PRACTICAL ACTIVE AND SEMI-ACTIVE STRATEGIES FOR THE CONTROL OF ROOM ACOUSTICS IN THE LOW FREQUENCY RANGE

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Abstract
Modal acoustics is one of the main sources of annoyance in the low frequency range in rooms. Active noise control techniques can be deployed to reach sufficient acoustic benefit, but limited to a single frequency, whereas broader efficiency would require complex systems with a large number of transducers. This paper introduces the semi-active/active concept of "electroacoustic absorbers", which are able to handle low-frequency room acoustics problems along significant frequency bandwidth, in the experimental context of the damping of the first modal frequencies of a test chamber. This study will focus on highlighting practical advantages of such techniques when compared to active strategies (performances, stability, complexity, integration, cost,…).

Keywords: electroacoustic absorbers, shunt loudspeakers, active acoustic impedance control, modal room acoustics.

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]. 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 semi-active aqnd
active use, allowing tuneable acoustic performances. This technique, denoted
“electroacoustic absorber”, is based on an electromechanical transducer, the electrical
terminals being loaded with a passive electrical circuit so that to obtain an optimal damping
around its resonance frequency. A general presentation of the technique will be provided,
together with numerical simulation of the acoustic behaviour in standard configurations
(impedance tube), and also experimental assessments with a semi-active concept (shunt
loudspeaker) 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. Several measurements have been performed near the Geneva Airport. Figure 1
illustrates the modal behavior 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).](image)

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 semi-active and active strategies described in the following.

3 Electroacoustic absorbers concept

3.1 General presentation

The electroacoustic absorber concept is based on the shunt loudspeakers techniques, consisting in loading the electrical terminals of an electroacoustic transducer with a dedicated electric load, so that to ascribe certain impedance to the voicing face. Thus, the damping of specific acoustic modes in a cavity, and more specifically in a room, can be addressed by correctly tuning the electroacoustic absorber so that its acoustic impedance matches the impedance of the air.

In the following, we will consider the specific case of a shunt moving-coil loudspeaker in a closed-box, the voicing face of which is radiating in a waveguide of adapted cross-section. The following discussions will focus on the different shunt possibilities at the electric terminals of the transducer, namely a passive dipole, a negative resistance (or motional feedback disposal), or a combined pressure/velocity feedback active impedance control. The system as a whole (the electroacoustic transducer and its electric load) is referred to as the electroacoustic absorber (EA).

An electrodynamic loudspeaker is a linear system that can be described with differential equations. From Newton’s law of motion, the mechanical dynamics, below the first modal frequency of the diaphragm, can be modeled with the following second-order differential equation:

\[ Sp_i(t) = -M_{ms} \ddot{v}(t) - R_{ms} v(t) - \frac{1}{C_{ms}} \int v(t) dt - (Bl)i(t) \]

where \( M_{ms}, R_{ms} \) and \( C_{ms} \) are the equivalent mass, mechanical resistance and compliance of the moving bodies of the loudspeaker, \( Bl \) the force factor, \( v(t) \) the diaphragm velocity (opposed to total air velocity), \( i(t) \) the driving current, \( S \) the effective piston area, and
the total acoustic pressure acting on the loudspeaker diaphragm. This pressure accounts both for the exogenous sound source \( p_s(t) \) and the self-radiated pressure \( p_r(t) \) of the loudspeaker diaphragm, due to the diaphragm’s velocity. This self-radiated pressure can be expressed in most cases as:

\[
Sp_r(t) = M_{mr} \dot{v}(t) + R_{mr} v(t) + \frac{1}{C_{mr}} \int v(t) dt
\]

(2)

where \( M_{mr}, R_{mr} \) and \( C_{mr} \) represent the radiated mass, mechanical resistance and compliance of the loudspeaker voicing face [2]. This self-radiated pressure accounts for both front and rear faces. Combining Eq. (1) and (2) yields to the generalized Newton’s law for mechano-acoustic transducers:

\[
Sp(t) = -M_{meq} \dot{v}(t) - R_{meq} v(t) - \frac{1}{C_{meq}} \int v(t) dt - Bl i(t)
\]

(3)

where

\[
M_{meq} = M_{ms} + M_{mr}, \quad R_{meq} = R_{ms} + R_{mr}, \quad C_{meq} = \frac{C_{ms} C_{mr}}{C_{ms} + C_{mr}}
\]

(4)

Eq. (3) being formally identical to Eq. (1), we will arbitrarily use the denotations of Eq. (1) in the following, for ease of understanding, assuming the total radiation impedance is embedded within the \textit{in vacuo} impedance of the loudspeaker. We will also denote \( p \) the acoustic pressure at the voicing face of the loudspeaker.

The electrical dynamics can also be modeled by a first order differential equation given as:

\[
u(t) = R_e i(t) + L_e \frac{di(t)}{dt} - (Bl) v(t),
\]

(5)

where \( u(t) \) is the voltage applied at the electrical terminals, \( R_e \) and \( L_e \) the dc resistance and the inductance of the voice coil, and \( (Bl) v(t) \) the back emf induced by the motion of the voice coil within the magnetic field. Eq. (1) and (5) form a coupled set of differential equations describing the loudspeaker system. Expressing the preceding relationships with the use of Laplace transform yields the characteristic equations of the generalized electrodynamic loudspeaker, given as:

\[
\begin{align*}
SP(s) &= -\left(s \cdot M_{ms} + R_{ms} + \frac{1}{s C_{ms}}\right) V(s) - (Bl) I(s) \\
U(s) &= -(Bl) V(s) + (R_e + s \cdot L_e) I(s)
\end{align*}
\]

(6)

where \( P(s), V(s), U(s) \) and \( I(s) \) are the Laplace transforms of \( p, v, u \) and \( i \).

**3.2 Acoustic absorption capability of the voicing face**

It is always possible to derive Eq. (6) in order to write the normalized acoustic admittance of the loudspeaker’ voicing face, whatever the load at its electrical terminals, as:

\[
Y(s) = -\rho c \frac{V(s)}{P(s)},
\]

(7)

where \( \rho \) is the density of the medium and \( c \) is the celerity of sound in the medium, as a function of the electric voltage \( U(s) \) and current \( I(s) \). The minus sign is justified by the fact that \( V(s) \) is defined as the diaphragm’s velocity, opposed to the total particle velocity at the
diaphragm. The corresponding reflection coefficient, representing the ratio of the reflected and incident sound pressures, can be derived after:

\[ R(s) = \frac{1 - Y(s)}{1 + Y(s)}. \]  

(8)

Then, the extraction of the magnitude \( |R| \) of \( R(s) \) yields the frequency-dependant absorption coefficient \( \alpha(f) \):

\[ \alpha(f) = 1 - |R(f)|^2; \]  

(9)

Eq (9) indicates that the choice of the electric load, namely the transfer functions \( H(s) \) between electric voltage \( U(s) \) and current \( I(s) \) in the following, imposes certain absorption characteristics at the voicing face of the transducer. The denotation “electroacoustic absorber” is then legitimized when this transfer function \( H(s) \) is tailored in such a way as to exhibit positive values of the acoustic absorption coefficient along the frequency bandwidth of interest. It is also possible to obtain negative values of the absorption coefficient [3, 4], these particular cases being out of the scope of this presentation.

### 3.3 Electroacoustic absorbers formulation

In order to describe a general case, we will consider the voltage at the terminals of the loudspeaker, as the combination of:

1) a feedback voltage proportional to diaphragm’s velocity,
2) a feedback voltage proportional to sound pressure at the diaphragm,
3) the voltage applied at the source resistance.

The total input voltage provided by the electric source is then:

\[ U(s) = U_v(s) - R_v I(s) = \Gamma_v V(s) + \Gamma_p P(s) - R_v I(s), \]  

(10)

where \( \Gamma_v \) and \( \Gamma_p \) represent the feedback gains (respectively in V.m\(^{-1}\).s and V.Pa\(^{-1}\)), including the sensors’ sensitivities.

### 3.3.1 Acoustic admittance and absorption capabilities

By replacing voltage \( U(s) \) in Eq. (6) with the expression of Eq. (10), including the voltage at the source resistance \( R_v \), yields the controlled normalized acoustic admittance, which can be expressed, after simplifications (when considering the electroacoustic absorber behavior around its natural resonance frequency and below the cut-off frequency of the electrical filter induced at its terminal, so that electric inductance can almost be neglected), as:

\[
Y(s) = Z_{mv} \cdot \frac{s}{s^2 M_{mEA} + s R_{mEA} + \frac{1}{C_{mEA}}}, \quad \text{with}
\]

\[
\begin{align*}
M_{mEA} & \approx M_{mv} \cdot \frac{(R_v + R_y)}{(R_v + R_y) + \Gamma_p \frac{Bl}{S}} \\
R_{mEA} & \approx \frac{\left[ (R_v + R_y) R_{mv} \right] \left( \frac{L_v}{C_{mv}} - Bl + \Gamma_v \right)}{(R_v + R_y) + \frac{\Gamma_p Bl}{S}} \\
C_{mEA} & = C_{mv} \left[ 1 + \frac{\Gamma_p Bl}{S(R_v + R_y)} \right]
\end{align*}
\]  

(11)
where $Z_{mc} = \rho c S$ is the mechanical equivalent to characteristic medium impedance $Z_c = \rho c$. This result legitimates the denomination "electroacoustic absorbers", since the normalized admittance of Equation (11) exhibits a characteristic resonant behavior, the component of which depend on the electroacoustic transducer and its electrical conditioning. We will show in the following that this conditioning can always be considered as an electrical load.

### 3.3.2 Equivalent electric load

Moreover, it derives from Eq. (6) and Eq. (10) that both velocity $V(s)$ and sound pressure $P(s)$ can be expressed as functions of intensity $I(s)$ such that:

$$U(s) = \Gamma_v V(s) + \Gamma_p P(s) - R_s I(s) = -H(s) I(s),$$

where $H(s)$ represents the equivalent electric load of the total control feedback voltage against current intensity. In the general case, this electric transfer function is (after simplifications):

$$H(s) = -R_s \left[ \frac{s^2 M_{ms} + s \left( R_{ms} + \frac{B l^2}{R_s} \right) - \frac{S}{\Gamma_p + \frac{B l S}{R_s}} + \frac{1}{C_{ms}} \right] \right].$$

(13)

This result indicates that feedback-based active acoustic impedance control can be replaced by an equivalent electric load, accounting for both pressure and velocity sensing. The design of such electrical impedance will be further developed in a future communication.

### 4 Case studies

#### 4.1 Shunt loudspeaker

Let’s consider the simplest case of electroacoustic absorber, also called shunt loudspeaker, where a positive resistor $R_s$ loads the electric terminals of the loudspeaker. Then, the normalized acoustic admittance can thus be expressed as follows:

$$Y(s) = Z_{mc} \cdot \frac{s}{s^2 M_{ms} + s \left( R_{ms} + \frac{B l^2}{R_s + R_s} \right) + \frac{1}{C_{ms}}}.$$  

(14)

In this expression, the mechanical resistance of the transducer can be enhanced so that the equivalent resistance $R_{ms} \rightarrow R_{ms} + \frac{B l^2}{R_s + R_s}$ expresses the total losses in the electroacoustic absorber. This said, it is now obvious that the absorption coefficient at the resonance frequency of the transducer can be easily varied, so as to take different values from $R_{ms}$ up to infinity. An optimal shunt value can be set so as to have perfect acoustic absorption at the transducer’s resonance. The corresponding resistance can be expressed as
\[ R_{\text{opt}} = \frac{B l^2}{Z_{mc} - R_{ms}} - R_v. \]  

(15)

### 4.2 Active acoustic impedance control

Let’s now consider the case where a combination of pressure and velocity feedback is performed, so that to result in the normalized acoustic admittance of Eq. (11). It is demonstrated that such normalized admittance can also be expressed similarly to Equation (15), replacing \( R_v \) with the equivalent electrical load of Equation (14):

\[ Y(s) = Z_{mc} \cdot \frac{s}{s^2 M_{ms} + s \left( R_{ms} + \frac{B l^2}{R_v + H(s)} \right) + \frac{1}{C_{ms}}}. \]  

(16)

This result confirms the formal analogies between semi-active concept such as shunt loudspeakers, and more complex active acoustic impedance control strategies, which can always be represented as “active” shunt at the loudspeaker electrical terminals.

### 5 Assessment of acoustic performances

#### 5.1 Assessment under normal incidence plane waves in an impedance tube

A Monacor SPH 300-TC low-range loudspeaker has been chosen for simulations and experimental assessment in an impedance tube. The acoustic absorption coefficient of such loudspeaker has been processed out of analytical expressions in Matlab®, for different case of control:

- shunt loudspeaker with an optimal electrical resistance (\( R_v = 4.7 \, \Omega \)) at the electrical terminals,
- and active acoustic impedance control with actual feedback on velocity and pressure (feedback gains \( \Gamma_v = 74 \, \text{V.s.m}^{-1} \) and \( \Gamma_p = 0.2 \, \text{V.s.m}^{-1} \)).

The experimental assessment is based on the ISO-10534-2 standard, with the two microphones technique [5].

![Figure 3: simulation and experimental setup for the assessment of acoustic performances of the electroacoustic absorber (here for an active impedance control)](image)

The results of simulations and experiments are given in Figure 4, for the two cases of control:
It is obvious that this technique practically allows an optimal damping of certain resonances in closed spaces, even with the semi-active case (namely the shunt loudspeaker), in a 1D configuration. The comparison between the two cases shows that the active case is capable of enhancing the absorption capabilities of the electroacoustic absorber, for which impedance matching is performed over more than 2 octaves between 20 Hz and 100 Hz, which is sufficient for significantly damping the whole modal content of a standard room. The next section will show first results obtained in a reverberant chamber, with only the semi-active case (the active case is under study at the present time).

5.2 Assessment in a reverberant chamber

In order to perform experimental assessment within the facilities of the Laboratory of Electromagnetics and Acoustics at EPFL, we have chosen the reverberant chamber, with a volume of about 200 m³. Figure 5 illustrates the experimental setup, including an array of 10 Monacor SPH-300 TC loudspeakers, in individual closed boxes of 50 l, and with an optimal electric shunt of \( R_s = 4.7 \Omega \). Each loudspeaker has been previously modified so that to present a resonance frequency of 34.9 Hz, corresponding to the room’s mode (1,1,0) to be damped.

Figure 5: the reverberant chamber at LEMA (left: CAD model in Comsol Multiphysics®, right: photograph taken from the entrance)
5.2.1 Experimental setup
The source that has been used for exciting the room is a home-made bass-reflex loudspeaker system, placed at one corner of the room, designed to provide the requested acoustic power in the requested frequency bandwidth, and 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 provides better results than other excitation as explained in a prior study [6]). A Brüel & Kjær Type 4165 electret microphone (sensitivity 50 mV/Pa) senses the acoustic pressure at one corner of the room (corner #4 in the illustration).

5.2.2 Experimental results
The results of assessment of the influence of the array of electroacoustic absorbers (shunt loudspeakers) on the identified mode at 34.9 Hz is given in Figure 6, and compared to the same measurement with no absorber (“hardwalls”).

![Graph showing sound pressure levels](image)

Figure 6: Stationary 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)

The array of 10 shunt loudspeakers is then capable of damping the chosen mode of the reverberant chamber, with an acoustic attenuation of 14 dB. This results highlights that even semi-active concepts are capable of performing such modal control. Further assessment with active loads are about to be performed to assess the same acoustic capabilities than those being assessed in a 1D configuration (see 5.1).

6 Conclusions
It has been proven that electroacoustic absorbers are 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. The next steps will be the assessment of active loads for broadening the modal damping capabilities. 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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References


