Active control of sound transmission loss through a single panel partition using distributed actuators. Part II : Experiments.

W. Dehandschutter†, K. Henrioulle, P. Sas
Department of Mechanical Engineering, division PMA, K.U.Leuven, Belgium
e-mail : Kris.Henrioulle@mech.kuleuven.ac.be

† author is now with Lernout & Hauspie Speech Products

Abstract
This paper discusses a series of experiments which have been performed in order to validate the potential of various configurations with which the transmission of sound through a single panel partition is actively controlled. The experimental set-up which is used consists of a single steel panel of size $300 \times 400 \times 1$ mm which is subject to a plane wave acoustic excitation. The control system consists of a distributed acoustic actuator which is placed near the steel panel. Two configurations, with the control actuator placed at the "receiving" side and at the "radiating" side have been tested. The control actuator is driven by a controller which aims at reducing the acoustic energy at the radiating side. Various control approaches have been tested, using different control strategies for both of the control actuator configurations.

1. Introduction

The research work which is presented in this paper aims at developing efficient distributed actuators for active control of sound transmission through single or double wall partitions. Typical applications where these actuators would be used are : to enhance the insertion loss of industrial sound encapsulations at low frequencies, to control the sound field inside car or aircraft cabins, to reduce the transmission of sound through the walls of poorly insulated buildings, etc.

Active control of sound and vibration relies on using a set of control actuators which are driven such that they minimise the vibration or sound pressure level due to a primary disturbance at specified locations. To this aim, a "classical" active noise control system will typically employ a number of loudspeakers with which the sound field, originating from the primary disturbance, is controlled. Active control systems have already been demonstrated to perform well for the kind of applications envisaged here [1 - 3]. However, the practical or commercial realisation of such an active control system is often constrained by practical considerations, such as the fact that standard permanent magnet loudspeakers cannot easily be fit into the air gap of double wall partitions.

In [4], a light-weight distributed acoustic actuator is presented which is designed specifically to be used under these circumstances. The actuator consists of a honeycomb core (the "carrier" structure) which is covered by a PVDF foil on both sides. By controlling the driving voltage which is applied to the PVDF foils, an out-of-plane motion can be induced in the actuator. The dimensions of the actuator and the design of the PVDF foils has been optimised [4] as to generate the highest sound power output possible in the frequency range considered (100 - 500 Hz for typical industrial applications).

This paper is a sequel of a first paper [5] in which numerical simulations are presented of various configurations in which the control actuator is used to control the transmission of sound through a single panel partition. The aim of the present paper is twofold : *(i)* to experimentally validate the simulation model which is used in [5], and *(ii)* to provide additional insight in the control mechanisms that determine the control performance of the various control approaches tested.

The paper starts with describing the experimental test set-up and the apparatus which was used in the tests, after which the results achieved for the different control configurations are presented and discussed.

2. The test set-up

2.1 Description of the test set-up

Figure 1 presents the test set-up which was used in the experiments. A transmission wall was installed above the cellar of a semi-anechoic test room. A 300 $\times$ 400 mm opening was provided in the transmission wall in which a passive plate and the control actuator could be mounted. The primary disturbance source was a loudspeaker, mounted in the cellar, below the transmission wall. The aim of the active control system was to minimise the transmission of sound from the cellar (the sending room) to the semi-anechoic test chamber (the receiving room).

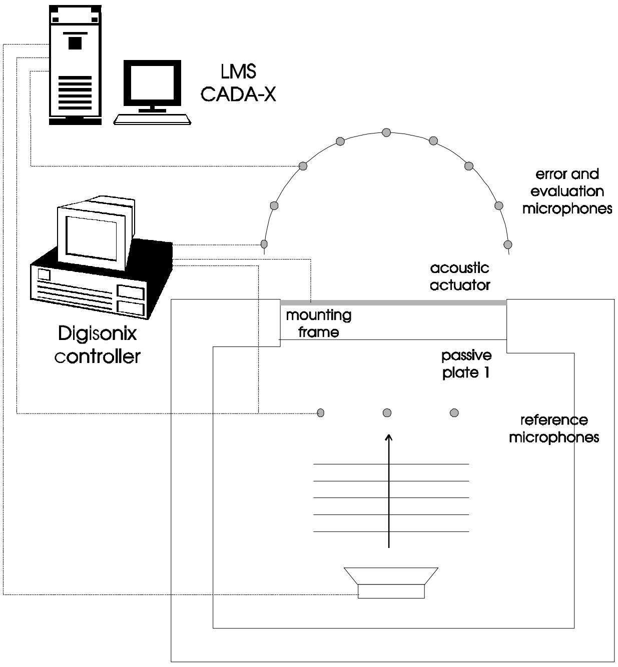

Figure 1a. Schematic overview of the test set-up.

The primary disturbance was driven by a Cada-X® data acquisition system. The primary excitation was controlled as to realise a constant (i.e. identical in all experiments) sound pressure level measured at the microphones located close to the single panel partition at the incident side. The control performance was monitored by measuring the acoustic intensity above the single panel partition, with and without control.

The test set-up was constructed carefully as to

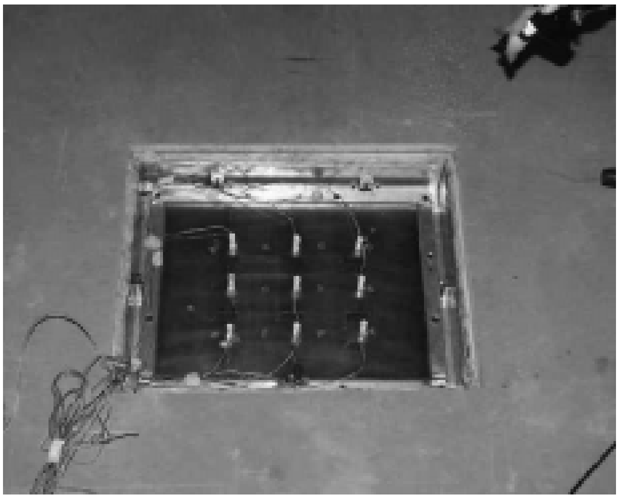

Figure 1b. View on the control actuator mounted in a frame in the transmission wall. The frame allows to mount a steel plate above or below the actuator. The vibration level of the actuator is measured by means of 9 accelerometers.

Minimise flanking noise transmission. This was necessary as the active control system proved to be able to realise an increase of the transmission loss in the order of 20 dB.

2.2 Actuator configurations tested

All experiments were carried out using the same control actuator, shown in figure 1b. As explained extensively in [4], the actuator principle is based on the effect of extension/contraction of the PVDF foils when a voltage is applied to them. By bonding the foils to a carrier structure, this extension/contraction can be converted into a bending motion of the carrier, and hence into vertical motion. The control force applied to the carrier structure has been maximised by completely covering the carrier with a PVDF foil. For this configuration, the distributed force applied by the foils can also be modeled by 4 line moments, each of them acting on the borders of the carrier structure. This implies that only the [odd, odd], modes of the carrier can be controlled with the current PVDF foil configuration. This means that in circumstances where the sound waves, incident on the single panel partition, are not perpendicular plane waves, some of the [odd,even], [even,odd] and [even,even] modes will be excited while their contribution cannot be controlled by applying a control voltage to the control actuator. It must be noted that due to the dimension of the actuator, plane waves with an angle of incidence up to 45° will excite these uncontrollable modes only to a very small extent (actually the dimensions of the actuator were limited to achieve this). In the case of a diffuse primary sound field, however, the occurrence of uncontrollable modes in the frequency band of interest can limit the achievable control performance. As illustrated by the results presented in [5], the deterioration of the control performance is limited in this case because the uncontrollable modes are all dipole modes and hence have a poor radiation efficiency.

Three control configurations have been tested and will be discussed in this paper. In a first set-up, only the control actuator was mounted in the opening of the transmission wall. Afterwards, two configurations were tested in which a passive steel panel is combined with the control actuator. These configurations simulate applications where the passive (steel) plate is a functional system component (e.g. the fuselage of an aircraft constitutes a pressurised passenger's compartment) of which the passive sound insulation properties are enhanced by mounting a control actuator close to it. Two such configurations are tested : in configuration A, the actuator is mounted at the incident (sending room) side of the passive plate ; in configuration B, the actuator is mounted at the radiating (receiving room) side of the passive plate.

2.3 Control strategies

All control experiments have been performed using an adaptive Filtered-X LMS feedforward controller to drive the control actuator. The controller was implemented on a Digisonix® dX-200 system. The reference signal for the controller is taken directly from the signal generator. The adaptive Filtered-X LMS feedforward controller generates a control signal with a frequency content equal to that measured in the reference signal by feeding the reference signal through an adaptive filter. The adaptive filter is updated every time sample as to optimise the amplitude and phase of all frequency components of the control signal. The adaptive filter is updated such that the sum of the squares of a predefined set of error signals is minimised.

Different control strategies can thus be realised by constructing different error sensor configurations. In the experiments presented here, it was investigated whether the control performance achieved with error microphones located in the receiving room (the best but often not-so-practical configuration) could as well be achieved using accelerometers located on the control actuator and/or passive plate as error sensors.

3. Test results

First a few general remarks are given with respect to the test results presented in this paper :

1. All error signals, vibration levels, sound pressure levels, and control voltages, are presented as an FRF with the sound pressure measured at the incident side as a reference. This was done mainly because it was virtually impossible to realise a flat frequency spectrum for the incident sound pressure level. This was due to the standing wave phenomena occurring in the sending room, which leads to sharp peaks at the acoustic resonance frequencies of the sending room in all measured spectra. The first three acoustic resonance frequencies of the cavity constituted by the cellar are 79, 119 and 121 Hz and hence they are well within the frequency range of interest. By "normalising" the measured spectra with the incident sound pressure level, yields a frequency response function of which all resonance peaks are due to the dynamics of the single panel partition itself. It must be noted that the sound intensities were not measured as an FRF and hence the sending room resonances show up in these spectra.
2. All spectra, with and without control, have been measured using a broadband excitation signal for the primary source. Feedforward control of broadband disturbances implies that the causality constraint must be satisfied [7]. In a control configuration such as the experimental set-up which is presented here, this implies that it is assumed that the primary source is located at large enough a distance from the transmission wall to allow sufficient time for the controller to measure and process the reference signal and to apply the control action at the appropriate time instance. In cases where this condition was not satisfied, the control performance was degraded seriously.
3. For the same reasons, the controller was run at a sample frequency of 5000 Hz, since this reduces the time delay through the control path.

3.1 Control of sound transmission through the control actuator

In a first series of tests, only the actuator was installed in the transmission wall opening. This configuration is only relevant for applications where honeycomb panels are already used as functional system components (e.g. aircraft trim panels). The experimental results show that the transmission loss through the panel can be increased significantly by restructuring the vibration patterns of the panel. The test results reveal some important properties of the control actuator.

Figure 2 presents the acoustic intensity -with and without control- radiated by the actuator when it is excited by the primary source. Two control systems are compared, the first employing 4 accelerometers as error sensors, the second one using 2 error microphones. One error microphone was located close to the actuator, the second at 1 m distance above the center of the actuator. The 4 accelerometers have been located on the actuator surface such that all structural modes which participate to the structural response in this frequency band could be observed. However, figure 2 shows that the transmission loss cannot be increased at frequencies above 250 Hz with this control system.

This is explained by figure 3 an 4. Figure 3 presents the vibration level, measured in the centre of the actuator for both control systems. Figure 4 presents the control voltage, sent to the control actuator, for both cases. Figure 3 shows that the vibration level is indeed reduced significantly when using accelerometers as error sensors. In the case of error microphones, however, the control system increases the vibration level substantially, while at the same time the radiated acoustic intensity is reduced (figure 2). This result shows that it is more efficient to excite some of the structural actuator modes and to properly control the phasing between these modes such that their contributions cancel, rather than to reduce the amplitude of all the structural modes. This control mechanism is present in the complete frequency band under consideration, except at low frequency. This is explained by the fact that the control actuator, due to the way it is constructed, can only control one structural actuator mode independently from all other modes. At low frequency, the response of the actuator is dominated by only one mode, namely the [1,1] -mode (resonance frequency at 125 Hz), which radiates very efficiently. Thus it is most efficient to reduce the mode amplitude of this mode only in order to reduce the radiated sound power. At higher frequencies, however, various efficiently radiating modes are excited at the same time such that it becomes more effective to control their relative phasing rather than to control the individual amplitude of just one mode.

3.2 Configuration A : passive plate at the radiating side

In this configuration, three different control systems were tested. The first employs the same two error microphones as in the previous test. The two other systems use accelerometers as error sensors : in the first case they are located on the passive plate, in the second case they are located on the actuator.

The acoustic intensity, radiated by the passive plate with and without control, is presented in figure 5 for the three control systems. A slightly different control mechanism is observed in this case : for this configuration and with this type of actuator it seems to be most efficient to reduce the vibration level of the passive plate. Both the control system with error microphones and the system with accelerometers on the passive plate yield almost identical control performance. The observation that controlling the vibration level of the control actuator is a less efficient control approach is explained by figure 6 which shows that from 225 Hz the vibration level of the passive plate can only be reduced by increasing vibration level of the actuator.

For this configuration, it must be noted that the control voltage sent to the actuator is significantly larger as compared to the first test (figure 7). This is explained by the fact that the actuator now has to “drive” the acoustic impedance of the double panel partition constituted by the actuator and the passive panel. This impedance is significantly larger as the impedance which the actuator “sees” when operating in the free field.

3.3 Configuration B : actuator at the radiating side

Two control systems have been compared for this configuration : the first one using the same two error microphones, the second one using accelerometers located on the actuator.

Figure 8 presents the acoustic intensity radiated by the actuator without and with control for both control systems. The figure shows that above 200 Hz no increase of transmission loss is achieved by merely controlling the vibration level of the actuator. Figure 9 shows that, in order to increase the transmission loss at all frequencies (which is realised by using the microphones as error sensors) the vibration level of the actuator is increased at all frequencies. This result seems to be somewhat in contradiction with the results presented in section 3.1, as in both experiments, the control actuator is

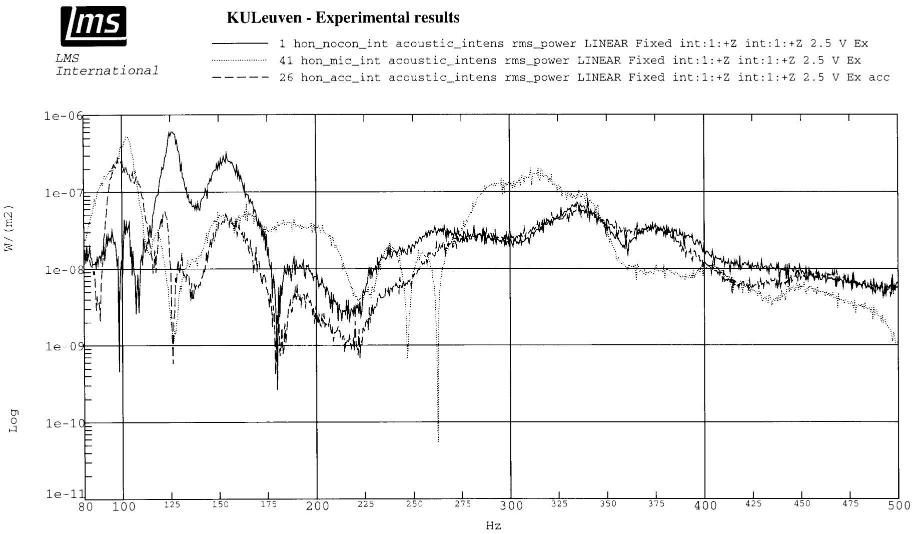

Figure 2. Acoustic intensity radiated by the actuator in the case where only the actuator is mounted in the transmission wall. Solid : without control ; dashed : control using 4 accelerometers as error sensors ; dotted : control with 2 error microphones.

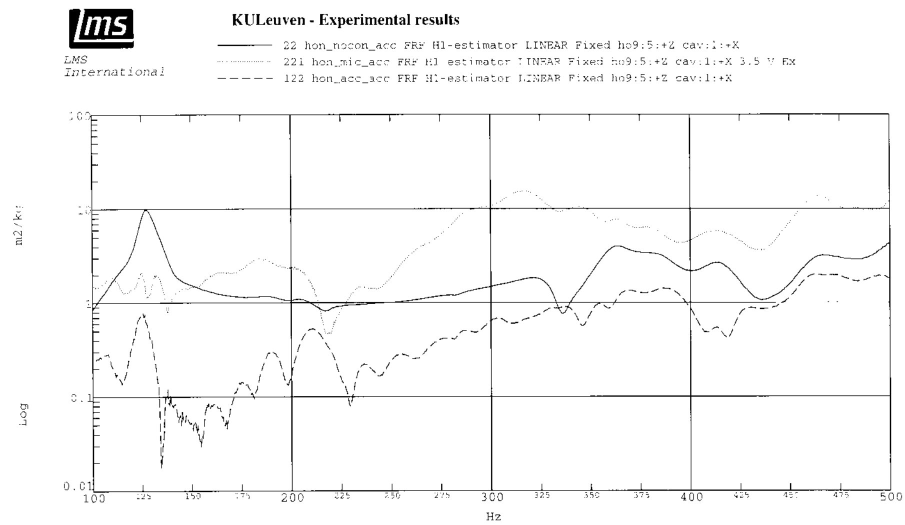

Figure 3. Vibration level measured in the centre of the actuator in the case where only the actuator is mounted in the transmission wall. Solid : without control ; dashed : control using 4 accelerometers as error sensors ; dotted : control with 2 error microphones.

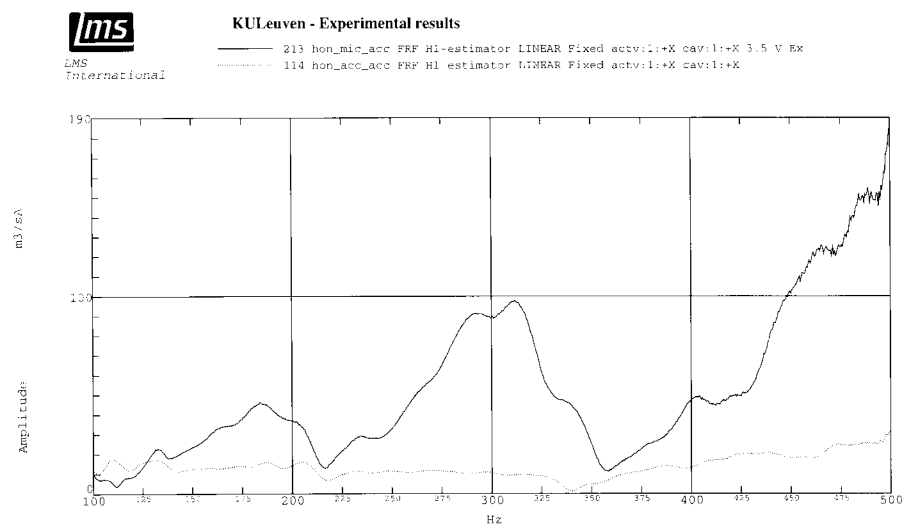

Figure 4. Control voltage sent to the actuator in the case where only the actuator is mounted in the transmission wall. Solid : control with 2 error microphones ; dotted : control using 4 accelerometers as error sensors.

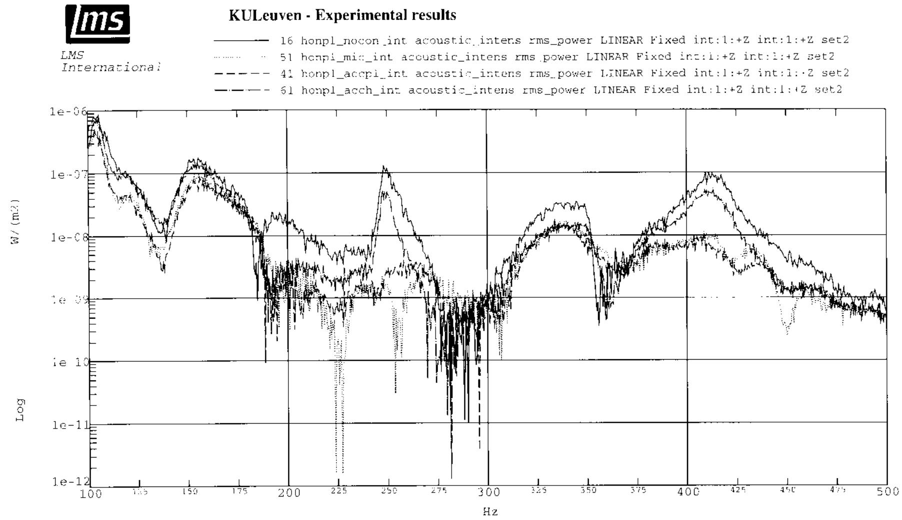

Figure 5. Acoustic intensity radiated by the panel in configuration A. Solid : without control ; dashed : control using accelerometers on the passive plate as error sensors ; dash-dot : control using accelerometers on the actuator as error sensors ; dotted : control with 2 error microphones.

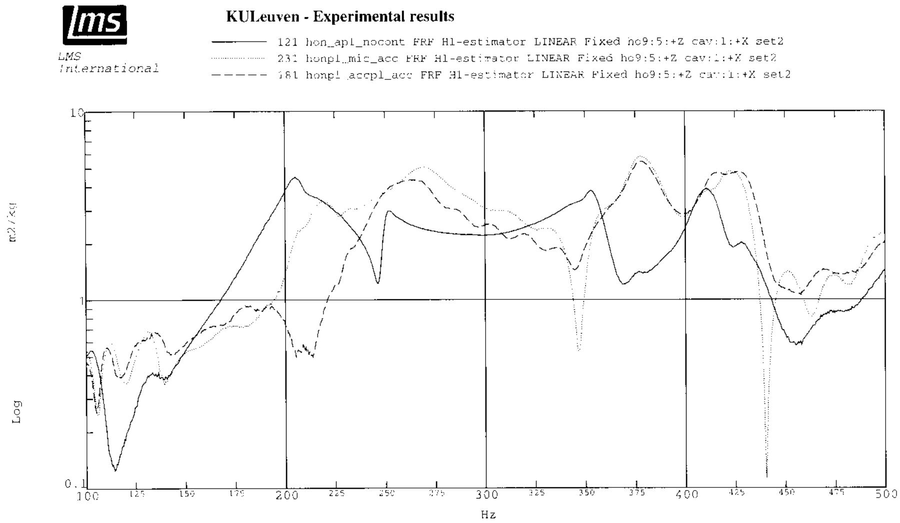

Figure 6. Vibration level measured in the centre of the actuator in configuration A. Solid : without control ; dashed : control using 4 accelerometers on the passive plate as error sensors ; dotted : control with 2 error microphones.

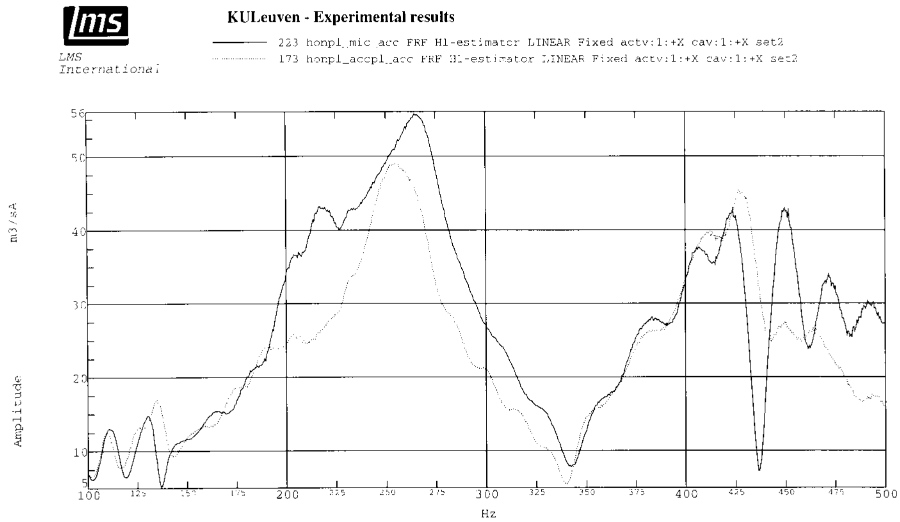

Figure 7. Control voltage sent to the actuator in configuration A. Solid : control with 2 error microphones ; dotted : control using 4 accelerometers on the passive plate as error sensors.

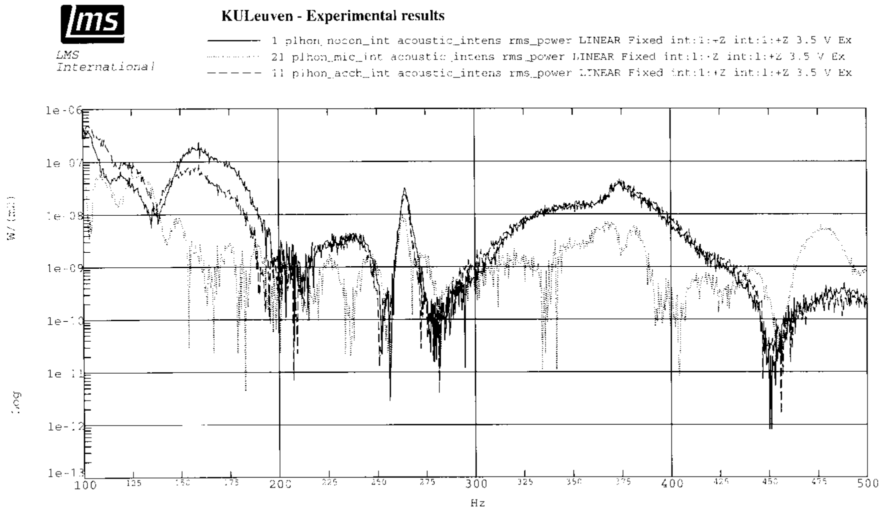

Figure 8. Acoustic intensity radiated by the panel in configuration B. Solid : without control ; dashed : control using accelerometers on the passive plate as error sensors ; dash-dot : control using accelerometers on the actuator as error sensors ; dotted : control with 2 error microphones.

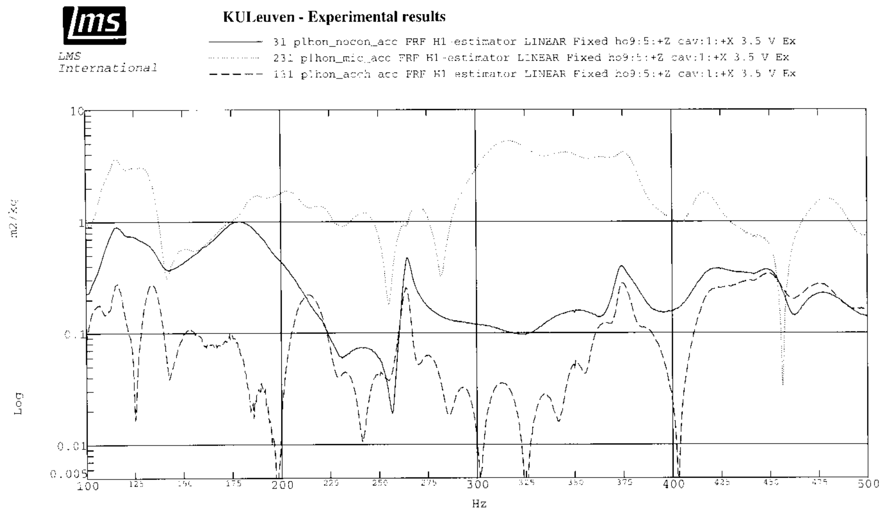

Figure 9. Vibration level measured in the centre of the actuator in configuration B. Solid : without control ; dashed : control using 4 accelerometers on the passive plate as error sensors ; dotted : control with 2 error microphones.

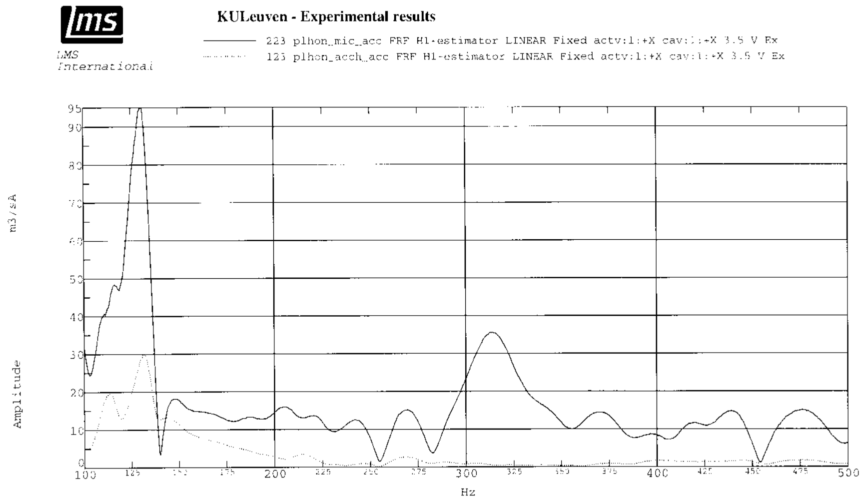

Figure 10. Control voltage sent to the actuator in configuration B. Solid : control with 2 error microphones ; dotted : control using 4 accelerometers on the passive plate as error sensors.

mounted at the receiving room side. In section 3.1, it was observed that the control system achieves the best performance by reducing the actuator vibration level at low frequencies, rather than increasing it, as it is the case in configuration B. In the configuration presented here, however, the sound field which excites the control actuator is more complicated due to the coupling between the actuator and the passive plate. As a result, most of the [odd,odd] actuator modes are excited at the same time at all frequencies (whereas only the [1,1] mode was found to be dominant at low frequency in the experiments presented in section 3.1). The most efficient control mechanism in this case is to rearrange the phasing between these modes.

Despite the apparent difference in achieved control performance, it is interesting to note that the vibration level of the passive plate is almost identical for both control systems tested here, i.e. the same level as in the case without control is maintained (except at low frequency).

4. Conclusions

The main aim of the experiments which are presented in this paper was to show that the distributed acoustic actuator, which was specifically designed to be a lightweight multi-purpose ANC actuator, can indeed be used to effectively control the transmission of sound through single panel partitions. In this sense, the experiments confirm the conclusions drawn from the simulations, presented in Part I [5].

In this respect it must be noted that in many potential applications, the actuator would be used to control periodic sound fields. The results which have been presented here only partially serve as a prediction of the achievable control performance under such circumstances. First of all, the experiments have been carried out using broadband excitation. The control performance achieved at each frequency line will only match the control performance achieved for a harmonic primary disturbance at that frequency iff *(i)* the control system satisfies the causality constraints applying to feedforward control systems [8] and *(ii)* the order of the controller is large enough to adequately model the optimal controller transfer function over the whole frequency range which is excited by the broadband primary disturbance. Secondly, the design of the actuator could be tailored for the specific application in which it would be used. The currently used actuator has its first resonance (the very efficiently radiating [1,1] mode) at around 125 Hz, which is ideal for control of low frequency sound sources. The first anti-resonance, however, appears at around 300 Hz (i.e. the [2,1] mode, which cannot be controlled with the current design of the PVDF foils). By varying the structural stiffness of the actuator in different directions, it could possible to avoid anti-resonances in the frequency range of interest. These additional design constraints may require other carrier structures, e.g. a multilayer composite structure as in [9].

As a secondary result, the experiments presented here provide additional information to answer the question “which of both configurations (A or B) is most optimal ?” Based upon the objective figures, configuration B clearly seems to be the “best” configuration, since a higher control performance is achieved at the expense of less control energy. The latter should not sound too surprising as the passive plate partly reflects the impinging sound field before it excites the control actuator which needs less control energy to control the partly reduced sound field. However, it is important to note that in this configuration, the best control performance can only be achieved by using error microphones located in the receiving room. For many applications this will be impossible. Furthermore, in configuration B, the control actuator is exposed to the receiving room side which is, at least, undesirable from a durability point of view. In configuration A, on the other hand, very good control performance can be achieved by using accelerometers on the passive plate as error sensors. Such a configuration is more compact, more robust against hostile environments, and hence much more suited for practical application.

Acknowledgements

The work reported herein was related to the EC Brite/Euram Research Project “DAFNOR” (under contract BRPR-CT96-0154). The project is supported by the Directorate-General for Science; Research and Development of the CEC. Partners in this project are : KULeuven (B), FFA (SE), ISVR (UK), Thomson (F), VTT (FI), CRF (I) and G+H MONTAGE (G).

References

1. P. Sas, W. Dehandschutter, A. Vecchio, R. Boonen, 1998, “Active control of sound transmission through an industrial sound encapsulation,” *Proc. of ICA/ASA Meeting*, 21 - 24 June 1998, Seattle, Washington, USA.
2. P. Sas, W. Dehandschutter, 1997, “Design of

active structural and acoustic control systems for the reduction of road noise in a passenger car," *Proc. of 6th International Conference on Recent Advances in Structural Dynamics*, ISVR, Southampton, 1997, 21-44.

3. P. De Fonseca, P.Sas, H. Van Brussel, 1998, "Active control of sound transmission through an aircraft fuselage test section," *4th AIAA/CEAS Aeroacoustics Conference Toulouse*, paper #98-2233.
4. K. Henrioulle, W. Dehandschutter, P. Sas, 1998, "Design of an active noise control system using a distributed actuator," *Accepted for publication in Journal of Applied Scientific Research*.
5. K. Henrioulle, W. Dehandschutter, P. Sas, 1998, "Active control of sound transmission through a single panel partition. Part I : Simulations," *Proc. ISMA23 Noise and Vibration Eng. Conf.,* 16 - 18 Sept. 1998, KULeuven, Leuven, Belgium.
6. P.A. Nelson, S.J. Elliott, 1992, "Active control of sound", *Academic Press*, San Diego, CA.
7. W. Dehandschutter, 1997, "The reduction of structure-borne noise by active control of vibration," *PhD Thesis*, KULeuven, Dept. Mech. Eng., Leuven, Belgium.
8. R.A. Burdisso, J.S. Vipperman, and C.R. Fuller, 1993, "Causality analysis of feedforward-controlled systems with broadband inputs," *J. Acoust. Soc. Am* **94**(1), 234-242.
9. M. Resch, M. Jeger, and W.J. Elspass, 1996, "Optimal design of laminated plates for active vibration control," *Proc. ISMA21 - Noise and Vib. Eng.*, Leuven, Belgium, 18 - 20 September 1996, 283 - 294.