The effects of damping treatment on the sound transmission loss of honeycomb panels

Sathish Kumar

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Vinnova Centre of Excellence for ECO² Vehicle Design
The Marcus Wallenberg Laboratory for Sound and Vibration Research
Department of Aeronautical and Vehicle Engineering

Postal address
Royal Institute of Technology
MWL/AVE
SE-100 44 Stockholm
Sweden

Visiting address
Teknikringen 8
Stockholm

Contact
Tel: +46 8 790 93 75
Email: sathish@kth.se
Abstract

In the industry, all passenger vehicles are treated with damping materials to reduce structure-borne sound. Though these damping materials are effective to attenuate structure-borne sound, they have little or no effect on the air-borne sound transmission. The lack of effective predictive methods for assessing the acoustic effects due to added damping on complex industrial structures leads to excessive use of damping materials. Examples are found in the railway industry where sometimes the damping material applied per carriage is more than one ton. The objective of this thesis is to provide a better understanding of the application of these damping materials in particular when applied to lightweight sandwich panels.

As product development is carried out in a fast pace today, there is a strong need for validated prediction tools to assist in the design process. Sound transmission loss of sandwich plates with isotropic core materials can be accurately predicted by calculating the wave propagation in the structure. A modified wave propagation approach is used to predict the sound transmission loss of sandwich panels with honeycomb cores. The honeycomb panels are treated as being orthotropic and the wave numbers are calculated for the two principle directions. The orthotropic panel theory is used to predict the sound transmission loss of panels. Visco-elastic damping with a constraining layer is applied to these structures and the effect of these damping treatment on the sound transmission loss is studied. Measurements are performed to validate these predictions.

Sound radiated from vibrating structures is of great practical importance. The radiation loss factor represents damping associated with the radiation of sound as a result of the vibrating structure and can be a significant contribution for structures around the critical frequency and for composite structures that are very lightly damped. The influence of the radiation loss factor on the sound reduction index of such structures is also studied.
Part I

Overview and Summary
Chapter 1
Introduction

Low noise inside cabins of passenger transport vehicles is an important quality criterion. Manufacturers work hard to improve the comfort standards of their products, while, at the same time try to keep the costs down. The design of rail vehicles is driven by a number of functional requirements. The components used in vehicles have been and still being designed, produced and assembled separately, each fulfilling different functions. The idea of multifunctional design is to design/select a component so that it can have multiple functionalities to reduce the number of total components. Solutions have to be obtained for several design criteria such as static and dynamic stiffness, thermal insulation, acoustic insulation, partition thickness, weight and production costs. In the final design, the various functional requirements should be met while keeping the material and production costs low and avoiding overly complex structures.

Traditionally, railway cars are designed with structural body-shells in steel or aluminium with acoustically de-coupled wall panels and walking floors, making up the interiors. Aluminium car bodies can be lighter than corresponding steel designs and may also be made to a high degree of pressure tightness making them ideal for high-speed trains. The inner flooring of a passenger compartment made of sandwich panels with aluminium face sheets and honey comb core material have certain advantages over floor panels made of wood. However, these panels exhibit very poor acoustic qualities necessitating the use of external damping treatments.

The sound levels in a modern passenger compartment is between 55-74 dB(A) depending on the type of car and the speed (Andersson, 2002). For trains, it is common to distinguish between the air-borne noise emission and structure-borne sound emission. According to Carlsson (1992), the various noise transmission paths into a passenger compartment for both air-borne and structure-borne noise emissions are as shown in Figure 1.1.

1) Air-borne noise through floors, walls, windows, roofs and auxiliary equipments like fans, motors, gears, HVAC units.
2) Structure-borne noise from bogie, diesel engines.
Vibrational energy from these sources can be transmitted to the passenger compartment. For most passenger trains, floating floors are applied to obtain sufficient noise reduction from the sources under the floor \textit{e.g.} the bogie. In addition, lightweight and thin floor designs are desired for increased weight reduction. For these reasons traditional floor panels made of wood is being replaced by light weight sandwich structures. Some of the advantages of these constructions are low weight, good moisture properties, fire resistance and high stiffness-to-weight ratio \textit{etc}. Within the scope of the project the air-borne sound transmission through these floor panels used in railway passenger compartments are carried out.

In the automotive industry, all passenger vehicles are treated with damping materials to reduce structure borne sound and its effectiveness mainly depends upon parameters such as materials, location and size of the damping treatment. Traditional damping treatments using viscous damping layers are typically of two types, unconstrained and constrained layer damping (Cremer, 2005). Constrained layer damping (CLD) treatments have provided an effective means to impart damping to the structure (Beranek, 1992) and (Nashif et al., 1985).

Due to the shear deformation occurring in the visco-elastic layer, CLD treatments are known to yield significantly larger system damping compared to unconstrained layer damping, for the same mass of damping material used (Kerwin, 1959). Though these damping materials are effective to attenuate vibration, their effect on the air borne sound transmission is limited due to the increase of the radiation efficiency (Cremer, 2005) and (Heckl, 1981). The lack of effective predictive methods for assessing the acoustic effects...
due to added damping on complex industrial structures leads to excessive use of damping materials. Examples are found in the railway industry where sometimes the damping material applied per carriage is more than one ton (Orrenius, 2001).

As product development is carried out in a fast pace today, there is a strong need for validated prediction tools to assist in the design process. In this thesis, the wave propagation approach is used to numerically predict the sound transmission through sandwich plates subjected to a diffuse sound field. Constrained layer damping treatments are applied to these panels and the sound transmission loss is predicted. The predicted wavenumbers and the sound reduction index are validated through laboratory measurements.
Chapter 2

Sandwich Structures

Sandwich structures have been the subject of many studies, a large amount of literature have been devoted to the development of theories to study their static and dynamic behaviours through the use of analytical and numerical methods (Nilsson, 1990) and (Nilsson & Nilsson, 2002). The Sound transmission loss of sandwich panels has been discussed by Dym (1974) and Kurtze (1959). Kurtze (1959) suggested that using a sandwich panel rather than a homogeneous panel might increase the sound insulation between partitions. However, for certain types of sandwich plates the acoustical properties can be very poor. Nilsson (1990) proposed a method to predict the sound transmission loss of sandwich partitions by studying the wave propagations in the sandwich plates.

A *classical* sandwich structure consists of two stiff, strong, thin face sheets bonded to either side of a relatively thick, weaker, lightweight core material as shown in the Figure 2.1. The faces are usually made from a high performance material such as steel, aluminium or fibre composite, whereas the core is usually a structural solid foam, balsa wood or honeycomb (this can again be made of aluminium, kraft paper, etc). The structural properties of the face sheets and the core are less significant as individual panels but when glued together to form a sandwich, they produce a structure of high stiffness and high strength-to-weight ratio, a property which is of great interest to the industry.

![Figure 2.1: Sandwich Cross-Section](image)

### 2.1 The Sandwich Effect

The good stiffness properties of a sandwich construction can be illustrated by the following example. A structure made up of a homogeneous material with a given Young's
Multifunctional Body Panels

modulus and strength having unit width and thickness ‘t’ will have a certain bending stiffness which is normalized as ‘1’. Then the beam is cut into two halves of thickness ‘t/2’ and a core material of thickness ‘2t’ is bonded between these two halves and the corresponding stiffness and strength is ‘12’ and ‘6’ times more than the homogeneous beam respectively. The core material is assumed to have a low surface density than the face sheets and therefore any addition in weight to the structure is considered negligible. This is called the sandwich effect.

\[ \begin{array}{ccc}
\text{Weight} & \text{Flexural Rigidity} & \text{Bending Strength} \\
\text{t/2} & 1 & 1 & 1 \\
\text{t/2} & 2t & 12 & 6 \\
\text{t/2} & 4t & 48 & 12 \\
\end{array} \]

Figure 2.2: The Sandwich Effect. Figure from Zenkert (1995).

### 2.2 Honeycomb Panels

A honeycomb panel is a lightweight sandwich panel with a honeycomb core of hexagon cell. Honeycomb cores can be manufactured in a variety of cell shapes but the most commonly used shape is the hexagonal shape as shown in Figure 2.3. Three different honeycomb panels are investigated in this thesis based on two different core thicknesses and two different core structures as shown Figure 2.3 with dimensions and properties as shown in Table 2.1. The panels with 6.4\text{mm} cell were selected to represent a common design solution for railway floor structures and the panel with 19.2\text{mm} was chosen to study the effect of a softer core.

<table>
<thead>
<tr>
<th>Panels</th>
<th>Honeycomb Cell Size [\text{mm}]</th>
<th>( t_f/t_e/t_f )</th>
<th>Surface Density bare panel [\text{kg/m}^2]</th>
<th>Surface Density with CLD [\text{kg/m}^2]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Panel 1</td>
<td>6.4</td>
<td>1.5/18/1.5</td>
<td>10.4</td>
<td>18.4</td>
</tr>
<tr>
<td>Panel 2</td>
<td>6.4</td>
<td>1/10/1</td>
<td>6.8</td>
<td>10.4</td>
</tr>
<tr>
<td>Panel 3</td>
<td>19.2</td>
<td>1/10/1</td>
<td>6.2</td>
<td>9.8</td>
</tr>
</tbody>
</table>

The damping treatment applied to these panels consists of a 1\text{mm} thick viscoelastic layer with a constraining layer. Two different types of constraining layers were used. Assuming the core to be weak, the bending stiffness of a sandwich plate is mainly
influenced by the face separation distance. Panel 1 having a face separation distance of 18mm has a high initial flexural stiffness. Therefore, a 1\text{mm} thick steel plate (surface density 7.8kg/m$^2$) was used as a constraining layer for panel 1 and 1\text{mm} thick aluminium plate (surface density 2.7kg/m$^2$) was used for panel 2 and panel 3 which had relatively low initial bending stiffness.
Chapter 3

Wave Propagation in Sandwich Structures

The sound reduction property of a panel is governed by the properties of the constituent materials, the boundary conditions and the wave motions in the panel. A model has been developed by Nilsson (1990) to predict the sound reduction index of sandwich panels using the classical equations of motion for wave propagation, as well as theories for excitation and radiation from plates. The wavenumbers in a structure is an indicator for the response of a sandwich plate to an external excitation. In particular, the waves determining the lateral bending of the plate is of importance, since these have the greatest influence on excitation and radiation. In the low-frequency, the propagation constants are equal to

\[ k_B = \sqrt{\frac{\omega^2 m}{D}} \]  \hspace{1cm} (3.1)

For increasing frequencies the lateral motion of the plate can no longer be described by bending alone, the shear and rotation of the core will influence the deflection of the plate and its apparent bending stiffness. It is assumed that the displacement of the facesheet is determined by the flexural and longitudinal waves and the displacement in the thick core is due to bending, rotation, shear as well as longitudinal deflection.

According to Nilsson (1990), for a sandwich panel with isotropic, symmetric face sheets and isotropic core as shown in Figure 3.1, the wavenumbers can be derived as a solution to the following expression

\[ F(k) = (U_1 - U_2)(V_3 - V_4) - (U_3 + U_4)(V_1 + V_2) \]  \hspace{1cm} (3.2)

The solutions for \( k \) are obtained when \( F = 0 \) and the functions \( U \) and \( V \) are given as
Figure 3.1: Geometry and material parameters of a sandwich panel with isotropic core.

\[ U_1 = (1 - e^\alpha)L_1[\zeta_L^2 - k^2v_2/(1 - v_2)]; \quad U_2 = (1 + e^\alpha)\zeta_L[k^4 - k_1^4] \]
\[ U_3 = (1 - e^{-\beta})\zeta_T k_1 L_1[(1 - 2v_2)/(1 - v_2)]; \quad U_4 = (1 + e^{-\beta})k(k^4 - k_1^4) \]
\[ V_1 = (1 - e^\alpha)\zeta_L[k^2_1 - k^2]; \quad V_2 = (1 + e^\alpha)\zeta_L M_1 k \]
\[ V_3 = (1 - e^{-\beta})\zeta_T[k^2_1 - k^2]; \quad V_4 = (1 + e^{-\beta})2M_1[k^2 - k^2_2]/(1 - v_2) \]

The wavenumbers for the flexural waves in the face sheets are given as

\[ k_1 = \left\{ \frac{\rho_1 \omega^2 12(1 - v_1^2)}{E_1 t^2_1} \right\}^{\frac{1}{2}} \] (3.3)

and the wavenumbers for longitudinal waves in the face sheets and core are given by the expressions

\[ k_{L1} = \left\{ \frac{\rho_1 \omega^2 (1 - v_1^2)}{E_1} \right\}^{\frac{1}{2}} ; \quad k_{L2} = \left\{ \frac{\rho_2 \omega^2}{E_2} \right\}^{\frac{1}{2}} \] (3.4)

where \( E_2 = E_2(1 - v_2)/[(1 + v_2)(1 - 2v_2)] \), \( E_2 = 2G_2(1 + v_2) \) and the various other functions used to solve for the wave propagation constant \( k \) are given in the appendix.

Due to manufacturing process of honeycomb cores, the cell wall thickness doubles in the direction of cell orientation and the actual shape of the honeycomb cell can be irregular making the core anisotropic. For simplicity, the honeycomb core can be treated as orthotropic and the whole panel can be treated as an orthotropic sandwich panel, if the frequency concerned is not too high. The wavenumbers \( k_{Bx} \) and \( k_{Bz} \) in the \( x \) and \( z \) directions as shown in Figure 3.2 can be solved by using an averaged value of the loss factor measured between 100 Hz and 1000 Hz and by applying different values for the core shear modulus \( G_{2x} \) and \( G_{2z} \) for the \( x \)- and \( z \)-directions respectively. Along with the geometrical properties in Table 2.1, the following parameters were used to calculate the wavenumbers. The shear moduli in Table 3.1 were measured according to DIN53294 and provided by the manufacturer.

For a honeycomb panel as shown in Figure 3.2 the motion is governed by (Heckl, 1981)

\[ D_x \frac{\partial^4 \nu}{\partial x^4} + 2D_{xx} \frac{\partial^4 \nu}{\partial x^2 \partial z^2} + D_z \frac{\partial^4 \nu}{\partial z^4} - \omega^2 m \nu = j\omega \Delta p \] (3.5)
Table 3.1: Parameters used for wavenumber calculations

<table>
<thead>
<tr>
<th>Panels</th>
<th>$G_{2x}$ $N/mm^2$</th>
<th>$G_{2z}$ $N/mm^2$</th>
<th>Loss factor bare panel</th>
<th>Loss factor with CLD of core</th>
<th>Poisson Ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>Panel 2</td>
<td>430</td>
<td>220</td>
<td>0.005</td>
<td>0.04</td>
<td>0.33</td>
</tr>
<tr>
<td>Panel 3</td>
<td>201</td>
<td>54</td>
<td>0.005</td>
<td>0.04</td>
<td>0.33</td>
</tr>
</tbody>
</table>

Figure 3.2: Cutaway of a honeycomb sandwich panel

where $D_x$ and $D_z$ are the bending stiffness in $x$– and $z$– directions respectively and $D_{xz}$ can be approximated by $D_{xz} \cong \sqrt{D_x D_z}$.

As an approximation, $D_x$ and $D_z$ can be defined as the bending stiffness of a simple homogeneous beam having the same dynamic properties as the honeycomb structure at certain frequencies.

According to Heckl (1981), for a given angle of incidence $\theta$, the transmission coefficient of an infinitely long orthotropic plate is given by

$$
\tau_{\theta \varphi} = \left| \frac{1 + j \omega m \cos \theta}{2\rho_0 c_0} \left[ 1 - \left( \frac{\sin^2 \varphi - \cos^2 \varphi}{k_{Bx}^2} \right)^2 \frac{k_0^4 \sin^4 \theta}{k_{\varphi,eqv.}^4} \right] \right|^{-2} \tag{3.6}
$$

where $k_{Bx} = \frac{4}{\sqrt{\omega^2 m / D_x}}$ and $k_{Bz} = \frac{4}{\sqrt{\omega^2 m / D_z}}$ are the wavenumbers in $x$– and $z$– directions respectively.

The structural damping can be included by replacing $D_x$ and $D_z$ in Equation 3.6 as $D_x(1 + j\eta)$ and $D_z(1 + j\eta)$ respectively. Assuming a constant loss factor in all the directions in the panel, Equation 3.6 can be rearranged as

$$
\tau_{\theta \varphi} = \left\{ \left[ 1 + \frac{\omega m \cos \theta}{2\rho_0 c_0} k_0^4 \sin^4 \theta \right]^2 + \left[ \frac{\omega m \cos \theta}{2\rho_0 c_0} \left( 1 - \frac{k_0^4 \sin^4 \theta}{k_{\varphi,eqv.}^4} \right) \right]^2 \right\}^{-1} \tag{3.7}
$$

where $k_{\varphi,eqv.}$ is the equivalent wavenumber in direction $\varphi$ and it is given by

$$
\frac{1}{k_{\varphi,eqv.}^2} = \frac{\sin^2 \varphi}{k_{Bx}^2} - \frac{\cos^2 \varphi}{k_{Bz}^2} \tag{3.8}
$$
The diffuse sound reduction index can then be calculated by averaging the transmission coefficient over all possible angles $\theta$ and $\varphi$ as

$$ R_{\theta\varphi} = 10 \log \left( \frac{1}{\tau_{\theta\varphi}} \right) $$ (3.9)

Radiation loss factor is a parameter indicating to what extent the vibration of a structure are damped due to radiation to the surrounding fluid. According to Cremer (2005), the radiation loss factor can be defined as

$$ \eta_{\text{rad}} = \frac{\rho_0 c_0 \sigma}{\omega_m} $$ (3.10)

where $\eta_{\text{rad}}$ is the radiation loss factor and $\sigma$ is the radiation efficiency which has been investigated by Beranek (1992) & Maidenik (1962). According to Leppington et al. (1982), a better estimation of the averaged radiation efficiency of a rectangular plate can be given as

$$ \bar{\sigma} \approx \frac{a + b}{\pi \mu kab (\mu^2 - 1)} \left\{ \ln \left( \frac{\mu + 1}{\mu - 1} \right) + \frac{2 \mu}{\mu^2 - 1} \right\} \quad \mu > 1 $$

$$ \bar{\sigma} \approx \sqrt{k a H(x)} \quad \mu = 1 $$ (3.11)

$$ \bar{\sigma} \approx 1/\sqrt{1 - \mu^2} \quad \mu < 1 $$

where $a$ and $b$ are the width and length of the plate with $a < b$, $\mu = k_B / k = \sqrt{f_c / f}$ with $f_c$ being the critical frequency of the plate, and $H(x)$ is a function of the panel aspect ratio and in our case it can be approximated to 0.41 (Leppington et al., 1982). For an isotropic panel, there is one critical frequency and the radiation loss factor can be applied as shown by Nilsson (1990). But for orthotropic panels, there exists different critical frequencies depending on the angle $\varphi$. There exists no expressions for calculating the wavenumbers and critical frequency as a function of $\varphi$. As an compromise, the ‘equivalent critical frequency’ in the orthotropic directions can be calculated by Equation 3.8 and the related radiation loss factor can be obtained.

The loss factor is normally measured in $1/3^{rd}$ octave bands because of the large fluctuations in the measurements. However, it is shown by Leping (2009) that by using a loss factor measured in $1/3^{rd}$ octave bands, the sound reduction index will be underestimated around the critical frequency because of the extremely large radiation loss factor concentrated in a very small frequency band as indicated in Equation 3.11. As a consequence, the radiation loss factor calculated in narrow band frequency is added to the measured loss factor for better sound transmission loss predictions around the critical frequency as shown by Leping (2009). The total loss factor can be expressed a sum of the measured loss factor and the radiation loss factor.

$$ \eta_{\text{total}} = \eta_0 + \eta_{\text{rad}} $$ (3.12)
Chapter 4

Measurements

4.1 Loss Factor

The loss factor is a good measure of the structural damping present in a system. From the several methods available, a simple and robust decay method was used to measure the loss factor of the structures in 1/3rd octave bands. The panel was suspended in springs to achieve free-free boundary conditions minimizing boundary losses apart from radiation. For the same reason, impulse excitation was used to avoid unwanted external damping due to a shaker mounting. The decay method uses the fact that free vibrations decay with time and by measuring the reverberation time of the structure, the loss factor can be calculated by (Cremer, 2005)

\[ \eta = \frac{2.2}{f_n T} \]  

(4.1)

where \( f_n \) is the frequency in Hz and \( T \) is the reverberation time in seconds.

Here, random accelerometer positions and excitation points were used to get a spatial average of the measured vibration response and to reduce the influence from individual modes.

After the damping layers were attached, the reverberation time decreased significantly making decay measurements difficult. Therefore, power injection method was used to measure the loss factor of the structures with CLD treatment. This method is not ideal for complex built-up structures but works very well for freely hanging simple structures as in our case. For a point excited system, the input power is related to the spatial average of the vibration velocity as \( P = 0.5Sm\omega\eta\nu^2 \) (Cremer, 2005). This can be rearranged to calculate the loss factor as

\[ \eta = \frac{2P}{Sm\omega\nu^2} \]  

(4.2)
where $P$ is the input power and $S$, $m'$, $v_{\Delta}^2$ the area, surface density and the mean square vibration velocity of the test structure respectively.

In practice, using harmonic excitation, the loss factor is determined according to Leping (2007) as

$$\eta = -\frac{\text{Im} g(G_{fa})}{S m' G_{aa}}$$

with $G_{fa}$ being the cross-spectrum of the excitation force and the response acceleration and $G_{aa}$ being the power spectrum of acceleration.

### 4.2 Sound Reduction Index

The sound reduction index is a measure of the sound insulation of a partition. Sound reduction index was measured using the sound intensity technique by fixing the sandwich panel between a reverberation room and an anechoic room as shown in Figure 4.1. Loudspeakers were used to generate white noise in the reverberation room in the frequency range 100 Hz to 5000 Hz to create a diffuse sound field. A condenser microphone attached to a rotating boom was used to measure the sound pressure level inside the reverberation room. The sound intensity was measured on the panel surface facing the anechoic room.

![Figure 4.1: Setup for Sound reduction Index Measurement](image)

Great care was taken to minimize flanking transmission thus ensuring that the sound transmission was only through the test panel. Then the air-borne sound reduction
index was calculated according to ISO15186-1:2000 as given below

\[ R_I = L_P - 6 - (L_I + 10 \log(S_m/S)) \] (4.4)

where \( R_I \) is the sound reduction index, \( L_P \) is the sound pressure level in the reverberation room, \( L_I \) is the sound intensity measured in the anechoic room, \( S_m \) is the measured surface area and \( S \) is the area of the test specimen. Weighted (apparent) sound reduction index \( R'_W \), a single numbered quantity used to describe the sound insulation of partitions was calculated according to ISO1717 – 1:2006.

4.3 Bending Stiffness

For homogeneous panels, the bending stiffness is frequency independent but for multilayer panels like honeycomb panels, the bending stiffness becomes frequency dependent. Simplified modal tests of freely hanging honeycomb beams were made to measure their natural frequencies to calculate the apparent bending stiffness. Sections \((2050\,mm \times 70\,mm)\) of panel 2 and panel 3 were cut out and suspended using rubber strings to achieve free-free boundary conditions. The measurement setup is shown in Figure 4.2.

![Figure 4.2: Setup for Bending stiffness Measurement](image)

An impact hammer was used to excite the beam and an accelerometer was used to measure the response. Different accelerometer positions were selected and measured to make sure to find all relevant modes. From the measured natural frequencies the bending stiffness was calculated by (Zenkert, 1995)

\[ D_x = \frac{\omega_n^2 m' L_x^4}{n \pi + \pi/2}; D_z = \frac{\omega_n^2 m' L_z^4}{n \pi + \pi/2} \] (4.5)

where \( \omega_n \) is the natural angular frequency, \( L \) the length of the beam, \( m' \) the beam surface density and \( n \) the order of the natural frequency. From the measured bending stiffness, the flexural wavenumbers of the beams were calculated from the relation

\[ k_x = \left( \frac{\omega_n^2 m'}{D_x} \right)^{1/4}; k_z = \left( \frac{\omega_n^2 m'}{D_z} \right)^{1/4} \] (4.6)
Chapter 5

Results and Discussion

Figure 5.1 shows the measured loss factor of panel 1 without and with the damping treatment. It can be clearly seen that there is a significant increase in the structural loss factor of the panel when the damping layer is attached.

Figure 5.2 and Figure 5.3 shows the measured loss factors of panel 2 and panel 3 respectively without and with the damping treatment. As observed for panel 1, there is an increase in the loss factors for both these panels after the damping treatment.

The loss factors of the bare panels were measured using the decay method as explained in Section 4.1. The measurements were limited to a frequency of 1000 Hz above which the reverberation time was too short to get a useful result. Power injection method was used to measure the loss factor after the damping layer was attached.

Figure 5.4 shows the measured sound reduction index of panel 1 with and without the CLD treatment. It can be seen that just like the loss factor, the sound reduction index of the panel increases with the damping treatment. This increase in SRI is partially attributed to the mass added to the structure due to the damping treatment.

For this reason, mass normalization was done to remove any effect of added mass
Figure 5.2: Measured loss factor of Panel 2

Figure 5.3: Measured loss factor of Panel 3

Figure 5.4: Measured Sound Reduction Index of Panel 1 with and without Damping
on the results. This was done by subtracting the mass term \( \Delta = 20 \log\left(\frac{m'}{m'_0}\right) \) from the measured sound reduction indices in each frequency band, where \( m' \) and \( m'_0 \) are the mass densities with and without damping layers attached. Figure 5.5 shows the measured sound reduction index compared to the mass normalized sound reduction index of the panel with damping treatment. It can be seen that the influence of damping material on the SRI of panel 1 is very little.

![Graph showing measured sound reduction index comparison](image)

**Figure 5.5:** Measured Sound Reduction Index of Panel 1 with and without Damping (mass normalized)

Figure 5.6 and Figure 5.7 shows the measured SRI compared to the mass normalized SRI of panel 2 and panel 3 respectively. It can be seen that the damping treatment has a significant influence on the SRI of panel 3 than panel 2.

![Graph showing measured sound reduction index comparison](image)

**Figure 5.6:** Measured Sound Reduction Index of Panel 2 with and without Damping (mass normalized)

Weighted sound reduction index \( R'_{W} \), a single number quantity characterising the sound insulation properties of a partition was calculated according to ISO717–1 : 2006.
Table 5.1 presents the weighted sound reduction indices of the three panels with and without damping treatments.

<table>
<thead>
<tr>
<th>Panels</th>
<th>$R'_W$ [dB]</th>
<th>$R''_W$ [dB]</th>
<th>$\Delta$ [dB]</th>
</tr>
</thead>
<tbody>
<tr>
<td>bare panel</td>
<td>25</td>
<td>30</td>
<td>4.9</td>
</tr>
<tr>
<td>Panel 1</td>
<td>22</td>
<td>27</td>
<td>3.6</td>
</tr>
<tr>
<td>Panel 2</td>
<td>19</td>
<td>27</td>
<td>3.9</td>
</tr>
</tbody>
</table>

It should be noted that the only structural difference of panel 2 and 3 is the honeycomb cell size affecting the shear stiffness of the core. It can be seen in Table 5.1 that without the damping treatment, panel 2 has a weighted sound reduction index 3 dB more than panel 3. When the damping material is attached, both the panels have the same sound insulation rating.

The bending stiffness of beam 2 and beam 3 in the transverse direction $D_x$ is shown in Figure 5.8. The static bending stiffness of a sandwich structure with thin faces and a weak core can be approximated to $D = E_f T_f d^2 / 2$, where $E_f$ and $T_f$ are the young's modulus and thickness of the face respectively and $d$ is the distance between the center of the two face sheets. When a weak core is assumed, the static bending stiffness is mainly influenced by the face separation distance. Panel 2 and panel 3 having the same separation distance of 10 mm, have the same static bending stiffness as seen in Figure 5.8 and it decreases with increasing frequency as observed by Kurtze (1959). The bending stiffnesses were determined from the measured natural frequencies as discussed in Section 4.3. The same measurements were repeated with the damping materials attached and it can be observed that there is an apparent increase in the static bending stiffness for
both the beams. The dotted and solid lines in Figure 5.8 are a polynomial fit to the measured data.

![Figure 5.8: Measured Bending Stiffness of the Beams](image)

From the measured bending stiffness, the bending wave numbers are calculated and Figure 5.9 shows the measured bending wavenumbers of beam 2 and beam 3 compared with the wavenumber in air. The lowest frequency at which the wavelength of bending waves in plate is equal to the wavelength in air is known as the critical frequency.

![Figure 5.9: Measured Bending Wavenumbers in the Beams and Air](image)

For the honeycomb panels analysed, this phenomenon is illustrated in Figure 5.9 at the frequencies where the dotted lines (plate wavenumbers) intersect the wavenumbers in air.

It is clear from Figure 5.9 that panel 3 displays a much wider coincidence range than panel 2. This extended range is due to the lower shear stiffness of the core of panel 3 which leads to coincidence in a wide frequency range (Orrenius, 2001). It can also be seen that the added damping has little effect on the wavenumbers in both the beams.
even though an increase in the bending stiffness can be noticed. This can be attributed to the mass added due to the damping material. Figure 5.10 shows the measured and calculated wave numbers of beam 2 and beam 3. It can be seen that there is a very good agreement between the two values and therefore the calculated ‘equivalent wavenumbers’ were further used to predict the sound transmission loss of the panels.

The ‘equivalent wavenumbers’ calculated as described in Chapter 3 were later used to predict the sound reduction index (SRI) of panel 2 and panel 3. Figure 5.11 shows the measured SRI of panel 2 compared with the predictions made with an averaged structural loss factor and the predictions made with the total loss factor which includes the radiation loss factor.

We can see that for panel 2, the predictions with the average loss factor holds good for \( f < f_c \) but for \( f = f_c \) and \( f > f_c \) the SRI is under predicted. According to Heckl
(1981), the radiation loss factor can have a significant contribution for structures such as composite structures that are very lightly damped similar to the one investigated in this thesis. As a consequence, the radiation loss factor calculated in narrow band frequency is added to the measured loss factor for better sound transmission loss predictions around the critical frequency as shown in Leping (2009). A good agreement can be seen between the measured curve and the predictions when the theoretical radiation loss factor is added. The same comparison is made for panel 3 and the results are as shown in Figure 5.12.

![Figure 5.12: Sound Reduction Index of Panel 3 without Damping](image)

Similar calculations were done for panel 2 and panel 3 with the damping materials attached and the comparisons are as shown in Figures 5.13 & 5.14. Although the agreements are not as good for the case in Figure 5.11, the accuracy of the predictions can be improved around and above the coincidence frequency.

![Figure 5.13: Sound Reduction Index of Panel 2 with Damping](image)
Figure 5.14: Sound Reduction Index of Panel 3 with Damping
Chapter 6

Conclusions and Future Work

The results in this thesis illustrate that a modified wave propagation approach can be used to predict the sound transmission loss of honeycomb sandwich panels. Radiation loss factor can be a significant contributor for composite structures that are very lightly damped. Structural loss factor alone is not enough to characterise the sound transmission properties of a lightweight panel around the critical frequency. Apart from the structural loss factor, the radiation loss factor calculated in narrow bands must also be included in the model to accurately predict the sound transmission loss of lightweight panels like the ones discussed in this thesis.

The wave numbers and the sound reduction indices calculated using the wave propagation approach agree very well with the measurements. The effect of damping layers on the sound reduction indices of panel 1 and 2 is very small, apart from that related to the added mass. However, for panel 3 a significant increase in the sound reduction index is obtained. This is most likely related to the extended coincidence range of panel 3 as demonstrated by the dispersion curves determined from the measured resonance frequencies. The only structural difference of panel 2 and 3 is the honeycomb cell size affecting the shear stiffness of the core. Without damping treatment, the weighted sound reduction index is 3 dB lower for panel 3 but with the damping treatment both these panels have the same sound insulation property. In-short, the visco-elastic damping treatment increases the sound insulation of an acoustically bad panel (in our case panel 3). Whereas, the same treatment on an acoustically better panel (panel 1 and 2) does not improve the sound insulation. Therefore, during the initial stages of product design, panels can be chosen accordingly to reduce weight if the application of damping material is considered necessary.

Optimisation of these damping treatments are of great interest. For this, a better understanding of the influence of distributed damping on a homogeneous or a sandwich panel is required. Vibrations propagating in structures in the form of mechanical waves of different types are the cause for structure-borne noise. Damping treatments to attenuate these vibration propagations in a built-up structure will be studied.
Chapter 7

Summary of Appended Papers

7.1 Paper A

On Application of Radiation Loss Factor in the Prediction of Sound Transmission loss of a Honeycomb Panel

Leping Feng and Sathish Kumar.

This article shows the importance of radiation loss factor in predicting the sound transmission property of lightly damped panels, especially around the critical frequency region. When the sound transmission properties of a panel is investigated, the radiation loss factor is often neglected for panels unless the structural damping is very less or when the sound transmission around the critical frequency of the panel is of interest. This article shows that by including the theoretical radiation loss factor in narrow bands to the measured structural loss factor, the sound reduction index of honeycomb sandwich panels can be predicted with good accuracy around the critical frequency. The wave propagation constants are calculated as given in Nilsson (1990) and the sound reduction index of the panels are predicted using the orthotropic panel theory. The predictions are validated through laboratory measurements.

7.2 Paper B

Predicting the Sound Transmission Loss of Honeycomb Panels using the Wave Propagation Approach

Sathish Kumar, Leping Feng and Ulf Orrenius.

The wave propagation approach can be used to predict the sound transmission properties of sandwich panels having an isotropic core. In this article, the wave propagation approach is used to predict the sound transmission loss of honeycomb panels. A honeycomb panel is a lightweight sandwich panel with a honeycomb core of hexagon cell. Due to the manufacturing process of honeycomb cores, the cell wall thickness
doubles in the direction of cell orientation and the actual shape of the honeycomb cell can be irregular making the core anisotropic. As an approximation, the honeycomb core can be treated as orthotropic and the whole panel can be treated as a orthotropic sandwich panel. The wave numbers are calculated for the two principle directions. The orthotropic panel theory is used to predict the sound transmission loss of panels with two different core structures. Visco-elastic damping with constraining layers (CLD) are attached to these panel and the sound transmission loss is calculated. The predicted wave numbers and the sound transmission loss of the panels agree well with laboratory measurements.
Chapter 8

Appendix

Functions used to solve for $k$

\[
L_1 = 12E_2(1 - \nu_2)(1 - \nu_1^2)/[E_1t_1^2(1 + \nu_2)(1 - 2\nu_2)]
\]

\[
M_1 = E_2(1 - \nu_1^2)/[2E_1t_1(1 + \nu_2)]
\]

\[
\alpha = \zeta_L\lambda_L ; \quad \beta = \zeta_T\lambda_T
\]

\[
\zeta_L = \left\{ k^2 - \frac{\omega^2\rho_2(1 + \nu_2)(1 - 2\nu_2)}{E_2(1 - \nu_2)} \right\}^{0.5} ; \quad \zeta_T = \left\{ k^2 - \frac{2\omega^2\rho_2(1 + \nu_2)}{E_2} \right\}^{0.5}
\]

\[
\lambda_L = \sqrt{(\omega^2\rho_2/E_2' - k^2)} ; \quad \lambda_T = \sqrt{(\omega^2\rho_2/G_2 - k^2)}
\]
Bibliography


Personal Communication with researchers in Bombardier Transportation.
