DESIGN AND TEST OF A FEEDBACK CONTROLLER FOR ATTENUATING LOW FREQUENCY NOISE IN A ROOM

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A single-channel feedback control system is presented for global noise attenuation inside a room. The controller design is based on the compensation filter approach of classical control theory. To demonstrate the global noise attenuation capability of this simple control scheme experimental results of noise control in an office are presented. They show that the positioning of the error microphone relative to the control loudspeaker greatly affects the global noise attenuation performance. If the microphone is placed too close to the control source, the microphone signal is dominated by the near field and little global noise reduction is achieved. If the error sensor is placed further away where the near field has little effect on the microphone signal, noise reductions of about 10 dB can be obtained over low frequency ranges.

INTRODUCTION

Active noise control technology is an attractive solution for the attenuation of low frequency noise in enclosures. Over the past decade, it had many successful applications including the control of sound pressure level in 1-D acoustic ducts [1], inside the fuselage of passenger jets [2], vibration suppression that reduces the structural acoustic coupling and therefore reduces the interior noise in cars [3], and active control for noise suppression in payload fairings [4]. In terms of control strategies, feedforward control has been widely used. While feedforward control has many advantages, its success relies on the availability of causal reference signals which have to be highly correlated to the noise to be cancelled [5]. For some applications such as the attenuation of random noise in office spaces or bedrooms, such reference signals are either not available or very expensive to obtain. In these situations, feedback control can be an alternative solution.

Progress on feedback control of the sound field in enclosures has been made in the area of prediction and control of noise radiated into enclosures using structural sensing [6]. Such a control scheme was developed for applications where neither a coherent reference signal, nor the sound field to be controlled were available. A method for the selection of the sensor and actuator positions was given based on a transformation of the problem into radiation modes. An optimal feedback control approach which allows the control of radiated pressure into a defined subvolume of the cavity using only structural actuators and sensors was also demonstrated. The main drawbacks of the method were its inability to model complex structures with a large number of modes accurately, failure to include robustness due to uncertainties and the need for system identification.

Techniques of state feedback control of sound fields in enclosures were reviewed by Samejima [7], who also investigated theoretically and experimentally the use of state feedback to achieve a desired modal distribution of the enclosed sound field. Pole allocation was employed to obtain the state feedback gain vector such that the roots of the closed-loop system have the desired modal distribution. This method can be difficult to employ in irregular shaped rooms and was used mainly for changing the acoustic resonances inside a cavity.

A method was also proposed by Yuan [8] to improve active noise control in enclosures. A virtual sensing technique by using two judiciously placed microphones was suggested in order to predict a virtual signal. However, an exact mathematical model between the virtual and physical sensors was required over the entire frequency band of interest for broadband control in a lightly damped enclosure. This can be impractical for rooms. No discussion of the global attenuation characteristic of the control strategies was provided in the paper.

Al-Bassyiouni and Balachandran [9] proposed a zero spillover scheme for active structural acoustic control inside a three-dimensional rectangular enclosure into which noise is transmitted through a flexible boundary. Piezoceramic patches mounted on the flexible boundary were used as actuators and microphones placed inside and outside of the enclosure were used as sensors. The technique took into account the effect of inherent acoustic feedback in the design of the control scheme. The results showed that significant attenuations can be obtained at the error microphone and near the collocated microphone locations, and that a good attenuation can be obtained over a large area of the enclosure in the presence of tonal and broadband disturbances. The experiments also demonstrated that the energy levels in the flexible panel increased significantly when applying the control scheme. The control algorithm used did not take into account the robustness of the control system to any possible changes in ambient conditions and other factors.

More recently, several feedback control schemes have been applied to control noise inside cavities. For minimising sound radiation, Hong and Elliott [10] examined closely spaced local feedback control systems on a honeycomb panel using an accelerometer and a piezoceramic actuator. It was found that the global control performance was affected by local coupling of the control channels, and the multichannel system did not yield a significant improvement in performance because of a decreased gain margin. Creasy et al. [11] described a method for adaptive energy absorption in acoustic cavities based on an adaptive scheme consisting of a self-tuning regulator that has the ability to target multiple modes with a single actuator. The inner control loop of the regulator used positive position feedback in series with a high and low-pass Butterworth filters for each controlled mode. The outer loop consisted of an algorithm that locates the zero frequencies of the collocated signal and uses these values to update the resonance frequency of the positive position feedback filter and the cut-off and cut-on frequencies of the filters. Experimental results show the robustness of the method in the presence of changes in the resonance frequencies of the system and the reduction of spillover. de Oliveira et al. [12] proposed a methodology to derive a fully coupled mechatronic model that deals with both the vibro-acoustic plant dynamics as well as the control parameters. The inclusion of sensor and actuator models was investigated since it can cause limitations to the control performance. The proposed methodology provided a reduced state-space model derived from a fully coupled vibro-acoustic finite element model. Experimental data on a vibro-acoustic vehicle cabin mock-up were used to validate the model reduction procedure. A collocated sensor/actuator pair was considered in a velocity feedback control strategy. The results showed that an optimal design could only be achieved when considering structure and control concurrently. Although the method is useful at the design stage its application to active noise control in existing rooms is impractical.

In this paper, a practical feedback control system for noise attenuation in a room is presented. It is aimed towards the development of a simple active noise control unit for household use (such as bedrooms) and offices in workplaces. The control system is designed based on classical control theory [13], and the controller can be realised with analogue electronic circuits developed for feedback noise control ear defenders, or with digital signal processors employed for traditional active noise control. The objective of the paper is to demonstrate the feasibility of active control of low frequency noise in rooms using classical feedback control theory and to investigate the effect of room acoustics on the performance of the noise control. Control performance measures such as system stability, control bandwidth and global controllability are studied and supported by experimental results.

1. DESIGN OF THE FEEDBACK CONTROLLER

Figure 1 shows the feedback control system considered in this paper. The aim of the control is to achieve global noise attenuation in a room with a control system consisting of a single loudspeaker, a single microphone and a controller.

The heart of the control system is the controller in which a control signal is generated using the error signal from the microphone and a compensator to be designed. There are several ways to design a controller in a feedback control system, ranging from a traditional PID (Proportional-Integral-Derivative) control approach based on classical control theory to a more advanced state-variable approach based on modern control theory [13].

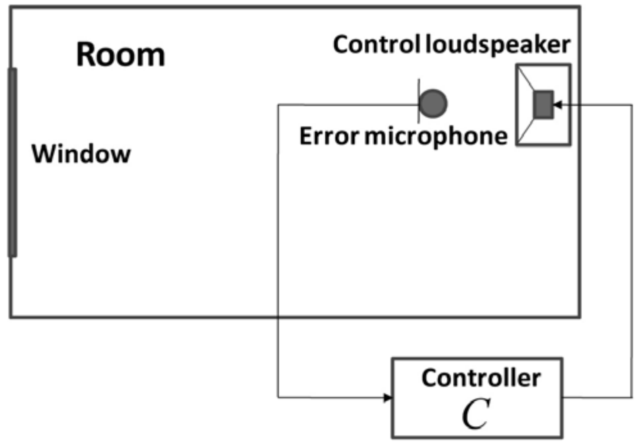

Figure 1 Schematic of the feedback control system.

In this paper, a compensation filter approach based on classical control theory is employed, as it is easy to implement with less expensive analogue circuits. The compensator to be used as the controller has a basic form

$$C(s) = K\left(\frac{s^2 + 2\xi_z\omega_z s + \omega_z^2}{s^2 + 2\xi_p\omega_p s + \omega_p^2}\right), \tag{1}$$

characterised by an angular frequency ω_z and damping ratio ξ_z for the zeros, and ω_p and ξ_p for the poles. In Eq. (1), K is the frequency independent gain and s is the Laplace variable. This form of compensator enables the gain of the open-loop system to be made high in the frequency region where attenuation is wanted while the phase recovers to zero at high frequencies. In order to obtain the best possible performance, the compensator parameters have to be optimised.

In the optimisation of the compensator parameters, the objective function to be minimised can be chosen as an energy term representing the amount of acoustic energy at the error microphone. However, for a practical system the minimisation of energy is not the only performance criterion that has to be taken into account. In fact there are two other important factors that need consideration. These are the Nyquist stability criterion and a term associated with fluctuations (uncertainties) in the open-loop frequency response caused by any changes in the physical system. A simple approach to the optimisation problem is to use multi-objective optimisation which enables a clear and easy problem formulation as well as preferences to be entered into the numerical design. The three objectives to be minimised in the compensator design make a vector of objectives which must be traded off in some way.

The Goal-Attainment method [14] is used here since it is very practical and requires less guessing on the part of the designer than other methods. This method involves expressing a set of design goals $\mathbf{f}^*=\{f_1^*, f_2^*, ..., f_m^*\}$ which is associated with a set of objectives $\mathbf{f}(\mathbf{x})=\{f_1(\mathbf{x}), f_2(\mathbf{x}), ..., f_m(\mathbf{x})\}$. The formulation of the problem allows the under or over-achievement of the objectives. This enables the designer to be relatively imprecise about initial design goals. A vector of weighting coefficients, $\mathbf{w}=\{w_1, w_2, ..., w_m\}$ controls the amount of under or over-achievement of the goals and lets the designer select the relative trade-offs between objectives. Before the Goal-Attainment method is used, the objective functions that determine the performance of the feedback system have to be defined. The three objective functions are terms related to the energy at the error microphone, the Nyquist stability criterion and the stability margins.

Energy-related objective function

In the optimisation of the compensator parameters, the open-loop transfer function of the control system without the compensator, H, is measured. Using the measurement data and Eq. (1), the energy-objective function can be written as [15]

$$f_1\left(K,\xi_z,\omega_z,\xi_p,\omega_p\right)=\sum_{i=1}^{N}\frac{W_i}{\left|1-C\left(\omega_i\right)H\left(\omega_i\right)\right|^2}, \quad (2)$$

where W_i is a frequency weighting window which allows emphasis at important frequencies.

Stability-related objective function

The second objective function takes into account the stability of the closed-loop system which needs to satisfy the Nyquist stability criterion. For the case at hand and for systems which are stable in open-loop, it states that systems whose open-loop loci do not encircle the (1,0) point in the complex plane will be closed-loop stable. The stability-related objective function can be defined by using an exponential function as [16]

$$f_2\left(K,\xi_z,\omega_z,\xi_p,\omega_p\right)=e^{\left[\alpha\left(\mathrm{Re}_{\max}-\beta\right)\right]}, \quad (3)$$

where Re_{max}, a function of the compensator parameters, is the maximum of the positive intercepts with the real axis of the Nyquist plot (of the compensated open-loop transfer function), and α and β are positive constants adjusted empirically. Typical values used are α=3 and β=0.5 [16], which indicates that the maximum positive intercept of the real axis in the Nyquist plot is desired to be 0.5.

Fluctuation-related objective function

In order to prevent any instability due to any fluctuation in the system response, a fluctuation-related objective function has to be minimised. This term is based on the gain and phase margins chosen as safety limits by which the system behaviour can deviate from a mean behaviour without causing instability. The fluctuation objective function is chosen as [16]

$$f_3(K,\xi_z,\omega_z,\xi_p,\omega_p)=\sum_{j=1}^{M}e^{\frac{\left(\Phi-\left|\phi_j\right|\right)}{\gamma}} \quad (4)$$

where Φ is the predefined phase margin, φ_j are the phase shifts with magnitude less than or equal to the phase margin and γ is a constant which allows the magnitude of f_3 to be adjusted so that it becomes comparable to the values of the two other objective functions when the optimisation is successful. In a practical situation a phase margin of 45° and a gain margin of 6 dB are often used. The weighting constant γ is then chosen to be 45 [16].

The three objectives are minimised simultaneously in order to obtain the optimal coefficients of the second order minimum phase filter. Several such filters can be cascaded together to improve the performance of the control system if needed.

2. EXPERIMENTAL INVESTIGATION

In order to demonstrate the feasibility of feedback control of noise using the controller designed in Section 1, experiments were conducted in an office of volume 4.0×2.9×3.0 m^3. These experiments also allowed a study of the effect of room acoustics on the locations of the sensor and actuator and hence on the control performance. The office is furnished with two desks and two filing cabinets and its floor is covered with carpet. The averaged reverberation time of the room below 200 Hz is about 1.5 seconds. There is a full-size window on one of the walls through which noise is transmitted from a nearby workshop. In the experiments, the primary noise was either generated internally by a loudspeaker standing next to the window or transmitted into the room from the workshop. The control loudspeaker was located at the opposite end of the room. Figure 2 shows a typical plot of the magnitude of the low frequency acoustic response of the room. It can be seen that the room modes (or room resonances) are all well damped except for the first mode at 42 Hz. Thus, according to Nelson and Elliott [17], global noise attenuation with a single control source is only possible around that frequency region. Or quantitatively, the required control bandwidth is about 50 Hz.

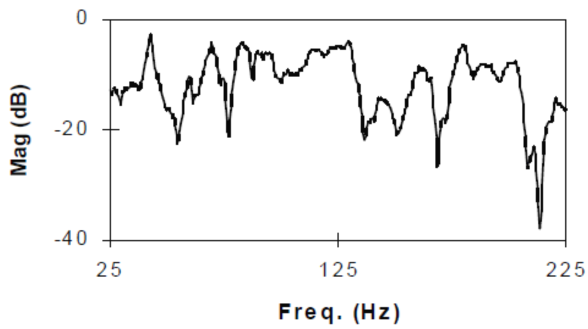

Figure 2 Typical acoustic frequency response of the room.

The performance of the control system is evaluated based upon the sound pressure measurements. The sound pressure spectra were measured at 15 locations distributed evenly along the diagonal line of the room. The measured spectra were then averaged over these 15 locations to form a global index. The comparison of the index with and without control indicates the global control performance of the system.

The locations of the control loudspeaker and error microphone are always important for effective control. It is well known that in order to meet the requirements of controllability and observability, the control loudspeaker and error microphone should not be located on the node-lines of those room modes to be controlled. However, in order to have effective global attenuation with feedback control, other considerations are also required. For instance, in order to minimise the effect of control spillover, it is desirable to have the control loudspeaker located on the node-lines of the room modes which cannot be controlled.

One of the issues to be investigated in the experiments is the relationship between stability, control bandwidth and global controllability. From the stability and control bandwidth point of view, the error microphone should be placed as close as possible to the control loudspeaker. However, this often leads to local control rather than the required global control due to a very strong near field in the vicinity of the control loudspeaker. This can be illustrated by the following examples.

In the first example, the error microphone was located 13 cm away from the control loudspeaker. Figure 3 shows the open-loop frequency response of the uncompensated system of this arrangement. It can be seen that the two phase cross-overs (phase cross-over being 0° in the convention adopted here) in the frequency region of interest are well apart (30 and 740 Hz). This provided a good margin for the compensator design, as the required control bandwidth is merely 50 Hz. As a result, it was possible to design a compensator consisting of two second order filters cascaded together.

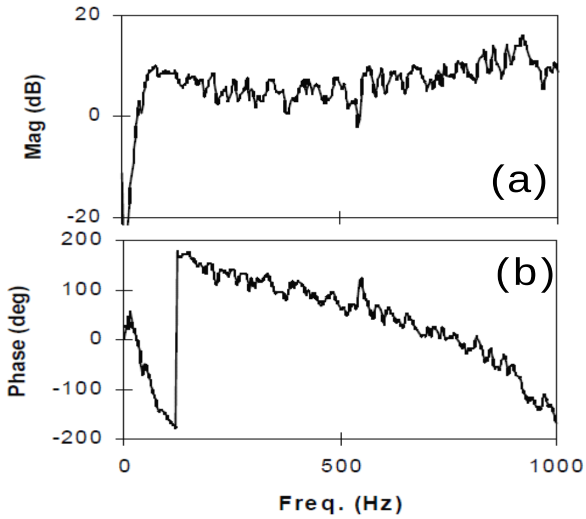

Figure 3 Magnitude (a) and phase (b) of the measured open-loop frequency response of the uncompensated system in Example 1.

Figure 4 shows the control result achieved from the 4th order compensator design. The primary noise in this example was generated by a loudspeaker inside the room. As expected, very good attenuation was obtained over the frequency range from 25 to 75 Hz at the location of the error microphone (see Fig.4.a). Around 42 Hz, an attenuation of more than 30 dB can be seen. However, because the error microphone was very close to the control loudspeaker the near field dominated the sound field and significant global attenuation was not achieved (see Fig.4.b). This is because in the near field the pressure and particle velocity are in quadrature and the sound power radiated is not necessarily reduced by minimising the sound pressure at the error microphone. In fact, it is the radiated sound power that affects the global control result and this is best minimised by an error microphone further from the control loudspeaker. It is also important that the error microphone is not too far so that the reverberant field dominates the measured sound field. The following examples illustrate these points.

In the second example, the error microphone was moved 28 cm away from the control loudspeaker. Figure 5 shows the open-loop frequency response of the uncompensated system.

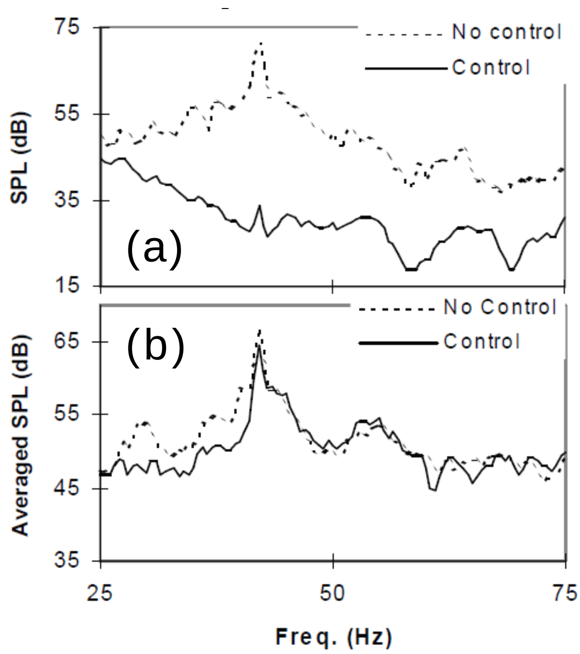

Figure 4 Control results with the error microphone 13 cm from the control loudspeaker. (a) SPL at the error microphone, (b) Averaged SPL in the room.

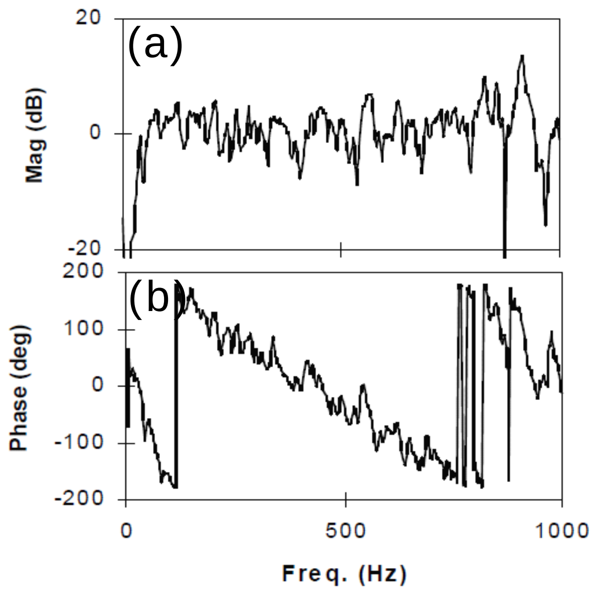

Figure 5 Magnitude (a) and phase (b) of the measured open-loop frequency response of the uncompensated system in Example 2.

It can be seen that the frequency span between the phase cross-overs (30 and 380 Hz) becomes smaller but nevertheless is still wide enough to accommodate a compensator consisting of two second order filters. Thus, good attenuation of more than 10 dB was still obtained over the frequency range from 30 to 65 Hz at the location of the error microphone (see Fig.6.a). As the error microphone was now further away from the control loudspeaker, the near field played little part in the sound field at the error microphone. In this case the radiated sound power is minimised. Consequently, global attenuation was obtained over the frequency range from 30 to 45 Hz (see Fig.6.b). Around 42 Hz, a global attenuation of more than 10 dB was achieved. Figure 7 shows the control result using the same configuration but with the primary noise coming from the nearby workshop through the closed window. Again, good attenuation was obtained at the location of the error microphone and some global attenuation was achieved below 45 Hz.

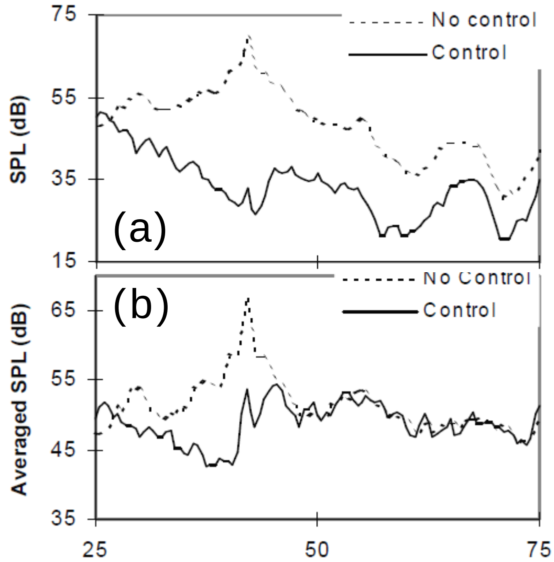

Figure 6 Control results with the error microphone 28 cm from the control loudspeaker. (a) SPL at the error microphone, (b) Averaged SPL in the room.

Moving the error microphone further away from the control loudspeaker can possibly extend the bandwidth of global attenuation to a higher frequency. However, this extension is limited by two factors.

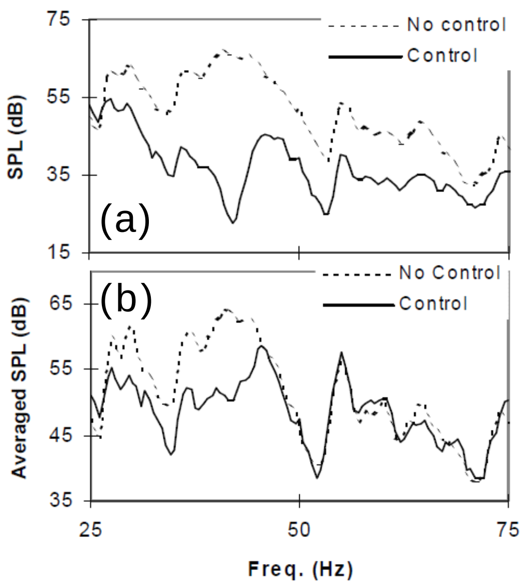

Figure 7 Control results with the error microphone 28 cm from the control loudspeaker and the primary noise from the nearby workshop. (a) SPL at the error microphone, (b) Averaged SPL in the room.

First, the bandwidth is confined by the acoustic characteristics of the room (e.g. modal overlap). In this particular case, the upper frequency limit of global attenuation achievable with a single control source is about 50 Hz. Secondly, as the error microphone is located further away from the control loudspeaker, the stable bandwidth (the frequency span between phase cross-overs) of the uncompensated system decreases. This will reduce the margin for the compensator design thereby limiting the control bandwidth of the compensator and its achievable attenuation as well. This can be illustrated by a last example.

In this example, the error microphone was located 170 cm away from the control loudspeaker, thereby eliminating the near field effect of the loudspeaker on the control. Figure 8 shows the open-loop frequency response function of the uncompensated system.

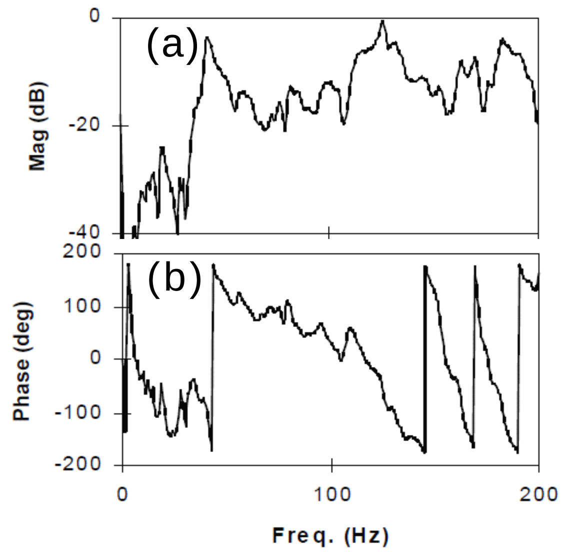

Figure 8 Magnitude (a) and phase (b) of the measured open-loop frequency response of the uncompensated system in Example 3.

It can be seen that the frequency span between the phase cross-overs (7 and 104 Hz) became smaller. This greatly reduced the margin for the compensator design. In this case, the order of compensator had to be limited to two to have a reasonable result.

Figure 9 shows the control result obtained from the 2nd order compensator design. It is clear that the bandwidth and the amount of attenuation were greatly reduced at the location of the error microphone, compared with Figs. 4.a and 6.a. However, as far as global attenuation is concerned, the result was still significant. The bandwidth and the amount of attenuation were quite similar to those at the location of the error microphone. Indeed, some global attenuation can now be seen between 40 and 50 Hz. The lack of global attenuation at lower frequencies is clearly due to the fact that the bandwidth of the uncompensated system is not wide enough. Between 50 to 60 Hz an increase of noise level can be noticed. This can be expected from practical feedback control systems around the phase cross-over frequencies of the compensated system as a direct result of Bode's integral theorem and spillover [18]. The increase of noise can be reduced by reducing the overall gain in the open-loop frequency response. However, this will also reduce the peak noise reduction as well as the control bandwidth.

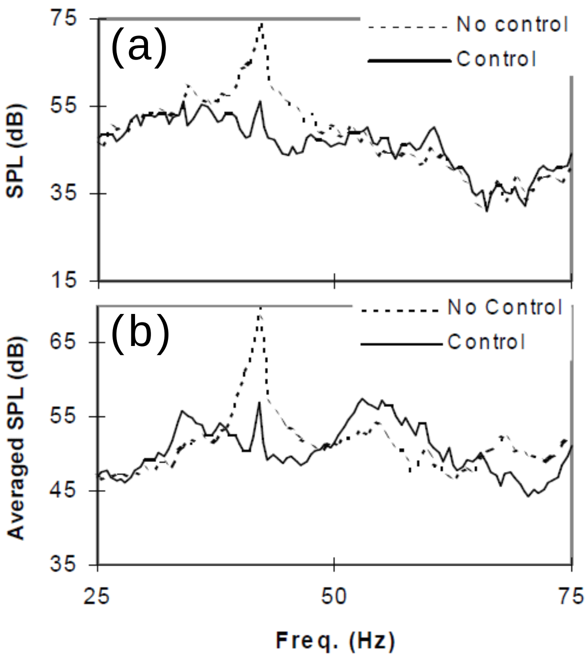

Figure 9 Control results with the error microphone 170 cm from the control loudspeaker. (a) SPL at the error microphone, (b) Averaged SPL in the room.

3. CONCLUSIONS

The single channel feedback control system presented in this paper can achieve a global noise reduction of the order of 10 dB. However, the control bandwidth depends on the relative position between the error microphone and control loudspeaker.

When the error microphone was placed 13 cm away from the loudspeaker significant global noise reduction was not achieved. This is despite excellent local control at the error microphone. In this case the near field dominates the sound field and minimising the sound pressure did not lead to attenuation of the radiated sound pressure in the far field.

When the error microphone is moved 28 cm from the loudspeaker the effect of the near field on the sound field is reduced. The radiated sound power is now minimised and both local and global control can be obtained. The peak attenuation at 42 Hz was about 10 dB and the control bandwidth between 30 to 45 Hz.

Finally when the error microphone is moved further away at 170 cm from the control loudspeaker the effect of the near field is eliminated. A peak global reduction of about 10 dB is achieved and the control bandwidth reduced to between 40 and 50 Hz with small increases adjacent to this band.

REFERENCES

[1] J.B. Bisnette, A.K. Smith, J.S. Vipperman, D.D. Bundy, "Active noise control using phase-compensated, damped resonant filters", *Journal of Vibration and Acoustics* **128(2)**, 148–155 (2006)

[2] S.A. Lane, R.L. Clark, S.C. Southward, "Active control of low frequency modes in an air-craft fuselage using spatially weighted arrays", *Journal of Vibration and Acoustics* **122(3)**, 227–234 (2000)

[3] C.K. Song, J.K. Hwang, J.M. Lee, J.K. Hedrick, "Active vibration control for structural–acoustic coupling system of a 3-D vehicle cabin model", *Journal of Sound and Vibration* **267(4)**, 851–865 (2003)

[4] S.A. Lane, M. Johnson, C. Fuller, A. Charpentier, "Active control of payload fairing noise", *Journal of Sound and Vibration* **290(3–5)**, 794–819 (2006)

[5] C. H. Hansen and S. D. Snyder, *Active Control of Noise and Vibration*, E. & FN Spon, Lon don (1997)

[6] S. Griffin, C. Hansen and B. Cazzolato, "Feedback control of structurally radiated sound into enclosed spaces using structural sensing", *J. Acoust. Soc. Am.* **106(5)**, 2621-2628 (1999)

[7] T. Samejima, "Modifying modal characteristics of sound fields by state feedback control", *J. Acoust. Soc. Am.* **110(3)**, 1408-1414 (2001)

[8] J. Yuan, "Virtual sensing for broadband noise control in a lightly damped enclosure", *J. Acoust. Soc. Am.* **116(2)**, 934-941 (2004)

[9] M. Al-Bassyiouni and B. Balachandran "Control of enclosed sound fields using zero spill-over schemes", *Journal of Sound and Vibration* **292**, 645–660 (2006)

[10] C. Hong and S. J. Elliott, "Local feedback control of light honeycomb panels", *J. Acoust. Soc. Am.* **121(1)**, 222-233 (2007)

[11] M.A. Creasy, D.J. Leo and K.M. Farinholt, "Adaptive positive position feedback for actively absorbing energy in acoustic cavities", *Journal of Sound and Vibration* **311**, 461–472 (2008)

[12] L.P.R. de Oliveira, M.M. da Silva, P. Sas, H. Van Brussel and W. Desmet, "Concurrent mechatronic design approach for active control of cavity noise", *Journal of Sound and Vibration* **314**, 507–525 (2008)

[13] C. H. Hansen and S. D. Snyder, *Active Control of Noise and Vibration*, E. & FN Spon, London (1997)

[14] F. W. Gembicki, *Vector optimisation for control with performance and parameter sensitivity indices*, Ph.D Thesis, Case Western Reserve Univ., Cleveland, Ohio (1974)

[15] R. Paurobally, *A study of feedback control of noise in enclosures*, Ph.D Thesis, The University of Western Australia (1997)

[16] C. Carme, *Absorption acoustique active dans les cavités*. Ph.D Thesis, Université D'Aix-Marseille II (1987)

[17] P. A. Nelson and S. J. Elliott, *Active Control of Sound*, Academic Press, London (1992)

[18] J. Hong and D.S. Bernstein, "Bode integral constraints, colocation, and spillover in active noise and vibration control", *IEEE Transaction on Control Systems Technology* **6(1)**, 111–120 (1998)