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Virtual testing of sound insulation panels

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ABSTRACT

This paper presents capabilities of virtually performing acoustical testing procedures specified per ISO 140-3:1995 and ISO 140-4:1998 standards. This test setup consists of a sound source room, an insulation panel, and a receiving acoustic room which are described in terms of parameters of practical interests. A number of sound sources can be placed at any desired locations in the source room. The insulation panel is an elastic plate loaded with features of design interest such as layers of acoustical materials, structural reinforcements, holes, etc. It can also be used to predict the effects of installation conditions on sound insulation performance of a sound insulation panel. The sound transmitted into the receiving room can be calculated based on the sound pressures "measured" at a number of locations and/or the sound intensity on the surface of the panel. This virtual testing system can also be used to determine the sound transmission characteristics via an opening of different shapes. An advantage of this testing system is that it is capable of determining the acoustical characteristics of a sound insulation panel or panel-like structure which is not yet constructed. Several examples are presented to demonstrate the reliability and ease-of-use of this virtual testing system.

Keywords: Virtual testing, Sound insulation, Vibroacoustic systems
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1. INTRODUCTION

Determination of acoustic characteristics (such as sound reduction index) of panels or panel-like structures is of interest to many applications like automotive, ship cabins, office spaces, and aerospace, etc. The fundamental theories for sound transmission through panels are typically derived under the condition that the sound fields in both rooms are ideally diffuse [1]. In practice, panels are often tested by subjecting them to acoustic excitation in finite sized reverberant chambers. In such a test, an acoustic field is generated by several loudspeakers in the source room and the sound is transmitted into the receiving room via the panel under test. As the frequency of excitation increases, the modal density and modal overlap in the enclosures also increases. In a "random incident" sound field, the sound is incident on the separating partition from all angles with approximately equal probability [2]. The diffuse field assumption is only valid in medium and high frequency ranges, since at low frequencies the sound field in the reverberation chambers is dominated by a small number of lower order modes. It has been noticed that the sound test standards did not provide adequately accurate results at very low frequencies [3].

To better understand the sound transmission through the two rooms via the panel, analytical and numerical methods have been used. Modal analysis [4] is adopted to study the errors in sound transmission measurements. Maluski and Gibbs [5] have used an FEM model to investigate the sound transmission between dwellings at low frequencies. They found the sound-level difference between rooms is strongly influenced by the modal characteristics of the rooms, as well as of the partition wall, producing large variance in data below 100 Hz. Jo and Elliott [6] have been developed a theoretical description of sound transmission between rooms , in terms of individual acoustic modes of the room and structural modes of the panel. Dijckmans [7] developed a model that combined the transfer matrix method (TMM) and the wave based method (WBM) to predict the sound transmission loss through finite-sized multilayered structures. Papadopoulos [8] proposed a virtual laboratory to represent a real laboratory consisting of two reverberation rooms to study the sound transmission loss of the panel based on the procedures as specified in relevant acoustic measurements and standards.

In this paper, a parametric coupled vibroacoustic model is presented to predict the acoustic behaviors (such as sound reduction indexes) of a panel-like structures. The model consists of two acoustic rooms and a flexible panel. The panel can be mounted elastically onto the windowed wall, which thus allows considering, among others, the effects on sound transmission of mounting conditions. With this modeling means, one can virtually measure the sound transmission characteristics of sound insulation panels, and study the effects on their acoustic performance of structural design modifications, acoustical treatments, and field installation conditions. Numerical examples are presented to demonstrate the reliability of this virtual testing approach and its usefulness in guiding design modifications regarding, for instance, panel mounting conditions, application of decoupling layers to improve sound insulation performance, and so on.

2. THEORETICAL FORMULATIONS

2.1. Description of the coupled cavity-panel-cavity system

The virtual testing environment depicted in Figure 1, consistings of a source room of dimensions $L_{x1} \times L_{y1} \times L_{z1}$, a sound transmission panel of dimensions $L_x \times L_y$, and a receiving room of dimensions $L_{x2} \times L_{y2} \times L_{z2}$. To simplify the descriptions, different local coordinate systems are used. The two acoustic rooms are not necessarily of the same sizes. The panel is placed between two rooms and lies in the $z_1 = L_{z1}$ and $z_2 = 0$ in source room and receiving room. Several point sources can be placed at any desired locations in the source room. The walls of the source and receiving rooms are assumed rigid.

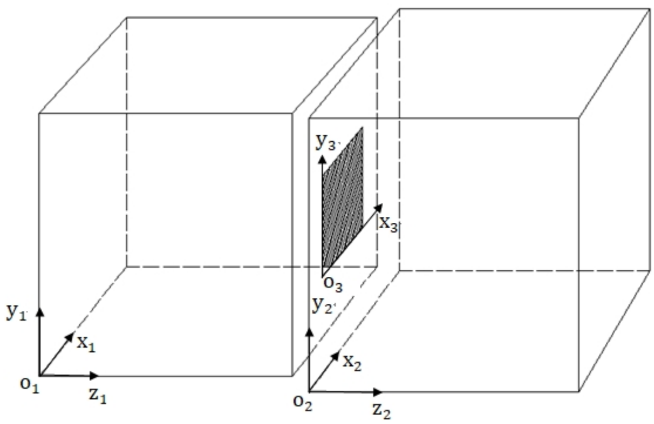

Figure 1: Source room-panel-receiving room system

2.2. General series representations of the displacement and sound pressure solutions

The transverse displacement of the partition panels with general elastic boundary supports will be invariably expanded into a modified Fourier series as [9]

$$w(x,y) = \sum_{m=0}^{M}\sum_{n=0}^{N} P_{mn} cos\lambda_m x cos\lambda_n y + \sum_{j=1}^{4}(\xi_y^j(y)\sum_{m=0}^{M} c_m^j cos\lambda_m x + \xi_x^j(x)\sum_{n=0}^{N} d_n^j cos\lambda_n y), \quad (1)$$

where$\lambda_m = m\pi/L_x$, $\lambda_n = n\pi/L_y$,and P_{mn}, c_m^j and d_n^j represent the Fourier coefficients for the panel displacement. The supplementary functions $\xi_x^j(x)$ and $\xi_y^j(y)$ are introduced to account for all the potential discontinuities encountered when the displacement function and its lower-order derivatives are periodically extended onto the entire x-y plane as mathematically implied by the Fourier expansion.

The acoustic pressure in the source room P_1 and the receiving room P_2 can also be similarly expressed as modified Fourier series expansions as [10]

$$\begin{aligned} P_1(x,y,z) = & \sum_{m_x=0}^{M_{x_1}}\sum_{m_y=0}^{M_{y_1}}\sum_{m_z=0}^{M_{z_1}} A^1_{m_x m_y m_z} cos\lambda_{m_x} x cos\lambda_{m_y} y cos\lambda_{m_z} z + \\ & \zeta_1(z)\sum_{m_x=0}^{M_{x_1}}\sum_{m_y=0}^{M_{y_1}} B^1_{m_x m_y} cos\lambda_{m_x} x cos\lambda_{m_y} y, \end{aligned} \quad (2)$$

$$\begin{aligned} P_2(x,y,z) = & \sum_{m_x=0}^{M_{x_2}}\sum_{m_y=0}^{M_{y_2}}\sum_{m_z=0}^{M_{z_2}} A^2_{m_x m_y m_z} cos\lambda_{m_x} x cos\lambda_{m_y} y cos\lambda_{m_z} z + \\ & \zeta_2(z)\sum_{m_x=0}^{M_{x_2}}\sum_{m_y=0}^{M_{y_2}} B^2_{m_x m_y} cos\lambda_{m_x} x cos\lambda_{m_y} y, \end{aligned} \quad (3)$$

where $A^i_{m_x m_y m_z}$ and $B^i_{m_x m_y}$ denote the complex Fourier expansion coefficients, $\lambda_{m_x} = m\pi/L_{x_i}$, $\zeta_1(z) = L_{z_1}(z/L_{z_1})^2(z/L_{z_1} - 1)$ and $\zeta_2(z) = L_{z_2}(z/L_{z_2})(z/L_{z_2} - 1)^2$ in which i=1, 2.

2.3. A panel covered by a decoupling layer

A typical configuration of the vibroacoustic coupling model of an elastically restrained plate that covered by a decoupling layer is shown in Figure 2. The decoupling layer is described as the locally reacting model [11], The thickness deformation of the decoupling layer implies that the base panel and the outer surface of the decoupling layer have distinct transverse displacements. Thus w is defined to be the transverse displacement of the base panel and $\bar{w}$ the transverse displacement of the outer surface of the decoupling layer. The details of w are the same as those defined in the previous section 2.2.

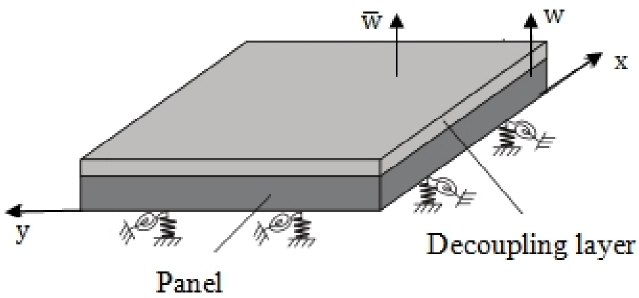

Figure 2: Panel covered by a decoupling layer

Assuming that the decoupling material behaves as evenly distributed massless springs on the panel, the surface acoustic pressure exerted by the fluid on the coating is the same as the normal stress exerted by the coating on the structure. In this paper, the pressure of the outer surface of the decoupling layer is the sound pressure $P_2(x, y, 0)$ in the receiving room.

The equation of continuity of the structural and acoustic normal accelerations on the outer surface of the decoupling material is

$$\frac{\partial P_2(x, y, 0)}{\partial z} = \rho_0 \omega^2 \bar{w}(x, y), \tag{4}$$

where ρ_0 is the density of the fluid, ω is the circular frequency.

From Eq. (4) one can have $\bar{w}$, then the potential energy of the decoupling layer can be derived as

$$W_{layer} = \int_0^{L_x} \int_0^{L_y} \frac{1}{2} Z_c (w - \bar{w})^2 dxdy, \tag{5}$$

in which Z_c is the impedance of the decoupling layer. Based on the locally reacting model, Z_c can be expressed as

$$Z_c = \frac{B_c(1 + j\eta_c)}{h_c}, \tag{6}$$

where B_c, η_c and h_c are, respectively, the bulk modulus, the damping factor, and the thickness of the decoupling layer on the panel and $j = \sqrt{-1}$.

2.4. Truncation criteria

The expansion Fourier series have to be truncated for numerical calculations. From a computational point of view, it is more appropriate to adopt a frequency dependent truncation rule. The truncation rule enables the current model to be used more effectively for high frequency calculations. The truncation criterium is based on the physical observation that acoustic and structural waves couple well when their wave numbers match better. Accordingly, the truncation criterium is described as

$$T_{lower} \leq \frac{\lambda_n}{\lambda_a} \leq T_{upper}, \tag{7}$$

where λ_n is the wavelength component in the coupled system and λ_a is the natural wavelength. T_{lower} is the lower limit and $T_{lower} \in [0, 1]$. T_{upper} is the upper limit and $T_{upper} \in [1, 2]$.

These criteria will lead to an appropriate selection of only a number of waves to be used in the series expansions of the solutions.

2.5. Solution procedure of the coupled cavity-panel-cavity system

The Rayleigh-Ritz procedure will be employed to calculate the unknown expansion coefficients in Eqs. (1), (2) and (3). The Lagrangian for the panel structure can be written as

$$L_{panel} = U_{panel} - T_{panel} + W_{r\&p} - W_{layer}, \tag{8}$$

where U_{panel} is the total potential energy associated with the transverse deformation of the panel and the total potential energy stored in the restraining springs; T_{panel} denotes the total kinetic energy of the panel; $W_{r\&p}$ is the work done by the sound pressure in source room.

The Lagrangian for the source room and receiving room are

$$L_{source} = U_{source} - T_{source} - W_{p\&r} - W_s, \tag{9}$$

$$L_{receiving} = U_{receiving} - T_{receiving} + W_{layer}, \tag{10}$$

where U_{source} and T_{source} are the potential energy and the kinetic energy for the source room, $U_{receiving}$ and $T_{receiving}$ are the potential energy and the kinetic energy for the receiving room. $W_{p\&r}$ is the work due to the panel vibration. The continuity conditions on the solid-fluid interface implies a reciprocity relationship, that is, $W_{r\&p} = W_{p\&r}$. W_s is the work done by the acoustic source.

In this paper, a monopole source is placed in the sorce room. The work done by the sound source inside cavity can be represented as

$$W_s = -\frac{1}{2}\int_v \frac{P_1 Q_0}{j\omega} dV, \tag{11}$$

where Q_0 is the volume velocity amplitude of the monopole source.

Substituting Eqs. (1) (2) (3) into Eqs. (8) (9) (10) and applying the Rayleigh-Ritz procedure against each of unknown Fourier series coefficients will lead to a set of couple system

$$\left\{\begin{bmatrix} \mathbf{K}_{r1} & \mathbf{0} & \mathbf{0} \\ \mathbf{C}_{r\&p} & \mathbf{K}_{panel} & \mathbf{0} \\ \mathbf{0} & \mathbf{W}_{L1} & \mathbf{K}_{r2} \end{bmatrix} - \frac{1}{\omega^2}\begin{bmatrix} \mathbf{0} & \mathbf{0} & \mathbf{0} \\ \mathbf{0} & \mathbf{0} & \mathbf{0} \\ \mathbf{0} & \mathbf{0} & \mathbf{W}_{L2} \end{bmatrix} - \omega^2 \begin{bmatrix} \mathbf{M}_{r1} & -\mathbf{C}_{r\&p}^T & \mathbf{0} \\ \mathbf{0} & \mathbf{M}_{panel} & \mathbf{0} \\ \mathbf{0} & \mathbf{0} & \mathbf{M}_{r2} \end{bmatrix}\right\} \times \left\{\begin{matrix} \mathbf{P}_1 \\ \mathbf{W} \\ \mathbf{P}_2 \end{matrix}\right\} = \left\{\begin{matrix} \mathbf{F}_s \\ \mathbf{0} \\ \mathbf{0} \end{matrix}\right\}, \tag{12}$$

where $\mathbf{P}_1$, $\mathbf{P}_2$ and $\mathbf{W}$ are, respectively, the vector of the Fourier expansion coefficients for the sound pressure in source room, in receiving room and the panel displacement. $\mathbf{F}_s$ is derived from Eq. (11), which is expressed as $\mathbf{F}_s = -j\omega\rho_0 Q_0 cos\lambda_{m_{x1}} x_s cos\lambda_{m_{y1}} y_s cos\lambda_{m_{z1}} z_s$, where (x_s, y_s, z_s) is the location of the monopole source.

3. RESULTS AND DISCUSSIONS

3.1. Validations of the coupled cavity-panel-cavity model

In this section, the prediction model will first be validated against some existing results in the literature. In the literature, the sound reduction index (SRI) in a coupled cavity-panel-cavity system was investigated by Dijckmans [12], using the Wave Based Method (WBM). Assume that the dimensions of the source and receiving rooms are respectively: $L_{x_1} \times L_{y_1} \times L_{z_1} = 4 \times 3 \times 5m$ and $L_{x_2} \times L_{y_2} \times L_{z_2} = 4 \times 3 \times 6m$, These two rooms are connected via an elastic panel of dimensions $1.5 \times 1.5m$. The panel is positioned centrally between the two rooms with the (x, y) coordinates of the corner of the panel closest to the origin of (1.75 m, 1.25 m) for the source room, and of (1.75 m, 1.25 m) for the receiving room. The speed of sound is $c_0 = 340m/s$, air density $\rho_0 = 1.2kg/m^3$ and loss factor 0.03. The panel has a thickness $0.025m$, a Young's modulus $3530MPa$, a density $1200kg/m^3$ and a Poisson's ratio of 0.5. A volume point source is placed in the corner of the source room at $0.5m$ distance of the walls. The panel is assumed to be simply supported (SSSS) which is considered a special case of elastic restraints by setting the transverse and rotational springs to infinity and zero, respectively. The lower limit of the truncation is $T_{lower} = 0.9$ and the upper limit of that is $T_{upper} = 1.1$. Figure 3 shows the sound reduction index of the panel and the current results agree well with those in [12].

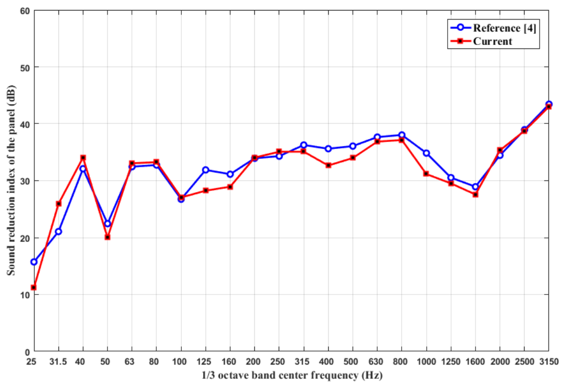

Figure 3: SRI of present method versus reference [12]

3.2. SRI of the panel with different boundary conditions

The parameters of the cavities are kept the same, but the panel is now modified to have: thickness $0.005m$, Young's modulus $70.3GPa$, mass density $2700kg/m^3$ and Poisson's ratio 0.3. The panel is asssumed to be simply supported. Figure 4 shows the sound reduction indices of the panel with / without a decoupling layer ($B_c = 10^4 Pa$, $h_c = 0.01m$ and $\eta_c = 0$). This results clearly show that the addition of a decoupling layer help improve the sound insulation performance of the panel.

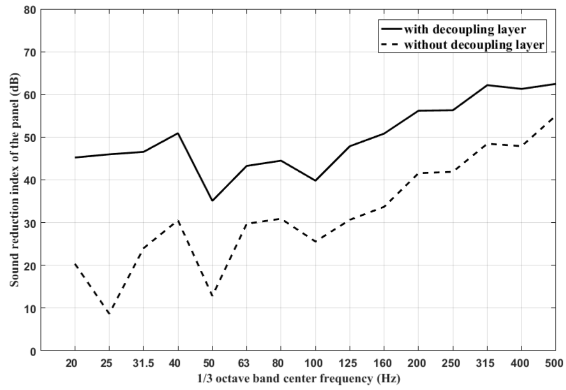

Figure 4: SRI of the panel with and without a decoupling layer

To understand the effects of the mounting conditions on the sound reduction index of the panel, different mounting conditions are applied along its edges. The familiar mounting conditions, simply supported (S), clamped (C), and guided (G), are easily considered as the special cases of the general elastic restraints. From Figure 5, it is seen that sound reduction index from the guided panel is noticeably higher than those for the simply supported and clamped boundary conditions. The sound reduction index is often used as a measure of sound insulation performance of a panel. The results in Figure 5 clearly shows that the sound insulation of a panel can be meaningfully affected by its mounting condition. Thus, the mounting conditions should be considered a meaningful acoustical design factor in practical applications.

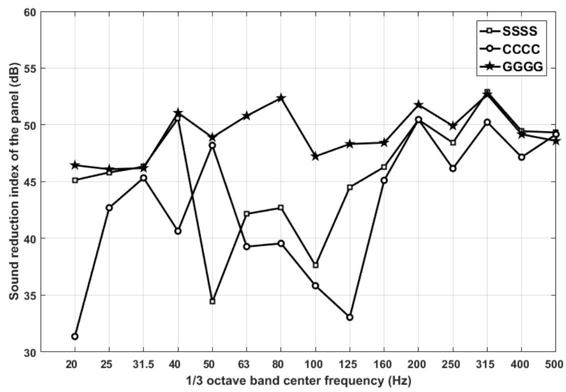

Figure 5: SRI under diferent mounting conditions (SSSS:simply supported; GGGG:guided; and CCCC:clamped)

3.3. SRI of the panel with different bulk modulus of the decoupling layer

From equation (6) one can expect that increasing the bulk modulus of the decoupling layer is equivalent to reducing its thickness. By keeping the thickness of the decoupling layer 0.01m. The bulk modulus for the decoupling layer is chosen to have three different values, and the results are plotted in figure 6. It is seen that as the bulk modulus of the decoupling layer increases, the sound insulation performance of the covered panel is significantly deteriorated. This is understandable: when B_c is increased, the decoupling layer is "hardened" and its ability of decoupling the acoustic particle and structural velocities is accordingly reduced.

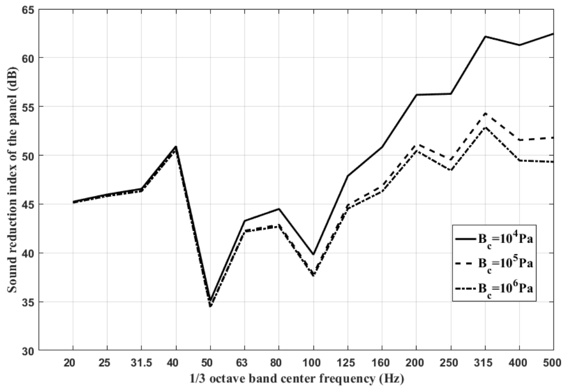

Figure 6: SRI of the panel with different bulk moduli of the decoupling layer

4. CONCLUSIONS

A parametric vibroacoustic model is developed to simulate (per ISO 140-3:1995 and ISO 140-4:1998 standards) the testing capability of measuring the acoustic characteristics (such as the sound reduction index) of a panel-like structure. The various acoustical capabilities of this virtual testing system have been extensively validated including determining Sound reduction index of a panel-like structure. While this system allows virtually and effectively testing various acoustical aspects of a sound insulation panel or structure, it is equally important to use it as a means to guide acoustic designs or find better design options for improving sound performance. For instance, it can easily help understand the effects of mounting conditions in field applications which suggests that properly selecting mounting conditions represents a viable design option for improving the transmission characteristics of sound insulation panels. Also, it is demonstrated that SRI for a panel-like can also be meaningfully affected by the bulk modulus of an applied decoupling layer. Other important design parameters, such as the properties of the sound insulation panels, impedance characteristics of acoustical materials, geometrical properties and locations of openings in a panel, structural modifications/reinforcements and son on, can also be readily tested in terms of their effects on the acoustical characteristics of the resulting sound insulation designs.

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