# Improving Absorption of Sound Using Active Control

E. Friot, A. Gintz, P. Herzog and S. Schneider

**Abstract** Absorption of sound is a common problem especially at low frequencies. Absorbing materials available today perform well at medium and high frequencies but are much less performing at low frequencies at least when considering layers of realistic thickness. By contrast active control of sound is the most powerful at low frequencies where the sound field that is to be controlled is rather simple. Hence a combination of passive materials and active control seems to be a promising way to improve the efficiency of sound absorbing acoustic linings. The paper reflects upon two main directions. First, it studies the elimination by active control of sound of a sound field reflected by an absorbing layer. Such a procedure may be applied to improve the quality of acoustic testing facilities like anechoic chambers around or below its cut-off frequency. Secondly, the paper considers the design of hybrid absorbing materials consisting of a passive materials whose sound absorption is improved using either acoustic or mechanic actuators. Both studies are characterized by a strong link of numerical studies and experimental verification.

## 1 Introduction

Absorbing materials available today, perform well at medium and high frequencies but are much less performing at low frequencies at least when considering layers of realistic thickness. By contrast active control of sound is the most powerful at low frequencies where the sound field that is to be controlled is rather simple. Hence a

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combination of passive materials and active control seems to be a promising way to improve the efficiency of sound absorbing acoustic linings.

The first part of this paper studies the estimation of the remaining reflexions in anechoic chambers below its cut-off frequency.A real-time active control procedure to suppress a scattered sound field has been presented in [2]. This study forms the basis of the considerations dealt with here. The second part focuses on the development of a hybrid panel used for sound isolation purposes in buildings. The performance of existing passive panels is limited by the properties of the absorbing material and its thickness. Using acoustic actuators, the low frequency properties of the absorbing materials is improved to yield higher absorption of sound, see [3, 4].

## 2 Estimation of a Reflected Sound Field

At low frequencies acoustic linings of anechoic chambers do not sufficiently absorb the incident sound field. The remaining reflections perturb measurements and define the cut-off frequency of such a chamber. Using todays absorbing materials anechoic chambers with a cut-off frequency below 80 Hz are difficult to design because of the significant thickness of the lining required to avoid reflexions at low frequencies. Here we consider a possibility of constructing an anechoic chamber with a cut-off frequency below 80 Hz using an active control of sound procedure. The objective of the study is to estimate and finally to cancel out the sound field reflected by the lining at a position within the chamber where measurement are to be carried out.

The sound field reflected at the boundary $\Gamma_F$ of the fluid domain $\Omega_F$, see Figure 1, can be evaluated using the boundary integral method. The sound field within the domain $\Omega_F$ at a position $y$ can be expressed by the sound pressure $p$ and the normal velocity $v_\nu$ at the boundary $\Gamma_F$ using

$$p(y) + \int_{\Gamma_F} \frac{\partial \phi(x,y)}{\partial \nu} p(x)\, d\Gamma_F - a \int_{\Gamma_F} \phi(x,y) v_\nu \, d\Gamma_F = p^{\mathrm{inc}}(y) \quad y \in \Omega_F \tag{1}$$

with an incident sound field $p^{\mathrm{inc}}$ and the fundamental solution

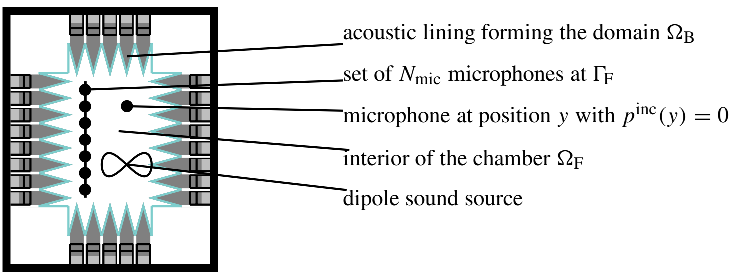


**Fig. 1** Diagramm of the set-up used to estimate the sound field reflected at the boundary $\Gamma_F$.

$$\phi(x,y) = \frac{e^{jk|x-y|}}{4\pi|x-y|} = \frac{e^{jkr}}{4\pi r}, \quad r = |x-y|, \quad x \neq y\,. \tag{2}$$

of the Helmholtz equation in three dimensions, the wave number $k = \omega/c$ and $a = \mathrm{j}\omega\rho^{\mathrm{f}}$. Therein $c$ denotes the speed of sound and $\rho^{\mathrm{f}}$ denotes the density of the fluid. If the point $y$ in Eq. (1) is chosen such that $p^{\mathrm{inc}}(y) = 0$ holds true then this equation can be used to estimate the scattered sound field $p^{\mathrm{scat}}$ using

$$p^{\mathrm{scat}}(y) = -\int_{\Gamma_{\mathrm{F}}} \frac{\partial\phi(x,y)}{\partial\nu} p(x)\, d\Gamma_{\mathrm{F}} + a\int_{\Gamma_{\mathrm{F}}} \phi(x,y) v_\nu\, d\Gamma_{\mathrm{F}} \quad y \in \Omega_{\mathrm{F}} \tag{3}$$

from the measured sound pressure $p$ and the fluid velocity $v_\nu$ at a certain boundary $\Gamma_{\mathrm{F}}$. Performing measurements for $N_{mic}$ positions at that boundary of the source with $p^{\mathrm{inc}}(y) = 0$ enables the estimation of the diffraction filter $H^{\mathrm{s}}$ such that the sound pressure scattered at the boundary $\Gamma_{\mathrm{F}}$ can be obtained from the measured sound pressures and velocities as

$$p^{\mathrm{scat}} = H^{\mathrm{s}} \begin{bmatrix} p \\ v_\nu \end{bmatrix}.$$

Disadvantage of such an approach is that the sound pressure and the fluid velocity must be measured. But the sound pressure and the surface velocity on $\Gamma_{\mathrm{F}}$ are not independent. Their relation is determined by the properties of the domain $\Omega_{\mathrm{B}}$. Supposing that $k$ is not an eigenvalue of the operator describing the sound propagation in $\Omega_{\mathrm{B}}$, a unique relation between the sound pressure and the fluid velocity can be found such that

$$Z v_\nu = p \quad \text{on}\,\Gamma_{\mathrm{F}} \tag{4}$$

holds true. The operator $Z$ in Eq. (4) is often referred to as the acoustic impedance boundary operator [5]. Supposing further that this operator is not singular we have

$$v_\nu = Z^{-1} p = Y p \quad \text{on}\,\Gamma_{\mathrm{F}} \tag{5}$$

with the acoustic admittance $Y$ of the boundary $\Gamma_{\mathrm{F}}$. Using Eq. (3) with Eq. (5) yields

$$p^{\mathrm{scat}}(y) = -\int_{\Gamma_{\mathrm{F}}} \left( \frac{\partial\phi(x,y)}{\partial\nu} - a\phi(x,y)Y \right) p(x)\, d\Gamma_{\mathrm{F}} = H^{\mathrm{s}} p \quad y \in \Omega_{\mathrm{F}}\,. \tag{6}$$

Eq. (6) shows that the scattered sound field can be obtained from measuring the sound pressure $p$ on $\Gamma_{\mathrm{F}}$ only. The estimation of these filters requires a dipole sound source that is placed such that the reference microphone does not receive any direct sound field from the source. The diffraction filters $H^{\mathrm{s}}$ can now be used to estimate the reflected sound field out of the measured sound field at the boundary $\Gamma_{\mathrm{F}}$ for any sound source.

The above proposed method has been used to estimate the reflections in a rectangular cavity measuring $2\times1.1\times1.2$ m$^3$ in the 20 to 400 Hz frequency range experimentally. The cavity was made of Siporex porous concrete and the walls were considered perfectly sound reflecting. The sound field close to the walls has been

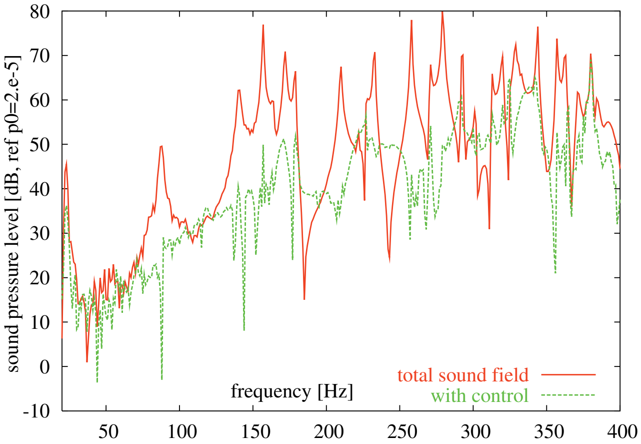


**Fig. 2** Sound pressure measured in the cavity and sound pressure obtained after the estimated reflected sound field has been removed from the data.

measured at 32 positions that were equally distributed over the walls. A dipole sound source was rotated in the cavity and the sound pressure at the walls and the scattered field $p^{\mathrm{scat}}$ were recorded for each rotation. Using Eq. (6) the diffraction filter $H^{\mathrm{s}}$ was estimated out of the experimental data. Figure 2 shows the sound pressure measured in the cavity and the sound pressure obtained when the estimated reflected sound field has been removed from the data. At least up to 300 Hz the resonance peaks have been reduced by 20 dB. These promising results motivate the application of the proposed method to estimate the wall reflexions in the 40 to 160 Hz frequency range observed in the large anechoic chamber at the Laboratoire de Mécanique et d'Acoustique (LMA) in Marseille. Once the reflected sound field can be estimated, this information can be used to either feed active control procedures to cancel out this sound field in the chamber or to post-process experimental data.

## 3 Hybrid Absorbing Panels

The objective of the second part of this study was the numerical study of the feasibility of a hybrid absorbing panel to be used in buildings. Unless using absorbing materials of unrealistic thickness, passive acoustic wall treatments are not efficient in the low frequency range. In what follows we consider therefore to increase the sound absorption of these panels by the use of active control of sound. The basic concept of these hybrid panels consists of using an acoustic actuator (loudspeaker) behind the absorbing material to influence the acoustic properties of the absorbing material. A sample panel with dimensions $0.6\times0.6\times0.25$ m$^3$ containing four loudspeakers and an absorbing material of 7.5 cm thickness was build at the LMA. So far this panel was used to verify the validity of the numerical model later on used to

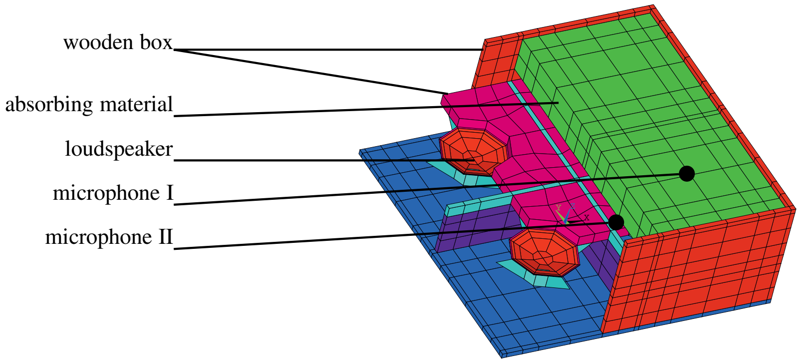


**Fig. 3** Meshed geometry of the hybrid cell.

test several control strategies. Figure 3 shows the finite element model for this panel with the assembly of the different components. The absorbing material, a glass wool ISOVER PB 38, was modeled using the equivalent fluid model. The material parameters, $\sigma = 2.0e4$ Ns/m$^4$ and $\phi = 0.97$, have been identified using an impedance tube experiment. The passive absorption coefficient $\alpha$ of a layer of 7.5 cm thickness is below $\alpha = 0.3$ up to a frequency of 200 Hz. The electro-mechanical behavior of the loudspeakers has been modeled using the model proposed by Thiele and Small [6].

To influence the acoustic properties of the panel two strategies were considered in the present study:

1. A given sound pressure behind the absorbing layer
   The impedance of the layer at the surface facing the cavity is modified by prescribing an appropriate pressure behind the layer. From the definition of the flow resistivity $\sigma$ of a porous material with a thickness $e$, see for example [1], $\sigma = (p_1 - p_2)/v_v/e$, with the sound pressure $p_1$ and $p_2$ on each side of the panel, it follows that for $p_2 = 0$, we have for the surface impedance Z of the panel $Z = \sigma e$. With an appropriate choice of the flow resistivity and the thickness of the material the surface impedance of the panel can be made equal to the impedance $Z = \rho^f c$ of a plane wave. Such a panel will be perfectly absorbent for plane waves in normal direction to the panel.
2. The control of the reflected sound field
   Under the assumption that the sound field $p^{scat}$ reflected by the panel is known, this quantity can be used directly as input for the active control algorithm. If and how this sound pressure can be measured is still an open question. Note that possible microphone position used to estimate the scattered sound field are restricted to be within the panel. Microphones within the cavity are not allowed.

The performance of these two strategies was studied using a numerical model for a cavity measuring 1.2×1.8×2.4 m$^3$. The walls are equipped with 2(2×3+2×4+3×4) = 52 panels covering the walls completely. In front of each membrane of the loudspeakers two points, one on each side of the absorbing ma-

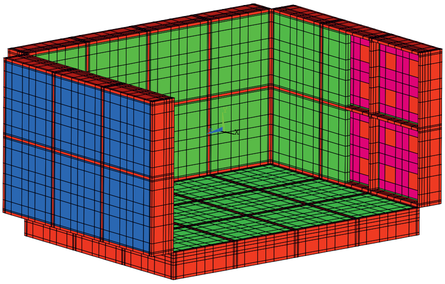

**Fig. 4** Model of a acoustic cavity equipped with hybrid wall panels.

terial, see Figure 3, were chosen as possible microphone positions. Microphone I served to evaluate the reflected sound pressure and microphone II was used with control strategy one. A part of the meshed walls, including the hybrid panels, is shown in Figure 4. The matrix of the transfer functions of the microphones at positions I and II, $H^{\mathrm{I}}$ and $H^{\mathrm{II}}$ respectively, and the supply voltage $U_q$ of the loudspeakers has been calculated in the 20 to 200 Hz frequency range using the computer code AKUSPOR developed at the LMA. A monopole sound source was used to generate an incident sound field $p^{\mathrm{inc}}$ in the cavity. The direct sound field at the microphones in front of the absorbing layer $p^{\mathrm{I}}$ and the total sound field behind the absorbing layer $p^{\mathrm{II}}$ were used to calculate the supply voltages $U_q^{\mathrm{I}}$ and $U_q^{\mathrm{II}}$ subsequently used to simulate the active control procedure. To asses the different control strategies two criteria were defined. The first

$$\eta_1 = \frac{\sum_{i\in\mathcal{I}} |p^i_{\mathrm{ctrl}}|}{\sum_{i\in\mathcal{I}} |p^i_0|} \tag{7}$$

compares the sound pressure $p^i_{\mathrm{ctrl}}$ at a set of points $\mathcal{I}$ obtained with control with the sound pressure $p^i_0$ obtained without control. Hence the reduction of the sound pressure is rated regardless of the incident sound field. The second criterion

$$\eta_2 = \frac{\sum_{i\in\mathcal{I}} |p^i_{\mathrm{inc}} - p_{\mathrm{ctrl}}|}{\sum_{i\in\mathcal{I}} |p^i_{\mathrm{inc}}|} \tag{8}$$

The sound pressure at various positions in the cavity with respect to the frequency are shown in Figure 6. We observe that both strategies reduce the significant impact of an acoustic mode of the cavity at 60 Hz. Hence the quite simple strategy of creating a zero sound pressure behind the absorbing material yields satisfactory results.

compares the sound pressure with control $p^i_{\mathrm{ctrl}}$ with the incident sound field and hence measures to what extent the sound field reflected by the walls has been removed by the active control procedure. Results for both criteria are shown in Figure 5. Both strategies yield a significant reduction of the total sound pressure in the cavity, see left sub-figure in Figure 5. The second control strategy became unstable

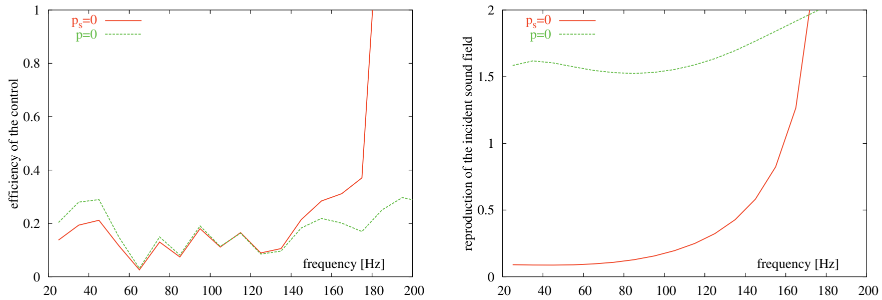


**Fig. 5** Efficiency of the active control procedures. Criterion $\eta_1$ in the left sub-figure and $\eta_2$ in the right sub-figure.

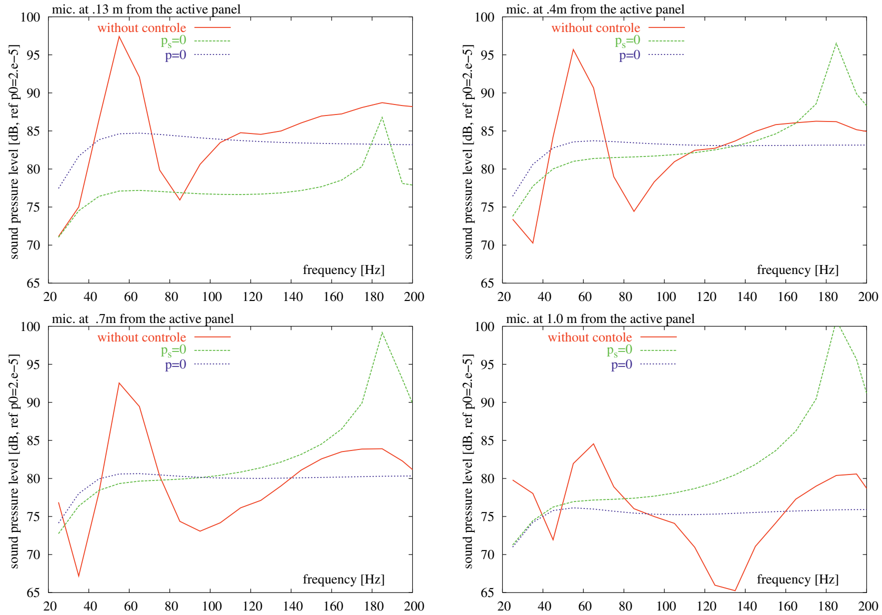


**Fig. 6** Sound pressure in the cavity with and without active control.

for frequencies above 160 Hz, however. But only the control strategy two, which directly controls the reflected sound field, was able to reproduce the incident sound field, see right sub-figure in Figure 5. The sound pressure distribution at 45 Hz on the side of the absorbing material facing the cavity and in a part of the cavity are shown in Figures 7 and 8. Strategy two reproduces quite well the incident sound field everywhere on the surface of the absorbing material, but also within the cavity. Strategy one does not reproduce the sound field obtained under free-field conditions, as with this strategy the panels are perfectly absorbent only for plane waves, but the

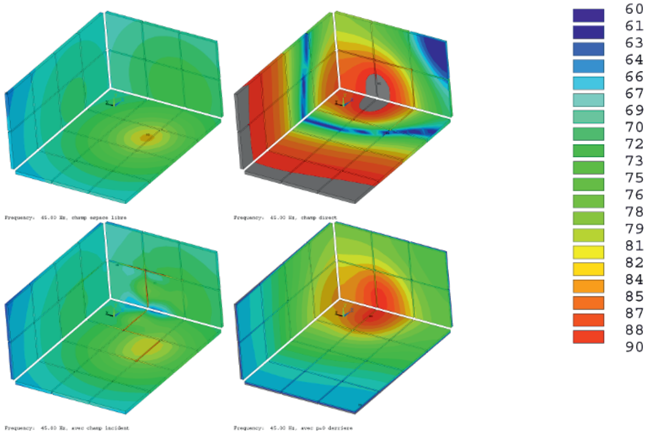


**Fig. 7** Sound pressure level on the absorbing material. Incident sound field (upper left sub-figure), total sound field in the cavity without control (upper right sub-figure), sound field with control strategy two (lower left sub-figure) and with strategy one (lower right sub-figure). Frequency: 45 Hz. The gray scale gives the sound pressure level in dB [ref. 2e-5].

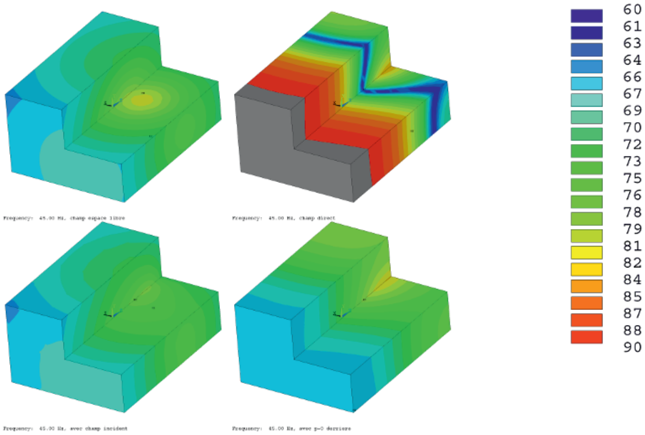


**Fig. 8** Sound pressure level in the cavity (one quater of the cavity has been cut out). Incident sound field (upper left sub-figure), total sound field in the cavity without control (upper right sub-figure), sound field with control strategy two (lower left sub-figure) and with strategy one (lower right sub-figure). Frequency: 45 Hz. The gray scale gives the sound pressure level in dB [ref. 2e-5].

sound pressure level within the cavity is reduced to the level of the direct sound field.

# 4 Conclusion

Two situations have been considered where the sound field scattered by an acoustic wall treatment is to be determined and subsequently removed.

First, we have presented a method to estimate the reflexions of the acoustic lining of an anechoic chamber in the very low frequency range. Using estimated diffraction filters the sound field reflected by the lining can be evaluated using the results of sound pressure measurements close to the lining. The estimation of these filters requires a dipole sound source that is placed such that the reference microphone does not receive any direct sound field from the source. The estimated diffraction filters do not depend on the sound source and allow therefore the estimation of the reflexions at the walls occurring when measurements are performed in the anechoic room at low frequencies.

Secondly, we have studied hybrid wall panels with an improved low-frequency absorption. Two control strategies were considered: (a) zero sound pressure behind the absorbing material and (b) direct cancellation of the reflected sound field. Using numerical experiments we have shown that the quite simple strategy (a) yields a significant improvement of the sound absorption of the panels.

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