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Transmission loss of a panel with an array of tuned vibration absorbers

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ABSTRACT

This paper presents a numerical model for the calculation of the transmission loss of a panel with an array of tuned vibration absorbers attached. The transmission loss of the panel is calculated for the case of the bare panel, with tuned vibration absorbers attached, and equivalent blocking masses. The theoretical predictions of transmission loss are compared with experimental measurements.

INTRODUCTION

Many noise control applications require panel partitions with high transmission loss and low weight. Partitions with high transmission loss are characterised by a high surface density, which can result in excessive weight. For weight-critical aerospace applications, the judicious use of stiffening ribs is often employed. The optimisation goal for these applications is to obtain a configuration that achieves the highest transmission loss for an acceptable weight penalty.

It was shown in previous conference papers by the author (Howard & Kidner 2006, Howard 2007) that the attachment of discrete masses to a panel can result in transmission loss results greater than merely increasing the thickness, and hence the surface density, of the panel by an amount that would result in the same total weight. The work presented here is an extension of the previous work, by including comparisons of theoretical and experimental results of the transmission loss of a panel with an array of cantilever beams attached that act as vibration absorbers. A Polytec Scanning Laser Vibrometer was used to measure the vibration of the cantilever absorbers during transmission loss tests.

The first part of this paper describes a mathematical model to predict the transmission loss of a panel with discrete masses and / or single degree of freedom oscillators. This model is used to predict the transmission loss for three cases namely a bare panel, a panel with an array of 49 discrete 'blocking' masses, and a panel with 49 single degree of freedom oscillators. In the second part of this paper, numerical predictions of transmission loss are compared with experimental results for these three cases.

PREVIOUS WORK

Previous work has considered the vibration reduction of panels due to the addition of an array of blocking masses and a literature review covering this topic is described in Howard (2007).

The use of secondary oscillators attached to a primary structure to reduce vibration or improve transmission loss has also been considered. Ormodroyd and den Hartog (1928) describe the theory of the optimisation of dynamic vibration absorbers, and this vibration control technique remains popular today. More recently, Liu and Liu (2005) describe alternative methods for the optimisation of their parameters. Nishida and Koopman (2007) describe methods for designing an absorber with multiple resonances to reduce the vibration of a beam. Relevant to the work presented here, Huang and Fuller (1997) used dynamic absorbers to reduce interior noise within a cylindrical shell. The results showed that the addition of dynamic absorbers if correctly positioned can successfully reduce the vibration of the shell and the interior acoustic pressure of the sound field enclosed by the shell. This technique has been used in real aircraft (Halvorsen and Emborg 1983, Waterman et. al. 1989). Su et.al (1996) used statistical energy analysis to examine the attachment of six dynamic vibration absorbers to a stiffened aircraft panel, all tuned to 101Hz, and found that there was improvement in the transmission loss. However the work here involves the investigation of vibration absorbers to improve the transmission loss due to broadband acoustic excitation, of which there has been little work.

The following section describes the development of a mathematical model to enable the prediction of the transmission loss of a panel with tuned vibration absorbers.

MATHEMATICAL MODEL

The mathematical model for the prediction of the transmission loss of a panel incorporating the effects of the addition of an array of lumped masses or oscillators is similar to previous work by Howard (2007) and involves calculating the:

- modal forcing vector due to pressure loading on a panel from an incident plane wave;
- vibration response of the panel, including the effects of the attached devices;
- sound pressure radiated from the panel and integrating the results to determine the radiated sound power; and
- transmission loss of the panel as the ratio of the incident to radiated sound power.

Incident Plane Wave

Consider an acoustic plane wave of pressure amplitude P_i incident at angles θ and ϕ on a simply supported panel with edge lengths L_x and L_y, as shown in Figure 1. The pressure that is incident on the panel $P(x,y)$ is given by

$$P(x,y) = P_i \exp[\,j(\omega t - kx\sin\theta_i\cos\phi_i - ky\sin\theta_i\sin\phi_i)] \quad (1)$$

This can be written as a modal pressure that acts on the panel and is calculated by multiplying the pressure distribution by the mode shape matrix of the structure $\mathbf{\Psi}$, and dividing by the modal mass matrix of the structure $\mathbf{\Lambda}$ (for consistency with later equations) as $\mathbf{\Psi}P(x,y)/\mathbf{\Lambda}$, and can be written as $p_{m,n} = L_x L_y \overline{Y}_m \overline{Y}_n$ where

$$\overline{Y}_m = (m\pi)\frac{1-(-1)^m e^{-j\alpha}}{(m\pi)^2 - \alpha^2} \quad (2)$$

$$\overline{Y}_n = (n\pi)\frac{1-(-1)^n e^{-j\beta}}{(n\pi)^2 - \beta^2} \quad (3)$$

$\alpha = kL_x\sin\theta\cos\phi$, $\beta = kL_y\sin\theta\sin\phi$, $k = \omega/c$ is the wavenumber, ω is the frequency, c is the speed of sound in air. This modal force (pressure) can be applied to the dynamics of the panel to calculate the structural modal participation factors w_p.

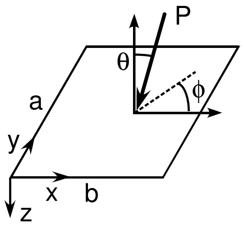

Figure 1: Coordinates for an incoming acoustic plane wave striking a simply support rectangular panel.

Vibration of a Simply Supported Panel

The displacement w of a simply-supported panel at position (x,y) can be written as an infinite sum of its vibration modes as

$$w(x,y) = \sum_{m=1}^{\infty}\sum_{n=1}^{\infty} w_{m,n}\sin(m\pi x/L_x)\sin(n\pi y/L_y) \quad (4)$$

where $w_{m,n}$ are the modal participation factors. The equations of motion for the panel can be written as (Soedel, 1993)

$$\ddot{w}_p + 2\xi_p\omega_p\dot{w}_p + \omega_p^2 w_p = \Gamma_p \quad (5)$$

where w_p is the p^{th} modal participation factor, ξ_p is the viscous damping coefficient of the shell at the p^{th} mode, ω_p is the resonance frequency of the p^{th} mode, and Γ_p is the p^{th} modal force which is applied to the panel for and is defined as

$$\Gamma_p = \frac{1}{\rho_p h N_p}\int_0^{L_x}\int_0^{L_y}\left[q_z U_{zp} + T_x U_{zp} + T_y U_{zp}\right]dy\,dx \quad (6)$$

where q_i and T_i represent the point forces and point moments applied along each axis, which could be due to point forces, point impedance due a lumped mass, or point impedance due to an attached absorber, and is defined as

$$q_{zJ} = F_{zJ}\,\delta(x-x_J)\delta(y-y_J)\,e^{j\omega t} \quad (7)$$

$$T_{iJ} = \frac{M_{iJ}}{R^2}\delta(x-x_J)\delta(y-y_J)\,e^{j\omega t} \quad (8)$$

where F_{iJ} and M_{iJ} are the forces and moments applied to the panel at locations (x_J, y_J) in the directions $i = x,y,z$, δ is the Dirac delta function, U_{ip} is the modal response in the i^{th} direction, and for the vibrations of the panel considered here where only the out-of-plane transverse vibration is considered, the expressions can be written as

$$U_{xp} = 0 \quad U_{yp} = 0 \quad U_{zp} = [\psi]\mathbf{w}_p \quad (9)$$

and

$$N_s = \int_0^{L_x}\int_0^{L_y} U_{zs}^2\;dy\,dx = L_x L_y/4 \quad (10)$$

The force and moment loads on the panel are assumed to be point loads, which can be described with Dirac delta functions. Making use of the relationship

$$\int_\alpha F(\alpha)\frac{\partial}{\partial\alpha}\left[\delta\left(\alpha-\alpha^*\right)\right]d\alpha = -\frac{\partial F(\alpha^*)}{\partial\alpha} \quad (11)$$

the integral in Eq. (6) can be evaluated as

$$\Gamma_s = \frac{1}{\Lambda_s}\left[[\psi_J]^{\mathrm{T}}F_J - \frac{\partial[\psi_J]^{\mathrm{T}}}{\partial y}M_{Jx} + \frac{\partial[\psi_J]^{\mathrm{T}}}{\partial x}M_{Jy}\right] \quad (12)$$

The rotations of the panel are given by (Leissa 1973, Soedel, 1993)

$$\theta_s = \frac{v}{R} - \frac{1}{R}\frac{\partial w}{\partial\theta} \quad (13)$$

$$\theta_\theta = -\frac{1}{R}\frac{\partial w}{\partial s} \quad (14)$$

The partial differentials of the mode shapes $[\psi]$ with respect to the spatial co-ordinates in Eq. (12) are the mode shapes in the rotational directions. Hence Eq. (12) can be written as

$$\Gamma_s = \frac{1}{\Lambda_p}\left[[\psi_J]^{\mathrm{T}}F_J - [\psi_{J\theta x}]^{\mathrm{T}}M_{Jx} + [\psi_{J\theta y}]^{\mathrm{T}}M_{Jy}\right] \quad (15)$$

where $[\psi_{J\theta x}]$ and $[\psi_{J\theta y}]$ are the rotational mode shapes about the θ_x and θ_y axes, respectively and are

$$\mathbf{\Psi}_{J\theta_x} = \frac{\partial\mathbf{\Psi}_J}{\partial y} = \sin(m\pi x/L_x)(n\pi/L_y)\cos(n\pi y/L_y) \quad (16)$$

$$\mathbf{\Psi}_{J\theta_y} = \frac{\partial\mathbf{\Psi}_J}{\partial x} = (m\pi/L_x)\cos(m\pi x/L_x)\sin(n\pi y/L_y) \quad (17)$$

The impedance of the J^{th} mass attached to the panel is included as point translational and rotational inertias by using Eqs. (7) and (8) where

$$F_J = \omega^2 m_J \quad (18)$$

$$M_{J\theta_x} = \omega^2 J_{J\theta_x} \quad (19)$$

$$M_{J\theta_y} = \omega^2 J_{J\theta_y} \quad (20)$$

where m_J is the mass of the block, $J_{J\theta_x}$, $J_{J\theta_y}$ are the rotational inertias of the blocks along the θ_x, θ_y axes, respectively. The attachment of an array of cantilever absorbers is modelled here as multiple single degree of freedom resonators or Tuned Vibration Absorbers (TVAs) (J=1… N_{TVA}), that have mass m_J^{TVA}, stiffness k_J^{TVA}, and are driven by a harmonic force at the attachment point of the spring to the structure. This framework can be used accommodate multiple modes of vibration along translational and rotational axes. However, it will be shown by comparison with the experimental results that for the system examined here it is sufficient to only consider vibration of a single translational mode normal to the panel. The equations for the vibration of the structure and the TVAs can be written in matrix form as

$$\begin{bmatrix} k_J^{\text{TVA}} - \omega^2 m_J^{\text{TVA}} & -k_J^{\text{TVA}}[\psi_J] \\ -[\psi_J]^{\text{T}} k_J^{\text{TVA}} & \begin{bmatrix} \Lambda_p\left(\omega_p^2 - \omega^2\right) + \\ [\psi_J]^{\text{T}} k_J^{\text{TVA}} [\psi_J] \end{bmatrix} \end{bmatrix} \times \begin{bmatrix} x_J^{\text{TVA}} \\ w_p \end{bmatrix} = \begin{bmatrix} \mathbf{0} \\ \mathbf{F}_p \end{bmatrix} \quad (21)$$

where $[\psi_J]$ is the structural mode shape vector evaluated at the J^{th} connection point of the TVA to the structure, T is the matrix transpose operator, and $\mathbf{F}_p$ is the vector of modal forces from Γ_p. The equations derived thus far have not included damping terms. Damping can be included by using a hysteretic structural loss factor, so that the stiffness value for the TMD becomes a complex number. Hence the complex stiffness can be written as $k_J^{\text{TVA}} = k_J^{\text{TVA}}(1 + j\eta)$, where η is the structural loss factor.

The resonance frequencies of a simply-supported panel $\omega_{m,n}$ are given by

$$\omega_{m,n}^2 = \omega_p^2 = \frac{D\pi^4}{\rho h}\left[\left(\frac{m}{L_x}\right)^2 + \left(\frac{n}{L_y}\right)^2\right]^2 \quad (22)$$

$$D = \frac{Eh^3}{12(1-\nu^2)} \quad (23)$$

where D is the bending stiffness of the panel, E is the Young's modulus, h is the thickness, ρ is the density of the panel, ν is the Poisson's ratio.

Sound Power Radiated from the Panel

Once the modal participation factors w_p are calculated, the transmitted pressure at a point remote from the panel due to the vibration of the panel is calculated using the Rayleigh integral and can be written as (Wallace 1972, Fahy 1994)

$$p_{m,n}^t = -j(j\omega w_p)k\rho c\frac{e^{jkr}}{2\pi r}L_x L_y Y_m Y_n \quad (24)$$

$$Y_m = (m\pi)\frac{1-(-1)^m e^{-j\alpha}}{(m\pi)^2 - \alpha^2} \quad (25)$$

$$Y_n = (n\pi)\frac{1-(-1)^n e^{-j\beta}}{(n\pi)^2 - \beta^2} \quad (26)$$

and the transmitted intensity is calculated as $I^t = |\sum_m \sum_n p_{m,n}^t|^2/(2\rho c)$. The total power Π^t that is radiated by the panel is calculated as the integral of the intensity over an imaginary far-field hemisphere as

$$\Pi^t = \int_{\phi_t=0}^{2\pi}\int_{\theta_t=0}^{\pi/2} I^t r^2 \sin\theta_t \, d\theta d\phi \quad (27)$$

Transmission Loss of the Panel

The transmission loss (TL) of a panel is the ratio of the incident to transmitted sound power and for an incident plane wave is given by

$$\text{TL} = 10\log_{10}(\tau(\theta_i,\phi_i)) = 10\log_{10}(\Pi^i/\Pi^t) \quad (28)$$

where the sound power incident on the panel is given by (Roussos 1985)

$$\Pi^i = (|P_i|^2 L_x L_y \cos\theta_i)/(2\rho c) \quad (29)$$

A diffuse field is characterised by an infinite number of uncorrelated plane-waves (Langley & Shorter 2002). The sound field inside the reverberation chamber used in the experimental part of the work conducted here is assumed to be a diffuse field. The transmission loss for a diffuse field is calculated as (Fahy 1994, Guigou-Carter & Villot 2003, Chiello et.al 2003, Wang et. al. 2005)

$$\begin{aligned} \text{TL}_{\text{diffuse}} &= \frac{\int_0^{2\pi}\int_0^{\pi/2}\tau(\theta_i,\phi_i)\sin\theta_i\cos\theta_i \, d\theta_i d\phi_i}{\int_0^{2\pi}\int_0^{\pi/2}\sin\theta_i\cos\theta_i \, d\theta_i d\phi_i} \\ &= \frac{\int_0^{2\pi}\int_0^{\pi/2}\tau(\theta_i,\phi_i)\sin 2\theta_i \, d\theta_i d\phi_i}{2\pi} \end{aligned} \quad (30)$$

Table 1. Geometry of the panel

Width L_x	1.0	m
Height L_y	1.5	m
Thickness t	0.0015	m
Density ρ	2700	kg/m^3
Young's Modulus E	70	GPa
Poisson's ratio ν	0.33	No units
Loss factor η	0.01	No units

EXPERIMENT SETUP

A panel with the properties listed in Table 1 was tested in The University of Adelaide's transmission-loss test facilities. The panel was also tested with an array of cantilever absorbers, shown in Figure 2(a), and with rigid 'blocking' masses of the same weights as the cantilever absorbers, shown in Figure 2(b) that had a total mass of 1.18kg, approximately 19% of the mass of the panel. High amplitude pink noise was played in the source reverberation chamber of the test facility which excited the test panels. The radiated sound power from the panels was measured in the receiving reverberation chamber by a traversing microphone.

The rigid masses or absorbers were attached to the panel in a regular pattern, as shown in Figure 3, with no regard given to optimising their locations for this initial study. The lightest blocking mass (also highest resonance-frequency oscillator) was placed in the top left corner, and the heaviest blocking mass (also the lowest resonance-frequency oscillator) was placed in the lower right corner of the panel. The weight of the rigid blocks and cantilever absorbers had an almost linear distribution. Each absorber had an equivalent rigid block of the same mass, as shown in Figure 4. The resonance frequencies of the absorbers had a distribution as shown in Figure 5. By restraining the beams at their midpoints creates two symmetric cantilever absorbers. Hence each device has two first bending mode resonance frequencies that are similar, as shown in Figure 5. The resonance frequencies of the absorbers are between 270-430Hz. It can be seen that the theoretically predicted resonance frequencies used in the design of the beams is higher than experimentally measured. This is to be expected as the beams were modelled as cantilevers having fully clamped ends, whereas in reality the beams were held at their midpoint with a nut and bolt resulting in non-ideal clamping conditions and hence lower resonance frequencies.

(a) (b)

Figure 2: (a) cantilever absorbers, and (b) rigid blocks attached to the panel.

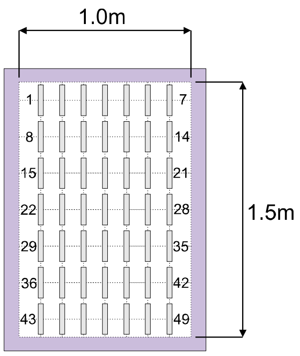

Figure 3: Location of the devices on the panel.

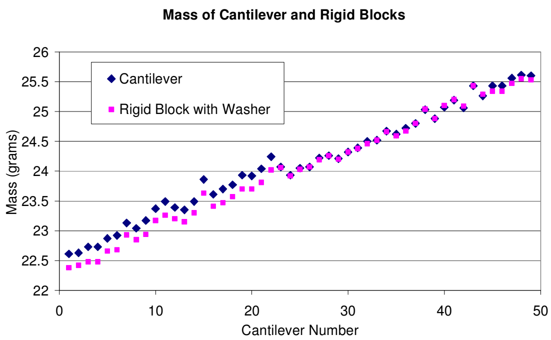

Figure 4: Mass of rigid blocks and cantilever absorbers.

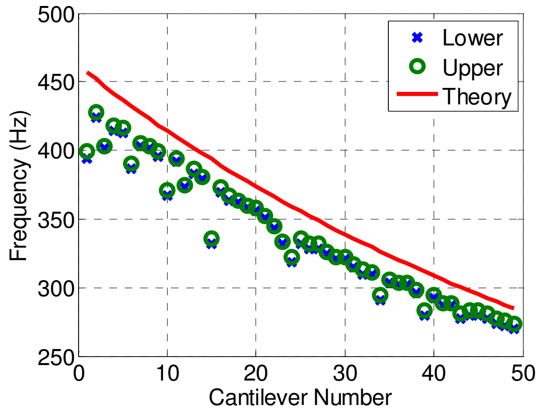

Figure 5: Resonance frequencies of the cantilever absorbers.

RESULTS

Figure 6 shows the experimentally measured transmission loss of the bare panel and theoretical predictions using the modal model described here, and the solid curve shows the predicted transmission loss for a finite panel based on the work by Sewell (1970), and discussed in Fahy (1994, p162). These results are from Howard (2007) and are included here for completeness. It can be seen that the measured TL above 10kHz does not follow the theoretical predictions as the measured T60 reverberation times were erroneous due to insufficient sound level in the receiver room above the background level. The low-frequency accuracy of the measurements is limited by the largest dimension of the chambers and is valid above 125Hz, hence comparisons between experimental results and theoretical predictions can be made at and above 125Hz. The predictions using the modal model are accurate from about 50-1000Hz. Above 1kHz it can be seen that the results diverge and the inaccuracy is caused by using an insufficient number of modes in the analyses; 2000 structural modes for these analyses. As the purpose of the work here is to investigate the improvement of TL at low-frequencies, the frequency range between 125-1000Hz will be considered.

Figure 7 shows the transmission loss results for experimental measurements and theoretical predictions with and without blocking masses from Howard (2007). The theoretical predictions show that the addition of the rigid blocking masses, which had a total mass of 1.18kg, increases the transmission loss of the panel between 100-315Hz, which was confirmed by experimental measurements. It was shown in Howard (2007) that if this added mass had been smeared over the panel, achieved by increasing the panel thickness, the TL would be 1.2dB greater than the bare panel. However the improvement in TL for the rigid blocking masses exceeds 1.2dB.

Figure 8 shows the transmission loss of the panel with and without the addition of the cantilever absorbers. The results show that the addition of the cantilevers improves the transmission loss in the frequency range between 125-200Hz and compares favourably with the theoretical predictions. It can be seen that the transmission loss between 250-400Hz is less than the case of the bare panel. In this frequency range it can be seen that the theoretical predictions under-predict the effect of the oscillators. The effect of the oscillators is modelled as single-degree-of-freedom resonators acting normal to the panel. However the absorbers have translational and rotational modes and the effects of the rotational modes were not included in the theoretical analysis.

A Polytec 3D Scanning Laser Vibrometer was used to measure the motion of the beams during the transmission loss testing. Figure 9 shows an image from the Scanning Laser Vibrometer system where the panel under test is in the background and the motion of the 49 absorbers were measured at 5 points along each absorber. The image shows the operating deflected shape of the absorbers at 220Hz and the rectangular box highlights two adjacent absorbers undergoing significant rotational motion.

If the effects of the rotational motion of the absorbers were included in the theoretical analysis, one would expect that the absorbers would impart rotational impedance to the structure limiting bending vibrations, and hence the transmission loss of the panel would be greater, closer to the experimentally measured transmission loss results.

Figure 10 shows the results from Figure 7 and Figure 8 super-imposed. The surprising result is that the greatest improvement in transmission loss occurred for the panel configuration with the rigid blocking masses and not for the case where the TVAs were attached to the panel. It was hypothesised that the use of the TVAs would provide significant benefits in transmission loss, and that by using 49 absorbers with closely spaced resonance frequencies would be sufficient to achieve a 'fuzzy' structure configuration, where the master structure, the panel, would have significant vibration reduction by the attachment of a large number of oscillators. It was shown in Howard et.al (2005) that the attachment of a large number of oscillators to a master structure could result in significant vibration reductions and improvement in transmission loss due to broadband acoustic excitation, and that the accurate placement of the oscillators on the master structure was not important, suggesting a robust noise control technique. However this was not the outcome for this project and further investigation of the laser vibrometer data is warranted.

The resonance frequencies of the cantilever absorbers, shown in Figure 5 corresponds to the frequency range where the transmission loss is no greater than the bare panel. This result suggests that the effect of the discrete masses acted to restrain the local motion of the panel, whereas the attachment of the cantilever beams did not. These results are similar to those found by Brennan and Dayou (2000) that investigated the addition of tuned vibration absorbers to reduce the vibration of a cantilever beam.

The mechanical impedance of an absorber at resonance is approximately given by $m^{TVA}\omega_r Q$, where m^{TVA} is the mass of the absorber, ω_r is its resonance frequency, Q is the quality factor (Fuller et.al. 1997). At resonance, greater mechanical impedance is achieved by increasing Q, the sharpness of the resonance peak. However at frequencies off-resonance, the impedance of the absorber is proportional to the mass, and thus a higher mass will therefore be more effective.

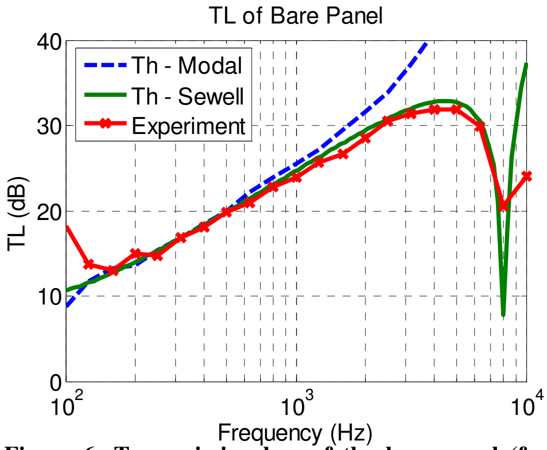

Figure 6: Transmission loss of the bare panel (from Howard 2007).

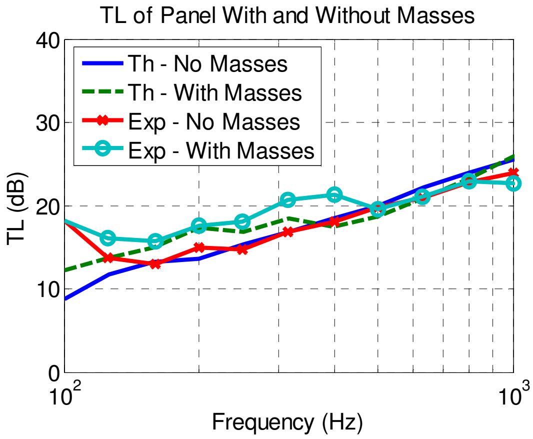

Figure 7: Transmission loss of the bare panel and with rigid blocks attached.

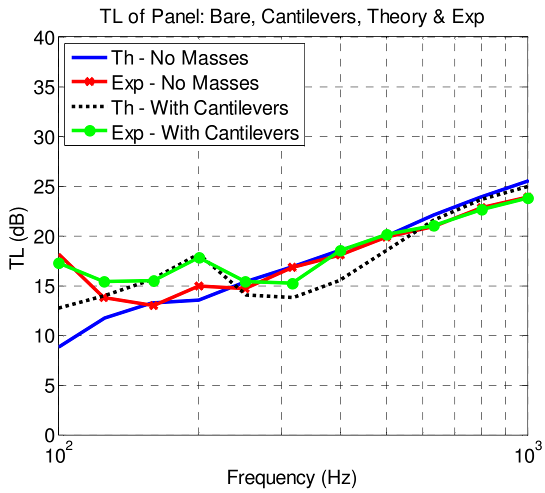

Figure 8: Transmission loss of the bare panel and with cantilever absorbers attached.

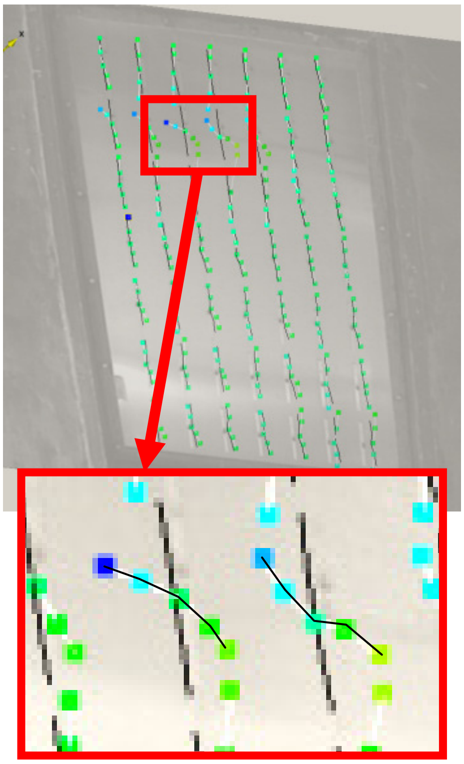

Figure 9: Vibration of cantilever absorbers attached to panel at 220Hz.

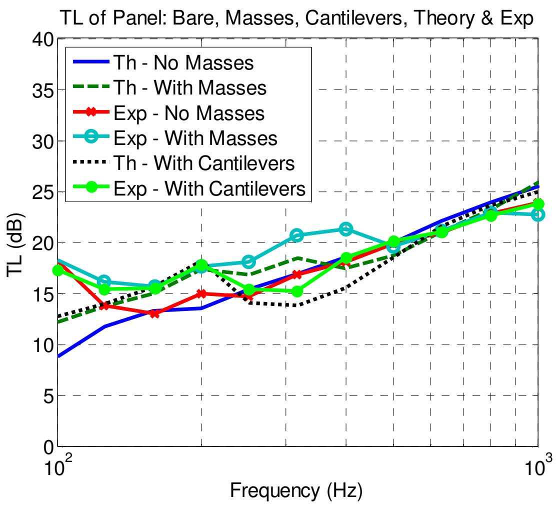

Figure 10: Transmission loss of the bare panel, with rigid blocks, and with cantilever absorbers.

CONCLUSIONS

A mathematical model was presented to enable the calculation of the transmission loss of a simply-supported panel with an array of discrete blocking masses attached or an array of single-degree-of-freedom oscillators attached. This model was an extension of the model presented in Howard (2007).

The calculation of the transmission loss described in this paper involved the analysis of 370 plane waves, which was achieved using a distributed computing network. Hence this calculation method is limited to investigators with extensive computational resources. Chazota and Guyader (2007) suggest an alternative calculation method using blocked patch pressures, which avoids the double integration calculation over all incident angles for the calculation of the transmission loss. Work will be conducted in the future to incorporate this calculation technique, which will reduce the calculation time and enable the optimisation of the locations of the rigid-masses and absorbers by using a genetic algorithm as described in Howard et. al (2005).

It was shown in Howard (2007) that the attachment of blocking-masses improved the transmission loss of the panel, greater than could be achieved by smearing the mass of the absorbers across the panel, which could be achieved by increasing the thickness of the panel. When cantilever absorbers replaced the blocking-masses, the transmission loss of the panel only improved in the frequency ranges outside the resonance frequencies of the absorbers, which was not expected. The likely reason for this is that the impedance that the absorbers present to the panel is less than the rigid-blocking mass presents to the panel. Further investigation is warranted for this noise control technique, as studies involving fuzzy structure theory point to potential benefits in vibration reduction of panels, and hence there is also likely to be corresponding improvement in transmission loss.

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