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Estimating the sound transmission loss of a single partition using vibration measurements

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A B S T R A C T

The sound transmission loss of a homogeneous, isotropic, thin panel under a diffuse acoustic field excitation is derived from a measurement of its airborne induced vibration field. Using this single dataset, the virtual fields method allows identifying the wall pressure field exciting the panel and estimating the corresponding incident acoustic power, provided that the differential operator governing the plate dynamics and its material properties are known *a priori*. Using the same dataset, the radiated acoustic power is calculated using the radiation resistance matrix method. For an aluminium plate, a comparison of transmission loss values obtained using this approach and a standardized measurement shows good agreement off resonance but large discrepancies on resonance and close to resonance, due to ill conditioning of the virtual fields method. A simple correction is proposed on resonance.

1. Introduction

The sound reduction index (R) or sound transmission loss (TL) of partitions is used to qualify a considerable range of structures, from fuselage panels to building elements (the acronym TL will be used in what follows). This ratio of incident to transmitted sound power of a partition is classically determined following two standardized methods [1,2]. Since incident and transmitted sound powers of a partition can hardly be directly measured, they are indirectly estimated either using the relation between the spatially-averaged mean square pressure and the sound intensity in a diffuse sound field on both reverberant emitting and receiving rooms in addition to reverberation time measurements in the receiving room [1], or using the same estimation on the emission side and sound intensity measurements on the reception side [2]. Controlled and large acoustic spaces are thus generally required. Among other parameters like panel mounting conditions, aperture size and position [3,4], the accuracy of sound pressure or sound intensity measurements influences repeatability and reproducibility of TL measurements [4,5].

As possible alternatives to these standardized approaches, the estimation of incident sound power using near field acoustic holography was examined in [6]. Possible improvement in the accuracy of transmitted sound power measurements at low frequencies was proposed in [7] using two microphones, a calibrated volume velocity source and an approach called peak envelope.

Estimation of the transmitted sound power using vibrations measurement was first proposed and investigated in the 1960s by Ward [8], Utley and Mulholland [9] and Burd [10] using accelerometers. This approach was mainly developed for tackling the problem of measuring transmission loss in presence of important flanking paths. It provided satisfactory results but has not been followed by additional works. Laser Doppler vibrometry (LDV) measurements and a Rayleigh integral approach [11] were later proposed to estimate transmitted sound power, and showed to be robust to the acoustic properties of the receiving space and thus more reliable at low frequencies compared with standardized microphone measurements [1]. Finally, in order to solve the limitations of the diffuse field assumption in source and receiving rooms in the low frequency domain, the combination of a Rayleigh integral approach with a synthetized diffuse excitation field by a loudspeaker array was considered [12]. Room acoustic effects were shown to be minimized below a frequency of 500 Hz but at the expense of a large measurement effort.

The main contribution of this note is to propose an approach for the calculation of the TL of a single and thin panel by estimating *both* incident and transmitted acoustic powers using a measurement of the vibration field of the panel. The radiated acoustic power is calculated using the radiation resistance matrix method [13,14], while the incident acoustic power is estimated thanks to the identification of the actual pressure field acting on the panel using the virtual fields method [15,16] (VFM). Using LDV measurements, this method has been applied to the identification of the autospectral and cross-spectral density functions of a diffuse acoustic field (DAF) acting on the surface of a thin plate [15]. A similar approach, the Force Analysis Technique, was tested using vibration measurements for identifying a DAF loading on a plate [17]. Both methods require that the differential operator governing the plate dynamics and its material properties are known *a priori*.

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Those could be estimated in a preliminary phase using controlled excitations [18,19].

2. Standard method and proposed approach for TL measurements

The transmission loss or sound reduction index of a partition is defined by

$$TL = 10\log_{10}\left(\frac{W_i}{W_t}\right), \tag{1}$$

where W_i and W_t are the incident and transmitted sound powers of the partition in the source and reception rooms, respectively. If the sound field in the reverberant source room is supposed to be perfectly diffuse, then the incident sound intensity I is related to the spatially-averaged mean square pressure $\langle p^2\rangle$ in the room

$$I = \frac{\langle p^2\rangle}{4\rho_0 c_0}, \tag{2}$$

where ρ_0 and c_0 are the density and sound speed in the air, respectively and $\langle p^2\rangle$ is usually estimated by using a rotating microphone or installing several microphones in the room.

Using the measurement procedure described in [2] in a transmission suite, the TL of a partition is determined using measurements of the spatially-averaged squared sound pressure level in the source room L_p and of the average sound intensity level L_i over the partition surface on the receiving side

$$TL = L_p - L_i - 6, \tag{3}$$

where the −6 factor arises from Eq. (2) and dB calculations using reference values of 2×10^{-5} Pa for sound pressure and 1×10^{-12} W/m^2 for sound intensity. This approach is schematically depicted in Fig. 1-a.

In the proposed approach, the TL is derived from the vibration response of the panel using LDV measurements (Fig. 1-b). The transmitted acoustic power W_t is obtained by [13]

$$W_t = \tilde{\mathbf{v}}^{\mathbf{H}}\mathbf{R}_{rad}\tilde{\mathbf{v}}, \tag{4}$$

where $\tilde{\mathbf{v}}$ is the $[N\times1]$ vector of measured complex vibration velocities at N points on a regular grid on the panel, $^{\mathbf{H}}$ is the Hermitian transpose, and $\mathbf{R}_{rad}$ is the element radiation matrix [13] of dimension $[N\times N]$ for a baffled panel, which elements are defined as

$$R_{rad_{i,j}} = \frac{\omega^2\rho_0 A^2}{4\pi c_0}\frac{\sin(k_0 r_{i,j})}{k_0 r_{i,j}}, \tag{5}$$

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. (4, 5) hold for any boundary conditions of the panel, do not depend on the velocity distribution on the panel and finally assume that it is baffled and that the receiving space is anechoic. The same approach is followed when using the Rayleigh integral, like in [11], the radiation resistance matrix being an application of the Rayleigh integral. The term $\frac{\sin(k_0 r_{i,j})}{k_0 r_{i,j}}$ also known as sinc function has a singularity when $r_{i,j}=0$, but a limit calculation provides unity value so that the diagonal terms $R_{rad_{i,i}}$ equal one [14]. Also, this matrix is symmetric. The incident acoustic power W_i is calculated using the spatially-averaged squared sound pressure on the panel surface $\langle p^2\rangle$

$$W_i = \frac{\langle p^2\rangle S}{8\rho_0 c_0}, \tag{6}$$

where S is the panel area and the factor 8 accounts for the doubling of pressure on the panel surface (blocked pressure hypothesis). In the following section, $\langle p^2\rangle$ is obtained from the vibration response of the panel using the VFM.

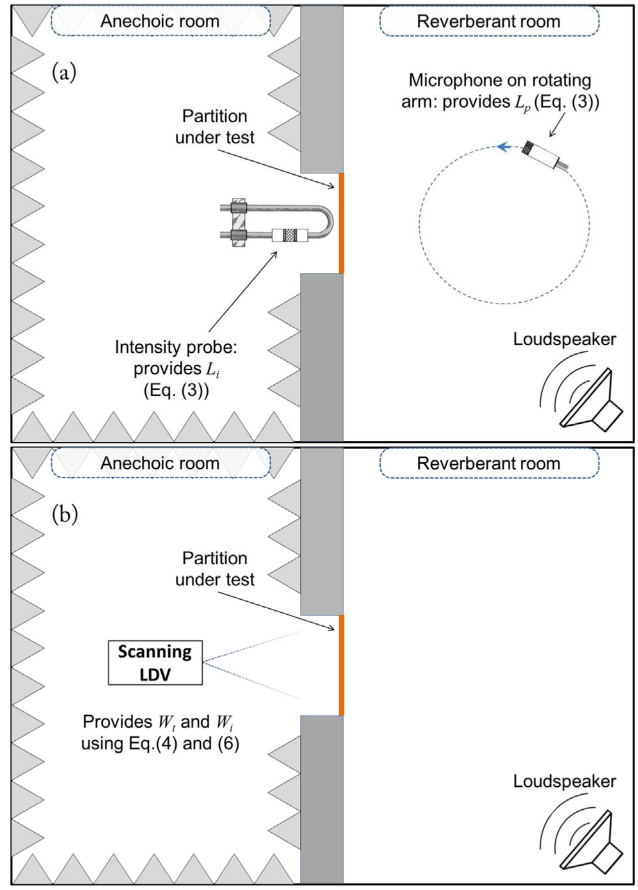

Fig. 1. (a) Standard TL measurement in a coupled reverberant-anechoic room facility [2] - (b) Proposed TL measurement.

3. Wall pressure estimation using the VFM

This section briefly describes the VFM formulation for identifying deterministic and random loadings on a bending plate. For further details, the reader is referred to [15]. For a flat bending plate of surface S, thickness h, mass density ρ, submitted to a harmonic transverse loading $\tilde{p}(\mathbf{x},\omega)$, the virtual work principle gives [16]

$$-\omega^2 h\int_S \rho w^v(\mathbf{x})\tilde{w}(\mathbf{x},\omega)d\mathbf{x} + \frac{h^3}{12}\int_S \mathbf{k}^{vT}(\mathbf{x})\mathbf{Q}(\mathbf{x},\omega)\tilde{\mathbf{k}}(\mathbf{x},\omega)d\mathbf{x} = \int_S w^v(\mathbf{x})\tilde{p}(\mathbf{x},\omega)d\mathbf{x}, \tag{7}$$

where $\mathbf{x}$ designates an arbitrary point of S, $\mathbf{Q}$ is the 3×3 plane stress stiffness matrix, $\tilde{w}(\mathbf{x},\omega)$ and $\tilde{\mathbf{k}}(\mathbf{x},\omega)$ are the transverse displacement field and the related curvature field, respectively, and $w^v(\mathbf{x})$ and $\mathbf{k}^v(\mathbf{x})$ are the virtual transverse displacement field and the related virtual curvature field. In this work, the elastic matrix is derived from the assumed, standard values of Young's modulus and Poisson's ratio of aluminium and by considering the structure to be a homogeneous flat structure of constant thickness. For pure bending, the curvature field is given by $\tilde{\mathbf{k}}(\mathbf{x},\omega) = -\left\{\frac{\partial^2}{\partial x^2}\frac{\partial^2}{\partial y^2}2\frac{\partial^2}{\partial x\partial y}\right\}^T\tilde{w}(\mathbf{x},\omega)$, with ^{T} the vector transposition and x, y cartesian coordinates on S; the same relation holds for virtual displacements and curvatures. In this work, *piecewise* virtual displacements defined over a limited region of S (a virtual window, denoted $S_{\mathbf{x}}$) are used in Eq. (7) to extract $\tilde{p}(\mathbf{x},\omega)$ over the corresponding region. The chosen virtual displacement (that can be any C^1 function over S) is based on the Hermite16 interpolation functions [16], for which both the virtual displacement and virtual slope are zero on the edges of $S_{\mathbf{x}}$ (continuity of the virtual displacement and slope at the boundary of the virtual window is thus ensured). Eq. (7) can then be

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