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Numerical method to predict active control efficiency for a double panel separating two rooms

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ABSTRACT

In this paper a numerical investigation to improve low frequency sound transmission performance of double wall systems by means of active noise control is presented. The double wall is placed in a laboratory situation, i.e. between an emission room and a reception room. The control sources are distributed in a plane surface inside the air cavity coupling the two plates. The double wall is excited by an acoustic field generated by a sound source placed in the corner of the emission room. The numerical double wall model is based on finite element method; the acoustic field in the reception room is obtained from an integral method using the receiving room Green's function. The model allows to investigate the effect of the number of control channels, the thickness of the double panel cavity, the distance between control sources and error microphones, the number of acoustical modes of the air cavity as well as effects of the active control inside the emission room.

1. INTRODUCTION

Most of new buildings are equipped with sheetrock (gypsum board) partition walls involving one or more layers of air which are responsible for a vibroacoustic coupling between successive sheets. Such layered walls are very efficient in terms of the sound transmission, except at the lower frequencies especially when walls and rooms resonances occur [1-7]. To get high acoustic insulation at low frequencies, such passive walls should require acceptable quantities of materials. As there is few hope of increasing significantly the performances of passive materials over a wide frequency band [1], active control has been investigated in previous work [2-5] as an alternative to improve double wall sound performances at low frequencies. This paper presents a numerical method for predicting sound transmission performances through double panels equipped with active control sources. The double panel separates an emission and a reception room and is composed of two gypsum plates sandwiching an air cavity. The panel is excited by an acoustic field generated by an acoustic source (monopole) placed in the corner of the emission room. In order to improve by active control the transmission loss of the double panel at low frequencies, 9 secondary sources (monopoles) are distributed on a vertical plane area located in the air cavity coupling the two plates. The complex amplitude spectrum of each monopole is introduced in the calculations so that the pressure level is minimized at error microphones positions. The numerical model of the double panel is obtained by finite element technique using Patran/Nastran Software. The acoustic field in the reception room is computed from an integral method using the receiving room Green function [6-9]. In this paper, a simulation method of the active control

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is described and effects of some parameters on active control efficiency such as the number of the secondary sources, the thickness of the air cavity, the number of the acoustic modes taken into account inside the cavity are presented and discussed. The effect of the active control on the pressure level in the emission room is also investigated.

2. SIMULATION METHOD

Consider a double panel separating two rooms as shown in Figure.1. In the case of passive walls, the acoustic field transmitted into the receiving room has been computed in previous works [6-7] by means of a Rayleigh-like integral which makes use of decoupled panel velocities and Green functions for isolated volumes. This method is valid in most practical cases where the plate's velocity is little affected by the pressure in the receiver domain [6]. The velocities are computed by finite element techniques and the Green functions by means of modal volume representations [7]. Results presented here have been obtained by using the GAIA software [6] which can be employed for the calculation of either the excitation pressure of the separating wall (located at z_1) in the emission room and of the radiated pressure in the reception room. The radiated pressure in the reception room P(M) is evaluated as follows:

$$P(M) = -jw\int_{S_V} V(Q)G_v(M,Q)dS(Q) \tag{1}$$

where S_V is the radiating surface located at z_4, V its velocity computed with the FEM software NASTRAN for the incident pressure excitation in the emission room at z_1. G_v is the Green function in the receiving volume computed for the vibrating wall considered to be motionless but keeping all other acoustical boundaries unchanged.

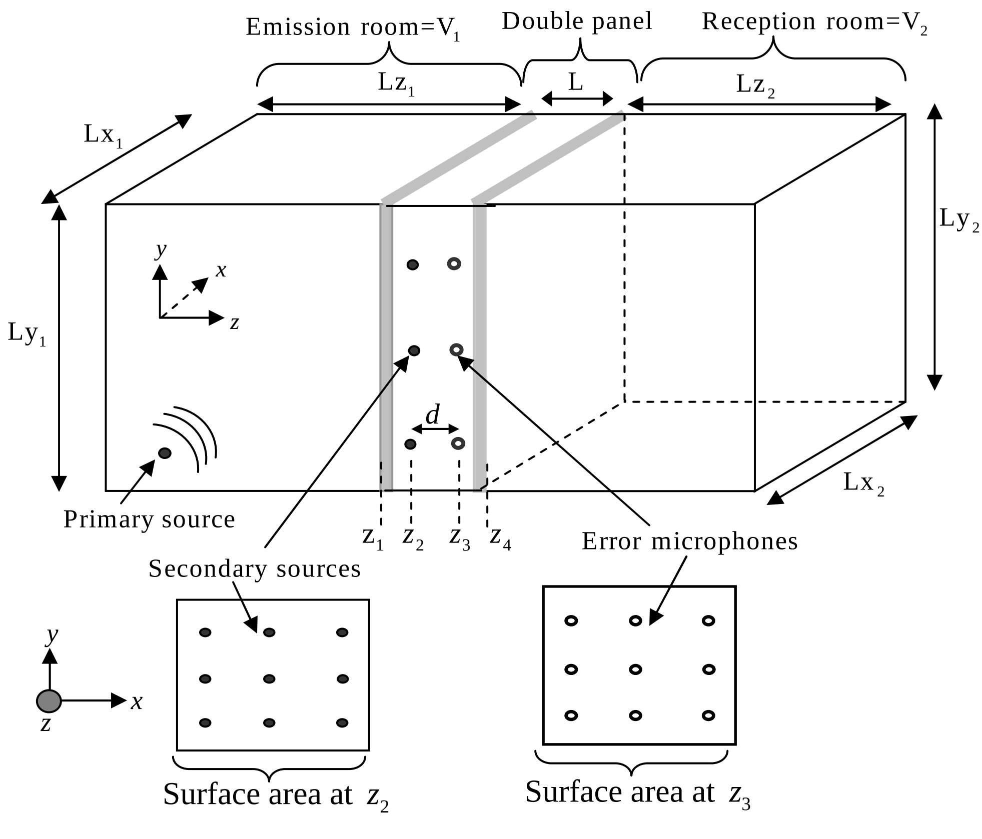

Figure1: Double panel including 9 active control sources separating an emission and reception rooms.

The simulation procedure of the active control is achieved through two principal steps:

- In the first steep, the complex amplitude spectrum of each of the 9 monopoles (secondary sources) enabling the cancellation of the acoustic pressure at the error microphones positions is determined.
- The second steep involves the computation of the transmitted field resulting from the action of both the primary source placed in the emission room and the 9 secondary sources (the monopoles with optimized complex amplitudes) located inside the double panel cavity.

A. Secondary sources amplitude calculation procedure

Computation of secondary or control sources amplitude spectrum procedure is divided into the following 6 steps:

1- The sound pressure in the emission room is computed for the noise source with GAIA software resulting in the acoustic pressure incident on the separating wall, i.e. a pressure field spectrum on a regular grid on the excited side of the partition wall at z_1.

2- A FEM meshing of the double panel separating wall is realized for a point-like mechanical excitation (since it is not possible at this stage to implement an acoustic excitation field) with PATRAN software, i.e. the standard mesher associated with NASTRAN.

3- In the input file for NASTRAN, the excitation is modified in order to include the distributed pressure excitation spectrum corresponding to the noise source excitation evaluated in Step 1. For each monopole control source placed in the cavity, an input file for NASTRAN is also created; at this stage each control source has unit amplitude over the frequency range selected.

4- NASTRAN is then run once with the input file corresponding to the noise source in the emission room, and then N times corresponding to the number of control sources in the cavity with the corresponding control source input file. The acoustic field spectrum at the nodes in the double panel air cavity is obtained for the noise source in the emission room and each of the control sources running independently.

5- The acoustic field spectra at the nodes in the double panel air cavity corresponding to the N error microphones placed in the cavity is extracted.

6- The complex amplitude spectra for the N monopoles control sources enabling the cancellation of the acoustic pressure at the N error microphones locations are deduced as follows:

$$\left\{Sc_j(f)\right\}_N = -\left[P_{kj}(f)\right]_{NxN}^{-1} \; x \; \left\{B_k(f)\right\}_N \; , \qquad (2)$$

Where P_{kj} is the acoustic pressure spectrum caused by the monopole control source j at error microphone position k, B_k is the acoustic pressure spectrum at error microphone position k, caused by the primary source (monopole) placed in the emission room.

It should be noted that to model the air cavity NASTRAN uses volume modes combining axial and transverse modes; the transverse modes are taken into account even though they are evanescent. Therefore care should be taken in selecting the number of these modes.

B. Transmitted pressure computation procedure

The transmitted acoustic field spectrum into the receiving room resulting from both the primary noise source and the N secondary control sources with their optimized complex amplitude is evaluated following the steps described below:

1- An input file for NASTRAN is created for an excitation combining the primary noise source and the N secondary control sources with their optimized complex amplitude.

2- NASTRAN is run and the velocity field at the nodes on the partition wall is obtained at each frequency considered.

3- The velocity field spectra at chosen nodes on the separation partition are extracted and formatted to use as input for the GAIA software.

4- GAIA is run with the extracted FEM velocity field. It should be noted that the FEM velocity field is interpolated to a new integration grid of varying size. The acoustic pressures and radiated power are then obtained according to the method described in [6-7]. The sound reduction index R is finally evaluated from the ratio between incident and radiated acoustic powers Winc and Wray.

3. RESULTS

Results from the different simulations are plotted versus frequency range between 10 and 400 Hz. The two plates are considered identical and lateral walls of the air cavity are assumed rigid. Physical properties as well as dimensions of the two rooms are listed below in Table 1. For Figures 3 to 5, the thickness L of the air cavity coupling the two plates is equal to 20 cm, and the distance ($d=z_3-z_2$) separating control sources plane and error microphones plane is equal to 10 cm. In Figures 6 and 7, the thickness L is reduced to 4 cm and the distance $d=z_3-z_2$ to 2 cm.

Figure2 represents the simulated sound reduction index R with and without active noise control. As expected, the sound reduction index R decreases for passive case at particular frequencies corresponding to plates resonances (near of 70, 82, 100, and 150Hz for example). The dip located near 45 Hz is due to the mass-spring-mass resonance of the double panel which is highly depending on the air cavity thickness L. Below 90 Hz, a gain on the sound reduction index R (more than 20dB) is generally obtained by active noise control. Unlike the passive case, R is not seriously affected by resonance frequencies. Indeed, around plate’s structural resonances and mass-spring-mass frequencies, the active control is more efficient (40 dB gained on R). Above 100 Hz, the active control efficiency decreases rapidly with increasing frequency. This is due to the pressure distribution at the error microphones plane which depends on the frequency. In fact, when the frequency increases the acoustic pressure field becomes more complex in terms of its distribution with respect to the x and y coordinates. As a result, the cancellation (by active noise control) of the acoustic pressure level at the 9 error microphone positions does not allow the cancellation of the pressure in the entire error microphones plane.

Figure 3 shows the influence of the control sources number on active noise control efficiency. It can be noticed that a 5 dB gain on the sound reduction index R is obtained with active control when the number of secondary sources is increased from 9 to 16. Furthermore, with 16 control channels (control sources), the frequency band on which the active noise control is efficient is enlarged, i.e. up to 200 Hz.

In order to investigate the effects of active noise control in the emission room, the sound pressure level caused by the primary source only, the 9 control sources (with optimized complex amplitudes)) and the primary plus 9 control sources is plotted in Figure 4. Except for frequencies close to the mass-spring-frequency, the active noise control has negligible effects on the sound pressure level in the emission room. However, in the case of thin plates with a low density, effects of active noise control on the sound pressure level in emission room could be important at resonance frequencies since for these frequencies the plate is acoustically transparent.

Figure 5 shows the sound reduction index R with and without active control cases for the case of a cavity depth of L=4cm with a distance between control source and error microphone of d=2cm. Compared to the previous results for L=20cm and d=10cm as seen in Figure 3), the efficiency of the active noise control is less important. Indeed, only a gain of 15 dB has been obtained for frequencies below 70 Hz ; this gain was of 20 dB for the previously considered case L=20cm and d=10cm. Indeed, the number of transverse modes taken into account in the cavity was the same for these two different cases (L=20cm - d=10cm and L =4cm - d=2cm). However, it appears when the cavity depth is small and when the distance between the control sources and the error microphones, the evanescent transverse modes influence greatly the results. As illustrated by Figure 6, the active control efficiency depends highly on the number of the transverse modes in the cavity even if these transverse modes are evanescent at a given frequency. Increasing the number of the evanescent transverse modes allows increasing active control efficiency for the reduced thickness cavity case. This phenomenon will be further investigated.

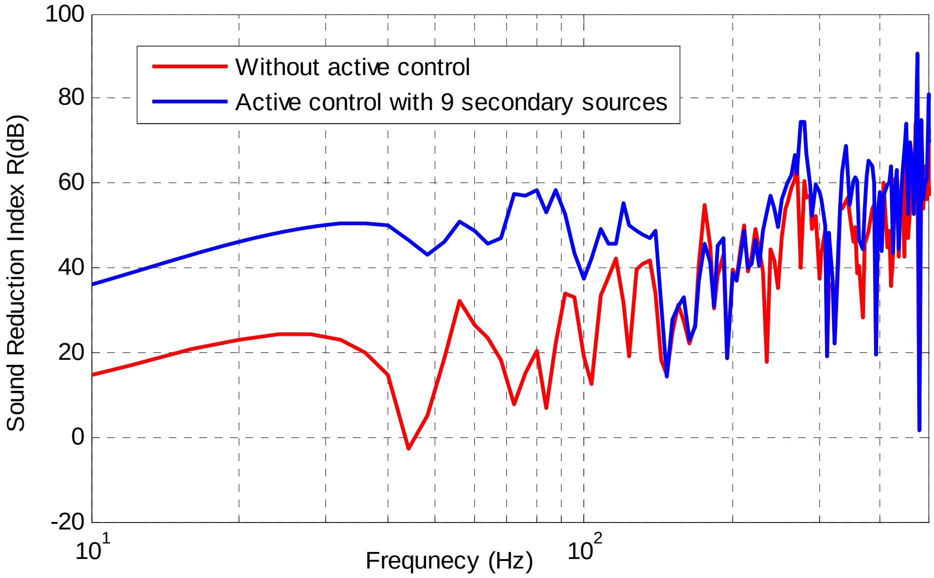

Figure 2: Predicted sound reduction index R with and without active control – Case L=20cm and d=10cm.

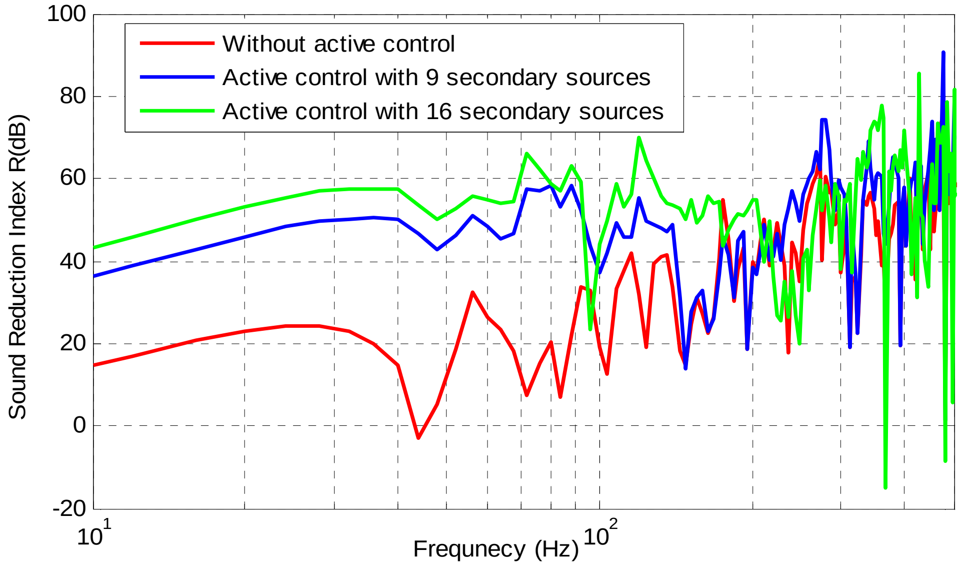

Figure 3: Comparison between predicted sound reduction indexes with either 9 or 16 control sources for active noise control - Case L=20cm and d=10cm.

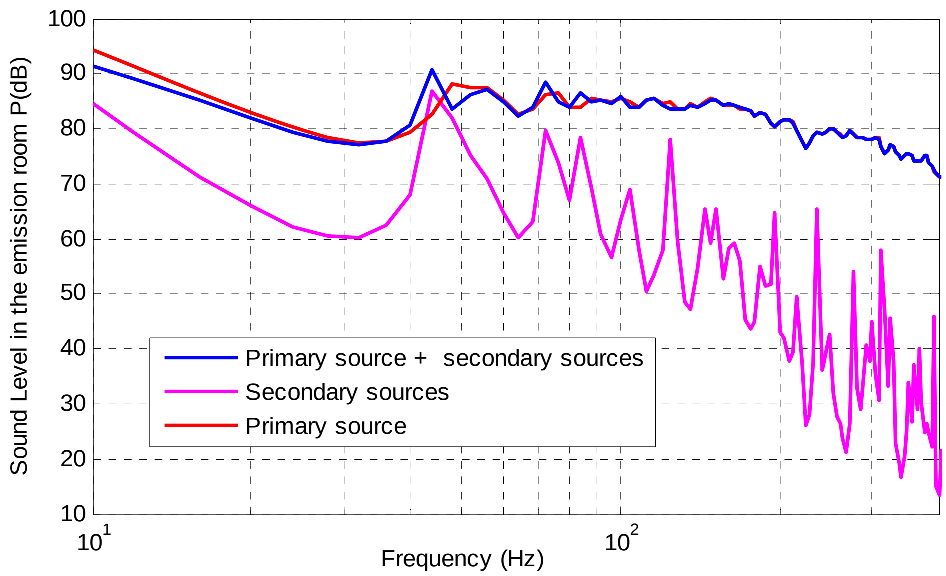

Figure 4: Predicted sound pressure level in the emission room – Case L=20cm and d=10cm.

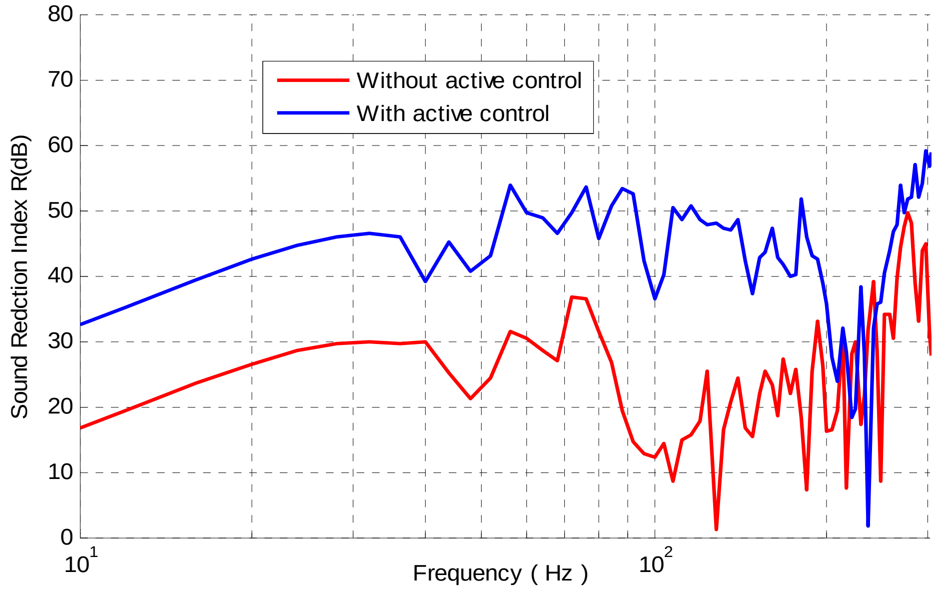

Figure 5: Predicted sound reduction index R with and without active noise control – Case L=4cm and d=2cm.

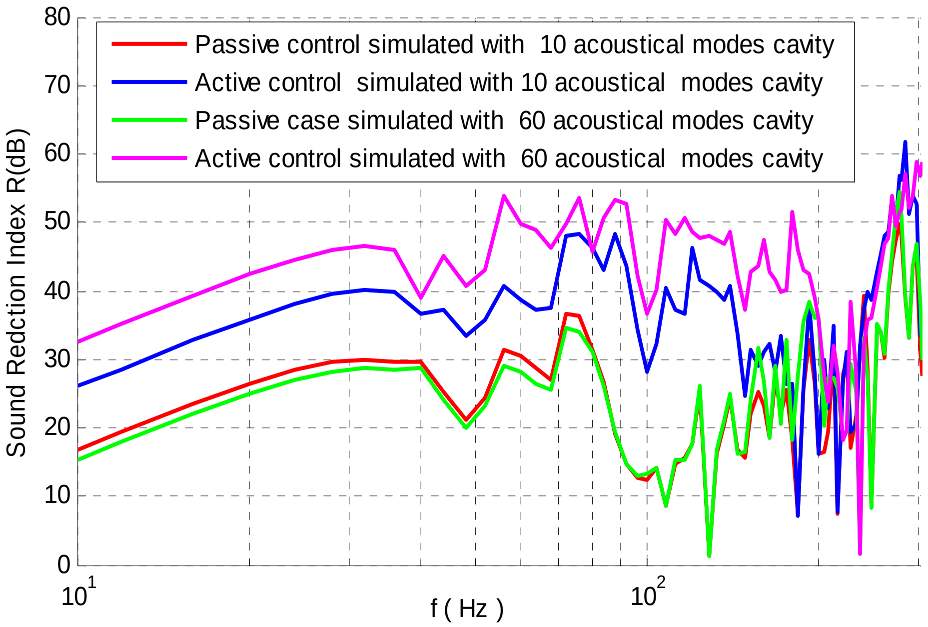

Figure 6: Comparison between predicted sound reduction indexes R with and without active noise control taking into account either 10 or 60 acoustical cavity modes (L=4cm, d=2cm).

Table 1: Physical properties and dimensions of the double plates, emission and reception rooms.

Plates	
Width	$Lx = 3$ m
Length	$Ly = 2.4$ m
Thickness	h = 2x12.5 mm
Density	$\rho = 725$ kg/m^3
Young's modulus	$E_{eq} = 0.684$ GPa
Poisson's ratio	$\nu = 0.1$
Emission and reception rooms	
Width	$Lx_1 = Lx_2 = 3$ m
Height	$Ly_1 = Ly_2 = 2.4$ m
Width	$Lz_1 = 3$ m $Lz_2 = 3.5$ m
Normal absorption coefficient of room walls	$\alpha=0.4$
Air cavities	
Density	$\rho_0 = 1.213$ kg/m^3
Speed of sound	$c_0 = 343$ m/s

4. CONCLUSIONS

In this work, a simulation method for predicting the sound reduction index of a double panel including active control sources is developed. Calculation steps have been described and effects of some parameters on active noise control efficiency have been simulated and presented.

The study emphasises that when the thickness L of the cavity is not too small (L>10cm), the active control through only 9 control sources improves seriously the transmission loss of the double panel (gain larger than 20dB) at low frequencies below

100 Hz. For frequencies higher than 200 Hz, active control effects are negligible. Results have also shown that increasing the control sources number increases the improvement of transmission loss as well as the controlled frequency band. Also, the study points out that when the cavity depth L is small, the evanescent transverse modes in the cavity influence greatly the results. Indeed, increasing the number of the evanescent transverse modes allows increasing active control efficiency for the reduced thickness cavity case.

In future works, effect of absorption coefficient of the cavity lateral wall, as well as the presence of a porous material will be investigated. Simulated results should be compared to experimental ones.

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