam (green plot), with a 2 in. thick HG blanket (red plot) and with a 3 in. thick HG blanket (yellow plot).

embedded masses as well as how to efficiently distribute their tuning frequencies.

Allard points out that such a prediction 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 media. By determining the transmission matrix, $[T]$, whose components are provided in Allard (1993), which relates the fluid 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 13. This transmission matrix then relates the

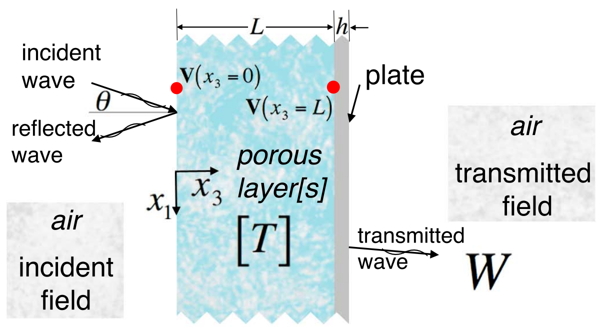


Figure 13: Porous material attached to an infinite plate with incident acoustic field and transmitted field on side of the plate.

stresses and velocities from the incident field, $x_3 = 0$, to those present at the plate, $x_3 = L$.

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

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

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

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 $\mathbf{V}(x_3 = L)$, 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.

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

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. Full derivation is available in his thorough text.

It was found 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 (Kidner 2004). 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.

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

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

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 number of masses N, the individual mass $m_{a,n}$ and its 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 loss through a poroelastic material with a large number of mass inclusions, similar to that of an HG blanket. A model was constructed, using both Allard's and Kidner's equations, to compare the diffuse field transmission loss through a sample 2 inch thick melamine foam sheet and the transmission loss through an HG blanket composed of the same melamine sheet with embedded masses tuned from 60–140 Hz. Considering the test plate that will be used for this work, the HG blanket of the model was designed so as to have a total mass ratio of 10%.

A comparison of the diffuse field transmission loss predictions for these treatments is shown in Figure 14. A significant increase in sound transmission loss is predicted for the HG blanket over the range of its embedded mass tuning frequencies, as much as 13 dB. However, above this bandwidth, the model predicts that the embedded masses will not produce a considerable change in TL compared to the foam only. Thus, to achieve the broadest tuning frequency range for a given poroelastic foam thickness, it is advantageous to embed masses over the full span of thickness available.

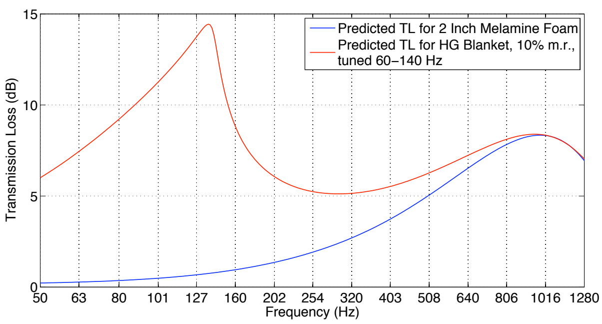
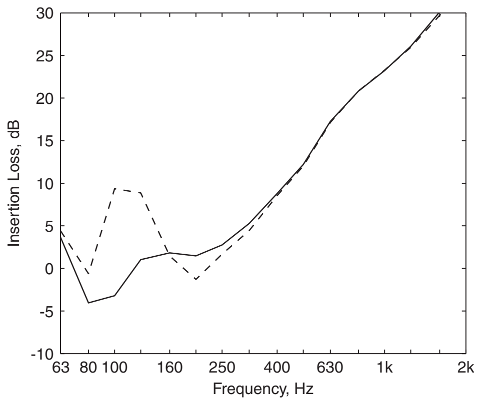


Figure 14: Predicted transmission loss comparison between melamine foam layer and HG blanket.

To validate the results of this model, an experiment has been carried out by Kidner et al. (2006) which measured the insertion loss through a panel when a poroelastic material was attached and when the same poroelastic material having 50 randomly embedded masses was attached. Figure 15 shows the significant increase in insertion loss over the range of tuning frequencies of the masses, about 80–140 Hz.



Source: Kidner et al. (2006)

Figure 15: Insertion loss of the foam layer and panel (solid line) and of the HG blanket and panel (dashed line).

The results of Figure 15 match closely with those predicted from the model, as plotted in Figure 14. Though the model was estimating transmission loss, it is known that values of transmission loss and insertion loss are very similar when the absorber treatments are primarily dissipative and not reflective (Bies and Hansen 2003). The increase in insertion loss in the 100 Hz band of about 15 dB is very nearly replicated by the model. This experimental validation of the model suggests it can serve as a design tool for future HG treatments.

Testing

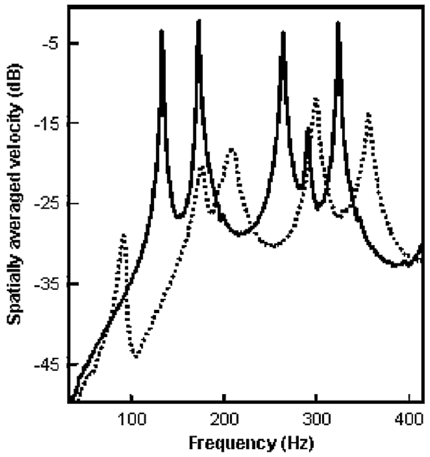
In addition to performing much of the fundamental work to determine HG blanket tuning characteristics and specific designing parameters, Idrisi (2008) completed a large number of tests to evaluate HG blanket performance. Of specific interest to Idrisi was the double panel structure design typical in aircraft applications.

The configuration of this double panel system was ordered as: fuselage, HG treatments, air cavity, interior trim panel, and the final interior acoustic field. The system was excited by point force on the fuselage panel. An HG blanket was constructed and tuned to 130 Hz and totaled just 10% additional mass to the fuselage panel weight. Figure 16 shows the transfer function comparison between the averaged fuselage panel velocity with and without the HG treatment.

Since the HG blanket was tuned to the 130 Hz resonance of the fuselage panel, a significant decrease in fuselage vibration results at that frequency when the HG is applied. In addition, resonances of the fuselage panel at higher frequencies are also attenuated despite the HG having been designed to target just 130 Hz. The attenuation at these higher frequencies can be attributed to the high frequency losses of the poroelastic material itself and that the embedded masses are capable of operating off-resonance due to this damping.

Product demonstration of HG Blankets

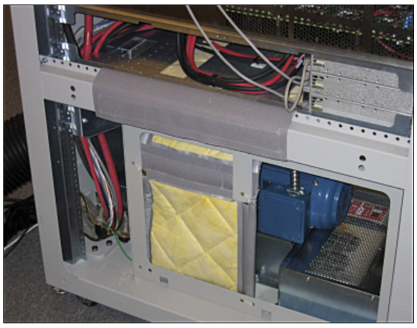
Tests were conducted to compare a commercial product based on HG technology with a conventional sound-proofing treatment (Kondylas et al. 2008). This HG design was ultimately termed "LoWave™" and was manufactured by DuPont™. The



Source: Idrisi et al. (2009)

Figure 16: Spatially average velocity of fuselage without treatment (solid line) and with HG blanket (dotted line).

conventional treatment was a mineral wool limp mass barrier. Both treatments were of comparable thickness and weight and were attached to a metal housing which contained an electric motor and fan assembly. Figure 17 shows the LoWave™ treatment attached to the component housing. The main objective of applying these treatments was to reduce radiated sound while not requiring the manufacturer of the equipment to alter the design of the metal housing.



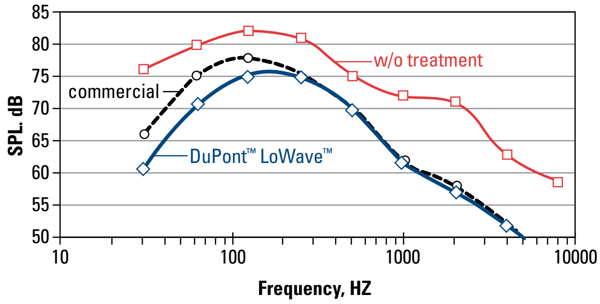
Source: Kondylas et al. (2008)

Figure 17: Electric motor housing with LoWave™ treatment attached at one location fitting with the existing form the case.

This specific LoWave™ was designed so as to enhance sound attenuation in the 50–200 Hz bandwidth. Figure 18 shows the results of measuring sound pressure level (SPL) in octave bands at a distance of 40 inches from the housing. At frequencies below 250 Hz, the LoWave™ treatment decreases sound radiation by an additional 3–5 dB when compared to the conventional sound proofing treatment. Since both the limp mass and LoWave™ treatments had similar total weight, this shows that the embedded masses present in the HG/LoWave™ design improve low frequency sound attenuation without additional weight.

CONCLUSIONS

Two new vibration absorber designs were considered for their low frequency passive sound attenuation. Distributed Vibration Absorbers [DVAs] which used two types of spring layers—continuous woven layer or a poroelastic material—have been constructed and thoroughly tested on a variety of real systems. DVAs were found to provide noticeable low frequency atten-



Source: Kondylas et al. (2008)

Figure 18: Test results comparing radiated sound levels from the housing without treatment (red plot), with the limp mass barrier (black dashed plot) and with the HG/LoWave™ design (blue plot).

uation, for instance, losses on the order of 10 dB at frequencies < 200 Hz, but also have been shown capable of providing global reduction of sound. Their compact and lightweight design makes them a robust solution for broadband noise and vibration control. DVAs have also demonstrated increased broadband damping above their tuning frequencies most likely due to an air squeeze damping effect.

Heterogeneous [HG] blankets constructed of randomly embedded masses in a poroelastic material have been studied and modeled. A prediction was made of the potential increase in transmission loss generated by the poroelastic material due to the presence of mass inclusions and this estimate was validated by experiment. HG blankets were then applied to real structures and were found to provide similarly great levels of low frequency attenuation, generally 10–15 dB of vibration reduction at the HG blankets' tuning frequencies. This vibration absorber design also contributed minimal mass to the main structure showing that they, too, are a lightweight noise control solution.

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