Alternative Methods for the Measurement of Panel Transmission Loss under Diffuse Acoustic Field Excitation

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ABSTRACT

Three different approaches are described that aim at providing alternative transmission loss measurements to standards, either using a classical coupled room arrangement or a different laboratory installation. The first one consists in reproducing a random excitation at a panel surface using a synthetic array approach, needing only a single monopole source in hemi-anechoic conditions. This approach removes the need of a reverberant room to generate a diffuse acoustic field excitation, and allows estimating the panel behaviour under other random excitations such as a turbulent boundary layer. The two others possibilities describes transmission loss estimations in a standard reverberant-anechoic rooms arrangement, but using 'single quantity' measurements. The transmission loss is first derived using sound intensity measurements only, performed as an insertion loss test (radiated and incident sound intensity are measured when a panel is installed or not). The transmission loss is then derived using vibration measurements only. The virtual fields method allows identifying the parietal pressure field exciting the panel and estimating the corresponding incident acoustic power, while the radiated acoustic power is calculated using the radiation resistance matrix method. For the three approaches, results for a small aluminium panel and a large composite panel are provided, illustrating their respective potentials.

Keywords: Transmission loss, coupled rooms, sound power
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1. INTRODUCTION

The Transmission Loss (TL) of a panel is defined as a ratio of incident sound power $\Pi_{inc}$ on a test specimen to radiated sound power $\Pi_{rad}$ transmitted through the test specimen, that is

$$ TL = 10 \log_{10}(\Pi_{inc}/\Pi_{rad}). \quad (1) $$

TL measurements are subject to variability and repeatability issues, especially in the low frequency domain [1, 2]. Another key issue in these measurements is 'how determining the two acoustic powers $\Pi_{inc}$ and $\Pi_{rad}$ than can hardly be evaluated in a straightforward manner?'. so that they are usually indirectly estimated via the measurement of other quantities like sound pressure and sound intensity (as made in [3, 4]).

Several approaches concerning different laboratory arrangements, or different methods for estimating incident and/or radiated sound powers can be found in the literature, and three others are presented in this paper. Such 'alternative' methods can be divided in two main categories :

(1) Those which suggest different laboratory setups than those used in standards, by synthetizing a Diffuse Acoustic Field (DAF) excitation on the surface of a partition to be tested as an example [5–7] (and thus removing the need of a reverberant room as a source room in this case).

(2) Those which rely on a classical coupled reverberant-anechoic room arrangement, by estimating incident and radiated sound powers by other means than the 'pressure-pressure' [3] or 'pressure-intensity' [4] standardized measurements. In this work, both incident and radiated sound power are recovered from the
measurement of the same quantity, either sound intensity or vibration field. Note that the incident sound power was estimated using near field acoustic holography in [8], while the transmitted sound power was evaluated using a peak envelope method [9] or laser Doppler vibrometer (LDV) and a Rayleigh integral approach in [10].

2. STANDARD MEASUREMENTS

For measurements using two coupled reverberant rooms [3] and assuming diffuse field conditions in both chambers, the incident power \( \Pi_{inc} \) is related to the spatially-averaged mean square pressure level in the source room, while transmitted power \( \Pi_{rad} \) is related to the spatially-averaged mean square pressure level in the receiving room and the equivalent sound absorption area:

\[
\Pi_{inc} = \frac{p_1^2 S}{4 \rho_0 c_0},
\]

(2)

and

\[
\Pi_{rad} = \frac{p_2^2 A}{4 \rho_0 c_0},
\]

(3)

where \( p_1^2 \) and \( p_2^2 \) are the spatially-averaged quadratic sound pressure in the source and receiving room, respectively, \( S \) is the panel area, \( A \) is the equivalent sound absorption area in the receiving room and \( \rho_0 c_0 \) is the specific acoustic impedance of air. Using sound pressure levels \( L_{p1} \) and \( L_{p2} \) in the source and receiving room respectively (in dB, reference \( 2 \times 10^{-5} \text{Pa} \)), and inserting Eqs.(2-3) in Eq.(1), the following relation is obtained

\[
TL = L_{p1} - L_{p2} - 10 \log_{10} \frac{A}{S}.
\]

(4)

When using coupled reverberant anechoic rooms, the incident power is still estimated following Eq.(2) but the radiated sound power is now determined by surface averaging the normal sound intensity \( I_n \) on a measurement surface \( S_m \), so that \( \Pi_{rad} \) now equals \( I_n S_m \). With the sound intensity level \( L_i = 10 \log_{10} (I_n/I_0) \), with \( I_0 = 10^{-12} \text{W/m}^2 \), and Eqs.(1,3), the Transmission Loss is now given by

\[
TL = L_{p1} - L_i - 6 - 10 \log_{10} \frac{S}{S_m}.
\]

(5)

The averaging surface \( S_m \) being generally nearly equal to the panel area \( S \), the last term in the last equation is generally neglected. Whatever the considered standard [3, 4], the global accuracy of sound pressure or sound intensity measurements influences repeatability and reproducibility of TL estimations, so as the way the sound pressure is spatially averaged in the considered room [2, 11]. At low frequency, TL measurements are especially dependent on the size of the test chambers and the receiving room equivalent sound absorption area.

3. TL ESTIMATION USING SYNTHETIZED DIFFUSE ACOUSTIC FIELD

3.1 Measurement principle

The use of a synthetic array, for which a small array element is moved to create a large array by post-processing [6, 7] is considered in this section. Bravo and Maury [5] also suggested the reproduction of a diffuse pressure field on the surface of a partition to be tested, using a near-field array of 16 loudspeakers. Marchetto et al. [12] recently proposed to use measured sensitivity functions (determined using a reciprocity principle) to experimentally characterize a panel response under DAF excitation. In all these cited works, the target pressure field (DAF) to be reproduced is defined by its Cross-Spectral Density (CSD) function, which was theoretically found to follow a sinc function behavior for a perfect diffuse acoustic field [13], which writes \( \sin(k_0 x)/k_0 x \) where \( k_0 \) is the acoustic wavenumber and \( x \) is the vector distance between two points.

The concept of the measurement is given in Fig. 1(a). The description is deliberately kept short, and additional details concerning the method can be found in [6, 7].

The incident power is calculated using transfer functions (between successive monopole positions and panel surface) and monopole amplitudes in anechoic conditions (no reverberation room further needed).
More precisely, the CSD matrix $S_{pp}^{\text{panel}}$ of reconstructed surface pressure on the panel is calculated using the matrix of theoretical Green’s functions $G$ and the reproduction source accelerations $CSD S_{q\dot{q}}$

$$S_{pp}^{\text{panel}} = \rho_0^2 G S_{q\dot{q}} G^H,$$

and finally the mean quadratic pressure on the panel $\langle p_{RMS}^2 \rangle = \frac{1}{N} Tr \left( S_{pp}^{\text{panel}} \right)$, where $Tr$ denotes the trace of a matrix. Since wall pressure fluctuations are considered instead of mean quadratic pressure at large distance from the panel (like in Eqs.(2-3)), the incident power is now calculated following

$$\Pi_{\text{inc}}^{\text{SYN}} = \frac{\langle p_{RMS}^2 \rangle S}{8 \rho_0 c_0},$$

where the factor 8 accounts for the pressure doubling at the panel surface (blocked pressure hypothesis).

Then the radiated power is calculated using scanning laser vibrometer measurements and a radiation resistance matrix method. The matrix of the $N \times M$ measured FRFs between the $M$ source volume accelerations and the $N$ panel velocity responses is noted $H(\omega)$ (see figure 1-(a)). Using this matrix, the calculation of the vibration velocity CSD of the panel with a CSD of the source volume accelerations (calculated with either a WFS, an holographic or a Least-Squares approach [6]) is straightforward and given by the equation

$$S_{vv}^{\text{panel}} = H S_{q\dot{q}} H^H.$$

The corresponding radiated acoustic power is calculated using

$$\Pi_{\text{rad}}^{\text{SYN}} = Tr \left( S_{vv}^{\text{panel}} R_{\text{rad}} \right),$$

where $R_{\text{rad}}$ is the element radiation matrix [14] of dimension $[N \times N]$, defined as

$$R_{\text{rad}} = \frac{\omega^2 \rho_0 A^2}{4\pi c_0} \frac{\sin k_0 r_{i,j}}{k_0 r_{i,j}},$$

where $k_0$ is the acoustic wavenumber, $A$ is the elemental area of each measurement point and $r_{i,j}$ is the distance from element $i$ to element $j$. Eqs. (9,10) hold for any boundary conditions of the plate and assume that the plate is baffled and that the receiving space is anechoic.

### 3.2 Laboratory measurements

Using a single acoustic monopole and a scanning laser vibrometer, the TL of a small aluminum panel and a large composite panel are obtained under a synthetized DAF excitation. The small aluminium was baffled and tested in a hemi-anechoic room. Pictures of this experiment are given in figure 2, and additional details can be found in [6, 7]. The representativity of simply supported boundary conditions realized on the panel
Figure 2: (a) Front view: The baffled panel stands in front of the laser vibrometer, and is surrounded by anechoic panels – (b) Rear view: A monopole source is on its stand, the microphone used for the measurement of the experimental transfer function between the wall pressure and the monopole can be seen (5 inches of glass wool are added on the floor to improve anechoicity).

Figure 3. Comparison between a FEM simulation result and an experimental result. These results are discussed and validated in [15]. A FEM calculation result and an experimental result for the transmission are given in Fig. 3, and both results are in close agreement.

Concerning the composite panel, measurements using a coupled room arrangement and a synthetized Diffuse Acoustic Field are compared. The panel is mounted in the existing niche between the two room (flush mounted on the reverberant room side), and the TL is first calculated following Eq.(5). The second method uses a synthetic antenna, the calculation of CSD reproduction source amplitudes and the measurement of the panel’s vibration response to calculate the TL of the panel at a post-processing step. The concept of this measurement is schematically depicted in Fig. 1(b), and pictures are given in Fig. 4. Note that a hemi-anechoic space was created in the reverberant room using rockwool dihedrons and rockwool panels on floor and ceiling, in order to obtain experimental conditions that allow calculating the reproduced pressure or reproduction source amplitudes using theoretical Green’s functions. Additional details concerning this experiment can be found in [7].

TI results for the standard calculation and the approach using a synthetized field are given in Table 1. Differences between the two measurements are distributed, with a maximum difference of 2.1 dB for the 1000 Hz third octave band. Compared with TL estimation discrepancies that can be obtained for different laboratories using the coupled rooms arrangement, the differences obtained in the present case between the two measurement methods are generally lower or comparable. This allows validating the synthetic antenna approach for measuring the TL of panels under a synthesized DAF excitation, and opens interesting perspec-
Figure 4: Pictures of the two experimental setups – (a) Panel seen from the anechoic room with the sound intensity probe – (b) Panel seen from the reverberant room with one of the acoustic sources and the microphone on its rotating arm – (c) Panel seen from the anechoic room with the scanning laser measurement – (d) Motorized monopole source on the reverberant room side, with dihedrons positioned in the room to create a hemi-anechoic space in the reverberant room.

Figure 5: TL measurement results using a 345 source synthetic antenna and a coupled rooms arrangement – (a) Narrow band result for the coupled rooms measurement is indicated by a thin dotted black line (continuous blue line for the synthetic antenna measurement) – (b) Third octave band results for the coupled rooms measurement are indicated by a thick gray line, with square markers (blue line and circle markers for the synthetic antenna measurement).
Table 1: Third octave bands results for the two methods (C.R. = Coupled Rooms ; S.A. = 345 source synthetic antenna).

<table>
<thead>
<tr>
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<th>Third octave band center frequencies [Hz]</th>
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<tbody>
<tr>
<td></td>
<td>200</td>
</tr>
<tr>
<td>1 - C.R. [dB]</td>
<td>15.4</td>
</tr>
<tr>
<td>2 - S.A. [dB]</td>
<td>16.4</td>
</tr>
<tr>
<td>Delta (1-2) [dB]</td>
<td>-1</td>
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</tbody>
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atives for such measurements. It is worth that the procedure also theoretically allows estimating the panel transmission loss under other random pressure fields such as a Turbulent Boundary Layer excitation [6].

4. TL ESTIMATION USING SOUND INTENSITY MEASUREMENTS ONLY

In this section, the possibility of estimating the TL of plane panels using sound intensity measurements only is considered. The test is now seen as an insertion loss test : mean sound intensity radiated by the panel or incident on the panel is measured when the panel is installed in the niche ($I_{with}$) or not ($I_{without}$), and surface averaged on a similar area $S_m$. Incident power $\Pi_{inc}$ equals $I_{without}S_m$ while radiated power is $I_{with}S_m$ (the same quantity than for standard TL measurements using coupled anechoic-reverberant room). Using the sound intensity levels $I_{without}$ and $I_{i}$ previously measured ($= 10 \log_{10}(I_{i}/I_0)$, with $I_0 = 10^{-12}$ W/m$^2$), the Transmission Loss is simply given by

$$TL = L_{without} - L_{i}.$$ (11)

The TL of two panels was evaluated using this approach : a small isotropic aluminium panel (previously described and used in Section 3.2), and a large composite panel of area $1.5 \times 1 m^2$ which properties are given in Tab.2. It is composed of a thick composite core between two thin carbon facesheets, and was previously characterized in [16]. A sound intensity probe composed of two 1/2 in. microphones and a 12 mm spacer was used for both panels. The pressure intensity index (PI index) value was systematically checked to verify the ‘smaller than 10 dB’ rule of thumb for this index. This rule was satisfied for all tests, with a maximum measured value of 7.4 dB.

Table 2. Physical properties of the orthotropic composite panel.

<table>
<thead>
<tr>
<th></th>
<th>$\rho$ [kg/m$^3$]</th>
<th>h [mm]</th>
<th>$E_1$ [GPa]</th>
<th>$E_2$ [GPa]</th>
<th>$E_3$ [GPa]</th>
<th>$G_{12}$ [GPa]</th>
<th>$G_{13}$ [GPa]</th>
<th>$G_{23}$ [GPa]</th>
<th>$\nu_{12}$</th>
<th>$\nu_{13}$</th>
<th>$\nu_{23}$</th>
</tr>
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<tbody>
<tr>
<td>Facesheet</td>
<td>1900</td>
<td>0.96</td>
<td>46</td>
<td>46</td>
<td>46</td>
<td>17.6</td>
<td>17.6</td>
<td>17.6</td>
<td>0.3</td>
<td>0.3</td>
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<tr>
<td>Core</td>
<td>64</td>
<td>25.4</td>
<td>1</td>
<td>1</td>
<td>179</td>
<td>1</td>
<td>26</td>
<td>56</td>
<td>0.45</td>
<td>0.01</td>
<td>0.01</td>
</tr>
</tbody>
</table>

Figures 6-7 report results obtained using the standard method and the sound intensity only method. For both panels, the two methods provide similar transmission loss estimations that are in agreement with numerical simulations.

5. TL ESTIMATION USING VIBRATION MEASUREMENTS ONLY

5.1 Theory

In this section, the TL is derived using only a measurement of the vibration field of the panel. The radiated acoustic power is calculated using the radiation resistance matrix method as in Section 3.1, while the incident acoustic power is estimated thanks to the identification of the actual pressure field acting on the panel using the Virtual Field Method (VFM) [17].

The radiated acoustic power $W_{rad}$ is again obtained by

$$W_{rad} = \tilde{v}^H R_{rad} \tilde{v},$$ (12)
Figure 6: Upper figure: Mean sound intensity levels $L_{\text{without}}$ and $L_{\text{with}}$ measured for the aluminum panel – Lower figure: Comparison of TL obtained following standard or using sound intensity only (a zoom in the $50 - 500$ Hz frequency range is shown, with a FEM simulation result similar to section 3.2 included).

Figure 7: TL results in third octave bandes in the $100 - 2000$ Hz frequency range for the standard calculation and the sound intensity only calculation (a simulation result taken from [16] in also included).
where $\tilde{v}$ is the $[N \times 1]$ vector of measured complex vibration velocities at $N$ points on a regular grid on the panel, $H$ is the Hermitian transpose, and $R_{rad}$ is the element radiation matrix previously defined in Section 2.

As in Section 3.1, the incident acoustic power is calculated using the spatially-averaged squared sound pressure level on the plate surface $< p^2 >$

$$W_{inc} = \frac{< p^2 > S}{8\rho_0 c_0},$$

(13)

where $S$ is the panel area and the factor $8$ accounts for the doubling of pressure on the panel surface (blocked pressure hypothesis). The main difference is that $< p^2 >$ is now obtained from the vibration response of the panel using the VFM.

The VFM was first applied to the identification of deterministic loadings on bending plates, and was recently extended to the identification of spatially correlated excitations [17] (further details concerning the formulation of the method and calculations can). It was shown that extraction of the loading power spectral density requires measuring power spectral density functions of transverse displacements and bending curvatures (which can be typically derived from contactless Laser Doppler Vibrometry measurements), that are then used as inputs to the general VFM framework. At frequencies distinct from natural frequencies of the plate, the method correctly identifies the autospectral density distribution of wall-pressure excitations generated by a DAF on a plate (which was independently measured by a microphone array). The biased force reconstruction at resonances was identified to result from the poorly conditioned problem of estimating applied force distributions from the response of one dominant structural mode. A correction was proposed in [17] to provide the spatially-averaged squared pressure on resonance under the assumption of well-separated, lightly damped modes

$$\langle p^2 \rangle \approx (\eta \omega_R h \rho)^2 \langle v_R^2 \rangle$$

(14)

where $\langle v_R^2 \rangle = \frac{1}{S} \int_S \tilde{v}_R \tilde{v}_R^* d\mathbf{x}$ is the spatially-averaged squared velocity of the panel on resonance, $\eta$ is the structural loss factor and $\omega_R$ is the natural angular frequency.

5.2 Experimental results

A simply-supported aluminium panel ($0.48 \times 0.42 \times 0.0032$ m$^3$, Young’s modulus = 70 GPa, mass density $\rho = 2740$ kg/m$^3$) was installed in the existing niche in the reverberant - anechoic transmission facility at Groupe d’Acoustique de l’Université de Sherbrooke. It was baffled and flush mounted on the reverberant room side. A double wall structure was then built around the panel to prevent direct and indirect (flanking) acoustic leaks. Regarding vibration measurements, a scanning laser vibrometer was used to measure the transverse velocity of the panel on a regular grid of $35 \times 29$ points in the 50 – 5000 Hz frequency range on the anechoic room side. The measured vibration velocity was converted to displacement by division by $j\omega$, interpolated on a finer $49 \times 43$ mesh, and spatially smoothed to remove small-wavelength measurement
noise, using the Matlab Gridfit function. The bending curvatures were then computed from the smoothed version of the displacement field using simple 3-point second-order finite differences of the displacements (a smoothing process is crucial to get reliable curvature fields to input into the loading identification procedure).

For brevity sake, only results concerning the incident power are provided in Fig. 8 (TL estimation will be shown at the conference presentation). From this figure, it is clear that the spatially averaged square pressure (needed for incident power calculation using Eq.(13)) is well estimated between structural resonances. As depicted with red circles, Eq.(14) provides satisfactory corrections at resonances, that can be nevertheless underestimated. It is interesting noting that with increasing frequency, the method provides better estimation of \( <p^2> \).

6. CONCLUSION

In this paper, three alternative methods for estimating the transmission loss of panels were described. The first allows estimating the TL without standard coupled rooms arrangement, and the two others suggest 'single quantity' methods for estimating the incident or radiated powers on or from a partition.

For the three proposals, measurements results were provided that proved their ability to obtain satisfactory estimations of either the transmission loss, or the incident or transmitted sound powers.

REFERENCES


