inter•noise 96
25th Anniversary Congress – Liverpool

INCE: 47.1

APPLICATION OF AN ELASTIC SANDWICH DAMPING TREATMENT AT HIGH TEMPERATURE ENVIRONMENT

R J Pinnington (1) & L C Chow (2)

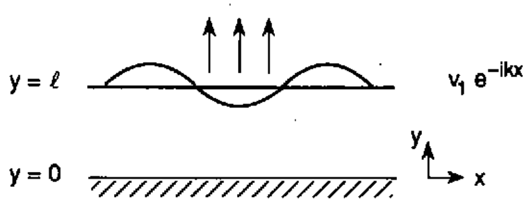
(1) ISVR, University of Southampton, Southampton, UK, (2) British Aerospace Airbus Ltd, Filton, Bristol, UK

1. INTRODUCTION

The most widely available damping treatments for the reduction of resonant vibration of components employ viscoelastic materials. These materials are rather limited for high temperature use, and so other damping treatments are sought. One method is to use ceramic fibre heat insulating material as a core of damping material between two parallel metal plates. The damping mechanism is probably friction between fibres and viscous losses due to gas pumping. In this study the damping of a plate is predicted when a sandwich of a ceramic fibre interlayer and a top plate are applied. Theoretical predictions are then compared with experiments.

2. THEORY

The plates are assumed to be of infinite extent to allow the loss factor to be extracted from the radiation of energy from a travelling wave into the treatment. This was quantified by the surface impedance of the treatment.



The interlayer was modelled as a linear elastic medium with both shear and bulk deformation. To simplify the prediction for a ceramic fibre core, a second simplified model was used in which shear (or rotational) effects were neglected. It was assumed that the top plate (or secondary plate) was thinner than the bottom plate (or primary plate) to which the excitation was applied. The free flexural wavelength difference in the two plates permitted the secondary plate to be regarded simply as a distributed mass.

It can be shown that the input impedance Z_{11} and transfer impedance Z_{12} of an elastic layer of infinite extent and of thickness ℓ required to couple to adjoining structures are [1]:

$$Z_{11} = i\omega e\left[k_\psi k^2 \sinh(k_\varepsilon \ell) - k_\psi^2 k_\varepsilon \sinh(k_\psi \ell)\right]/D \qquad (1)$$

$$Z_{12} = i\omega e\left[k_\psi k^2 \cosh(k_\psi \ell)\sinh(k_\varepsilon \ell) - k_\varepsilon k_\psi^2 \sinh(k_\psi \ell)\cosh(k_\varepsilon \ell)\right]/D \quad (2)$$

where $D = 2k^2 k_\varepsilon k_\psi\left[\cosh(k_\varepsilon \ell)\cosh(k_\psi \ell) - 1\right] - k_\varepsilon^2 k_\psi^2 \sinh(k_\varepsilon \ell)\sinh(k_\psi \ell) - k^4 \sinh(k_\varepsilon \ell)\sinh(k_\psi \ell)$,

$k_\varepsilon^2 = k^2 - k_1^2$, $k_\psi^2 = k^2 - k_2^2$

and $k_1^2 = \omega^2\rho/(\lambda + 2\mu)$ and $k_2^2 = \omega^2\rho/\mu$,

with $\lambda = \dfrac{\nu E}{(1+\nu)(1-2\nu)}$ and $\mu = \dfrac{E}{2(1+\nu)}$.

If the elastic material is relatively soft, the imposed wavenumber k will be much less than k_1 and k_2, and equations (1) and (2) will reduce to the impedance of one-dimensional waves in a rod of length ℓ, i.e.

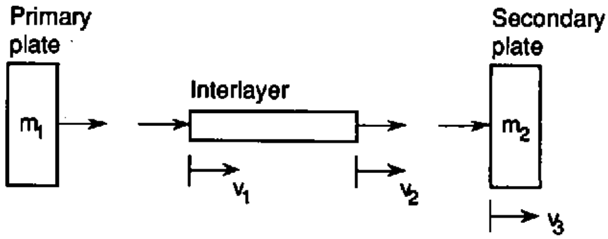
$$Z_{11} \approx Z_\ell \cos k_1\ell / i \sin k_1\ell \quad , \quad Z_{12} \approx Z_\ell / i \sin k_1 \ell \qquad (3), (4)$$

where $Z_\ell = \rho c_\ell$ and $c_\ell = \sqrt{\lambda + 2\mu/\rho}$. Z_ℓ is the surface impedance of an infinitely thick layer which is subjected to plane excitation over the plate surface. In these circumstances there are only dilatational waves present; as in a fluid medium, shear is unimportant.

The loss factor of the coupled sandwich system, in which parallel plates of mass per unit area m_1 and m_2 are separated by an interlayer, may be calculated by using the four-pole parameter representation as given in the block diagram shown below. It can be shown that the loss factor η of the system can be calculated as [1].

$$\eta = \frac{\mathrm{Re}\,\{Z_t\}}{\omega\left(m_1 + m_2\left|\frac{V_2}{V_1}\right|^2\right)} \qquad (5)$$

where $Z_t = Z_{11} - Z_{12}^2/(Z_{11} + i\omega m_2)$, and $\dfrac{V_2}{V_1} = -\dfrac{Z_{12}}{(Z_{11} + i\omega m_2)}$.



3. EXPERIMENTAL VALIDATION

An experimental rig was made to measure the dynamic properties of the ceramic fibre material used in the experiments. The sample dynamic stiffness was obtained from the transmitted blocked force and the input acceleration at the top. The real and imaginary parts of the dynamic stiffness $\bar{K}$ yield the longitudinal elastic modulus and loss factor, $\bar{K} = K + i\eta K$. The term 'elastic modulus' is used rather vaguely, as these ceramic materials have a low Poisson's ratio and so the elastic moduli of a rod and an infinite solid are similar. These data were used for the estimation of the loss factor.

Figure 1 shows the comparison of loss factor predicted using the exact impedance expressions (equations (1) and (2)) and the approximate expressions (equations (3) and (4)). The first peak at 100 Hz corresponds to the spring-mass resonance of the interlayer and secondary plate. The higher resonances are associated with standing waves in the interlayer. The difference occurs at low frequency (below 20 Hz) because of the allowance for shearing effects in the interlayer. At low frequencies the bending wavenumber k becomes comparable with the wavenumbers k_1 and k_2, and so the approximate solution is not strictly valid.

Figure 2 displays the measured and predicted loss factor of plates (with outer dimensions of 0.5 × 0.5 m) with ceramic fibre interlayer. The measurements do not display the resonant features of the prediction. It was thought that no resonances occurred because the ceramic fibre material density varied substantially over the area. This means that each part of the system would have local resonance frequencies leading to a net smearing of the total loss factor into a spatial average.

4. CONCLUSIONS

A model was developed to calculate the loss factors of two plates separated by an elastic interlayer. This model would be useful for any light core material such as elastomer or polymer foams. The core can support both dilatational and shear waves, but the dilatational waves were found to be more important. This permitted a simplified expression for damping to be developed, for which it was suggested that peak damping was associated with the resonances across the core thickness.

5. REFERENCE

[1] L.C. Chow et al, Journal of Sound and Vibration, 184(2), 299-310 (1995).

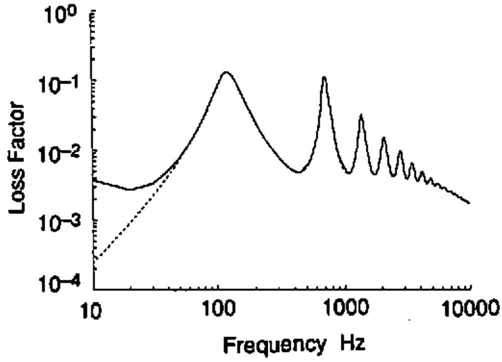


Figure 1. Theoretical loss factors according to the exact (—) and approximate (– – –) formulae. Layer thickness 12 mm; secondary plate thickness 1.5 mm.

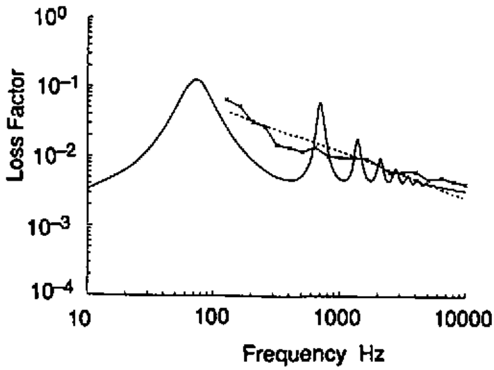


Figure 2. Measured and predicted loss factors of plates with ceramic fibre material as interlayer. -*-, measured; —, predicted values including radiation and material losses (progressive wave solution); – – –, predicted values for infinite medium. Ceramic layer thickness 16 mm; excited plate thickness 12.5 mm; attached plate thickness 5.0 mm.