The Sixteenth International Congress on Sound and Vibration
Kraków, 5-9 July 2009

# SHUNT LOUDSPEAKERS FOR MODAL CONTROL IN ROOMS

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Engineers dealing with noise reduction in habitations close to transportation traffic or industrial facilities encounter several problems to decrease noise level in rooms at low frequencies. Passive materials and current building construction knowledge enable to avoid noise transmission in habitations at medium and high frequencies and the regulations based on the dBA scale can often be respected. But these regulations do not take into account the real annoyance of noise for the inhabitants who are still disturbed by low frequency noise. Because of the modal behavior of rooms, air-borne and structure-borne noise generate high sound pressure level at the first modal frequencies, even with small amount of energy. In order to lower the annoyance induced by low-frequency behavior in rooms, huge compliant wooden panels can be used, though with quite low performances in terms of resonances damping. An alternative can be found with bass-traps, comprising a mechanical resonator and a closed-box, with fixed acoustic performance at specific frequency. Enhanced performances can even be obtained with closed-box shunt loudspeakers, where the acoustic impedance of the loudspeaker's diaphragm around its resonance can be modified (in terms of magnitude and also in frequency) by way of a simple passive electrical device, providing the requested complementary complex acoustic impedance for perfect absorption at the first modal frequencies. The present work describes the design of a small array of low-frequency shunt loudspeakers, dedicated to damp a certain amount of modes within a known room. Assessment in laboratory conditions (impedance tube, reverberant chamber) will then be presented and compared to simulations, leading to concluding remarks on practical issues on the applicability of the concept of shunt loudspeakers for low-frequency noise reduction in habitations.

## 1. Introduction

About 80 million people in the European Union are exposed to noise levels that are considered as unacceptable. Health, worker productivity, or comfort at home, are directly impacted by noise. Though, despite the complexity of noise annoyance, regulations are often based on indicators that obviously decrease the influence of low frequencies (roughly below 200 Hz) on the referenced level. The question has then raised whether these metrics are accurate when assessing the annoyance response to noise containing strong low-frequency components, and some alternative measures have been proposed to correlate the actual annoyance to objective assessments[1,2]. This situation might explain that only few low-frequency soundproofing solutions exist on the market.

In fact, usual acoustic treatments can be found with pretty good performances in the high and middle frequencies range, but present little performance for low frequencies. For the low frequency range, some panel absorbers[3] solutions exist, consisting of wide flat wooden panels (that can also be of metal, gypsum board, or plastic material), that are arranged in front of an enclosed air volume (partly or completely filled with porous material). Such a system has several resonance frequencies that can be excited by airborne sound. But their dimensions, as well as their weight, reduce their interest for usual rooms. These compliant structures are then mainly used to provide enhanced absorption in the low frequency range for demanding conditions such as concert halls. A more practical low-frequency absorber is the bass-trap, consisting of a small discrete membrane resonator, assembled within a closed-box, which resonance frequency is tuned so that to provide sufficient absorption within a narrow frequency band.

This paper aims at presenting a system dedicated to reduce low frequency noise, inspired by the bass-trap technique, but involving electroacoustic transducers in a passive resonator use, allowing tuneable acoustic performances. This technique is based on an electromechanical transducers loaded with a passive electrical circuit so that to obtain an optimal damping around its resonance frequency. A quick presentation of the technique will be provided, together with numerical simulation of the acoustic behaviour in standard configurations (impedance tube), and experimental assessments in actual room to highlight its performances.

## 2. Influence of rooms' modal behavior on indoor noise

In the low frequency range, a room's acoustic response is driven by the modal behavior which is characterized by high levels at eigenfrequencies that can be very annoying inside rooms[3,4]. Several measurements have been performed near the Geneva Airport. Figure 1 illustrates the modal behaviour in an office, located under the take off path of airplanes.

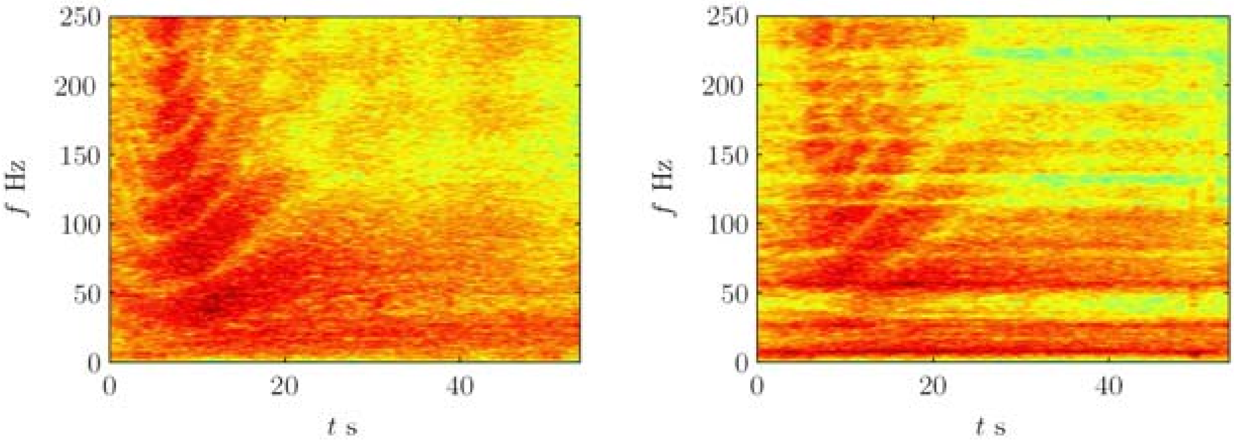


**Figure 1: spectrograph measured during aircraft flyover outside a room (left chart) and inside (right chart).**

When comparing the noise outside and inside the room, one can observe that the noise spectrum inside the room appears to be concentrated on modal frequencies (straight horizontal red lines), whereas the noise outside is strictly characteristic of the source (with the Doppler effect illustrated by the curved red areas). To reduce the annoyance in the room, the sound pressure level has to be reduced at these identified frequencies. If the modes of the room are excited with the broadband low frequency noise excitation of an air plane, it becomes obvious that the main energy concentrates on the eigenfrequencies in the room, and at the main annoying frequency, the noise level inside the room is higher than outside the building.

The former example highlights the necessity to perform a damping of the first modal frequencies in a room to reduce the noise annoyance. This can be performed with the shunt loudspeaker concept described in the following.

# 3. Shunt loudspeakers

## 3.1 Design of shunt loudspeakers

An electrodynamic loudspeaker is a physical system that is generally described with differential equations[5]. From Newton's law of motion, its mechanical dynamics assuming piston-like vibrations, can be modeled by a second-order differential equation, given as

$$m_{\mathrm{s}}\ddot{x}(t)+\frac{1}{C_{\mathrm{ms}}}x(t)+R_{\mathrm{ms}}\dot{x}(t)=\left(Bl\right)i(t)-Sp(t) \tag{1}$$

where $m_{\mathrm{s}}, C_{\mathrm{ms}}, R_{\mathrm{ms}}$ are the mass, compliance and mechanical resistance respectively of the loudspeaker's moving part, $\left(Bl\right)$ the force factor, $x(t)$ the diaphragm position, $i(t)$ the driving current, $S$ the effective piston area, and $p(t)$ the exogenous pressure disturbance acting on the loudspeaker diaphragm. The electrical dynamics can also be modeled by a first order differential equation after meshes law, given as

$$u(t)=R_{\mathrm{e}}i(t)+L_{\mathrm{e}}\frac{di(t)}{dt}+\left(Bl\right)\dot{x}(t) \tag{2}$$

where $u(t)$ is the applied voltage, $R_{\mathrm{e}}$ and $L_{\mathrm{e}}$ the dc resistance and inductance of the voice coil, and $\left(Bl\right)\dot{x}(t)$ the back emf induced by the motion of the voice coil within magnetic field. Equations (1) and (2) form a coupled set of differential equations describing the loudspeaker system. Expressing the preceding relations in complex quantities (phasor representation) yields the characteristic equations of the generalized electrodynamic loudspeaker, given as

$$\begin{cases} S\underline{p}=\left(j\omega m_{\mathrm{s}}+R_{\mathrm{ms}}+1/j\omega C_{\mathrm{ms}}\right)\underline{v}-\left(Bl\right)\underline{I} \\ \underline{U}=\left(R_{\mathrm{e}}+j\omega L_{\mathrm{e}}\right)\underline{I}+\left(Bl\right)\underline{v} \end{cases} \tag{3}$$

Under driving operation, $S\underline{p}=0$ and no external force is applied to the mechanical system; under generating operation $\underline{U}=0$ and there is no voltage source at the electric terminals of the loudspeaker. For simulating the dynamics of a loudspeaker we use an approach based on electroacoustic analogy. By taking account of the internal radiation impedance induced by the boxed environment we obtain the analytical description of the loudspeaker system, be it mechanical or acoustical, in the form of a synthetic circuit illustrated on the figure below.

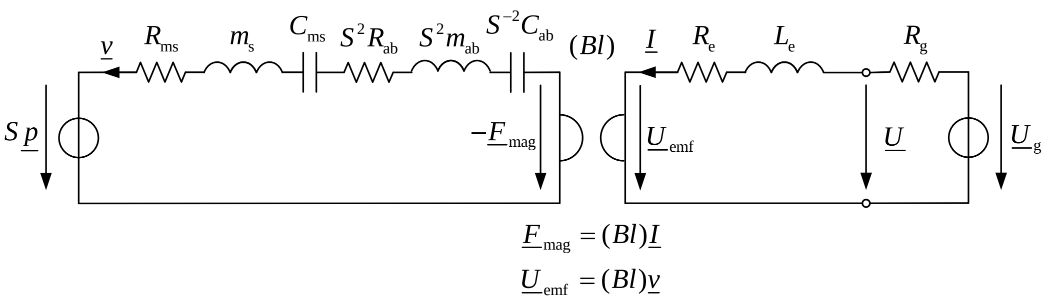


**Figure 2: Circuit representation of generalized closed-box loudspeaker system**

where $U_{\mathrm{g}}$ and $R_{\mathrm{g}}$ are the voltage source and internal resistance of the audio amplifier.

Starting from the circuit representation, if we remove the audio amplifier on the electric side to replace it by shunt impedance $\underline{Z}_{\mathrm{sh}}$, the driving current in the coil becomes equal to $\underline{I}=-\underline{U}_{\mathrm{emf}}/\left(\underline{Z}_{\mathrm{e}}+\underline{Z}_{\mathrm{sh}}\right)$ where $\underline{Z}_{\mathrm{e}}$ is the blocked electrical impedance of the coil ($\underline{v}=0$). Solving the equations (3) when the loudspeaker system is connected to a shunt impedance yields to the specific acoustic impedance $\underline{Z}_\mathrm{s}$, given as

$$\underline{Z}_\mathrm{s} = \frac{p}{\underline{v}} = S\underline{Z}_\mathrm{ab} + \frac{\underline{Z}_\mathrm{m}}{S} + \frac{\left(Bl\right)^2}{S\left(\underline{Z}_\mathrm{e} + \underline{Z}_\mathrm{sh}\right)} \qquad \textbf{(4)}$$

where $\underline{Z}_\mathrm{ab}$ is the acoustic impedance corresponding to the closed-box and $\underline{Z}_\mathrm{m}$ the mechanical impedance of the diaphragm.
At last, the acoustic absorption coefficient under normal incidence is defined as:

$$\alpha = 1 - \left|\frac{\underline{Z}_s/\rho c - 1}{\underline{Z}_s/\rho c + 1}\right|^2 \qquad \textbf{(5)}$$

where $\rho$ and $c$ are the density of air and the celerity of sound in the air.

### 3.2 Numerical simulations

The figures below illustrate the effect of adding a mass to the moving part of the loudspeaker (dotted line with cross markers curves), and then connecting a suitable resistive load to its electrical terminals (dotted line with circular markers curves), compared to the case of the open closed-box loudspeaker (continuous line curves). The results are expressed in terms of acoustic impedance (left chart), and absorption coefficient (right chart). These results give the evidence of the performances of the shunt loudspeaker technique for use as locally reacting low-frequency noise absorbers, for example in the scope of damping low-frequency modal behaviour in a room.

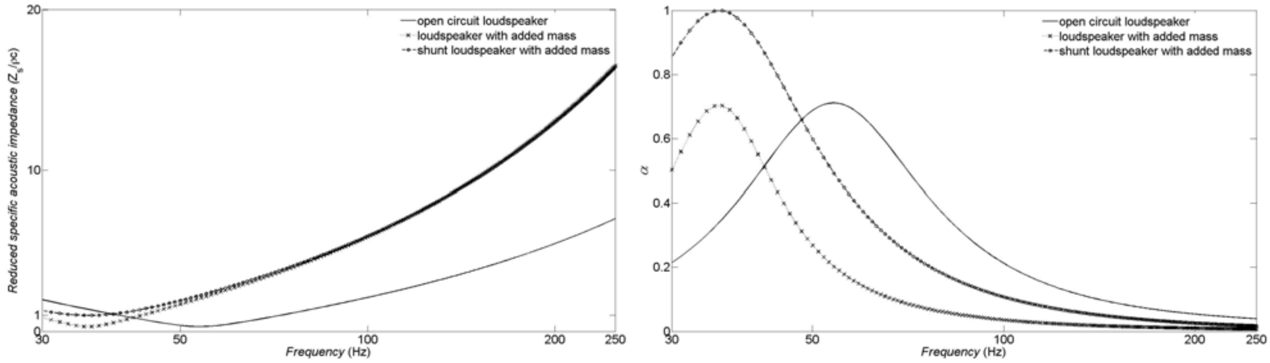


**Figure 3: computed normalized acoustic impedance $\underline{Z}_\mathrm{s}/\rho c$ (left) and absorption coefficient $\alpha$ (right) of the shunt loudspeaker system**

### 3.3 Observations

A basic analytical description of the loudspeaker in the frequency domain is able to model its dynamics in piston mode, and hence to predict its acoustic performance. By conducting a short sensitivity analysis on the loudspeaker parameters we can tune the resonant part of its frequency response toward a desired value. Then, by connecting suitable shunt impedance, the quality factor can be enhanced to provide an improved absorption[6].

## 4. Damping of rooms' modal behaviour with shunt loudspeakers

Having shown the capability of the shunt loudspeaker concept to adapt the acoustic impedance of a loudspeaker diaphragm to a desired value, it can be assessed as a solution for low-frequency absorber especially in the frame of the damping of modal behaviour in rooms. The following intends to demonstrate the performance of the shunt loudspeaker technique as low-frequency absorber for closed spaces.

### 4.1 Simulations

In order to perform both computational and experimental assessment within the facilities of the Laboratory of Electromagnetics and Acoustics at EPFL (LEMA), we have chosen the reverberant chamber, of about 200 m$^3$, the complex geometry of which does not allow the use of an analytical solution for the stationary waves description. It has then been chosen to design a specific finite-element model of the reveberant chamber, with the help of Comsol Multiphysics® 3.5a software.

#### *4.1.1 Simulation setup*

The following characteristics have been considered:

- the reverberant chamber geometry has been accurately modelled as illustrated on Figure 4, the 6 walls being denoted in the following $Sx_1$, $Sx_2$, $Sy_1$, $Sy_2$, $Sz_1$, $Sz_2$,
- the acoustic domain has been set up, with frequency dependant parametric solver,
- the walls of the reverberant chamber have been considered as having a uniform specific impedance $Z_{\text{walls}}= 10^6$ Pa.s.m$^{-1}$ (determined after a preliminary assessment of the quality factor of the mode “(1,1,0)” in the room),
- all loudspeakers in the model (source and absorbers) are considered as circular pistons with specific boundary conditions (velocity for the source, impedance for absorbers),
    - a circular piston of radius $a$=150 mm represents the acoustic source (loudspeaker), embedded within wall $Sx_1$, roughly at a corner of the room. The excitation consists of a velocity depending on frequency after the mechanical impedance of the loudspeaker, assuming a frequency independent electro-mechanical Laplace force at the loudspeaker diaphragm,
    - a horizontal array of 10 circular pistons of radius $a$=150 mm represents the variable absorbers, embedded within wall $Sx_2$, roughly along the bottom line of the face. They are computed as impedant boundary conditions, the value of which can be set to $Z_{\text{walls}}$ or to $\underline{Z}_s$, depending on the computation case (resp. for hard walls and shunt loudspeakers conditions).

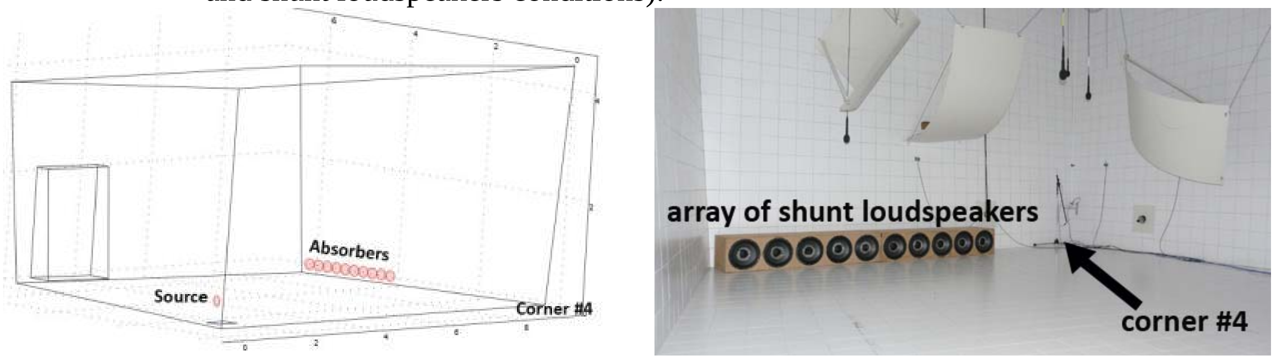


**Figure 4 : the reverberant chamber at LEMA (left: CAD model in Comsol Multiphysics®, right: photograph taken from the entrance)**

#### *4.1.2 Simulation results*

The sound pressure levels at corner #4 (as defined on Figure 4) computed with Comsol Multiphysics®, without any treatment (“hardwalls”), and with an array of 10 low-frequency shunt loudspeakers along one face of the room are given in Figure 5.

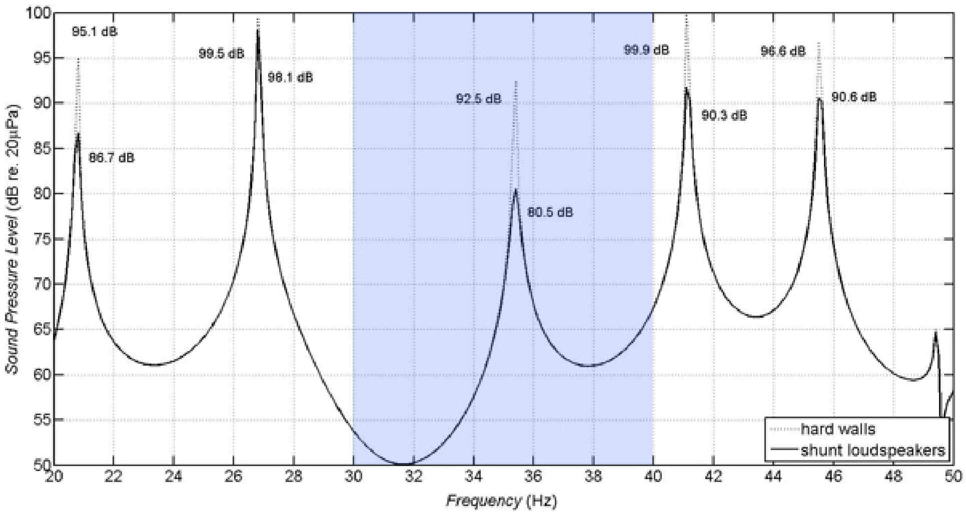


**Figure 5: Sound pressure levels computed between 20 Hz and 50 Hz, at corner 4, for different treatments (hard walls and shunt loudspeakers)**

It can be observed that a small surface of treatment (not more than 0.7 $m^2$) is sufficient to damp of about 12 dB the main resonance frequencies in the reverberant chamber. In the following, we will focus on the 34.9 Hz resonance frequency of the room, corresponding to mode (1,1,0) (this notation is not rigorous since the room is not a parallelepiped). The experimental assessment will then address the blue zone in the room response illustrated above.

## 4.2 Experimental assessment

In order to assess the accuracy of the model, and verify the performances of the shunt loudspeakers in rooms, it has been decided to set up an experimental setup with similar specification than the numerical computation. For the purpose of the present paper, we have only focussed on the mode 110 at 34.9 Hz, for which the shunt loudspeakers have been optimized. Then the following experimental assessments will only address the [30 Hz-40 Hz] frequency bandwidth.

### *4.2.1 Experimental setup*

- the source is a home-made bass-reflex loudspeaker system, designed to provide the requested acoustic power in the requested frequency bandwidth,
- the excitation is a discrete swept sine with 0.1 Hz frequency steps from 30 Hz to 40 Hz, each step lasting 30s, allowing the extinction of any prior mode, and the establishment of stationary sound field at each frequency step (this technique, illustrated on Figure 7 and Figure 8, provides better results than other excitation as explained in a prior study[3]),
- an array of 10 shunt closed-box loudspeakers (Monacor SPH-300 each in a 50 dm3 closed-box) is put parallel to the bottom line of wall $Sx_2$, the loudspeakers' face pointing along direction *x*,
- a Brüel & Kjaer Type 4165 electret microphone (sensitivity 50 mV/Pa) senses the acoustic pressure at corner 4 of the room.

### *4.2.2 Experimental results*

The results of assessment of the influence of the array of absorbers on the identified mode at 35 Hz is given in Figure 6, and compared to the same measurement with no absorber (“hardwalls”). It is also to be compared with computation results given in the preceding section.

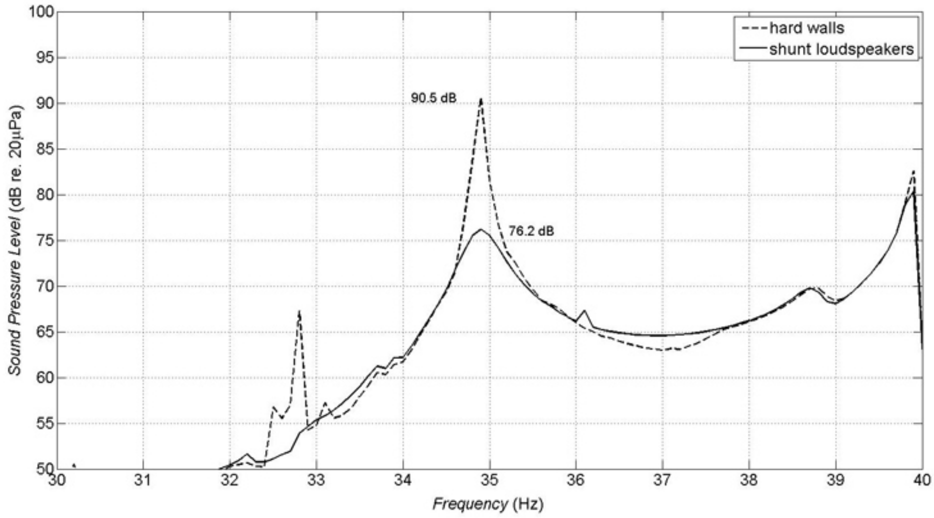


**Figure 6: Stationnary sound pressure levels measured in the LEMA reverberant chamber, between 30 Hz and 40 Hz, at corner #4, for 2 different treatments (hard walls and shunt loudspeakers)**

It can be observed that a simple array made of 10 passive shunt loudspeakers is sufficient to reduce of more than 10 dB the level of the 34.9 Hz resonance of the room. These performances are obviously strongly dependant on the position of the absorber, and a further sensitivity study should be performed for a better understanding of the concept within wide reverberant rooms.

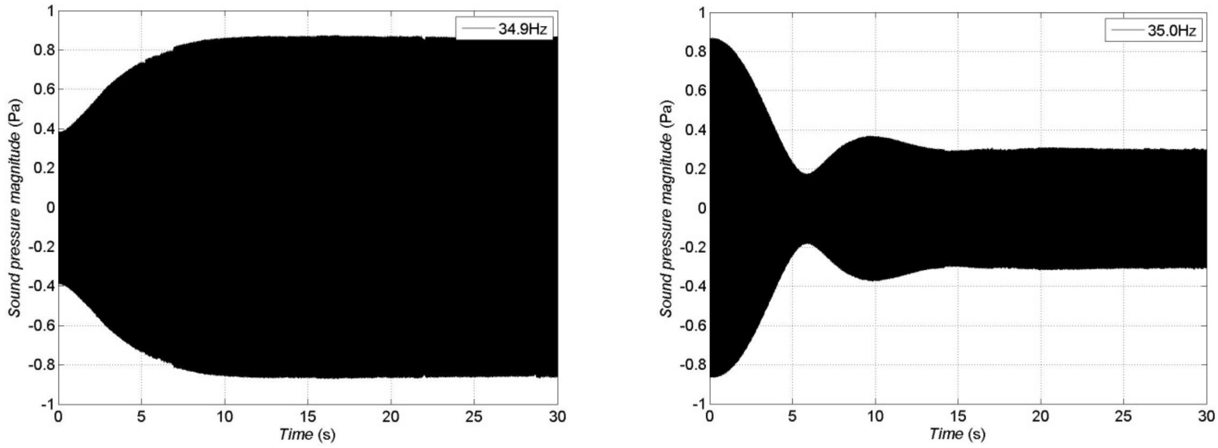


**Figure 7: temporal behaviour of the acoustic pressure in the vicinity of the mode, with hard walls (left: 34.9 Hz; right: 35.0 Hz)**

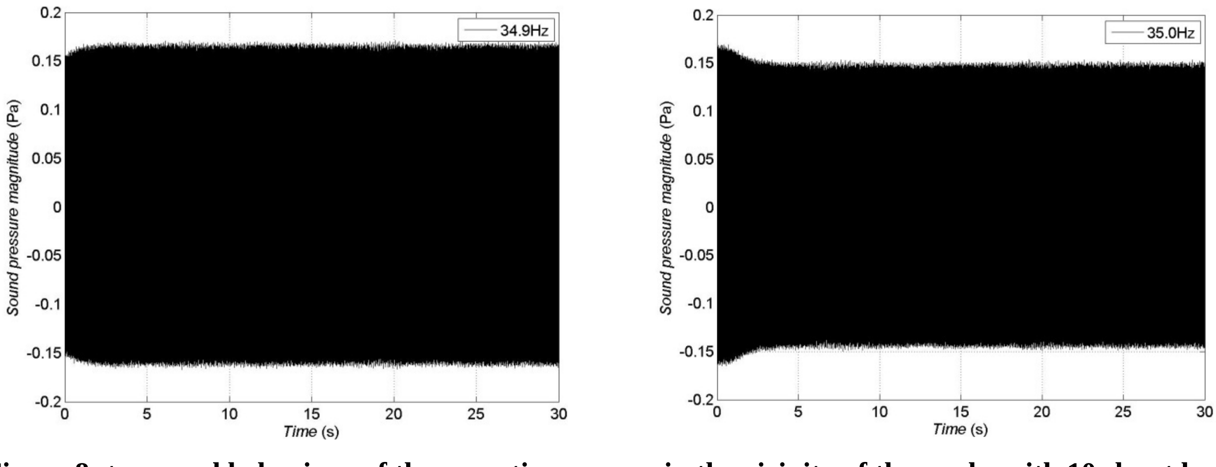


**Figure 8: temporal behaviour of the acoustic pressure in the vicinity of the mode, with 10 shunt loud-speakers (left: 34.9 Hz; right: 35.0 Hz)**

## 5. Conclusions

It has been proven that shunt loudspeakers is an efficient technique for providing enhanced acoustic absorption in the low-frequency range, and allowing a rather broad variability. An experimental proof of concept has been performed with an Impedance Tube assessment, after ISO 10534-2 standard, and the assessment in a 3D situation such as the reverberant chamber show the efficiency of this concept in an almost realistic condition, with a broadband noise excitation. Further experimental validations should be performed with less impedant walls, such as those of actual habitations, and with real low-frequency noise sources.

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