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Assessment of different low-frequency soundproofing systems for room acoustics

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ABSTRACT

Low frequency background noise (below 200 Hz) is perceptible in many situations such as in room acoustics or industrials issues. Generate by numerous type of source (traffic noise, aircraft, industrial plants, electricity transformer,…), this type of noise is under evaluate by regulations laws due to the used of the A-weighted decibel (dB(A)). In spite of that low frequency background noise can reach high levels which produce poor speech intelligibility and annoyance feeling. Unfortunately traditional acoustic treatments have little impact to decrease this kind of disturbance. New approaches have to be developed to give new efficient techniques in order to overcome the problem. Three noise cancellation systems dedicated to low frequencies are studied here. The first one is based on an electromechanical transducers loaded passively to get an optimal damping around the resonance frequency of the disposal. In the second system the passive load is substituted by an active control to enhance acoustic properties of device. The last strategy consists on a control of the first modal frequencies of a room to decrease the low frequency background noise level. This paper aims at presenting those three noise cancelation systems. Their advantages and drawbacks will also be discussed.

1 INTRODUCTION

Today, noise is an ever increasing cause of concern. About 80 million people in European Union are exposed to noise levels considered unacceptable. Health, worker productivity or comfort at home are directly impact by this noise. The cost of the associated damage is estimated at about 12 billion euro per year.

In this context, noise levels are more and more severely regulated by laws which decreased acceptable thresholds. Unfortunately regulation laws are based on the A-weighted decibel (dB(A)) for indicator which obviously decreased the effect of low frequencies (below 200 Hz). In the same time acoustic treatments are efficient for high and middle frequencies range but present little performance for low background noise. However, low frequency noise has a great impact on the acoustic quality of a room producing poor speech intelligibility and annoyance feeling.

This paper aims at presenting three systems dedicated to reduce low frequencies noise. The first one is based on an electromechanical transducers loaded passively to get an optimal damping around the resonance frequency of the disposal. In the second system the passive load is substituted by an active control to enhance acoustic properties of device. The last strategy consists on a control of the first modal frequencies of a room to decrease the low frequency background noise level.

A brief presentation of each disposal is made, followed by some results. The advantages and drawbacks of each technique are discussed in the last part.

2 ELECTROMECHANICAL PASSIVE DAMPING SYSTEM

2.1 Description

The overall behavior of the loudspeaker, seen as a resonator, can be described as represented in Figure 1.

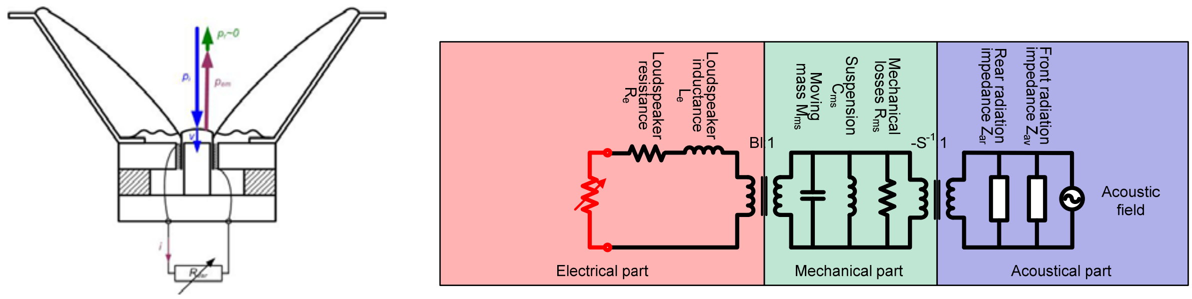


Figure 1: shunt electrodynamic loudspeaker (left) and it lumped-elements model (right) [1]

The mechanical part of the loudspeaker can be described by a moving mass M_{ms} (paper cone + moving coil), a compliance C_{ms} (annular suspension + spider) and a mechanical resistance R_{ms} (friction loss). Two different types of transduction occur within the loudspeaker: the electro-mechanical transduction with the force factor Bl, where B is the magnetic induction and l the electrical conductor's length, and the mechanic-acoustic transduction with a diaphragm apparent surface S.

The study of the properties of a loudspeaker can be done by using equivalent model method (Figure 2) [1]. The acoustical equivalent parameters of the loudspeaker are easily deduced from the mechanical ones using the equivalent acoustical mass M_{as}, equivalent acoustical compliance C_{as} and the equivalent acoustical resistance R_{as}:

$$\begin{aligned} M_{as} &= M_{ms} / S^2 \\ C_{as} &= C_{ms} \cdot S^2 \\ R_{as} &= R_{ms} / S^2 \end{aligned} \tag{1}$$

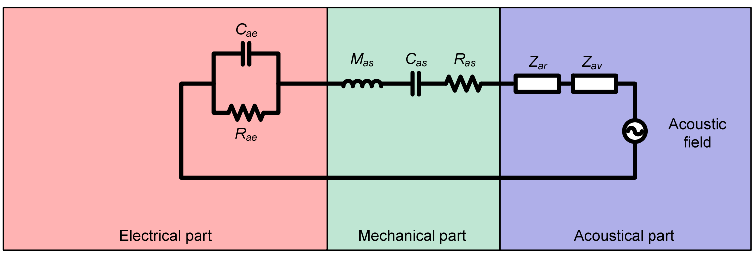


Figure 2: equivalent acoustic model of an electrodynamic loudspeaker [1]

The electric side of the transducer is loaded by a shunt resistor R_c. At low frequencies (i.e. C_{ae} in the equivalent acoustic model) can be neglected. R'_e is equals the global resistive value of the transducer comprising the internal resistance R_e of the coil and the eventual load R_c. The equivalent acoustic load can be written:

$$R_{ae} = \frac{1}{S^2} \cdot \frac{(Bl)^2}{R'_e} \qquad \text{with } R'_e = R_e + R_c \tag{2}$$

The electric load is seen on the acoustic side as an acoustic conductance. Thus, the lower the electrical load value, the higher the acoustic losses.

The acoustic impedance $\underline{Z}'_{as}$ presented by the loudspeaker in regards of the exogenous acoustic field is:

$$\begin{aligned} \underline{Z}'_{as}(\omega) &= (R_{as} + R_{ae}) + j\omega M_{as} + \frac{1}{j\omega C_{as}} \\ &= R'_{as} + j\omega M_{as} + \frac{1}{j\omega C_{as}} \end{aligned} \tag{3}$$

where ω is the pulsation of the acoustic excitation.

At the loudspeaker resonance the imaginary part of Z'_{as} is equal to 0. The apparent acoustic resistance R'_{as} of the loudspeaker can be easily modified by varying the electric load so that it reaches an optimum value. In particular, there is a specific value of R'_e such as R_{as} + R_{ae} equals the so-called characteristic acoustic impedance of the medium $Z_{ac}=\rho c/S$, where ρ and c are respectively the density and the sound speed in air. The corresponding value of the total electric load is then:

$$R'_{e0} = \frac{(Bl)^2}{\rho c / S - R_{as}} \tag{4}$$

This can be obtained either by varying the value of an electric load at the loudspeaker electrical input, or by shunting the loudspeaker and modifying the length of the electrical conductor of the coil so that its internal resistance reaches the optimum value of equation (4). This solution, even if smarter, has implications on the moving mass M_{ms}, and by way of consequence on the resonance frequency of the loudspeaker, as well as on force factor Bl. With the electrical resistance value verifying equation (4), the loudspeaker voicing face becomes totally absorbent in regards with the impinging acoustic propagation (the diaphragm is seen as if it was the air, with the same impedance than air, and becomes totally "transparent" acoustically).

2.2 Results

The normalized acoustic admittance $\underline{y}$ is defined as the ratio of the characteristic impedance of the medium $Z_{ac}=\rho c/S$ over the acoustic impedance $\underline{Z}'_{as}$ of the loudspeaker. Using equation (2) and equation (3) $\underline{y}$ can be written as follows:

$$\underline{y}(\omega) = \frac{\rho c}{S\left[\left(R_{as} + \frac{1}{S^2} \cdot \frac{(Bl)^2}{R'_e}\right) + j\omega M_{as} + \frac{1}{j\omega C_{as}}\right]} \tag{5}$$

The acoustic absorption coefficient α of the electroacoustic absorber, ratio of the acoustic energy dissipated by the absorber over the incident energy is given by:

$$\alpha(\omega) = 1 - \left| \frac{1 - \underline{y}(\omega)}{1 + \underline{y}(\omega)} \right|^2 \tag{6}$$

Comparison of computational and measurement results get with a standard Medium-range loudspeaker AUDAX HT210F0 are given in Figure 3 for 2 electrical loads (in green: optimal shunt; in blue: open circuit). Experimental results have been assessed with respect to standard ISO 10534-2 ([2], [3]) in an impedance tube. Measurement of absorption coefficient α versus frequency is represented in Figure 4 for a specific loudspeaker with seven different electrical loads.

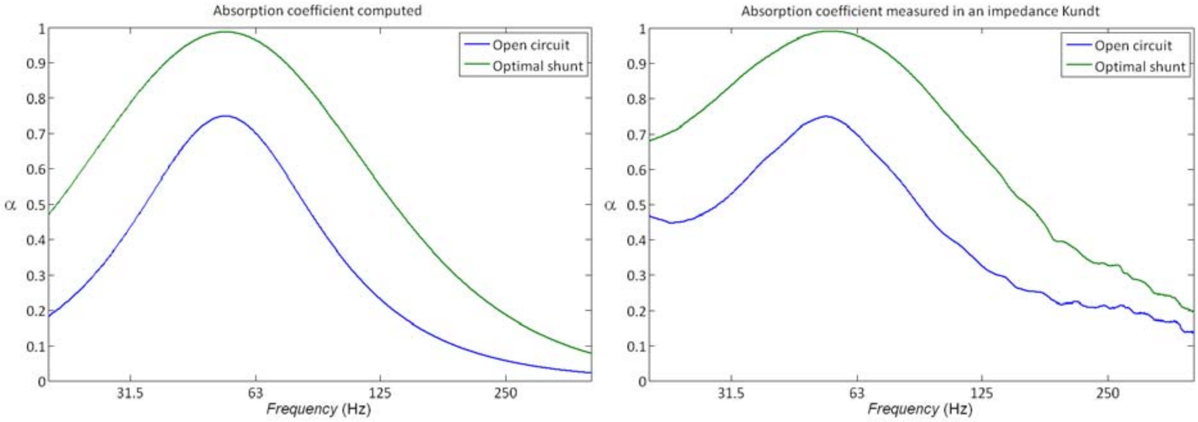


Figure 3: Absorption coefficient - left: simulations, right: measurements

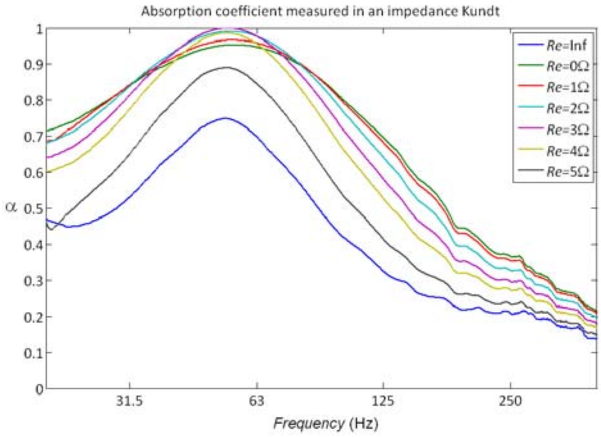


Figure 4: Variation of α versus frequency for a specific loudspeaker with several loads

The agreement between calculations and measurements is good. An optimal shunt can be found which increases the absorption of the loudspeaker at it resonance. It can be seen that the loudspeaker can be controlled to be totally absorbent over a certain frequency bandwidth around its resonance frequency ($\alpha > 0.9$ over one octave in our case).

3 ELECTROMECHANICAL ACTIVE DAMPING SYSTEM

3.1 Description

Electromechanical active damping system consists of a loudspeaker loaded by a dedicated control disposal on their electric side (Figure 5).

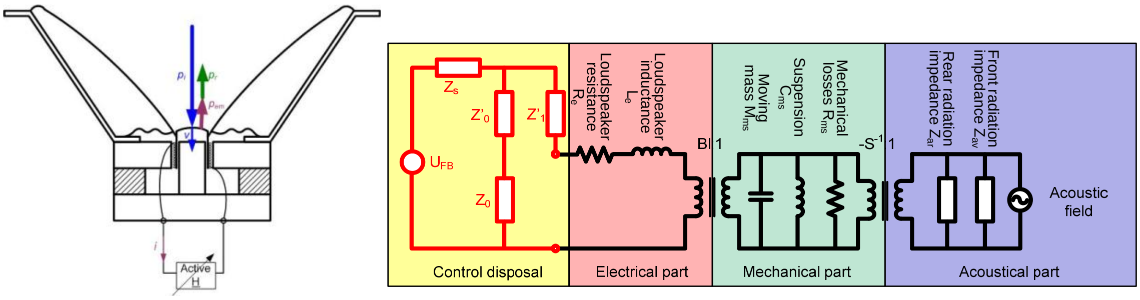


Figure 5: Electrical circuit equivalent to motional feedback.

This active impedance control is based on a combined pressure *p* - volume velocity *v* caption at the transducer's diaphragm. The pressure is measured with a microphone (sensitivity $\sigma_p(\omega)$ [V/Pa]). The velocity is determined with a resistance bridge disposal which the differential tension (sensitivity $\sigma_v(\omega)$ [$V/(m.s^{-1})$]) is proportional to the transducer's volume velocity [4]. This can be obtained for $Z_s = 0$ if

$$Z_0 \cdot Z'_1 = Z_e \cdot Z'_0 \tag{7}$$

with

$$Z_0 = Z'_0 \quad \text{and} \quad Z'_1 = \underline{Z}_e(\omega) = R_e + j\omega L_e \tag{8}$$

These two feedback signals, i.e. $\underline{U}_p$ for the pressure and $\underline{U}_v$ for the velocity, are the inputs of a double feedback disposal. They are then linearly combined using gains Γ_p et Γ_v to provide *feedback voltage* $\underline{U}_{FB}$ which control the transducer motion:

$$\underline{U}_{FB} = \Gamma_p \cdot \underline{U}_p + \Gamma_v \cdot \underline{U}_v \tag{9}$$

The loudspeaker can be represented by a two-port network where *P* is the pressure, *Q* the volume flow, *U* the voltage and *I* the current in the coil [5]:

$$\begin{bmatrix} U \\ I \end{bmatrix} = \begin{bmatrix} A & B \\ C & D \end{bmatrix} \cdot \begin{bmatrix} P \\ Q \end{bmatrix} = \begin{bmatrix} -Z_e \cdot \dfrac{S}{Bl} & -\dfrac{Bl}{S} - \dfrac{Z_e \cdot Z_{ms}}{S \cdot Bl} \\ -\dfrac{S}{Bl} & -\dfrac{Z_{ms}}{S \cdot Bl} \end{bmatrix} \cdot \begin{bmatrix} P \\ Q \end{bmatrix} \tag{10}$$

with Z_{ms} the mechanical impedance of the loudspeaker.

The acoustic impedance of the transducer $\underline{Z}'_{as}$ presented by the loudspeaker loaded by his control disposal in regards of the exogenous acoustic field can be now expressed by (if $\Gamma_v >> 2$) :

$$
\underline{Z}'_{as}(\omega) = -S \cdot \frac{P}{Q} = \frac{\underline{Z}_{ms}(\omega).\underline{Z}_e(\omega) + [\Gamma_v \sigma_v + (Bl)](Bl)}{S\underline{Z}_e(\omega) + \Gamma_p \sigma_p (Bl)}
\approx \frac{\underline{Z}_{ms}(\omega) + [\Gamma_v \sigma_v + (Bl)](Bl / R_e)}{S + \Gamma_p \sigma_p (Bl / R_e)} \tag{11}
$$

Except parameters Γ_p et Γ_v all others parameters of equation (11) are systems constant. This equation shows that the impedance at the loudspeaker voicing face, around it resonance, is function of those 2 gains. Thus, a good tuning of the motional feed-back allows getting wanted impedances. The loudspeaker voicing face can become totally absorbent (the diaphragm is seen as if it was the air, with the same impedance than air, and becomes totally "transparent" acoustically), or can become totally reflective (the diaphragm is seen as if it was a rigid perfect wall, and reflects totally the acoustic wave) in regards with the impinging acoustic propagation.

In the case where each feedback gain is equal to 0, the equation (5) becomes equivalent to equation (2) with an open electric circuit.

3.2 Results

The normalized acoustic admittance can be now expressed as follow:

$$
\underline{y}(\omega) = \frac{Z'_{as}}{\underline{Z}'_{ac}(\omega)} = \frac{\rho c}{S} \frac{S + \Gamma_p \sigma_p(\omega)(Bl) / \underline{Z}_e(\omega)}{\underline{Z}_{ms}(\omega) + [\Gamma_v \sigma_v(\omega) + (Bl)](Bl) / \underline{Z}_e(\omega)} \tag{12}
$$

The acoustic absorption coefficient α of the electroacoustic absorber is defined using equation (6). Comparison of computational and measurement results obtain with a standard Medium-range loudspeaker AUDAX HT210F0 are given in Figure 6 (in green: feedback control on; in blue: open circuit). The experimental results, have been assessed with respect to standard ISO 10534-2 ([2], [3]) in an impedance tube. Computed absorption coefficient α versus frequency is represented in Figure 7 for a specific loudspeaker with five different settings of the control.

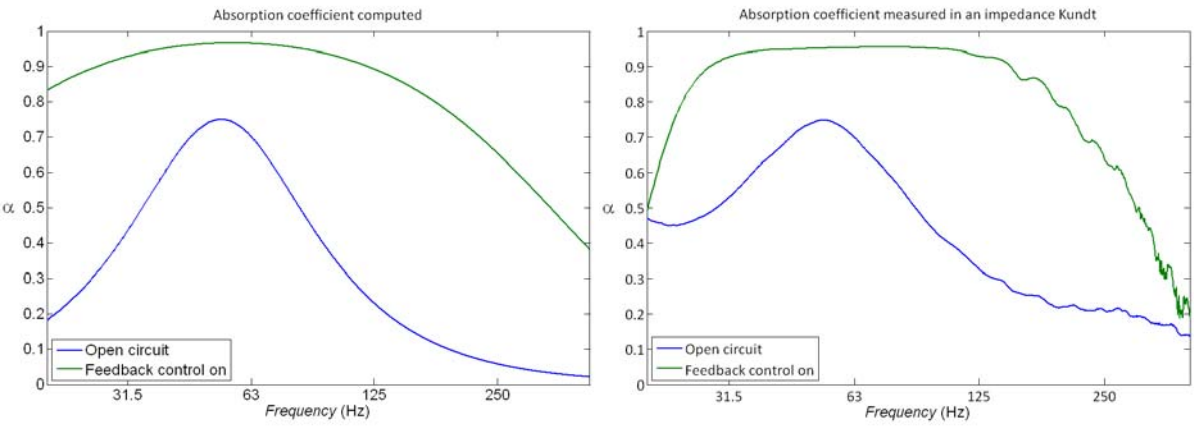


Figure 6: Absorption coefficient - left: simulations, right: measurements

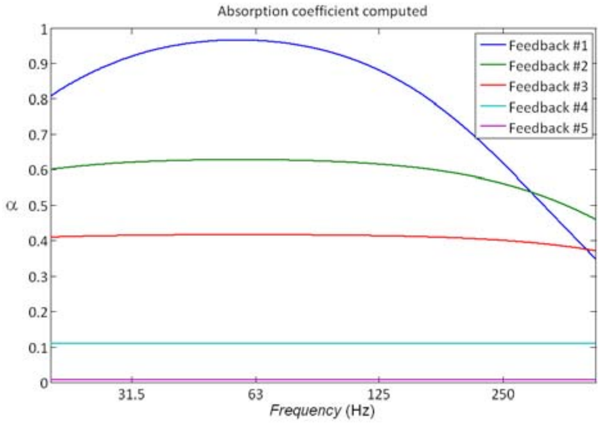


Figure 7: Variation of α versus frequency for a several feedback setting.

We can see in Figure 6 that results from simulations and measurements are very close. An absorption coefficient bigger than 0.9 can be reached over 3 octave bands around the loudspeaker resonance. The use of an amplifier with a cut off at 20 Hz for the control loops explains the differences observed at low frequencies between simulations and measurements results. Figure 7 shows than the active material is tunable. It can be a good absorber or to become totally reflective depending on the setting of the control loop.

4 ACTIVE MODAL CONTROL

4.1 Description

At low frequencies, the room acoustic response is driven by the modal behavior which is characterized by high levels at eigenfrequencies that can be very annoying inside rooms [6]. The aim of active modal control is to reduce the noise inside the rooms at these specific low frequencies by using the modal behavior itself.
Several measurements have been performed near the Geneva Airport. Figure 8 illustrate the modal behavior in an office, located under the take off path of airplanes.

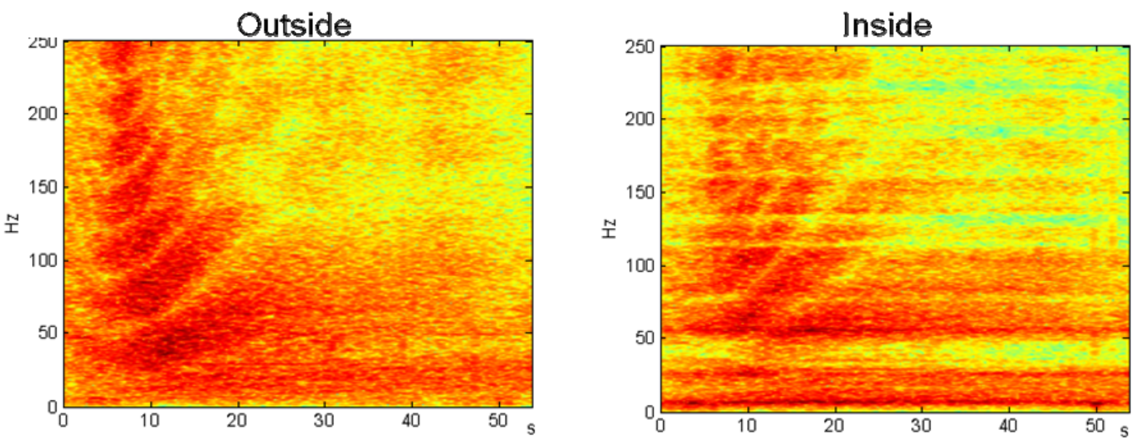


Figure 8: Time frequency response of the noise of a plane taking off – left: in front of the window, right:inside a room inside the building.

The comparison between the noise outside and inside the room shows that the noise spectrum inside the room is concentrated on modal frequencies. To reduce the annoyance, the sound pressure level has to be decrease at these specific frequencies.

If the modal response of the room when excited with a loudspeaker is compared with the large band low frequency noise excitation of an air plane, it becomes obvious that the main energy concentrates on eigenfrequencies in the room. In Figure 9, the modal response of the room shows the eigenfrequencies of the room. The difference between outside and inside noise is very low especially at modal frequencies. At the main annoying frequency, that is to say at 54 Hz, the noise level inside the room is higher than outside the building. The former example explains why it is a necessity to reduce the noise level at modal frequencies. Since the passive solutions are very often bulky with no versatility – one specific design for one frequency- modal control is a very interesting alternative. It has been tested in several configurations with only one microphone and one loudspeaker.

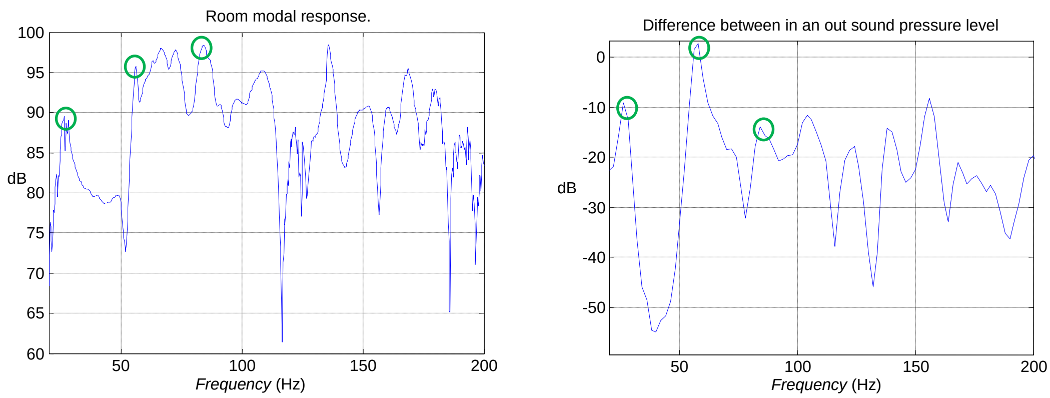


Figure 9: Frequency response of the room - left: excited with a pink noise, right: difference between outside and inside noise during an airplane flying over, the circles represent the eigenfrequencies of the room.

One of the main challenges of the modal control is the question of the reference microphone. The control should use a reference microphone and an error microphone. In many cases, it would be convenient to use only one microphone inside the room. This is possible for a stationary noise. For non-stationary noise, an inside reference microphone would provide too late the information on the noise to control since the modal time behavior is long. It appears very difficult to perform modal control for time varying noise source with only an inside microphone due to the time behavior of modes.

A high sound pressure level at a modal frequency means that the quality factor of the corresponding eigenmode is high. In other words the damping factor is small and consequently the modal time behavior is slow. Figure 10 illustrates the time behavior when a stationary source is turned on at a modal frequency in a room and then turned off. The transient and free sequences duration are fully determined by the damping factor of the mode. This damping factor can be measured or roughly estimated from the wall impedance and the furniture equipment.

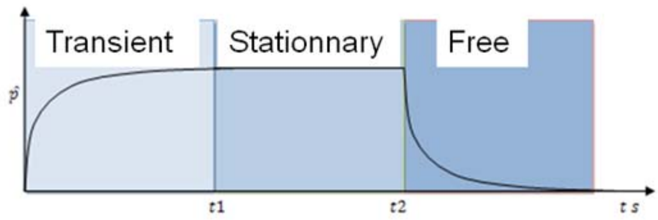


Figure 10: Time behavior in the reverberant room when a stationary source is turned on (t=0) and off (t=t2) at a modal frequency.

Since the stationary noise sources of the following examples are not fully known, the active modal control has been performed only with an inside microphone.

4.2 Results

The algorithm used by the controller is based on the forecast of the modal behavior of the room for a given measured excitation and a given control source. Before the control, the modal parameters of the room (i.e. eigenfrequencies and the corresponding damping factors) are automatically estimated by the control disposal. Then the controller automatically select the modal frequencies, it has to work on to reduce the most annoying noise.

The modal active noise control have been carried out in the 280 m3 reverberant room of the Laboratoire Electromagnétisme et d'Acoustique (LEMA) of EPFL for which the modal behavior is very strong below 60 Hz. The first audible eigenfrequency is at 35 Hz and corresponds to the (1,1,0) eigenmode. The control is performed at this specific frequency with a primary source and a control source in two opposite corners of the room. Figure 11 shows that an abatement of 42 dB can be reach at this first audible eigenfrequency. It appears that the control establishment time is fully determined by the damping factor of the mode.

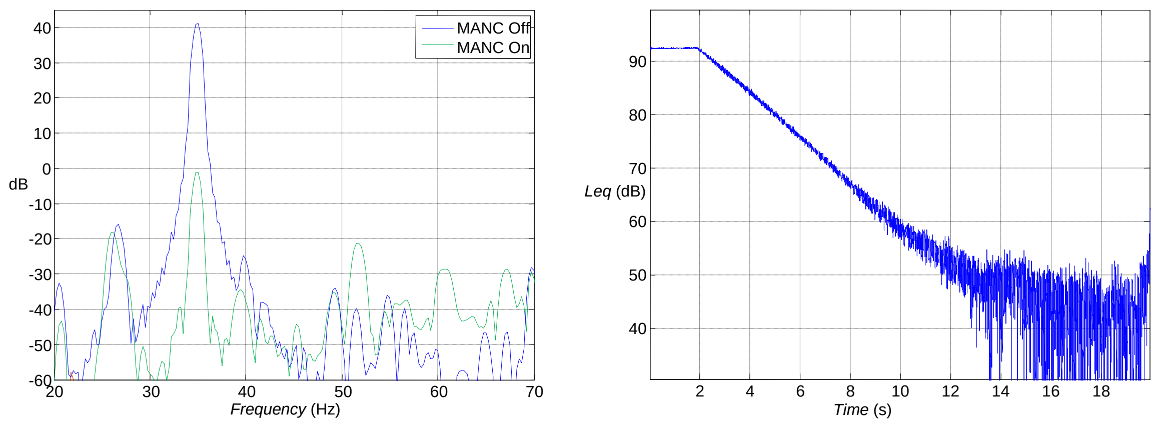
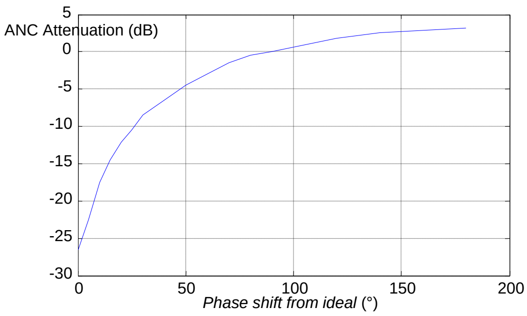


Figure 11: Frequency response without and with control (left), and time behavior when the control is turn on at *t=2s* (right) in the reverberant room.

The control has also been tested in a real case: a bedroom exposed to a high 50 Hz noise from inside building machinery. One of the main difficulties is to get the best amplitude and phase shift between the secondary source and the noise source. Figure 12 shows the performances of the modal control in function of the phase and the amplitude. For optimal values the noise reduction reaches 26 dB at the head of the bed (Figure 13).



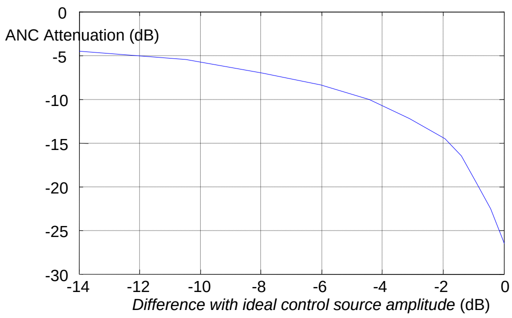
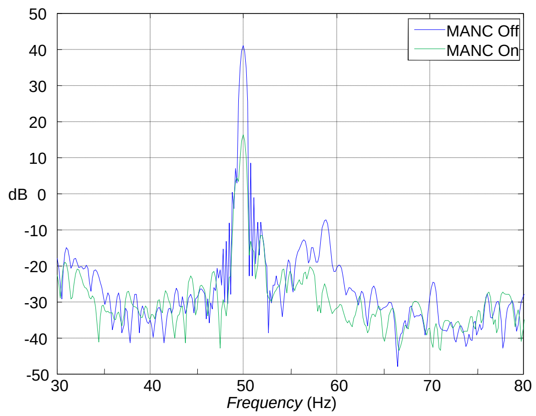


Figure 12: Modal control efficiency versus phase shift from the best value (left), versus amplitude difference of the control source with the best value (right).



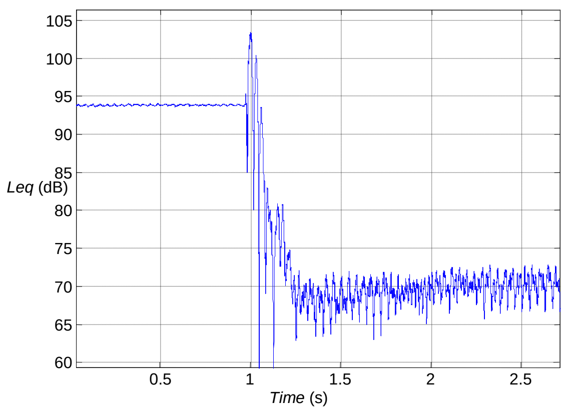


Figure 13: Frequency response without and with control (left), and time behavior when the control is turn on (right) in a bedroom.

Modal control is very efficient and easy to implement solution to reduce low frequency noise in rooms. The time behavior of eigenmodes can be a problem when controlling a non stationary source. The only solution to control efficiently non stationary sources is to measure this sources before the room is excited by it, and then forecast the control to be made knowing the transfer function between the source and noise inside the room. In all cases the accuracy of the amplitude and phase of the control source has to be very precise to get the maximum control performances.

5 COMPARISON

The comparison of the three disposals is sum up in Table 1.

Table 1: comparison of the three disposals

	Electromechanical passive damping system	**Electromechanical active damping system**	**Active modal control**
Active/passive	passive	active	active
Frequencies band	Narrow band	Broad band	Modal frequencies
Strategy	Absorption	Absorption/reflection	Modal interferences
Advantages	No need of power supply Easy to design	Works for a wide frequency band Tunable	Need only few loudspeakers Efficiency
Drawbacks	Need an important surface of treatment	Need an important surface of treatment	Time for modes to decrease

The electromechanical passive damping system is efficient for a narrow band around the loudspeaker resonance. It is dedicated to reduce low frequency noises created for example by rotating machinery or power supply. Needed no power supply, the disposal has to be tuned for a specific range of frequencies. Like any absorbing material its effectiveness depends directly on the treated surface.

The electromechanical active damping system has the same specificities than the electromechanical passive damping system except that the control loop extends the range of frequencies where the disposal is efficient. It allows two different strategies of control against low frequency background noise. In one hand, it can be applied for broad band absorption especially if it is coupled with traditional absorbent materials. On the other hand, a different set up of the control feedback make the disposal available for insulation to reduce for example transmission between two rooms. In both cases, the effectiveness depends directly on the treated surface like the electromechanical passive damping system. It is important to notice than the loudspeakers used for the electromechanical active damping system can be managed in the same time for active modal control.

The active modal control is an efficient method to reduce low-frequency background noise amplified by the modal behavior in rooms. This phenomenon appears when a room is excited by a noise close to the first eigenfrequencies of the room. It can be canceled with only few loudspeakers controlling the noise at the modal frequencies with appropriate magnitude and phase. The main challenge of this technique is to be able to control a non stationary noise. The only solution would be to measure the noise source before it excites the room, and then forecast the control to be made knowing the transfer function between the source and the noise inside the room. Indeed, the event exciting the room should not be shorter than the decay time of the mode.

6 CONCLUSION

Three smarts methods for low frequency noise damping have been presented here. Each of them has to be chosen according the type of noise to decrease. The passive damping system is efficient for low frequency narrow band noise absorption. The active damping system can be applied for absorption in a wider band or for noise insulation. The modal control reaches good performances in reducing low frequency background noise amplified by the modal behavior in rooms.

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