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One dimensional study of a module for active/passive control of both absorption and transmission

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ABSTRACT

Hybrid active/passive absorbers have proven to be efficient over a large frequency range. The next step consists in building up a system which can exhibit good absorption and insulation properties. To simulate such hybrid cell, active and passive behaviors of an electroacoustic loudspeaker have been modelized by using a one-dimensional approach. The rear acoustic load at the back of the membrane has been taken into account to obtain a reliable model. The proposed model has been validated with measurements performed in a 7 cm diameter tube. Then, a hybrid cell composed of a porous plate and a small thickness loudspeaker has been designed and numerically tested. It is shown that, when driving the loudspeaker for total absorption, the transmission losses are suppressed at lower frequencies. To overcome this problem, a dual actuator cell is designed to deal with both absorption and transmission. Simulations shows that this solution can lead to good results. It is also shown that interaction of the loudspeakers can be significantly reduced by using directive sources, thus lowering supplying voltages and condition number of the matrix inversion required by the control process.

1. Introduction

Many situations involve multiple volumes including sound sources; this is the case of buildings, transportations, hoods, etc. The boundaries of each volume plays a double role: it increases the radiated pressure on the source side, and it transmits part of the acoustic energy on the other side. The design of acoustically efficient walls is therefore a trade-off between both aspects, of course limited by practical constraints such as the resulting cost, thickness and weight.

Usually, walls are optimized for low acoustic transmission, as this is the major challenge in many applications. Passive approaches have been improved over the years, starting from the basic "law of mass" concept to reach multiple-layer designs [1–4]. A very typical example is the partition walls for buildings, involving two sheets of gypsum separated by a fluid layer, with a wide range of possible configurations [5–7]. The basic objective of such designs is to reduce the vibration of the receiver side of the wall, ideally leading to a perfectly motionless boundary.

Other applications consider absorption as their major concern; this is the case of the lining of nacelles in aircraft engine, or the lining of some test equipments like anechoic rooms. Acoustic absorption usually results from viscous and thermal phenomena at the interface between a structure and the fluid; it is improved by a suitable design of this interface. It may be tuned for a narrow frequency band (like in resonators [8,9]) or for broadband (like in porous or fibrous materials [10,11]). In any case, a major requirement is that the direct field of the source is not reflected at the interface, which must therefore ensure a fluid motion compatible with the incident pressure. A hidden requirement is also that the absorber structure (frame) must not vibrate, usually requiring to be installed on a stiff wall – although absorbing materials may be used to some extent with a limp frame configuration [12,13].

There are also applications requiring good performances both for absorption and transmission. Examples are compact machine hoods (for which the inside pressure must not increase too much as this would degrade the overall performance), and room acoustics where wall reflections must be controlled (*e.g.* to preserve speech intelligibility or the quality of musical events) while neighboring rooms require low background noise. A typical application could be a multiplex cinema, or a live music facility within an urban area. For such situations, the design must achieve both a low reflection coefficient on the source side (thus allowing suitable fluid motion) and a negligible motion of the wall on the receiver side. At first glance, it may seem that adding an absorbent layer to an existing wall would contribute to reduce the incident acoustic pressure on it, so leading to a reduction in transmission. This can be true only at higher frequencies, for which the inertia of the absorbing material frame is significant. At medium and lower

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frequencies the viscous constraints are transmitted by the frame to the supporting wall. Moreover, acoustic absorption decreases with frequency, even for a very thick lining, and passive solutions thus reach their limits both for absorption and transmission.

Active control of sound has its maximum efficiency at low frequencies [14], so it has been used for direct active absorption and transmission control, especially for room acoustics for which it provides a mean to tune the acoustic behavior of the wall, and so to adjust it for different needs [15–18]. Many researchers have also proposed hybrid solutions where high frequencies are controlled by passive means while active components are used to reduce noise at low frequencies. Passive and active effects are then combined into a single hybrid cell. For instance, pressure release [19–21] or impedance matching methods [22–24] yielded to large absorption coefficients at low frequencies. Other authors [25–27] chose to embed the active component into the passive media to create smart foams. Active control has been widely used to reduce sound transmission, by controlling the vibration of a single panel [28,29], or through a smart foam [30]. A single actuator has been used for acting on a double panel [31,32]. Some authors also used two piezoelectric actuators bounded on the two sides of a double panel or two loudspeakers to improve the transmission loss of such walls [33–35].

Recently, Sitel et al. proposed to control simultaneously the absorption and the transmission of a hybrid cell involving a single actuator and several plates or layers of absorbing material [36]. The present paper extends this work by using two separate actuators for such a simultaneous control, and investigates the potential of the principle of this double-channel hybrid cell. This is based on a 1D model using the scattering matrix formalism. The system involves the single-DOF model of a dynamic loudspeaker which represents an active panel of finite stiffness (active structure). Its model is given by Section 2, and experimentally assessed in Section 3. Single and dual actuator cells are then simulated and compared in Section 4. Finally, in Section 5, results are discussed and evaluated with suggestions made for further research.

2. Component models

The purpose of this work is not to provide an accurate model, but to compare the dual-channel principle to previous ones. This section thus proposes a model as simple as possible, based on a circular cell which diameter is much smaller than the wavelength (LF approximation), fitted in an impedance tube which ensures normal incidence on the cell surface.

The dual-channel principle is investigated by considering a composite cell featuring two active structures and an absorbing plate, separated by air layers. The model of such a simple configuration may be obtained by the cascade of a few components, using the scattering matrix formalism which provides straightforwardly the reflection and transmission coefficients. Three components types are involved: air layer, porous layer, and finite stiffness active structures.

Following the above assumptions, a 1D behavior is assumed throughout this paper. Therefore each component of the module may each be described by a 2×2 scattering matrix $\mathbf{S_x^n}$ as follows:

$$\begin{Bmatrix} p_x^- \\ p_{x+l}^+ \end{Bmatrix} = \begin{bmatrix} S_{11}^n & S_{12}^n \\ S_{21}^n & S_{22}^n \end{bmatrix} \begin{Bmatrix} p_x^+ \\ p_{x+l}^- \end{Bmatrix}, \tag{1}$$

Note that +/− denotes propagation in the positive/negative x direction. The two indices x and $x + l$ here stand for two consecutive locations along the axis, while the n exponent expresses the nature of the considered component (air or porous layer, structure, etc.). Matrix entries S_{11}^n and S_{22}^n represent the anechoic reflection coefficients on each side of the component while S_{12}^n and S_{21}^n are the anechoic transmission coefficients.

As in a previous work [37], a simplified model is used to describe the behavior of a plate of finite stiffness, here considered as a single-degree-of-freedom structure as the frequency band of interest is usually close to the first flexural mode of the plate. Following the 1D approximation, this flexural mode is equivalent to a suspended piston of reduced equivalent area embedded into a rigid plate. The actuator technology is not the focus of the present work, so only its action on the equivalent normal displacement of the structure is considered here. The resulting component is then a two-port system as shown in Fig. 1, and its behavior can be expressed as follows:

$$\begin{pmatrix} p_a^- \\ p_b^+ \end{pmatrix} = \begin{bmatrix} S_{11}^s & S_{12}^s \\ S_{21}^s & S_{22}^s \end{bmatrix} \begin{pmatrix} p_a^+ \\ p_b^- \end{pmatrix} + \begin{pmatrix} p_s^- \\ p_s^+ \end{pmatrix}, \tag{2}$$

where p_a^+ and p_b^- are the incident pressure fields while p_a^- and p_b^+ are the pressure fields reflected by the structure. Whereas the scattering matrix $\mathbf{S^s}$ depicts its passive behavior, p_s^- and p_s^+ are the active pressures radiated under the influence of the embedded actuator.

2.1. Passive components

Within the simplified model considered in this paper, two components are passive ones: an air layer, and a layer of absorbing material. They are both described by symmetric scattering matrices thereafter.

With a $e^{j\omega t}$ dependence, the scattering matrix $\mathbf{S_x^a}$ of an air layer is given by

$$\begin{Bmatrix} p_x^- \\ p_{x+l_a}^+ \end{Bmatrix} = \begin{bmatrix} 0 & e^{-jkl_a} \\ e^{-jkl_a} & 0 \end{bmatrix} \begin{Bmatrix} p_x^+ \\ p_{x+l_a}^- \end{Bmatrix}. \tag{3}$$

where l_a is the thickness of the air layer. This matrix simply expresses the propagation delay of each of the travelling waves involved into the scattering matrix formulation.

The absorbing layer is here described as a plate made of porous media. To simplify the model, we choose to consider a porous media with rigid frame [10], leading to the following scattering matrix coefficients:

$$\begin{aligned} S_{11}^p = S_{22}^p &= \frac{j\sin k_m l_p \left(\frac{z_m}{\phi} - \frac{\phi}{z_m}\right)}{2\cos k_m l_p + j\sin k_m l_p \left(\frac{z_m}{\phi} + \frac{\phi}{z_m}\right)}, \\ S_{12}^p = S_{21}^p &= \frac{2}{2\cos k_m l_p + j\sin k_m l_p \left(\frac{z_m}{\phi} + \frac{\phi}{z_m}\right)}, \end{aligned} \tag{4}$$

where k_m and z_m are respectively the wave number and the reduced characteristic impedance of the porous media, while ϕ is its porosity and l_p its thickness. Parameters of the porous plate required by the Johnson–Allard Model [10] are: the flow resistivity σ, the porosity Φ, the tortuosity α_∞ and the viscous and thermal characteristic lengths Λ and Λ'. Their values are given in Table 1. Note that such a simple model can barely be accurate, as it assumes that the material

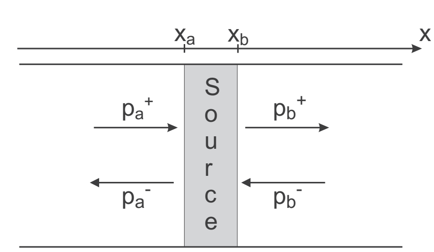

Fig. 1. Notation used for the active structure modelling.