ICA 2013 Montreal
Montreal, Canada
2 - 7 June 2013

Structural Acoustics and Vibration
Session 1aSA: Measurement and Modeling of Structures with Attached Noise Control Materials I

1aSA1. Tunneling effect on the sound transmission loss of a flat structure coupled with a porous material
Franck C. Sgard*, Noureddine Atalla, Mohammad Gholami and Hugues Nelisse

*Corresponding author’s address: Direction scientifique, IRSST, 505 Boulvd de Maisonneuve O, Montreal, H3A3C2, QC, Canada, frasga@irsst.qc.ca

It is well known that when measuring Sound Transmission Loss (STL) in a laboratory, among all test conditions, the location of a specimen in an aperture affects the results, due to the tunneling effect. Previous studies have considered this effect for flat single panels and double walls but the case of a panel with attached sound package seems to have received very little attention. This paper deals with the application of a modal approach to study the STL of a rectangular plate coupled with a porous material located inside a tunnel. The sound absorbing material is supposed to be either described by a modal approach or a transfer matrix calculated using a Transfer Matrix Method which relates interstitial pressure and total normal stress on both sides of the material. The model is validated by comparison with Finite Element/Boundary Element computations. Numerical results are shown to illustrate the validity of the proposed full modal and hybrid modal-TMM methodologies and their use to investigate the niche effect in presence of a sound absorbing material.

Published by the Acoustical Society of America through the American Institute of Physics
INTRODUCTION

One of the various reasons for the large deviations in reproducibility Sound Transmission Loss (STL) laboratory tests is the so-called niche or tunneling effect. In STL tests, the test sample is placed inside an opening which separates the two rooms. This opening behaves as an acoustic duct at low frequencies and alters the sound field on both sides of the sample. The niche effect has been shown to be significant below the coincidence frequency both experimentally and theoretically. Several experimental studies revealed that the values of STL depend on the location of the sample in the tunnel [1–5]. For subcoincidence frequencies, a lower STL is achieved when the sample is located in the middle position along the tunnel depth compared to the case without tunnel whereas a higher STL is obtained when the sample is flush mounted at one end of the tunnel. The STL is also affected by the lateral size and the depth of the tunnel. Several models have been proposed in previous studies to explain and predict the niche effect. Kim et al [6] considered a simply supported thin panel of finite height and infinite width placed in a rigid tunnel embedded in an infinite rigid baffle. They calculated the STL of this system using normal modes expansion for the sound pressure fields and the panel displacement field. They explained the niche effect by a modification of the vibration pattern and radiation efficiency of the plate which, in turn, modifies the transmission loss. Their results confirmed that the tunneling effect plays an important role in STL measurements. Vinokur [7] simulated the tunneling effect for a single panel inside a niche using a simplified mechanical model of an infinite extent triple partition in which the panel stands for the middle leaf and the air masses moving at the aperture edges work as the edge leaves. This model is approximately valid for frequencies \( f < c_0 / \sqrt{S} \), with \( c_0 \) the speed of sound in air and \( S \) the tunnel cross section area. The author claims that the difference does not depend on the parameters of the test panel, but rather on the aperture dimensions and frequency. Dijckmans and Vermeir [8] used a wave based method (WBM) to predict the STL of simply supported single and double leaf panels placed in a tunnel between two 3D rectangular reverberant rooms. They found the same trends regarding the niche effect for single and double walls but this effect is more pronounced for double walls especially at mid-frequencies since the sound transmission through such systems is highly dependent on angle of incidence in this frequency range. Sgard et al [9] presented preliminary results regarding the niche effect for flat panels coupled with porous materials using a mixed variational boundary/finite element approach. Previous studies have mainly considered this effect for flat single panels and double walls but the case of a panel with attached sound package seems to have received very little attention. This paper deals with the application of a modal approach to study the sound transmission loss of a rectangular plate coupled with a porous material located inside a tunnel. The sound absorbing material is supposed to be either described by a modal approach or a transfer matrix relating interstitial pressure and total normal stress on both sides of the material. The first objective of this paper is to quantify the niche effect when a porous material is attached to the test sample. The second objective is to evaluate if a simplified Transfer Matrix Method (TMM) approach can be used to predict the STL of a structure in a niche with and attached Noise Control Treatment instead of a more detailed model. Next section presents the theoretical background. Preliminary comparisons with Finite Element/Boundary Element computations are then presented to validate the model and illustrate the validity of the proposed approach.

THEORY

Figure 1 depicts the studied configuration and the used coordinate system. It consists of a simply supported homogeneous isotropic elastic thin panel located inside a rectangular cross section tunnel of size \( S = 2a \times 2b \) and depth \( L_n \). The tunnel ends are embedded in rigid baffles and radiate in semi-infinite fluid domains with respective sound speed \( c_1 \) and \( c_2 \), density \( \rho_1 \) and \( \rho_2 \). The panel subdivides the niche into two acoustic sub-cavities of respective length \( L_1 \) and \( L_2 \), sound speed \( c_{c,1} \) and \( c_{c,2} \), density \( \rho_{c,1} \) and \( \rho_{c,2} \). The panel is treated on one side with an absorbing material of thickness \( L_n \). It is excited by an oblique incidence plane wave or a diffuse sound field. In all the following, a \( e^{j\omega t} \) temporal dependency is assumed so that a variable \( X(M,t) \) is written as \( \Re[\hat{X}(M)e^{j\omega t}] \).
Panel Equations

The panel is assumed to vibrate in flexural motion. The associated weak integral form is given by:

\[
\int_{\Omega} \left[ \hat{\sigma}_e \left( \hat{u}_e \right) : \hat{\varepsilon}_e \left( \hat{\varphi} \right) - \rho_s \omega^2 \hat{u}_e \cdot \hat{\varphi} \right] d\Omega - \int_{\partial \Omega_e} \left[ \hat{\sigma}_{e,n} \hat{n}_e \right] \hat{\varphi} d\Gamma = 0 \quad \forall \hat{\varphi}
\]

where \( \hat{u}_e \) is the displacement field, \( \hat{\sigma}_e \) and \( \hat{\varepsilon}_e \) are the stress and strain tensors, \( \hat{\varphi} \) is a test function satisfying the kinematic boundary conditions of the problem, \( n_e \) is the outward normal of the panel. \( \rho_s \) is the plate’s density.

The transverse displacement field is expanded in terms of its in-vacuo eigenmodes:

\[
\hat{u}_e = \sum \hat{C}_y \varphi_n \left( x, y \right)
\]

with

\[
\varphi_n \left( x, y \right) = \varphi_{mn} \left( x, y \right) = \sin \left( \frac{m\pi}{2a} \left( x+a \right) \right) \sin \left( \frac{n\pi}{2b} \left( y+b \right) \right)
\]

\[FIGURE 1. Description of the problem\]

Subcavities Equations

In each subcavity \( \Omega_{x,j} \), the weak integral formulation is given by:

\[
\int_{\Omega_{x,j}} \left( \frac{1}{\rho_{c,j}} \nabla \hat{p}_j, \nabla \hat{\varphi}_j - \frac{1}{\rho_{c,j} c_{c,j}^2} \hat{p}_j \hat{\varphi}_j \right) d\Omega - \int_{\partial \Omega_{x,j}} \left( \frac{1}{\rho_{c,j} \omega^2} \hat{p}_j \hat{\varphi}_j \right) n_{c,j} d\Gamma = 0 \quad \forall \hat{\varphi}_j, j = 1, 2
\]

where \( \hat{\varphi}_j \) is a test function satisfying the kinematic boundary conditions of the problem. \( n_{c,j} \) is the outward normal of each subcavities. Each subcavity is supposed to have rigid lateral walls so that \( \frac{\partial \hat{p}_j}{\partial n_{c,j}} = 0 \) over \( \Sigma_j \).

The acoustic field can be written in terms of propagating and evanescent modes in each subcavity:
\[
\hat{p}_j(x, y, z) = \sum_p \left( \hat{A}_{j,p} e^{-jk_{p}^{(j)}(z-z_j)} + \hat{B}_{j,p} e^{jk_{p}^{(j)}(z-z_j)} \right) \phi_p(x, y) \quad z_0, j \leq z \leq z_j \quad j = 1, 2
\]  

where \(z_{0,1} = 0\), \(z_1 = L_1\) and \(z_{0,2} = L_1 + L_a\), \(z_2 = L_n\) and \(k_{p}^{(j)} = k_{pq}^{(j)} = \sqrt{k_{c,j}^2 - \left(\frac{p\pi}{2a}\right)^2 - \left(\frac{q\pi}{2b}\right)^2}\)

\[k_{c,j} = \frac{\omega}{c_{c,j}}, \phi_p(x, y) = \phi_{pq}(x, y) = \cos\left(\frac{p\pi}{2a}(x+a)\right) \cos\left(\frac{q\pi}{2b}(y+b)\right)\]

Sound Absorbing Material Equations

The porous material occupying the volume \(\Omega\) with boundary \(\partial \Omega\) is supposed to behave as an equivalent fluid (limp model) with effective density \(\bar{\rho}_a\) and bulk modulus \(\bar{K}_a\) [10]. Two approaches are considered. The porous material is either described using transverse modes or using a transfer matrix. Both descriptions are described in the following.

Modal Description

The interstitial pressure \(\hat{p}_a\) inside the porous material satisfies an equation similar to Eq(2) in which \(\Omega_{e,j}, \rho_{e,j}, c_{e,j}, \hat{Y}_j\) are replaced by \(\Omega_a, \bar{\rho}_a, \bar{K}_a, \hat{\rho}_a\) and \(\hat{Y}\) respectively. It can thus be expanded in terms of transverse modes:

\[
\hat{p}_a(x, y, z) = \sum_p \left( \hat{A}_{a,p} e^{-jk_{a}^{(p)}(z-z_a)} + \hat{B}_{a,p} e^{jk_{a}^{(p)}(z-z_a)} \right) \phi_p(x, y) \quad L_1 \leq z \leq z_a = L_1 + L_a
\]  

where \(k_{a}^{(p)} = k_{pq}^{(a)} = \sqrt{k_a^2 - \left(\frac{p\pi}{2a}\right)^2 - \left(\frac{q\pi}{2b}\right)^2}, k_a = \omega \sqrt{\frac{\bar{\rho}_a}{\bar{K}_a}}\)

Transfer Matrix Approach

The absorbing material is supposed to be alternatively described by a transfer matrix linking the total normal stress \(\hat{\sigma}_a\) and total normal displacement \(\hat{w}_{p,a}\) at point A on one side of the material to that at point B on the other side:

\[
\begin{bmatrix}
\hat{\sigma}_a \\
\hat{w}_{p,B}
\end{bmatrix} =
\begin{bmatrix}
D_{11}(k_i, \omega) & D_{12}(k_i, \omega) \\
D_{21}(k_i, \omega) & D_{22}(k_i, \omega)
\end{bmatrix}
\begin{bmatrix}
\hat{\sigma}_B \\
\hat{w}_{p,A}
\end{bmatrix}
\]  

where:

\[
D_{11} = T_{11} - \frac{T_{22}}{T_{22}}; D_{12} = j\omega \frac{T_{12}}{T_{22}}; D_{21} = \frac{j T_{21}}{\omega T_{22}}; D_{22} = \frac{1}{T_{22}}
\]  

\(k_i\) is the imposed transverse component (in plane (x,y)) of the wave vector in the sound absorbing material and is assumed to be equal to 0 in the following. Components, \(T\), have very simple analytical expressions [10].
External Pressure Fields

In medium 1, the total pressure is \( \hat{p}_s + \hat{p}_r \) where \( \hat{p}_s = \hat{p}_b + \hat{p}_r \) is the blocked pressure and \( \hat{p}_r \) is the sound pressure radiated by the baffled panel in medium 1. The pressure radiated in medium 1 \( (\hat{p}_r, M_0) \) can be written as:

\[
\hat{p}_r(M) = \int_{\Sigma} G^{(1)}(M, M_0) \frac{\partial \hat{p}_r}{\partial z} (M_0) d\Gamma(M_0)
\]

where \( G^{(1)}(M, M_0) \) is the baffled Green’s function given by \( G^{(1)}(M, M_0) = \frac{e^{-jkr}}{2\pi R} \) with \( R = \sqrt{(x-x_0)^2 + (y-y_0)^2 + z^2} \).

The transmitted pressure field in medium 2 \( (\hat{p}_t, M_0) \) satisfies the relation:

\[
\hat{p}_t(M) = -\int_{\Sigma} G^{(2)}(M, M_0) \frac{\partial \hat{p}_r}{\partial z} (M_0) d\Gamma(M_0)
\]

where \( G^{(2)}(M, M_0) = \frac{e^{-jkr}}{2\pi R} \) with \( R = \sqrt{(x-x_0)^2 + (y-y_0)^2 + (z-z_0)^2} \).

Interface Conditions

The coupling conditions between the panel and subcavity \( (\partial\Omega_{c,1}) \) read:

\[
\begin{align*}
\hat{\sigma}_c n_e &= -\hat{p}_t n_e \\
\frac{\partial \hat{p}_t}{\partial n_{e,1}} &= -\rho_{e,1} \omega^2 \hat{u}_e n_e
\end{align*}
\]

The coupling conditions between the panel and the sound absorbing material \( (\partial\Omega_{c,2}) \) are given by:

\[
\begin{align*}
\begin{cases}
\hat{\sigma}_e n_e &= -\hat{p}_t n_e \\
\hat{u}_e n_e &= \frac{1}{\rho_e \omega^2} \frac{\partial \hat{p}_a}{\partial n_e}
\end{cases}
\quad \text{or} \quad
\begin{cases}
\hat{\sigma}_e n_e &= -\hat{\sigma}_t n_e \\
\hat{u}_e n_e &= \hat{w}_p
\end{cases}
\end{align*}
\]

Eq(10-a) and Eq(10-b) correspond to the modal description and the transfer matrix approach of the porous material, respectively.

The coupling conditions between the sound absorbing material and subcavity 2 \( (\Sigma_a) \) for the modal description and the transfer matrix approach of the porous material are given by Eq(11-a) and Eq(11-b), respectively:

\[
\begin{align*}
\begin{cases}
\hat{p}_a = \hat{p}_a \\
\frac{1}{\rho_{e,2} \omega^2} \frac{\partial \hat{p}_a}{\partial n_{e,2}} &= \frac{1}{\rho_{a,2} \omega^2} \frac{\partial \hat{p}_a}{\partial n_{a,2}}
\end{cases}
\quad \text{or} \quad
\begin{cases}
\hat{p}_a = \hat{\sigma}_t' \\
\frac{1}{\rho_{a,2} \omega^2} \frac{\partial \hat{p}_a}{\partial n_{a,2}} &= -\hat{w}_p
\end{cases}
\end{align*}
\]
The coupling conditions between the sound pressure fields in the subcavities and the semi-infinite media 1 and 2 read:

$$(a) \begin{cases} \hat{p}_r + \hat{p}_b = \hat{p}_1 \\ \frac{\tilde{p}}{\rho_c} \frac{\tilde{p}}{\rho_c} = \frac{\tilde{p}}{\rho_c} \frac{\tilde{p}}{\rho_c} \end{cases} \text{ over } \Sigma_F \quad (b) \begin{cases} \hat{p}_1 = \hat{p}_2 \\ \frac{\tilde{p}}{\rho_c} \frac{\tilde{p}}{\rho_c} = \frac{\tilde{p}}{\rho_c} \frac{\tilde{p}}{\rho_c} \end{cases} \text{ over } \Sigma_g \quad (12)$$

**Systems to be solved**

Substituting the modal expansions of $\hat{w}, \hat{p}_1, \hat{p}_2, (\hat{p}_a$ for the porous material modal description) together with $\hat{q} = \varphi_j (x, y) e^{i k x}$, $\hat{Y}_i (M) = \phi_i (x_M, y_M)$, $(\hat{Y}_i (M) = \phi_i (x_M, y_M)$ for the porous material modal description) into the weak integral formulation of each domains, using the interface conditions and the orthogonality of normal modes lead to the following systems:

$$\begin{align*}
&\left( K_{MM} - \omega^2 M_{MM} \right) \hat{C}_r - \sum_p \left( \hat{A}_{r,p} + \hat{B}_{r,p} \right) L_{r,MM} + \sum_p \left( \hat{A}_{a,p} e^{i \omega t} + \hat{B}_{a,p} e^{-i \omega t} \right) L_{a,MM} = 0 \\
&\sum_p L_{MM} \hat{C}_r - \frac{1}{\rho_c} \sum_{M} (j M) \left( \hat{A}_{r,p} - \hat{B}_{r,p} \right) L_{r,MM} = 0 \\
&\left( \hat{A}_{a1,p} + \hat{B}_{a1,p} \right) N_{a,MM} + \frac{1}{\rho_p} \sum_p \hat{k}_{a,p} \left( \hat{A}_{a,p} - \hat{B}_{a,p} \right) N_{a,MM} = 0 \\
&\left( \hat{A}_{a2,p} + \hat{B}_{a2,p} \right) N_{a,MM} + \frac{1}{\rho_p} \sum_p \hat{k}_{a,p} \left( \hat{A}_{a,p} - \hat{B}_{a,p} \right) N_{a,MM} = 0 \\
&\left( \hat{A}_{a1,p} + \hat{B}_{a1,p} \right) N_{a,MM} + \frac{1}{\rho_p} \sum_p \hat{k}_{a,p} \left( \hat{A}_{a,p} - \hat{B}_{a,p} \right) N_{a,MM} = 0
\end{align*}$$

$$\begin{align*}
&\left( K_{MM} - \omega^2 M_{MM} + D_{11} (0, \omega) N_{a,MM} \right) \hat{C}_a - \sum_p \left( \hat{A}_{a,p} + \hat{B}_{a,p} \right) L_{a,MM} + D_{11} (0, \omega) \sum_p \left( \hat{A}_{a,p} e^{i \omega t} + \hat{B}_{a,p} e^{-i \omega t} \right) L_{a,MM} = 0 \\
&\sum_p L_{a,MM} \hat{C}_r - \frac{1}{\rho_c} \sum_{M} (j M) \left( \hat{A}_{a,r} - \hat{B}_{a,r} \right) L_{a,MM} = 0 \\
&\left( \hat{A}_{a1,r} + \hat{B}_{a1,r} \right) N_{a,MM} + \frac{1}{\rho_p} \sum_p \hat{k}_{a,r} \left( \hat{A}_{a,r} - \hat{B}_{a,r} \right) N_{a,MM} = 0 \\
&\left( \hat{A}_{a1,r} + \hat{B}_{a1,r} \right) N_{a,MM} + \frac{1}{\rho_p} \sum_p \hat{k}_{a,r} \left( \hat{A}_{a,r} - \hat{B}_{a,r} \right) N_{a,MM} = 0 \\
&\left( \hat{A}_{a1,r} + \hat{B}_{a1,r} \right) N_{a,MM} + \frac{1}{\rho_p} \sum_p \hat{k}_{a,r} \left( \hat{A}_{a,r} - \hat{B}_{a,r} \right) N_{a,MM} = 0
\end{align*}$$

(13)
Where

\[
N_{e,M}^2 = N_{e,mm}^2 = \int_{\Omega_{x,y}} \phi_{mm}^2 (x,y) \, dx \, dy = ab
\]

(15)

\[
K_{MM} = K_{mmmm} = D \left( \frac{m \pi}{L_x} \right)^4 + 2 \left( \frac{m \pi}{L_x} \right)^2 \left( \frac{n \pi}{L_x} \right)^2 + \left( \frac{n \pi}{L_x} \right)^4 \right) N_{e,mm}^2
\]

(16)

\[
M_{MM} = M_{mmmm} = \rho_s h N_{e,mm}^2
\]

(17)

\[
L_{PM} = L_{pmmm} = \int_{\Omega_{x,y}} \phi_{pq} (x,y) \phi_{mm} (x,y) \, dx \, dy
\]

(18)

\[
N_{a,M}^2 = N_{a,mm}^2 = \int_{\Omega_{x,y}} \phi_{mm}^2 (x,y) \, dx \, dy = \varepsilon_m \varepsilon_s ab \quad \varepsilon_0 = 2 \quad \varepsilon_m = 1 \text{ if } m \neq 0
\]

(19)

\[
F_{a,M} = \int_{\Sigma_p} \hat{p}_b (x,y) \phi_M (x,y) \, d\Gamma (M)
\]

(20)

\[
\hat{Z}_{MN}^{(i)} = \frac{j \alpha_p}{S} \int_{\Sigma_S} \phi_M (x_M, y_M) G^{(i)} (M, M_0) \phi_p (x_M, y_M) \, d\Gamma (M) \, d\Gamma (M) \quad i = 1,2
\]

(21)

Eq(13) corresponds to the case where the interstitial pressure field inside the porous material is expanded in terms of normal modes whereas Eq(14) is associated to the case where the porous material is replaced by a transfer matrix. \( \hat{Z}_{MN}^{(i)} \) denotes the modal radiation impedance matrix of subcavity \( i \) into the external fluid. It can be calculated as described in [11]. Eq(13) and Eq(14) are first solved in terms of modal amplitudes \( \hat{A}_{1,M}, \hat{B}_{1,M}, \hat{A}_{2,M}, \hat{B}_{2,M}, \hat{A}_{a,M}, \hat{B}_{a,M}, \hat{C}_M \). Next, the sound power transmitted by the system is readily computed together with the STL.

**RESULTS**

As preliminary results, consider a 8mm thick aluminum panel of dimension 0.6mx0.4m. It is placed in the middle of a niche of total length 8.8cm. A 2.54cm thick melamine foam slab can be coupled to the panel. The system is excited by a diffuse acoustic field. All the fluids (external and internal) are identical (air). The physical parameters used in the calculation are shown in TABLE1. FIGURE1 compares the diffuse field sound TL of the system calculated with the present approaches to an in-house variational direct boundary element/ finite element method based software. In the modal approach, modes up to a maximum frequency of \( 1.5 \times f_{\text{max}} \) for the structure and \( 2 \times f_{\text{max}} \) for the cavities and the porous material are kept in the modal expansions where \( f_{\text{max}} = 1500\text{Hz} \). The diffuse field transmitted acoustic power is calculated numerically using 16x16 Gauss points and a limit angle of 78deg. The finite element mesh consisted of 40 elements along \( x \), 30 along \( y \), 4 elements along \( z \) for subcavity1, 2 elements along \( z \) for the panel, 10 elements along \( z \) for the melamine and 2 elements along \( z \) for subcavity2. 8-noded hexahedron elements were used for the fluids, the porous material and the panel (modeled as a solid). 4-nodes quadrangle BEM elements were used at the two ends of the tunnel to estimate the radiation impedance matrices and the transmitted acoustic field. FIGURE1 shows that there is an excellent agreement between the analytical modal model and the numerical BEM/FEM approach in the case where the porous material pressure field is represented in terms of lateral modes. The STL of the panel without the attached porous material is also plotted to show the effect of the treatment. The effect of the melamine on the STL is particularly important above 1kHz where an increase is observed as expected (this increase is governed by the absorption coefficient of the foam). In the case of the TMM approach for the porous material using a transverse wavenumber \( k = 0 \) the agreement is not as good (maximum differences of 2-3dB in the frequency band [600-1200Hz]) but it still captures very well the effect of the porous material above 1.2kHz. However, a detailed parameter analysis (trace wavenumber, thickness, type of materials…) is needed to correctly assess the advantages vs. limitations of this approximate method.
TABLE 1. Parameters used in the calculation

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Panel density (kgm⁻³)</td>
<td>2742</td>
</tr>
<tr>
<td>Panel Young’s modulus (Pa)</td>
<td>6.9 10¹⁰</td>
</tr>
<tr>
<td>Panel Poisson’s ratio</td>
<td>0.33</td>
</tr>
<tr>
<td>Panel loss factor</td>
<td>0.03</td>
</tr>
<tr>
<td>Melamine Porosity</td>
<td>0.99</td>
</tr>
<tr>
<td>Melamine Flow resistivity (Nsm⁻⁴)</td>
<td>10900</td>
</tr>
<tr>
<td>Melamine Tortuosity</td>
<td>1.02</td>
</tr>
<tr>
<td>Melamine Viscous Characteristic Length (m)</td>
<td>10⁻⁴</td>
</tr>
<tr>
<td>Melamine Thermal Characteristic Length (m)</td>
<td>1.3 10⁻⁴</td>
</tr>
<tr>
<td>Melamine matrix density (kgm⁻³)</td>
<td>8.8</td>
</tr>
<tr>
<td>Air density (kgm⁻³)</td>
<td>1.213</td>
</tr>
<tr>
<td>Air sound speed (ms⁻¹)</td>
<td>342.2</td>
</tr>
</tbody>
</table>

FIGURE 2. Diffuse field Sound TL of an aluminum panel located inside a niche with or without attached 1” thick melamine foam. Comparisons between the results obtained from a finite element/variational boundary element approach and the proposed approaches (full modal and modal/TMM for the porous material (PM)).

CONCLUSION

This paper presented two models based on a modal approach to study the sound transmission loss of a rectangular panel coupled with a porous material located inside a tunnel. The sound absorbing material which was supposed to behave as a limp equivalent fluid, was either described by a modal approach (first model) or a transfer matrix calculated using a Transfer Matrix Method (second model). Preliminary results show that there is an
excellent agreement between the results obtained with the model based on the modal expansion of the interstitial pressure field in the porous material and those computed with a Finite Element/Boundary Element approach. The model based on a Transfer Matrix Method using a null transverse wavenumber captures the physics very well in the tested configuration especially above 1.2kHz but differences of levels are observed in some frequency bands. It is expected that the niche effect is decreased by the presence of the sound absorbing material. The presented model being fast and accurate, it will allow for parametric studies to evaluate quantitatively the sensitivity of the niche effect to the presence of a sound absorbing material. Also, it will be used to assess if a simplified TMM approach can be used to predict the STL of a structure with and attached Noise Control Treatment instead of a detailed model. A detailed analysis of both will be presented during the conference.

REFERENCES