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Study on room modal equalization at low frequencies with electroacoustic absorbers

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Nowadays, requirements for acoustic treatments in rooms are ever more demanding in terms of performance and compactness. A recurring issue in closed spaces is the occurrence of standing waves and acoustic interferences at low frequencies. These unwanted phenomena are likely to affect the frequency response of rooms such as offices, concert halls or home theaters. Unfortunately, state-of-the-art soundproofing solutions cannot efficiently dissipate sound energy, or their embodiments are so bulky that they become almost impracticable. To that purpose, electroacoustic absorbers employing direct-radiator loudspeaker systems can be used to improve listening quality and meet the specifications. Electrodynamic transducers are obvious candidate for this type of noise control application. They are controlled by the resistance around the mechanical resonance, typically of the order of a few tens of Hertz, which also lies in the frequency range of acoustic modes to be damped. This results in a strong interaction between both dynamic systems from which part of the incident acoustic energy of the sound field can be dissipated. A specific electrical load is also connected across the transducer terminals in order to further improve the passive dissipation through the internal damping within the loudspeaker. This paper investigates an optimized placement of electroacoustic absorbers in a reverberant room in order to damp the lowest acoustic resonances.

1 Introduction

Modal equalization is an experimental method specifically addressing modal resonances issues in enclosed spaces [1]. Generally speaking, the response of a room may result in a high sound pressure level at certain frequencies which coincide with the room's natural frequencies, even with small amount of sound energy [2, 3]. This leads to an uneven spatial distribution of acoustic energy in the room, namely the nodes and antinodes of sound pressure. When the sound stops the natural resonances of the room may even sustain. During a concert or conference, such room effect can be detrimental in terms of sound clarity and definition, or even harmful to intelligibility [4]. Consequently, low-frequency resonances have prejudicial audible effects in the rendering of sound in listening spaces. Unfortunately, the state-of-the-art soundproofing means such as porous materials or micro-perforated panels cannot efficiently dissipate sound energy in the frequency range of interest. To address the low frequencies sound absorption more specifically, resonant panels (or membrane) and Helmholtz resonators can be used [5]. Typical examples of passive resonant systems are referred to as bass traps. Their achievements are often too bulky, however, making this option almost impracticable in some situations.

The following discusses the modal equalization of a room through the use of electroacoustic absorbers [6]. The general idea is to control the acoustic impedance at discrete locations within the room where antinodes are particularly pronounced. By electroacoustic absorbers, we mean a loudspeaker the behavior of which can be significantly altered by connecting an appropriate electrical load [7, 8]. In this way, the sound absorption capability of the transducer diaphragm can be improved by extending the control bandwidth around the transducer resonance. The choice of the electrodynamic transducer is first justified as an obvious candidate for several reasons: the immediate availability, a relative low cost, and electromagnetic properties that can advantageously alter the transducer dynamics in a controlled fashion.

The remainder of the paper is organized as follows. The concept of electroacoustic absorber is first introduced from the characteristic equations of the electrodynamic loudspeaker. The intake of coupling a specific electrical load across the transducer terminals is discussed in terms of sound absorption capability. The methodology applied for damping some modal resonances of the room is then provided. As a conclusion, the overall performances are discussed from data measured in a test room, together with some concluding remarks on the optimized placement of electroacoustic absorbers.

2 Electroacoustic absorbers

2.1 Governing equations

For small displacements and below the first modal frequency of the diaphragm, the generalized governing equations of an electrodynamic loudspeaker can be obtained after Newton's second law and Kirchhoff's circuit law [4]. With the use of Laplace transform, the characteristic equations of the transducer can be expressed as

$$\begin{cases} S\,P(s) = \left(sM_{ms} + R_{ms} + \dfrac{1}{s\,C_{mc}}\right)V(s) - Bl\,I(s) \\ E(s) = (sL_e + R_e)\,I(s) + Bl\,V(s) \end{cases} \tag{1}$$

where $P(s)$ is the driving pressure acting on the transducer diaphragm, $V(s)$ is the diaphragm velocity, $I(s)$ the driving current and $E(s)$ is the voltage applied to the electrical terminals (cf. Fig. 1). For the model parameters, S is the effective piston area, Bl is the force factor of the transducer (product of B, the magnetic field amplitude and l, the length of the wire in the voice coil), M_{ms} and R_{ms} are the mass and mechanical resistance of the moving body, R_e and L_e are the dc resistance and the lossless inductance of the voice coil, respectively.

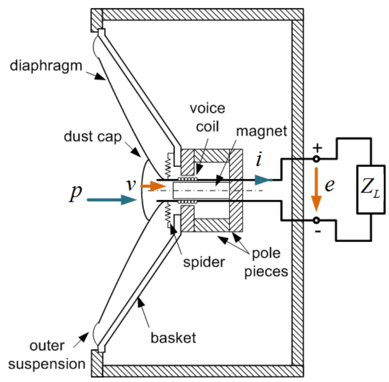

Figure 1: Schematic of a direct-radiator loudspeaker system when connected to an electrical load.

Here, $C_{mc} = (1/C_{ms} + \rho c^2/V_b)^{-1}$ is the equivalent mechanical compliance accounting for both the flexible edge suspension and spider of the loudspeaker C_{ms} and the enclosure, where ρ and c are the density and celerity of air and V_b is the volume of the enclosure. The coupling term $Bl\,I(s)$ represents the Laplace force induced by the current circulating through the coil and $Bl\,V(s)$ is the back electromotive force induced by the motion of the coil within the magnetic field.

2.2 Coupling to an electrical load

By connecting an electric load Z_L as depicted in Fig. 1, the voltage applied across the transducer terminals becomes

$$E(s) = -Z_L(s)\,I(s) \tag{2}$$

and the electrical current flowing through the coil can be written as

$$I(s) = -\frac{Bl}{Z_e(s) + Z_L(s)} V(s) \tag{3}$$

where $Z_e(s) = R_e + sL_e$ is the blocked electrical impedance of the voice coil. When designed properly, the shunt electrical impedance Z_L can make a functional relationship between the induced voltage $Bl\,V(s)$ and electrical current, thus taking precedence over the transducer dynamics.

2.3 Acoustic absorption capability

A closed form expression of the specific acoustic admittance at the transducer diaphragm can always be derived after Eqs. (1-2) regardless of the load connected across its terminals. Normalizing relative to the characteristic impedance of the medium ρc, the specific acoustic admittance ratio can be expressed as

$$y(s) = \rho c \frac{V(s)}{P(s)} \tag{4}$$

This dimensionless parameter reflects the motion (response) of the diaphragm that is caused by the driving acoustic pressure. By combining Eqs. (1-4), the generalized velocity response of the transducer diaphragm to any surrounding sound field can be expressed as

$$y(s) = \rho c S \frac{Z_e(s) + Z_L(s)}{Z_m(s)\big(Z_e(s) + Z_L(s)\big) + (Bl)^2} \tag{5}$$

where $Z_m(s) = sM_{ms} + R_{ms} + 1/(s\,C_{mc})$ is the mechanical impedance. The corresponding reflection coefficient under normal incidence can be derived after

$$r(s) = \frac{1 - y(s)}{1 + y(s)} \tag{6}$$

and the extraction of the magnitude $|r(\omega)|$ of $r(s)$, where $s = j\omega$, yields the sound absorption coefficient $\alpha(\omega)$ as

$$\alpha(\omega) = 1 - |r(\omega)|^2 \tag{7}$$

Figure 2 illustrates the measured sound absorption coefficient, assessed after ISO 10534-2 standard [10], when connecting a low-range Monacor SPH-300TC loudspeaker (see Tab. 2) mounted in sealed enclosure of volume $V_b = 23\,\mathrm{L}$ to a specific shunt electrical load designed to increase the control bandwidth, compared to the open circuit's case. As clearly shown in Fig. 2 an effective control of the acoustic impedance in the room is expected in the frequency range between 50 Hz and 100 Hz.

3 Modal equalization of the room

The present work considers a room with parallel pairs of walls, the pairs being perpendicular to each other. Although rather distant from the shapes of auditoria or concert halls, this basic geometry is a good example to understand the low-frequency distribution of acoustic energy in closed spaces.

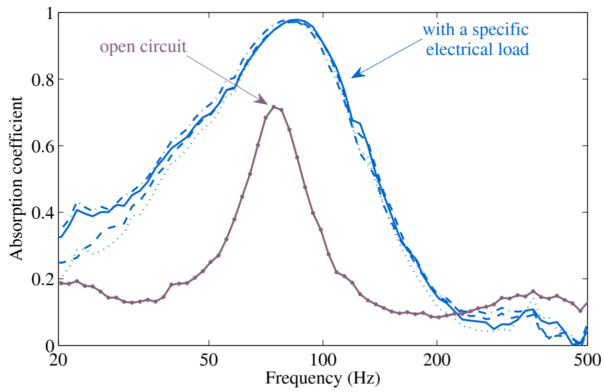

Figure 2: Measured sound absorption coefficient of the four electroacoustic absorbers used in the experiments in open circuit and connected to a specific electrical load.

The experiment is carried out in a technical room (width L_x = 3 m by length L_y = 5.6 m by height L_z = 3.53 m) used to store containers of waste sorting. The total area is 94.3 m^2 and the volume is 59.3 m^3. Such a room is characterized by a strong reverberation due to rigid walls and is likely to have isolated modes, particularly at low frequencies.

3.1 Eigenfrequencies identification

The first stage is to identify the modes of the room to be damped. The explicit evaluation of the eigenvalues and eigenfunctions of usual room is generally quite difficult and requires the application of numerical methods such the finite element method (FEM). From the wave theory of room acoustics, a closed form expression for these quantities can be derived after solving the eigenvalue [1, 9]. According to the Helmholtz equation

$$\nabla^2 p + k^2 p = 0 \tag{8}$$

where ∇ is the nabla operator and $k = \omega/c$ is the wave number, and for an ideal case of rectangular room with rigid wall, i.e. with associated boundary conditions such that

$$\begin{cases} \dfrac{\mathrm{d}p}{\mathrm{d}x} = 0 & \text{for } x = 0 \text{ and } x = L_x \\ \dfrac{\mathrm{d}p}{\mathrm{d}y} = 0 & \text{for } y = 0 \text{ and } y = L_y \\ \dfrac{\mathrm{d}p}{\mathrm{d}z} = 0 & \text{for } z = 0 \text{ and } z = L_z \end{cases} \tag{9}$$

the general solution for the sound pressure can be written, after applying the separation of variables, as

$$p_{n_x n_y n_z}(x, y, z) = C\,\cos\left(\frac{n_x \pi}{L_x} x\right) \cos\left(\frac{n_y \pi}{L_y} y\right) \cos\left(\frac{n_z \pi}{L_z} z\right) \tag{10}$$

where C is an arbitrary constant and n_x, n_y and n_z are integers that define the structure of the modes, and the eigenvalues can be expressed as

$$k^2 = k_x^2 + k_y^2 + k_z^2 = \pi^2 \left[\left(\frac{n_x}{L_x}\right)^2 + \left(\frac{n_y}{L_y}\right)^2 + \left(\frac{n_z}{L_z}\right)^2\right] \tag{11}$$

The corresponding eigenfrequencies are given by

$$f_{n_x n_y n_z} = \frac{c}{2} \sqrt{\left(\frac{n_x}{L_x}\right)^2 + \left(\frac{n_y}{L_y}\right)^2 + \left(\frac{n_z}{L_z}\right)^2} \tag{12}$$

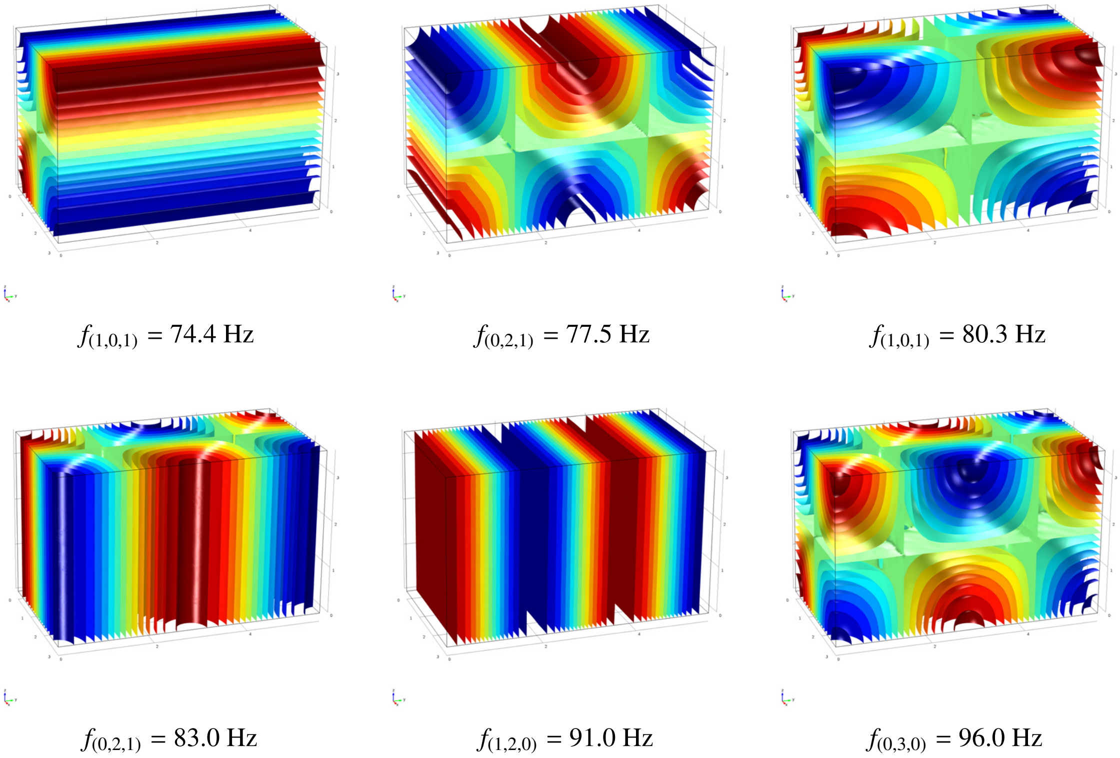

Figure 3: Isosurface of the total acoustic pressure field illustrating a selection of modes of the room between 70 Hz and 100 Hz.

Table 1 summarizes the eigenfrequencies of the room between 70 Hz and 100 Hz for c = 343 m.s^{-1}, together with the corresponding mode structure given by subscripts (n_x, n_y, n_z).

Table 1: Eigenfrequencies of the rectangular room with dimensions $3 \times 5.6 \times 3.53$ m^3 between 70 Hz and 100 Hz.

n_x	n_y	n_z	Eigenfrequency (Hz) Computed	Eigenfrequency (Hz) Measured	Type of mode
1	0	1	74.4	73.0	Tangential
0	2	1	77.5	76.0	Tangential
1	1	1	80.3	79.7	Oblique
1	2	0	83.0	83.9	Tangential
0	3	0	91.0	91.6	Axial
1	2	1	96.0	95.2	Oblique

Figure 3 shows the computed isosurface of the total acoustic pressure field in the room for a selection of eigenfrequencies. This graphical representation clearly illustrates the nodes and antinodes of pressure.

3.2 Location of the electroacoustics absorbers

The analysis of the acoustic energy distribution shows that the electroacoustic absorbers must be carefully placed in the room. At low frequencies, typically where the size of electroacoustic absorbers becomes small relative to the wavelength, the coupling with the room is inefficient when they are located on nodes of pressure [3]. For optimal performance, it is best to place them on pressure antinodes, while directing the transducers diaphragm according to the modes structure to be damped.

For ease of understanding, only two configurations will be discussed in the following. In configuration C$_1$, the electroacoustic absorbers are placed in corners 3, 4, 7 and 8, as depicted in Fig. 4 (up). The orientation is such that the diaphragm of each loudspeaker is facing edges along the y-axis. In configuration C$_2$, the loudspeakers are located on the ground in the corners 1, 2, 3 and 4, and the orientation of the diaphragms is facing edges along the x-axis (Fig. 4 (down)).

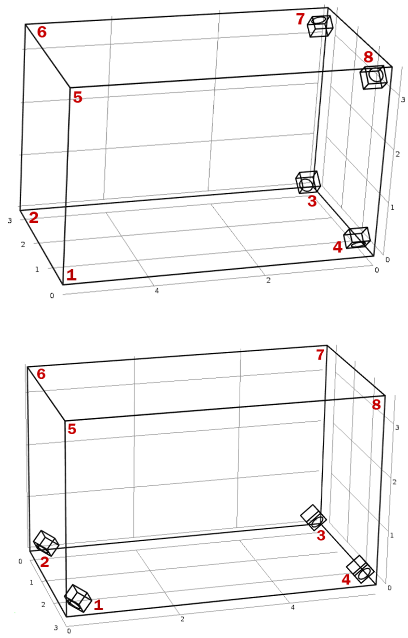

Figure 4: Location of electroacoustic absorbers in the room: configuration C$_1$ (up) and configuration C$_2$ (down).

4 Results and discussion

4.1 Experimental setup

The sound source used for measuring the frequency response of the room is a low-midrange loudspeaker in sealed enclosure, designed to provide the necessary acoustic power in the frequency range of interest. It is placed on the floor in the corner 1 of the room. The signal excitation is a random pink noise the frequency span of which is between 20 Hz and 500 Hz. For the spectral analysis we used a frequency resolution of 0.1 Hz in order to examine the frequency response of the room in more detail. The measured response of the room is picked up by a 1/2" microphone (Norsonic Type 1225 cartridges mounted on Norsonic Type 1201 amplifier), the sensitivity of which is $50\,\mathrm{mV\,Pa^{-1}}$, and processed through a 01dB-Metravib multichannel analyzer. In configuration C_1, the microphone is located in corner 8 and in corner 2 for the configuration C_2. Table 2 summarizes the small signal parameters of the low-range Monacor SPH-300TC loudspeaker used in the experiments. The pictures of the experimental setup are presented in Fig. 5.

Figure 5: Pictures of the configurations C_1 (up) and C_2 (down) tested in the room (corner ♯1 at the bottom right).

Table 2: Small signal parameters of the Monacor SPH-300TC.

Parameter	Notation	Value	Unit
dc resistance	R_e	6.3	Ω
Voice coil inductance	L_e	1	mH
Force factor	Bl	10.3	$\mathrm{N\,A^{-1}}$
Moving mass	M_{ms}	68	g
Mechanical resistance	R_{ms}	3.24	$\mathrm{N\,m^{-1}\,s}$
Mechanical compliance	C_{ms}	0.85	$\mathrm{mm\,N^{-1}}$
Effective area	S	495	$\mathrm{cm^2}$
Natural frequency	f_0	23	Hz

4.2 Damping of the low-frequency modes

The sound pressure level measured in the room (in corner 8 with C_1 and in corner 2 with C_2) is shown in Fig. 6. The maximum gains in decibels for the frequency range between 70 Hz and 100 Hz are summarized in Tab. 3.

Table 3: Measured gains for the C_1 and C_2 configurations.

Type of mode	$n_x\,n_y\,n_z$	Eigenfrequency (Hz)	C_1 (dB)	C_2 (dB)
Tangential	1 0 1	74.4	-11.7	-8.6
Tangential	0 2 1	77.5	-4.0	-6.2
Oblique	1 1 1	80.3	-6.8	-9.0
Tangential	1 2 0	83.0	-7.9	-9.4
Axial	0 3 0	91.0	3.9	-4.7
Oblique	1 2 1	96.0	-7.1	-13.6

The measured data clearly show the intake of acoustic impedance control in view of damping low-frequency modes of a room. It can be observed that a small treatment area is enough to significantly decrease the sound pressure level of the lowest natural resonances of the room. As given in Tab. 2, the effective area of the diaphragm for each electroacoustic resonator equals 0.0495 $\mathrm{m^2}$ only. Even with an equivalent absorption area of only $0.2\,\mathrm{m^2}$ (compared to the total surface area $94.3\,\mathrm{m^2}$ of the room), gains of over 10 dB can be measured on some modes. Note that the electroacoustic absorbers provide efficient damping for a large number of modes. The configuration C_1 is particularly effective in damping the tangential mode (1,0,1) at 80.3 Hz with a gain of -11.7 dB. And the configuration C_2 is optimized to damp the oblique mode (1,2,1) at 96.0 Hz with a gain of -13.6 dB. The slight shift of frequency that can be observed on some modes compared to rigid wall (see Fig. 6) can be attributed to the effects of coupling between the room and the electroacoustic absorbers. The measured gains are obviously strongly dependent on the location of electroacoustic absorbers in the room. This result is in accordance with the analysis of the acoustic energy distribution in the room (see Fig. 3). Through the proposed methodology, acoustic corrections can be made in the room, in a frequency range where the usual soundproofing treatments are ineffective or would be impractical to apply.

5 Conclusion

In this paper we discussed a practical realization of electroacoustic absorbers in view of controlling the low-frequency sound field in closed spaces. A simple engineering approach employing an arrangement of electrodynamic loudspeakers the terminals of which are connected to a specific electrical load has been presented. Through judicious control of acoustic impedance in a test room a significant damping of the dominant natural resonances can be achieved. It is shown that the magnitude of the low-frequency resonances of the room can be greatly reduced, even with a very small equivalent absorption area. The surface of acoustic treatment is only 0.2 $\mathrm{m^2}$, while the total area of the room is 94.3 $\mathrm{m^2}$. The measured gains are largely related to the location of the absorbers in the room as well as the orientation of the diaphragm relative to the modes structure. The proposed

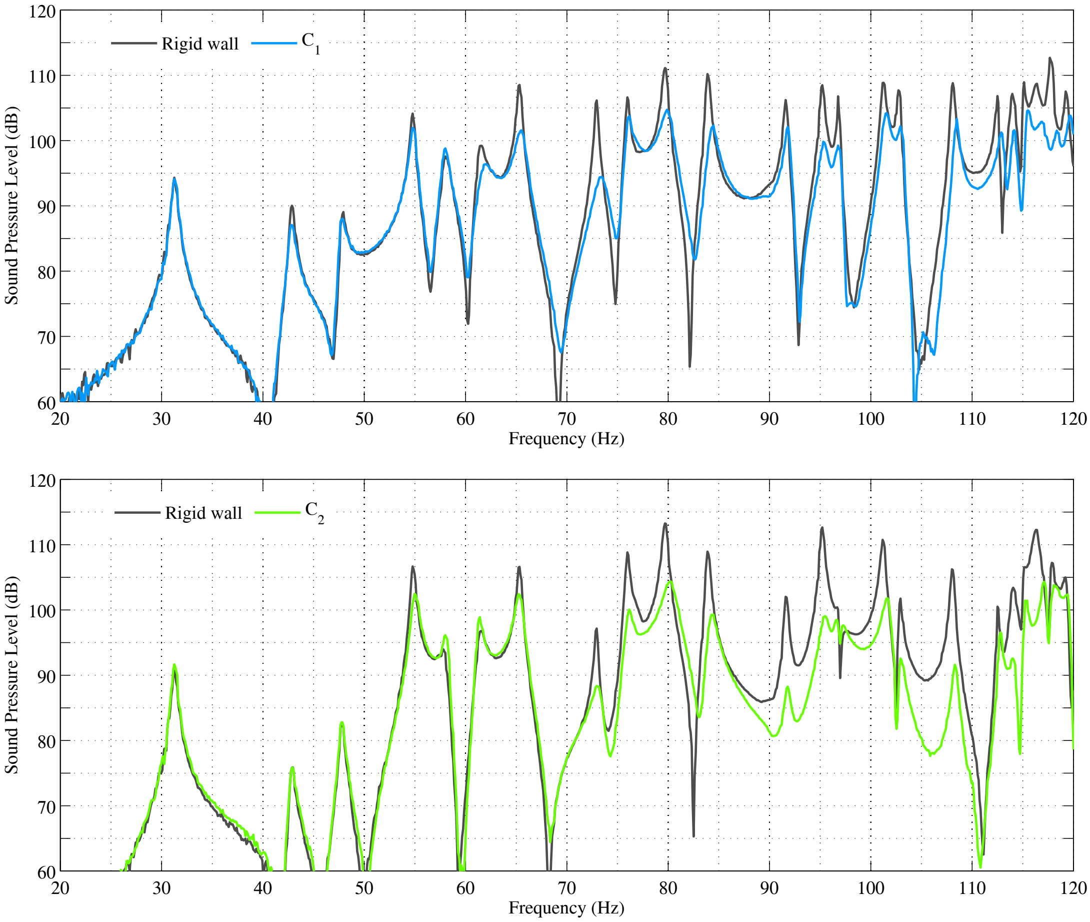

Figure 6: Measured sound pressure levels: in corner 8 with configuration C_1 (up) and in corner 2 with configuration C_2 (down)

methodology can be viewed as a complementary way to improve sound quality in listening spaces beyond what is attainable using a conventional soundproofing treatments. This study was mainly restricted to a frequency range between 70 Hz and 100 Hz. In order to address another frequency range, in the case of a larger room or smaller, the transducers should be changed to suit requirements but the expected results would be similar.

Acknowledgments

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