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Journal of Sound and Vibration 294 (2006) 466–472
JOURNAL OF SOUND AND VIBRATION
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Increase in transmission loss of single panels by addition of mass inclusions to a poro-elastic layer: Experimental investigation ☆

M.R.F. Kidner[a,*], C.R. Fuller[a], B. Gardner[b]

[a] Vibration and Acoustics Laboratory, Virginia Polytechnic Institute and State University, Blacksburg, VA 24060, USA
[b] ESI US R&D, 12555 High Bluff Drive, Suite 250, San Diego, CA 92130, USA

Received 4 April 2005; received in revised form 25 October 2005; accepted 7 November 2005
Available online 24 January 2006

Abstract

The insertion loss of standard acoustic blankets can be significantly improved at low frequencies by the addition of randomly placed mass inclusions to the poro-elastic layers. The improvement is much greater than that due to the mass effect alone. The mass inclusions act as resonant systems and so increase the structure impedance. This paper reports the results of experimental investigations into this phenomenon. Increases in insertion loss of 15 dB in the 100 Hz third octave band are reported.

1. Introduction

The transmission of noise into payload fairings has become an increasingly important issue in the design of launch vehicles. This is primarily due to the increase in size of fairings and reduction in mass of the faring walls [1,2]. This reduction in mass has come about through the use of composite materials. Acoustic blankets are placed inside the fairing of most launch vehicles to reduce the noise level to which the spacecraft or payload is subjected. These blankets provide three primary functions:

(1) Reduction of the transmission of energy into the payload bay (measured by transmission loss).
(2) Increase in the acoustic absorption of the payload bay (measured by absorption coefficient).
(3) Addition of damping to the fairing (measured by structural damping loss factor).

Many blankets currently used in launch vehicles use a mass-law and absorption approach, this has fundamental limits at low frequencies. The development of an advanced blanket, to specifically reduce the vibration level of the payload by 3 dB in the 200–250 Hz band, was reported by Hughes et al. [3]. Exterior take-off levels for the Titan IV of 140–143 dB from 40 to 300 Hz were reported. The level of the spectrum then drops to 130 dB at 400 Hz. Internal levels of 10–13 dB less from 80 to 200 Hz were reported with the blanket in

☆ This work was supported by the SBIR program under AFRL and The ESI Group.
* Corresponding author. ANVC group, Department of Mechanical Engineering, University of Adelaide, Adelaide, SA 5006, Australia.
E-mail address: michael.kidner@adelaide.edu.au (M.R.F. Kidner).

0022-460X/$ - see front matter
doi:10.1016/j.jsv.2005.11.022

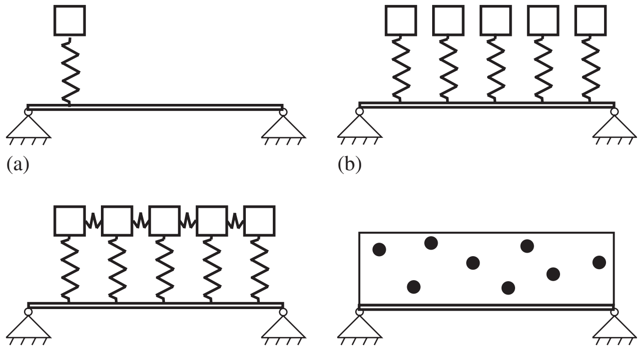

Fig. 1. Development of the mass-inclusion concept. The point absorber (a), was extended to multiple point absorbers (b), which was then extended to multiple coupled oscillators (c), and finally to a continuous distribution of fully coupled masses embedded in an elastic layer (d).

place. At 400 Hz the internal level had dropped to 20 dB below the untreated level. However, the level of the internal spectrum at lower frequencies was still in excess of 110 dB and as such emphasises that the improvement in the blanket designs are required in this frequency range.

The application of vibration absorbers and Helmholtz resonators to this problem has been investigated by Estève et al. [4]. It was shown that a significant decrease in the noise transmitted into the faring could be achieved with very small additional mass, compared to the existing acoustic treatments. This paper looks at extending the absorber approach to a random distribution of masses within an absorbent layer, hence fully integrating the discrete additional impedances with the mass law plus absorption concept.

In line with the concept proposed by Maidanik [5] the behaviour of a number of mass inclusions in a poro-elastic layer is considered. Fig. 1 illustrates the development of this idea from the traditional point absorber (Fig. 1a). The use of a single point absorber was extended to multiple absorbers acting over a distributed space (Fig. 1b). An example of this is the work done by Brennan [6]. Coupling between the absorber masses was then considered, as shown in Fig. 1c. This coupling has been discussed in work on the effective impedance of collections of oscillators (which is closely tied to work on statistical mechanics) [7,8]. The extension to continuous elastic layer with embedded masses (Fig. 1d) comes from work by Marcotte et al. [9] and Estève et al. [4] on distributed vibration absorbers (DVAs).

Following this introduction the second section of this paper investigates the behaviour of single and multiple, mass inclusions in a poro-elastic layer. In the third section the effect of the poro-elastic layer with mass inclusions on the insertion loss of a composite panel is discussed. Experimental results for the insertion loss are then presented.

2. Behaviour of mass inclusions

In this section, the variation of inclusion resonant frequency versus inclusion depth in layers of melamine foam is described. The work done on DVA [10] has shown that $f_n \propto 1/\sqrt{t}$ for a plate on a layer of foam. This variation in the resonant frequency, f_n as a function of the layer thickness, t, is to be expected from the longitudinal stiffness of a rod $k = EA/L$, which varies inversely to the rod length. E is the Young's modulus, A the area, and L the length of the rod. Using this simple model an equivalent mass–spring system was developed for the DVA. Fig. 2 shows the variation in resonant frequency and damping ratio as a function of layer thickness. Using this as a starting point it was assumed that the inclusion resonant frequency would vary in a similar way.

2.1. Experimental investigation of the properties of a single inclusion

The input apparent mass of a single inclusion in a 50 mm block of foam rubber was measured using the set-up shown in Fig. 3. Steel ball bearings with a mass of 8 g and diameter of 9 mm were used. The foam rubber

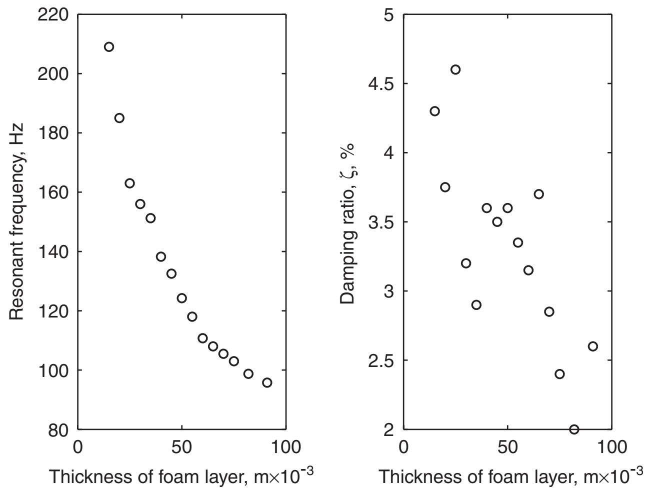

Fig. 2. Variation of resonant frequency and damping ratio as a function of layer thickness for a plate on a column of foam.

Fig. 3. Experimental set-up to measure inclusion apparent mass.

was Wiltec^{TM} type from Illbruck Corporation. The foam rubber cube was glued to a stiff honeycomb platform (2 g). A shaker with a B&K impedance head was used to drive the platform and the mass in the foam acted as a mass–spring oscillator. The stiffness element is dominated by the stiffness of the elastic frame. The impedance head had 28.8 g of mass above the force gauge (acting with the platform mass). The results were corrected for the mass of the platform and the downstream mass of the impedance head. The apparent mass of the foam cube with a single inclusion is shown in Fig. 4. The peak in the apparent mass occurs at 125 Hz, this corresponds to the resonant frequency of the inclusion in the foam layer. The damping ratio ζ, was estimated using the half-power point method and found to be approximately 3.5%.

2.2. Multiple inclusions

A number of inclusions were added to a foam layer to determine when the impedance became a smooth function of frequency over a wide bandwidth. The apparent mass of poro-elastic layer attached to a light and

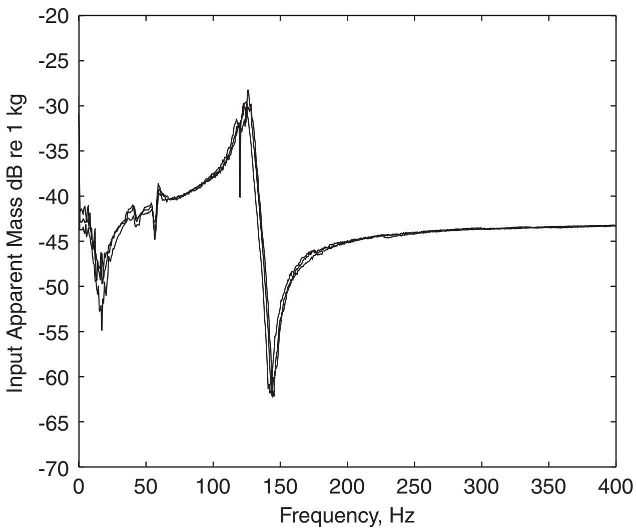

Fig. 4. Input apparent mass to five melamine foam cubes with 8 g ball bearing at their centre.

stiff base plate is shown by the dotted line Fig. 5. The figure also shows the effect of the addition of 10 mass inclusions with resonant frequencies distributed over the 95–105 Hz band (solid line). The added unsprung mass effect has been calculated and is shown by the dashed line. It can be seen that over the targeted frequency band, 95–150 Hz, the increase in apparent mass is in excess of that expected from the additional unsprung mass. Much less additional mass is therefore required to achieve attenuation over a specified bandwidth. Inclusions should be used to target a troublesome bandwidth in the transmission loss by increasing the structural apparent mass over that band. The resonance of the structure combined with the collection of oscillators causes a decrease in the structural apparent mass over a bandwidth at higher frequency. The net improvement over a band encompassing both regions is small. However, as was seen with the Cassini blanket [3], it is often required to focus the improvement in transmission loss at a certain bandwidth in which the launch vehicle fairing has poor performance compared to the required standards. In this context, the use of inclusions may be beneficial because of their band specific effect and low mass. By considering the excitation spectrum and the sensitivity of the payload to different frequencies an optimal transmission loss criteria can be developed.

Careful use of vibro-acoustic modelling techniques such as statistical energy analysis and the fuzzy structure model are important for this. For the mass inclusions to be included within such a modelling framework it is required that the validity of a statistical description of the mass inclusions is verified. The experimental results shown here are compared with the closed form solution for the apparent mass of a number of oscillators proposed by Strasberg [11]

$$A = \omega^2 \frac{m_0}{2\pi}\left[\mathrm{i}\Omega - \frac{1}{\sqrt{1+\mathrm{i}\eta}}\arctan\left(\mathrm{i}\Omega\sqrt{1+\mathrm{i}\eta}\right)\right]_{\Omega^-}^{\Omega^+}, \tag{1}$$

where $\Omega = \omega_0/\omega$, the ratio of the oscillator resonant frequency to the frequency of excitation, ω. The mass, m_0, is written as a mass per Hz, and η is the loss factor.

In this case the frequency range was from 95 to 150 Hz and a mass per Hz of 1.45 g/Hz was assumed. (The experimental result shown used 10 8 g ball bearings over a range of tuned frequencies from 97 to 142 Hz.) Assuming the resonant frequencies of the inclusions to be evenly distributed leads to a value of 5.5 Hz per oscillator. The damping ratio is 3% which gives a minimum and maximum half-power bandwidth of 5.5–8.5 Hz. This suggests that the Strasberg approximation should be valid over this frequency range. It can be seen from the figure, however, that with only 10 inclusions over the bandwidth the value of the apparent mass

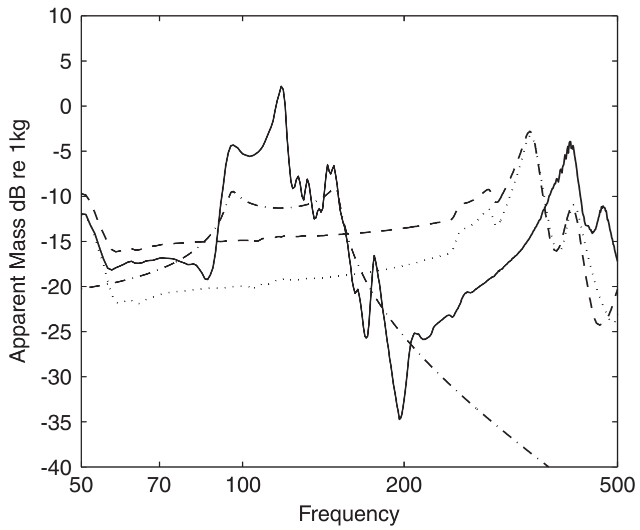

Fig. 5. Input apparent mass with and without multiple inclusions for melamine foam. Base plate and foam only, no inclusions (······). Base plate, foam plus 10 mass inclusions tuned from 97 to 142 Hz (——). Calculated effect of the inclusion mass only (– – ––). Prediction after Strasberg [11] (– · – · – · –).

has not converged to that predicted by Strasberg. This could be due to the resonant frequencies of the inclusions being unevenly distributed over the bandwidth.

The reduction in apparent mass from 150 to 400 Hz is expected for the attachment of a vibration absorber to a rigid mass. The bandwidth of this reduction is large. For a single vibration absorber on a rigid mass the frequency separation of the maximum and minimum apparent mass is a function of the mass ratio [12]. The mass ratio is the ratio of the absorber mass to the rigid mass.

3. Experimental results for changes insertion loss and plate vibration due to mass inclusions

The effect of the mass inclusions on insertion loss of a panel and foam layer was measured. The source room contained two loudspeakers both driven with the same white-noise signal, band limited from 40 to 3000 Hz. Two microphones in the source room were used to monitor the incident SPL to provide consistency between tests. On the receiving side, the plate radiated into a highly damped enclosure, $1.5 \times 1.5 \times 1.5$ m in size. The radiated intensity was measured over the surface of the plate using a B&K intensity probe (type 3520).

Measurements were made for the following three configurations; the bare plate, the plate plus a layer of poro-elastic material, the plate plus a layer of poro-elastic material containing 50 randomly located mass inclusions.

The 1.11×1.11 m composite plate, entirely filled the window between the two rooms. It has a mass of 3.4 kg. The poro-elastic layer was a 0.07 m thick layer of Wiltec^{TM} foam made by Illbruck. Density and bulk modulus are estimated to be 30 kgm^{-3} and 79×10^3 Nm^{-2}. A value of 11.89×10^3MKS Rayk/m was provided by the manufacturer as its flow resistivity. The mass inclusions were steel ball bearings each with a mass of 8 g.

3.1. *Insertion loss results*

The solid line of Fig. 6 indicates the insertion loss values for the poro-elastic layer. As expected the insertion loss of the layer increases with the frequency. The insertion loss at lower frequencies, the 100–200 Hz band, can be correlated to increased structural damping rather than dissipation of acoustic energy. The plate vibration results discussed in Section 3.2 support this.

The addition of the mass inclusions produce a peak in the insertion loss that covers the 80–125 third octave band. The increases in these bands are 4 dB, 15 dB and 9 dB, respectively. From 160 to 315 Hz the addition of

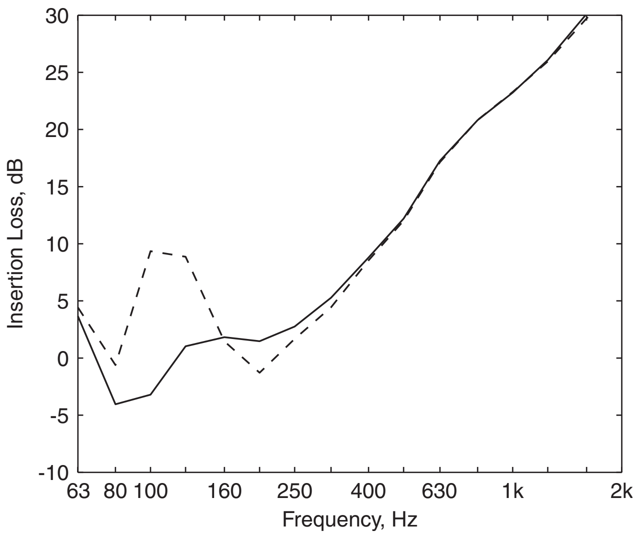

Fig. 6. Insertion loss for the foam layer (—–) and the foam layer with mass inclusions (– – ––).

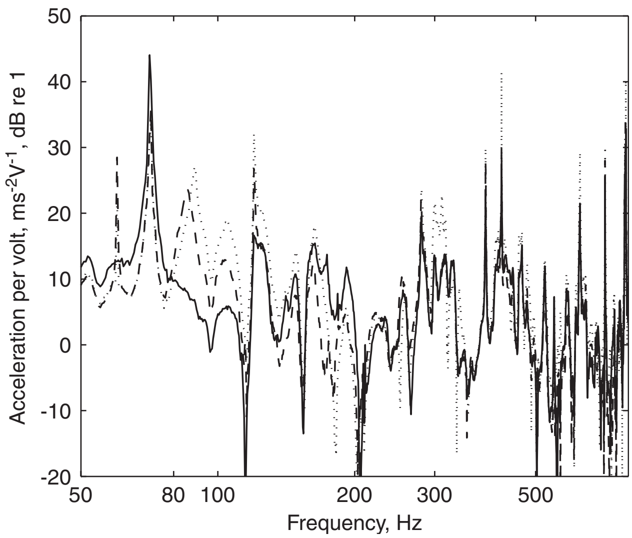

Fig. 7. Acceleration per volt for three plate configurations. Plate alone (······). Plate and foam layer (- - - -). Plate, foam layer and mass inclusions (—–).

mass inclusions reduces the insertion loss, but only by a small amount (3 dB at 200 Hz being the worst case). This is due to the increased mobility of the plate in this band due to the interaction with the mass inclusions. The vibration response data shown in Fig. 7 supports this conclusion.

3.2. Vibration response results

The plate was instrumented with 16 PCB accelerometers (type A352B65) at a spacing of 0.27 m. The transfer function from the loudspeaker drive signal to these was measured. Fig. 7 shows the level of these transfer functions as a function of frequency for the three measurement configurations. The addition of the poro-elastic layer, as shown by the dashed line, adds some damping to the plate. The plate vibration in the 70–100 Hz range is significantly reduced by the addition of mass inclusions to the poro-elastic layer, shown by the solid line. Reductions of 10–20 dB are shown. The peak reductions occur at the two plate modes within the targeted bandwidth. These peaks are at 80, 100 and 120 Hz and show reductions of 21, 13 and 15 dB.

The expected increase in vibration level occurs over the 150–200 Hz band. The level increases are smaller than the reductions achieved over the target bandwidth. Only 2–3 dB is added to the peaks, this counteracts the reductions achieved by the foam layer. The minima in the band are increased from −15 to −5 dB, although this contributes very little to the overall vibration level.

At frequencies above 200 Hz, the mass inclusions have little effect and the solid and dashed lines converge. The foam layer reduces the vibration level at resonant peaks by 6–10 dB.

4. Conclusions

An insertion loss increase of 15 dB has been shown in the 100 Hz third octave band. This corresponds to reductions in measured vibration of 20 dB. Significant improvements in the insertion loss of a blanket treatment over a bandwidth can be achieved by using resonant inclusions. Some reduction in insertion loss is apparent at higher frequencies but it is much less than the increases in the targeted bandwidth. Careful design of the poro-elastic layer may be able to reduce the effect. The behaviour of the mass inclusions have been experimentally determined. A simple relationship between inclusion position and resonant frequency has been shown. The effect of inclusion position on damping coefficient has shown to be less significant.

Acknowledgements

The authors would like to acknowledge the support of Dr. Steven A. Lane of the Air Force Research Lab Kirtland AZ.

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