Passive distributed vibration absorbers for low frequency noise control

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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. Heterogeneous blankets consist 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 natural 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. The second treatment, distributed vibration absorbers, spread mass and spring elements 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. The paper will summarize the concepts, development and testing of both devices. Applications of the new treatments to realistic structures will be considered.

Primary subject classification: 47; Secondary subject classification: 43

1 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 less than 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, the Distributed Vibration Absorber [DVA], was originally developed by Fuller and Cambou[1] 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 Fig. 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.

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[2] in depth and has been applied to a variety of structures, for instance large cylindrical shells[3,4].

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 Fig. 2.

The second treatment, known as a heterogeneous

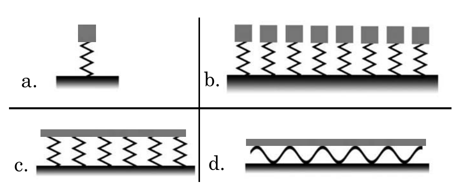

Fig. 1—Progress from point vibration absorber, (a) towards a continuous mass and continuous spring design as achieved in (d).

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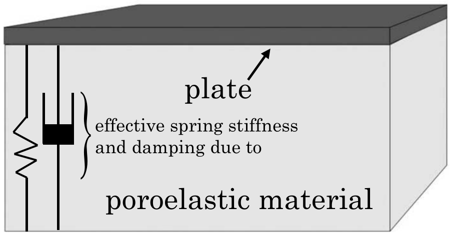

Fig. 2—Effective distributed vibration absorber resulting from interaction between continuous top mass and poroelastic material.

[HG] blanket, is composed of a poroelastic material into which a number of masses have been embedded. It was shown by Fuller et al.[5] 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 Fig. 3. Unlike a DVA designed with a poroelastic layer, HG blankets can target a range of tuning frequencies.

Idrisi[6] 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[5,7]. Kidner et al.[8] found that over 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.

While there are other works which summarize vibration absorber developments over the years, for example, Sun et al.[9], this paper aims to provide an organized review of the development and testing of passive DVA and HG technology to date. 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.

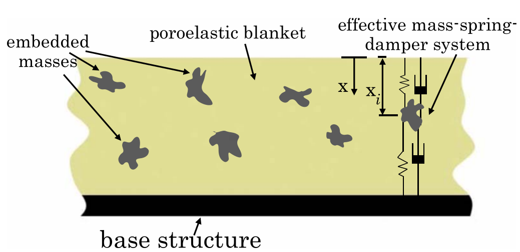

Fig. 3—Diagram of poroelastic with embedded masses, the HG blanket.

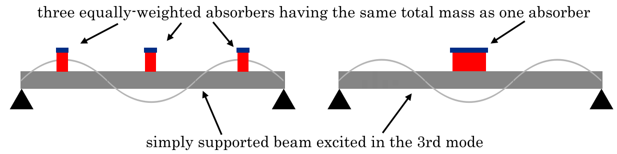

Fig. 4—Test of advantage of distributing vibration absorber mass.

2 DISTRIBUTED VIBRATION ABSORBERS

2.1 Development

The work of Smith et al.[10] was some of the earliest to discover and analyze the benefits of distributing dynamic absorbers over a continuous structure for increased attenuation. Cambou[11] 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 Fig. 5 shows the additional reduction of beam vibration resulting from the distributed absorber arrangement.

The DVA design Cambou[11] is composed of four, basic elements (Fig. 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

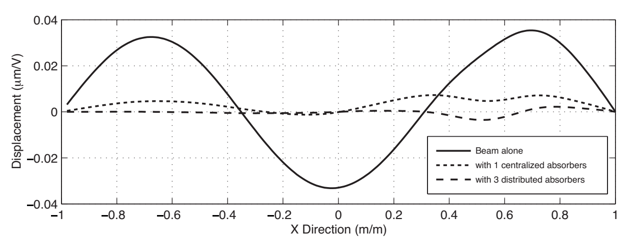

Fig. 5—Beam displacement without absorber (solid line), with one centralized absorber (dotted line) and with three distributed absorbers (dashed line)[11].

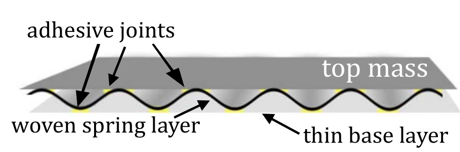

Fig. 6—The components of a DVA with woven spring layer.

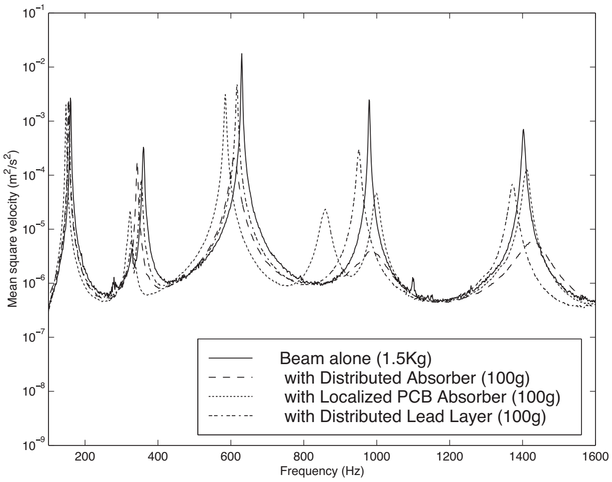

Fig. 7—Beam mean square velocity without treatment (solid line), with a DVA (dashed line), with a point vibration absorber (dotted line) and with the DVA mass simply adhered to the beam (dash-dash-dot line)[11].

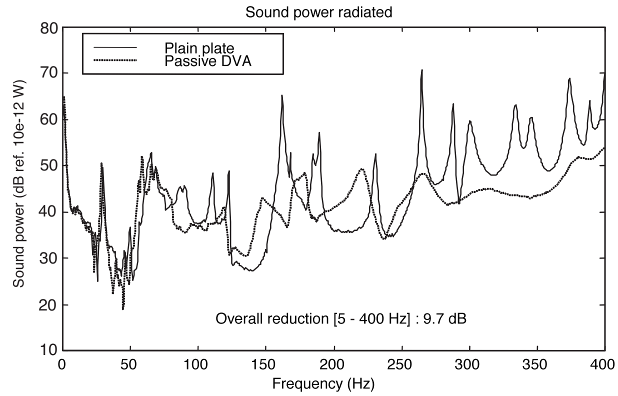

Fig. 8—Plate radiated sound power without treatment (solid line) and with DVA treatment (dotted line)[2].

the DAVA [Distributed Active Vibration Absorber] response, only passive versions will be considered here. 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 Fig. 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 Fig. 7 shows a dramatic decrease in beam vibration at this frequency while other beam modes are additionally attenuated.

Since the DVA was not tuned to all of the modes that it attenuated in Fig. 7, damping within the DVA design must play a role in allowing it to work off-resonance. Fuller et al.[12] 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. For example, welded joints would naturally be stiffer and less damped than if the DVA were bonded together with epoxy.

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[2] 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 Fig. 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".

2.2 Testing

An application of the DVA design using a poroelastic spring layer was attached to a vibrating plate in a transmission loss testing 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.

It is important to recognize that the DVA is itself a vibrating plate when it is serving to attenuate vibration of a host structure. However, given the much smaller dimensions of the DVA in relation to the main structure and given that the DVA is not baffled, the radiation efficiency of the DVA is very poor at low frequencies where it is vibrating the most due to re-actively canceling the structural motion beneath it. Thus, the DVA is

Fig. 9—Large payload shroud suspended in laboratory[3].

capable of reducing radiated sound power even though it is intended to serve primarily as a vibration absorber. These results are encouraging and the next logical extension of evaluating DVA performance was to apply a treatment to a real structure.

2.3 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 (Fig. 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, ref. 20 μPa.

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.[3] 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 predicted by Fuller et al.[12].

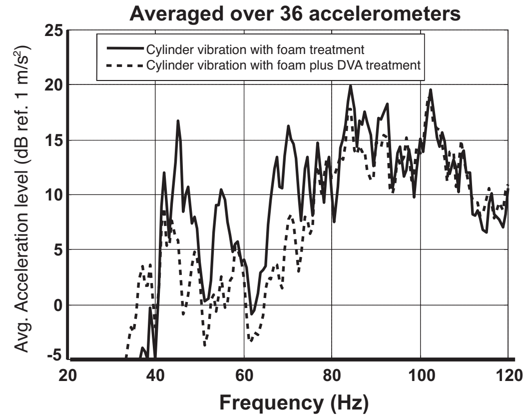

Fig. 10—Averaged acceleration level of the cylinder with the foam treatment (solid line) and with the additional DVAs (dashed line)[3].

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.

3 HG BLANKETS

3.1 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.[5] was some of the earliest to investigate the effects of embedded masses in an acoustic foam layer, the conception of which was termed a heterogeneous [HG] blanket. 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.

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 Fig. 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.

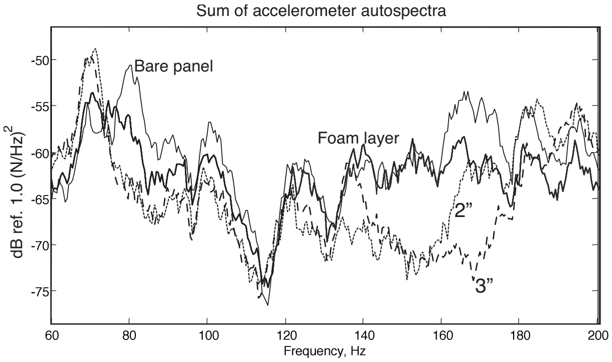

Fig. 11—Plate vibration without treatment (solid line), with a 3 in. layer of melamine foam (grey line), with a 2 in. thick HG blanket (dotted line) and with a 3 in. thick HG blanket (dashed line)[5].

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.

Practical efforts to assist in the design of HG blankets have been made. A parametric study of HG blanket features by Idrisi et al.[13] helped to decode many of the design variables in order to determine the resulting natural frequency of an individual embedded mass

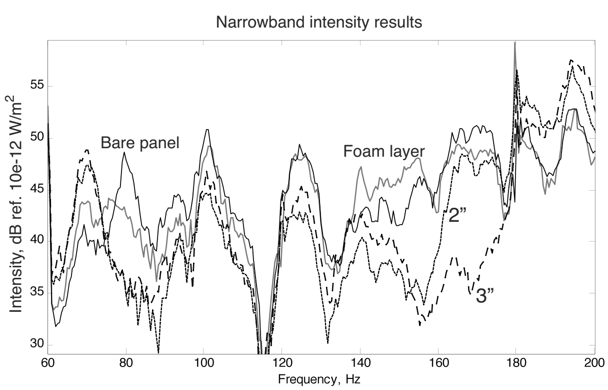

Fig. 12—Plate radiated sound intensity without treatment (solid line), with a 3 in. layer of melamine foam (grey line), with a 2 in. thick HG blanket (dotted line) and with a 3 in. thick HG blanket (dashed line)[5].

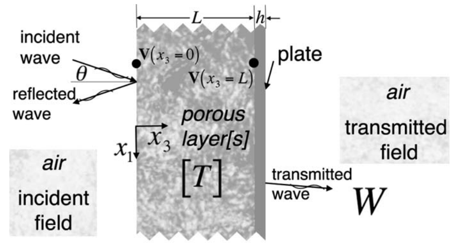

Fig. 13—Porous material attached to an infinite plate with incident acoustic field and transmitted field on side of the plate.

or array of masses. The study found that embedded masses interact locally within the poroelastic material with dependence on embedded depth, mass area "footprint", mass shape and the distance to nearby masses which all, in addition, help to determine unique tuning frequencies. These experimental results encouraged further evaluation of the embedded masses for optimum placement and form. Given the computational expense of a fully coupled 3D finite element model of poroelastic media with embedded masses[14], a simplified model is desired which may assist in HG blanket design and optimization[15].

In an effort to simulate an HG blanket, a model was developed from the work of Allard[16] to determine the sound transmission or insertion 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 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. The equations of motion used are those derived for the propagation of waves through fluid-saturated porous solids[17,18]. By determining the transmission matrix, [T], 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 Fig. 13.

This transmission matrix then relates the stresses and velocities from the incident field, $x_3=0$, to those present at the plate, $x_3=L$.

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

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

$$\mathbf{V}(x_3) = [v_1^s \quad v_3^s \quad v_3^f \quad \sigma_{33}^s \quad \sigma_{13}^s \quad \sigma_{33}^f]^T \quad (2)$$

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 operator. 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. For example, the diffuse field transmission loss can then be calculated,

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

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.

Extending the model to include the presence of embedded masses, Kidner et al.[19] found that one can assume the masses are an additional panel input impedance, Eqn. (4). This simplification can only be met by assuming that the embedded masses are evenly and densely distributed in both space and natural frequency, allowing their combined input impedance to be "spatially averaged", shown in Eqn. (5). This assumption therefore ignores localized effects of the masses but treats them as a lumped impedance.

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

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

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 $\eta_{a,n}$. This averaging method would not be compatible should the masses fail to meet the assumption of regular and even spacing in both geometric placement as well as in anticipated natural frequency of vibration.

By modifying the panel impedance present in $\mathbf{V}(x_3 = L)$, one can then estimate the sound transmission or insertion loss through a poroelastic material with a large number of mass inclusions, similar to that of an HG blanket attached to a plate. Then, a model was constructed to compare the insertion loss through a 70 mm thick foam sheet attached to a 1.6 mm thick, lightly damped panel and through an HG blanket composed of the same foam sheet with embedded masses tuned from 60–140 Hz, also attached to the same aluminum panel. The masses represented 11% of the mass of the aluminum panel. The foam material and panel had properties as provided in Table 1.

Table 1—Material properties as used for the simulation in Sec. 3.1.

Foam layer			
Thickness [m]	Density $[\mathrm{kg \cdot m^{-3}}]$	Bulk modulus [kPa]	Flow resistivity $[\mathrm{kPa \cdot s \cdot m^{-2}}]$
0.07	30	79	11.89
Panel			
Thickness [m]	Density $[\mathrm{kg \cdot m^{-3}}]$	Young's modulus [Pa]	Poisson's ratio
0.0016	2100	72e9	0.30

A comparison of the insertion loss predictions for these treatments is shown in Fig. 14. A significant increase in insertion loss is predicted for the HG blanket over the range of its embedded mass tuning frequencies. At the 100 Hz one-third octave band, the HG blanket yields an IL of 5.2 dB greater than the foam layer on its own. However, above this tuning frequency bandwidth, the model predicts a gradual convergence of insertion loss such that there exists little to no difference in IL at 1 kHz. Thus, to justify the added mass of the HG blanket for improving noise control performance, it is advantageous to embed masses over the full span of thickness available such that the broadest "tuning" of the masses is achieved.

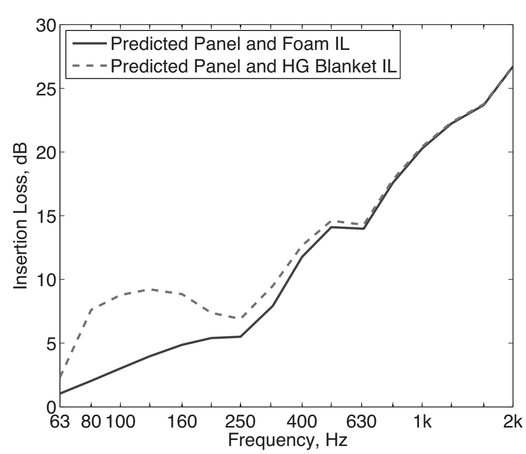

Fig. 14—Predicted insertion loss comparison between foam layer (solid line) and HG blanket (dashed line) attached to an aluminum panel.

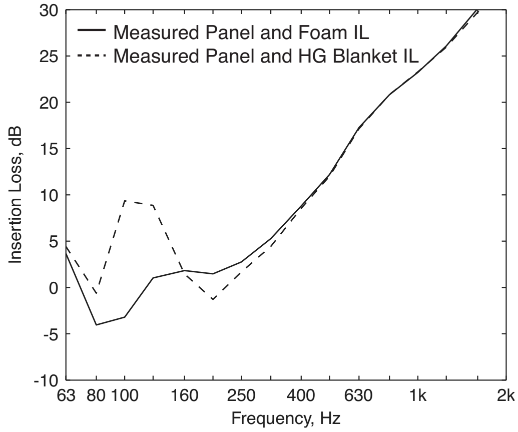

Fig. 15—Insertion loss of the foam layer and panel (solid line) and of the HG blanket and panel (dashed line)[8].

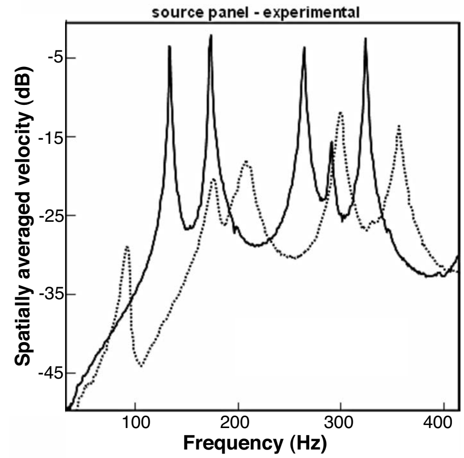

Fig. 16—Spatially average velocity of fuselage without treatment (solid line) and with HG blanket (dotted line)[7].

To validate the results of this model, an experiment was carried out by Kidner et al.[8] 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. The foam material and panel had properties given in Table 1, as was modeled earlier. Figure 15 plots the measured values of insertion loss for the two cases: the panel and foam material, the panel and HG blanket.

A decrease in insertion loss occurs for the panel and foam at the 80 and 100 Hz one-third octave bands but a steady increase in IL is evident as the measured frequency is increased. The HG blanket showed a dramatic improvement in IL over its range of tuning frequencies as compared to just the foam layer and panel. At the 100 Hz one-third octave band a 15 dB improvement in IL is measured. An interesting decrease in IL is measured for the 200 Hz one-third octave band suggesting there may be some coupling present between the embedded masses and foam material. Overall, a significant improvement in low frequency insertion loss results from the additional embedded masses which represented an 11% ratio of mass to the panel.

The results of Fig. 15 match closely with those predicted from the model, as plotted in Fig. 14. The model, in fact, under-predicted the usefulness of the additional masses of the HG blanket; for example, the experiment measured a 15 dB increase in IL at the 100 Hz one-third octave band while the simulation estimated an increase in 5.2 dB.

Nevertheless, the dynamics of the experiment are captured in the model with the exception of the potential coupling effect which occurred in the experiment in the 200 Hz one-third octave band. This difference aside, the in-depth study of natural or "tuning" frequency parameters by Idrisi et al.[13] and their relationship with embedded mass input impedance, Eqn. (5), allows the designer to quickly model the utility of added masses to a poroelastic sheet to serve as a low frequency noise control device.

3.2 Testing

In addition to performing much of the fundamental work to determine HG blanket tuning characteristics and specific designing parameters, Idrisi[6] 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.

3.3 Product Demonstration of HG Blankets

Tests were conducted to compare a commercial product based on HG technology with a conventional sound-proofing treatment[20]. This HG design was

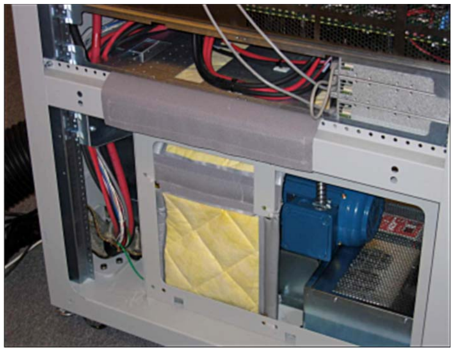

Fig. 17—Electric motor housing with LoWave™ treatment attached at one location fitting with the existing form the case[20].

ultimately termed "LoWave™" and was manufactured by DuPont™. LoWave™ uses a robust elastic layer for the embedded masses to avoid issues of wear by fatigue as the masses oscillate.

For the tests, the comparable conventional treatment was a mineral wool limp mass barrier. Both treatments were of equal thickness and weight and both 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.

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.

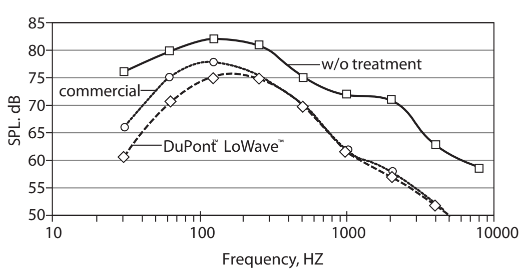

Fig. 18—Test results comparing radiated sound levels from the housing without treatment (solid line with squares), with the limp mass barrier (dotted line with circles) and with the HG/LoWave™ design (dashed line with diamonds)[20] *SPL ref 20 μPa.*

4 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—poroelastic materials or continuous woven layers—have been constructed and thoroughly tested on a variety of real systems. DVAs were found to provide noticeable low frequency attenuation, for instance, losses on the order of 10 dB at frequencies less than 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.

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.

5 ACKNOWLEDGMENTS

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