FIFTH INTERNATIONAL CONGRESS ON SOUND AND VIBRATION
DECEMBER 15-18, 1997
ADELAIDE, SOUTH AUSTRALIA

FEEDBACK CONTROL OF NOISE IN A ROOM

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Abstract

Active noise control technology is an attractive solution for attenuation of low frequency noise in enclosures. In terms of control strategies, feedforward control has often been 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. For some applications such as attenuation of random noise in office spaces or vehicle cabins, such reference signals are either not available or very expensive to obtain. In these situations, feedback control can be an alternative solution. In this paper, a single channel feedback control system for global noise attenuation in a room is presented. The controller is designed based on the compensation filter approach of the classical control theory. Experimental results are also presented.

1. INTRODUCTION

Active noise control technology is an attractive solution for attenuation of low frequency noise in enclosures. The most successful applications include the control of propeller noise in the passenger cabins of aircraft and the control of engine-induced noise inside cars. 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. For some applications such as attenuation of random noise in office spaces or vehicle cabins, such reference signals are either not available or very expensive to obtain. In these situations, feedback control can be an alternative solution.

In this paper, a practical feedback control system for noise attenuation in a room is presented. The control system is designed based on the classical control theory, as the system is to be implemented with simple analogue electronic circuits. The objective of the paper is to investigate practical issues when applying the classical feedback control theory to noise attenuation in rooms. The relation between stability, control bandwidth and global controllability is studied. Experimental results are also presented.

2. 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 and a single microphone.

The heart of the control system is the controller where the control signal is generated. 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 the classical control theory to a more advanced state-variable approach based on the modern control theory. In this paper, a compensation filter approach based on the classical control theory is employed, as it is easy to implement with cheaper analogue circuits.

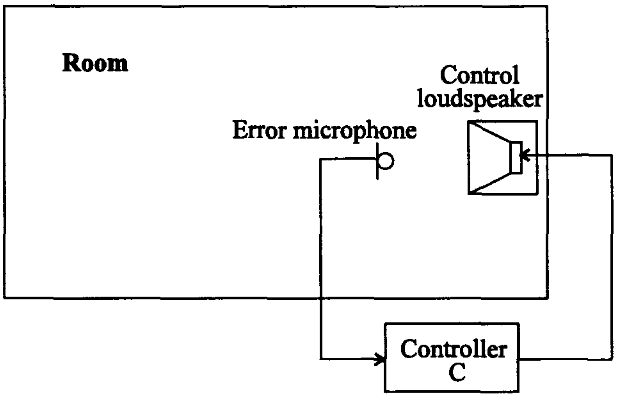


Fig.1. Feedback control system.

The compensator to be used as the controller has a basic form of

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

This form of compensators enables the gain 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 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 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 [1] 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 $\mathbf{f}^* = \{f_1^*, f_2^*, \dots f_m^*\}$ which is associated with a set of objectives $\mathbf{f}(\mathbf{x}) = \{f_1(\mathbf{x}), f_2(\mathbf{x}), \dots f_m(\mathbf{x})\}$. 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, $\mathbf{w} = \{w_1, w_2, \dots w_m\}$ 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 [2]

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

where W_i is a frequency weighting window which allows emphasis of more 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 defined by using an exponential function as [3]

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

where $\mathrm{Re}_{\max}$, a function of the compensator parameters, is the maximum of the positive intercepts with the real axis, and α and β are positive constants adjusted empirically. Typical values used are α=3 and β=0.5, 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 an instability. The fluctuation objective function is chosen as [3]

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

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 often chosen to be 45.

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.

3. EXPERIMENTAL INVESTIGATION

In order to study the practical issues such as sensor and actuator arrangement when employing the feedback controllers designed with the method presented in Section 2, experiments were conducted in a realistic situation. The control system was set up in an ordinary office of a volume of $4.0\times2.9\times3.0$ m^3. The office is furnished with two desks and two 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 which subjects the noise transmission 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 form the workshop. The control loudspeaker was at the opposite end of the room.

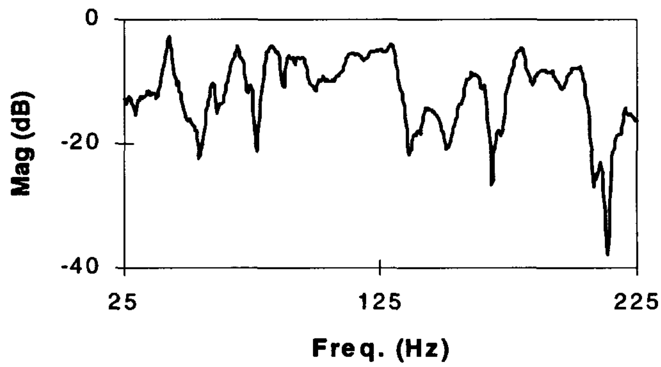


Fig.2. Typical acoustic frequency response of the room

Figure 2 shows a typical acoustic frequency response of the room. It can be seen that the room modes (or room resonances) are all well damped except the first mode at 42 Hz. Thus, according to Nelson and Elliott [4], 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.

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 and formed 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 relation 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 the local control rather than the required global control due to a very strong direct field in the vicinity of the control loudspeaker. This can be illustrated by the following examples.

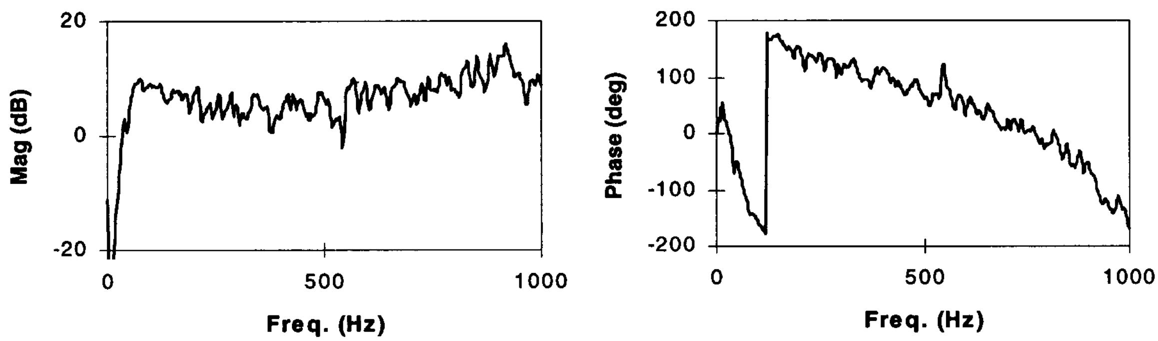
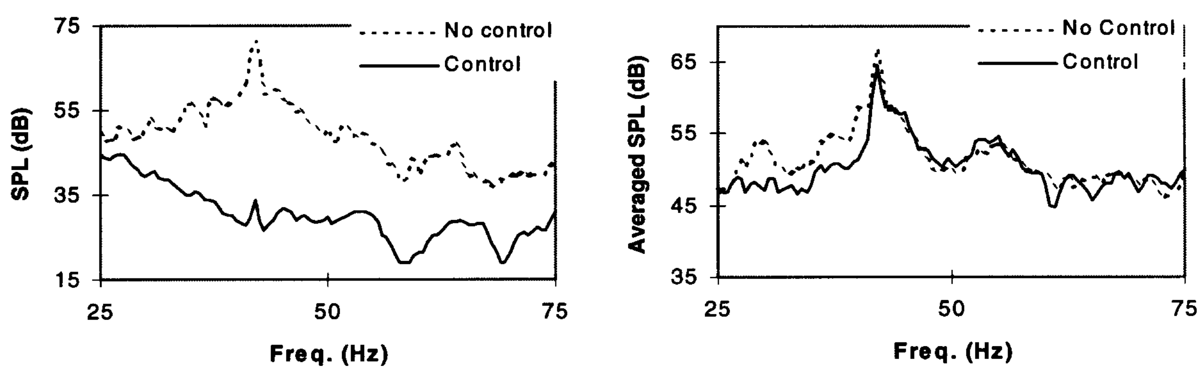


Fig.3. Measured open-loop frequency response of the uncompensated system in Example 1.



(a) SPL at the error microphone

(b) Averaged SPL in the room

Fig.4. Control results with the error microphone 13 cm from the control loudspeaker.

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

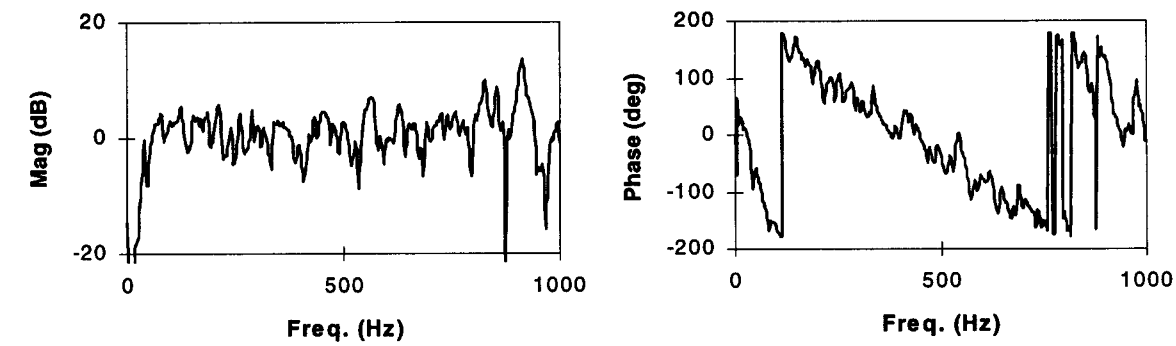
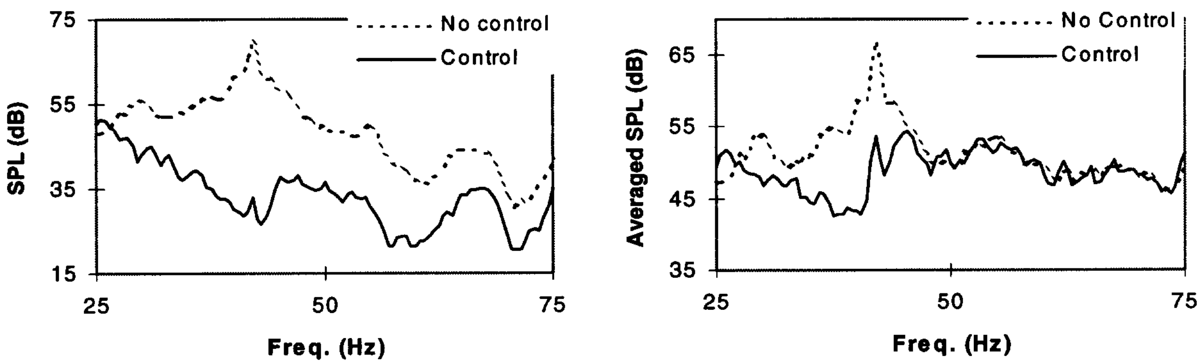
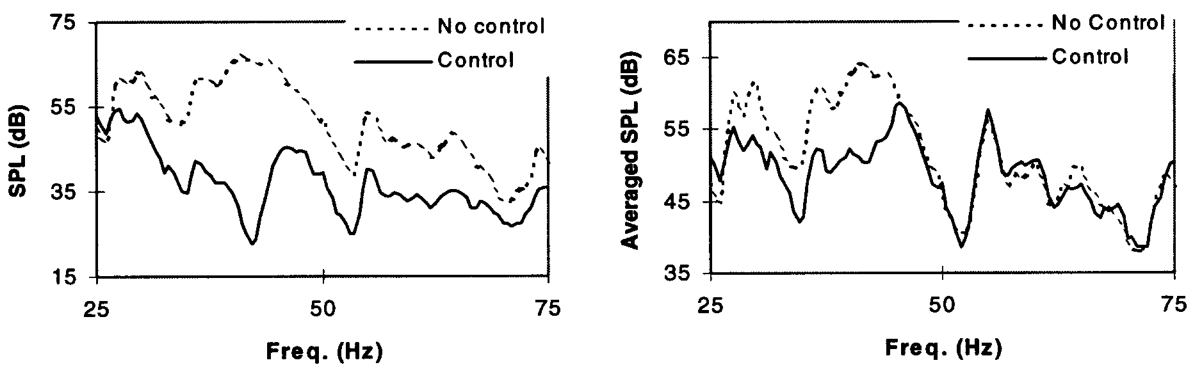


Fig.5. Measured open-loop frequency response of the uncompensated system in Example 2.



(a) SPL at the error microphone (b) Averaged SPL in the room

Fig.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

Fig.7. Control results with the error microphone 28 cm from the control loudspeaker and the primary noise from the nearby workshop.

Figure 4 shows the control result achieved from the above compensator design. The primary noise in this example was generated by the loudspeaker. 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, attenuation of more than 30 dB can be seen. However, because the error microphone was very close to the control loudspeaker, the output of the loudspeaker which attenuates the sound pressure at the error microphone was so small that it had little impact on the sound field elsewhere. Consequently, global attenuation was not achieved (see Fig.4.b).

In the second example, the error microphone was moved away from the control loudspeaker and the distance between the two became 28 cm. Figure 5 shows the open-loop frequency response function of the uncompensated system. 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 farther away from the control loudspeaker, the output of the loudspeaker which attenuates the sound pressure at the error microphone had some impact on the sound field elsewhere. Consequently, global attenuation was obtained over the frequency range from 30 to 45 Hz (see Fig.6.b). Around 42 Hz, 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. Again, good attenuation was obtained at the location of the error microphone and some global attenuation was achieved below 45 Hz.

Moving the error microphone further away from the control loudspeaker will extend the bandwidth of global attenuation to a higher end. However, this extension is limited by two factors. First, the bandwidth is confined by the acoustic characteristics of the room (eg, modal overlap). In this particular case, the upper frequency limit of global attenuation achievable with a single control source is 50 Hz. Secondly, as the error microphone moves 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 the example, the error microphone was located 170 cm away from the control loudspeaker thereby eliminating the direct field effect of the loudspeaker on the control. Figure 8 shows the open-loop frequency response function of the uncompensated system. It can be seen that the frequency span between the phase cross-overs (7 and 104 Hz) becomes very small. This greatly reduced the margin for the compensator design. In this case, the order of compensator has to be limited to two to have a reasonable result. Figure 9 shows the control result obtained from the compensator design. It can be seen 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 not too bad. The bandwidth and the amount of attenuation were quite similar to those at the location of the error microphone. And indeed, some global attenuation can now be seen beyond 45 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.

4. CONCLUSIONS

A single channel feedback control system for global noise attenuation in rooms has been presented. The controller has been designed based on the compensation filter approach of the classical control theory and implemented with analogue electronic circuits.

In the controller design, a multi-objective optimisation approach has been employed. This is a numerical design method which enables a clear and easy problem formulation and delivers a good result without much involvement from the designer.

The relation between stability, control bandwidth and global controllability has been investigated experimentally. It has been shown that the proper sensor arrangement in relation to the control source plays a vital part in achieving the satisfactory global control.

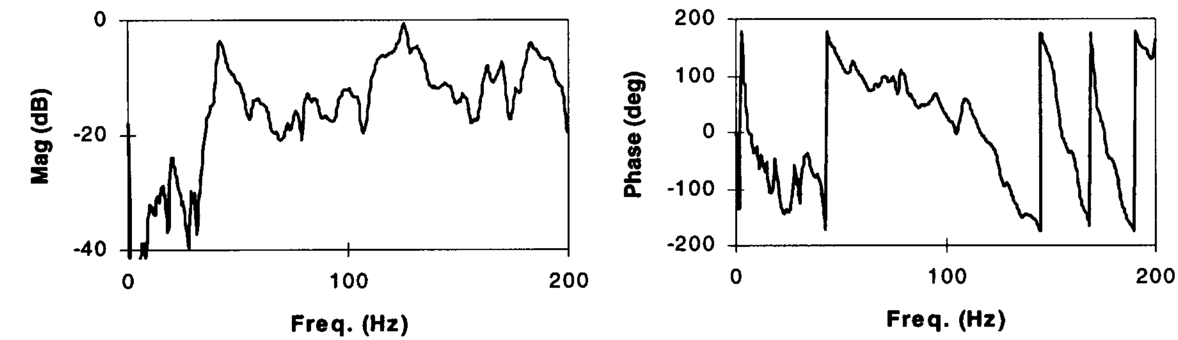
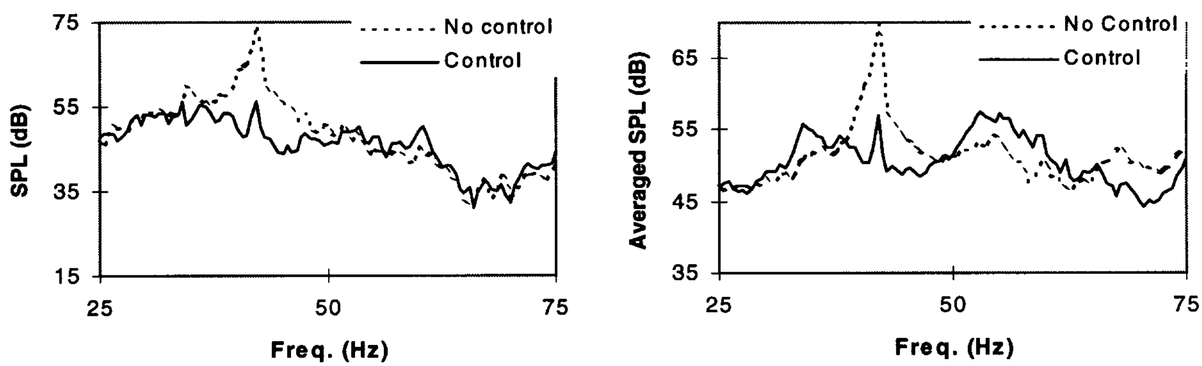


Fig.8. Measured open-loop frequency response of the uncompensated system in Example 3.



(a) SPL at the error microphone (b) Averaged SPL in the room

Fig.9. Control results with the error microphone 170 cm from the control loudspeaker.

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