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SUPPRESSING ROOM MODES USING ACTIVE ABSORBERS

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ABSTRACT

The acoustic modes of built spaces can be suppressed by the introduction of damping materials. Unfortunately, the performance of traditional acoustic absorbers is very poor at low frequencies, such that the first few modes cannot be effectively treated. This paper describes the application of an active acoustic absorber to the control of acoustic modes in a room. A brief treatment of the theory of active absorbers is presented to establish optimal low frequency performance of a circular active absorber, built around a conventional loudspeaker. The performance of an absorber controlled by a simple sub-optimal control system is then investigated theoretically. The theoretical material is supported by experimental evidence demonstrating the control of the first acoustic mode of a highly reverberant room. The prototype hardware is described in order to demonstrate the commercial viability of the concept.

INTRODUCTION

The active control of sound and vibration is now an established commercial engineering technique. The active manipulation of the boundaries of acoustic spaces remains, however, an emerging technology, with many potentially attractive applications in noise control, architectural acoustics etc.. Although the principle of the active absorption of acoustic energy at a surface is a well understood, practical realisations of the concept have been few and have been principally motivated by simple acoustical applications. It is the purpose of this paper to demonstrate that it is possible using contemporary technology to construct active absorbers which offer useful performance in controlling the low frequency modes of a room.

The experimental data in this paper was collected from active absorbers controlled by a simple sub-optimal control law. In order to establish a benchmark against which to compare the performance of these practical systems, the behaviour of theoretical optimal active absorbers is reviewed below.

FUNDAMENTALS OF ACTIVE ABSORPTION

A typical active absorber system is illustrated by the block diagram of Figure 1. The system is designed around a conventional low frequency electrodynamic loudspeaker, the motion of which is transduced by an accelerometer attached to the cone. The acoustic pressure at the diaphragm is detected by a microphone and some implementations require that a second microphone is provided to resolve incident pressure waves.

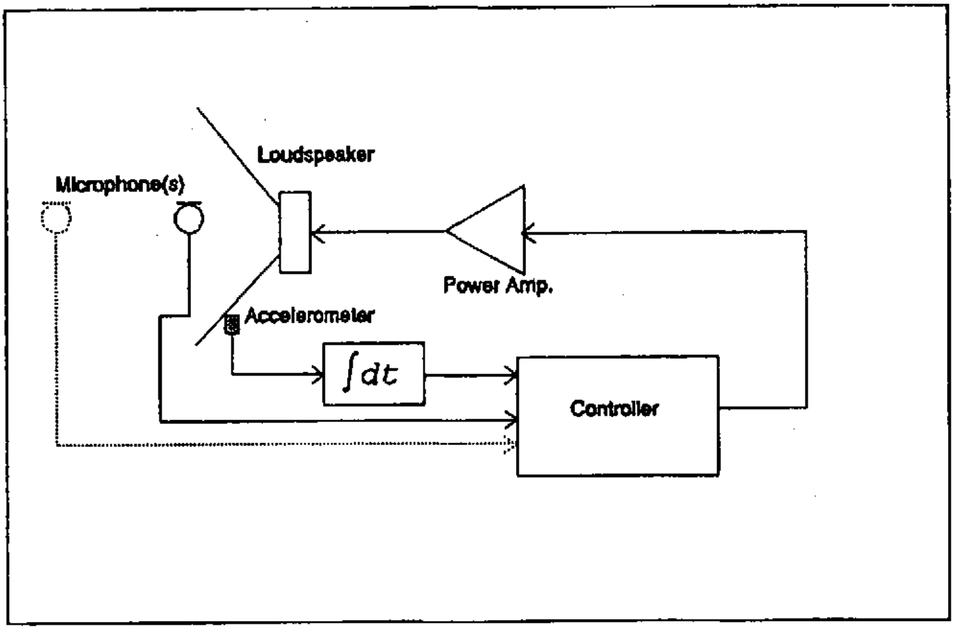

Figure 1 Block Diagram of an Active Acoustic Absorber

The measures of acoustic variables at and near the surface of the absorber are used as the inputs to a control system which produces an output signal designed to force the loudspeaker into motion such that the desired impedance is achieved at the surface. It has been demonstrated by a number of workers that the controller can be a fixed or adaptive system, using any one of several alternative structures [1,2,3]. Details of the controller are not discussed in the present paper.

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A Simple One Dimensional Application

Operation of the acoustic absorber is easily understood in the simple case when the system terminates an acoustic waveguide of cross sectional area equal to the active surface area of the absorber. Acoustic plane waves, lowpass limited to frequencies below the "plane wave cut-off" frequency of the waveguide, are normally incident upon the active surface of the absorber. The total acoustic pressure at the absorber's surface has three components;

incident pressure, $p_{incident}$
blocked reflected pressure, (which equals $p_{incident}$)
radiated pressure, caused by motion of the loudspeaker.

Specifically, the pressure at the surface is:

$$p_{total} = 2p_{incident} + Z_{rad}\,u(t) \qquad (1)$$

where Z_{rad} is the radiation impedance presented to the absorber by the waveguide (which would, for example, have magnitude $\rho_0 c$ for an anechoic waveguide) and u(t) is the velocity of the loudspeaker diaphragm.

The impedance at the surface of the absorber is:

$$Z = \frac{2p_{incident} + Z_{rad}u(t)}{u(t)} \qquad (2)$$

By rearranging equation 2 it is possible to define control laws, relating the cone velocity to acoustic pressures, which force the impedance at the surface to the characteristic value appropriate to an absorber in this application. If it is possible to transduce the incident pressure $p_{incident}$, then:

$$u(t) = \frac{2p_{incident}}{(\rho_0 c + Z_{rad})} \qquad (3)$$

If a single microphone is used to transduce total pressure, then:

$$u(t) = \frac{p_{total}}{\rho_0 c} \qquad (4)$$

Note that, *in the special case of an absorber terminating an anechoic waveguide with normally incident plane waves*, the control laws 3 and 4 are equivalent as $p_{total} = p_{incident}$ when the active termination implements characteristic specific acoustic impedance.

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A Three Dimensional Application - Absorption in a Room

If the same active absorber is applied to the control of acoustic modes in rooms a more generalised geometry should be considered. The lowest frequency modes of a typical room have antinodal pressure planes at the walls (which shall be assumed to be perfectly reflecting). If such a wall is insonified by plane harmonic waves associated with a mode between two parallel walls of a rectangular space the it would be appropriate to install active absorbers on this wall.

If it were feasible to cover the entire wall with active absorbing elements then the problem would reduce to the 1 dimensional situation described above, with the array of absorbing elements generating a plane radiated field and the control laws 3 & 4 describing the operation of each absorber element. In a more realistic installation a smaller number of discrete absorbers would be used, distributed across the wall's surface. These devices would not generate a plane radiated field and consequently the control laws 3,4 would no longer be appropriate. In order to investigate the control laws required in these more realistic 3-dimensional applications, the system depicted in Figure 2 is considered.

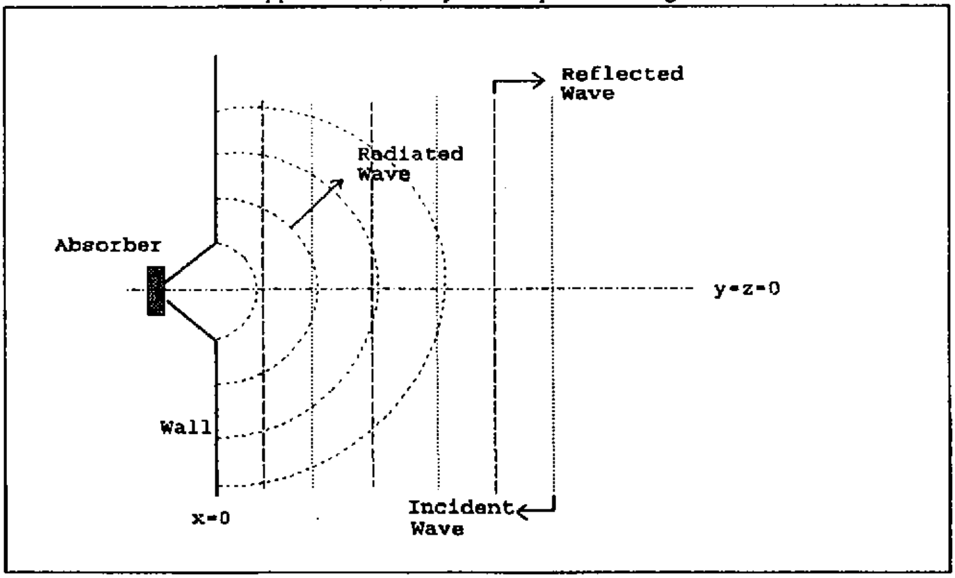

Figure 2 Circular Active Absorber on a Plane Hard Wall with Normally Incident Plane Waves

In Figure 2 a single finite source is placed in the middle of a plane perfectly reflecting wall. The wall is located at x=0 and the circular source is centred at x=y=z=0. The pressure field to the right of the wall is described by three components. Plane acoustic waves are normally incident on the wall:

$$p_{incident}(x,t) = P_{incident}e^{j(\omega t+kx)} \tag{5}$$

causing a reflected blocked pressure field which is also plane

$$p_{reflected}(x,t) = P_{incident}e^{j(\omega t-kx)} \tag{6}$$

Motion of the loudspeaker causes a radiated field which, at the low frequencies of interest, is approximately a hemispherical wave:

$$p_{radiated}(r,t) = \rho_0 c\ U\ \frac{a}{r}\ \cos(\theta_a)\ e^{j(\omega t-k(r-a)+\theta_a)} \tag{7}$$

where U is the complex velocity amplitude, a is the absorber radius, r is the distance from the absorber centre:

$$r = \sqrt{x^2 + y^2 + z^2}$$

and θ describes the normalised frequency:

$$\theta_a=\tan^{-1}(\frac{1}{ka}) \tag{8}$$

In this idealised 3 dimensional situation the total acoustic pressure across the surface of the absorber has value:

$$p_{total}=2p_{incident}+\rho_0 c\ U\ [\cos^2(\theta_a)+j\cos(\theta_a)\sin(\theta_a)]\ e^{j\omega t} \tag{9}$$

Note that, as the wall is perfectly reflecting, all acoustic energy dissipation occurs over the surface of the absorber. As the absorber is assumed to be moving with constant velocity over its surface and the pressure at the surface is assumed constant (9), the power flow into the absorber is given by:

$$W_{absorber} = \frac{1}{2}\ \pi a^2\ \Re e[\ p^*_{total}\cdot\ U\ ] \tag{10}$$

where the asterisk denotes conjugation. Substituting for the total pressure gives :

$$W_{absorber} = \frac{1}{2}\pi a^2 \Re e[(2P_{incident}+\rho_0 c\ U\ (\cos^2(\theta_a)+j\cos(\theta_a).\sin(\theta_a))^*.U] \qquad (11)$$

Optimal Controller for the Active Absorber

The optimal controller for the active absorber is obtained by designing the cone velocity to maximise power flow into the absorber, a condition defined by the negative extremum of eqn 11. If the controller derives the cone velocity U from linear operations on the incident pressure :

$$U = H.P_{incident} \qquad (12)$$

then the acoustic power at the absorber reduces to:

$$W_{absorber} = \frac{\pi a^2}{2}\ |P_{incident}|^2[\ 2\ \Re e(H)\ +\ \rho_0 c\ \cos^2(\theta_a)\ \ |H|^2] \qquad (13)$$

Limiting the control law H to be a real scalar does not limit the power which can be absorbed at a single frequency and yields:

$$W_{absorber} = \frac{\pi a^2}{2}|P_{incident}|^2[2.H + \rho_0 c\ \cos^2(\theta_a)\ \ H^2\ ] \qquad (14)$$

from which it is possible to calculate the scalar controller which results in maximum acoustic power absorption :

$$H_{opt} = \frac{-1}{\rho_0 c.\cos^2(\theta_a)} \qquad (15)$$

The maximum power which can be absorbed is :

$$W_{opt} = H_{opt}\ \frac{\pi a^2}{2}|P_{incident}|^2 \qquad (16)$$

Absorption Cross Section of the Optimal Absorber

Given the power absorber by the optimally controlled active absorber above, it is useful to consider the area of perfect passive absorbent which would be required to remove the same power from the acoustic field - this amounts to calculating the absorption cross section of the active system. The intensity absorbed by a perfect plane absorber from normally incident plane waves is:

$$I_{passive} = \frac{-|P_{incident}|^2}{\rho_0 c} \qquad (Watts/m^2) \tag{17}$$

where the minus sign indicates that power is dissipated in the passive absorber. The equivalent (perfectly absorbing) area of the optimally controlled active system is :

$$Absorption\ Area = \frac{\pi a^2}{2\cos^2\theta_a} \tag{18}$$

which, substituting for cos(θ) from Eqn. 8, gives:

$$Absorption\ Area = \frac{\pi}{2\ k^2\ \sin^2\theta_a}$$

which approaches:

$$Absorption\ Area = \frac{\lambda^2}{8.\pi} \qquad m^2$$

at low frequency.

The efficiency of the absorber is seen to reduce with the square of frequency and, as has been observed by Nelson and Elliott [4], the equivalent perfect absorber in the application described by Figure 2 is a circle of circumference equal to the wavelength divided by the square root of two.

As an example, the area of perfect absorber which absorbs equal power to an ideal active device in the situation described in Figure 2 is plotted, as a function of frequency, as Figure 3. The efficient absorption of low frequency sound is clearly seen in Figure 3, although it is emphasised that the very high levels of power absorption suggested at low frequencies would require very large source strengths in a practical implementation. Such source strengths would not be available from conventional loudspeakers and would consequently limit the low frequency performance of a practical active absorber.

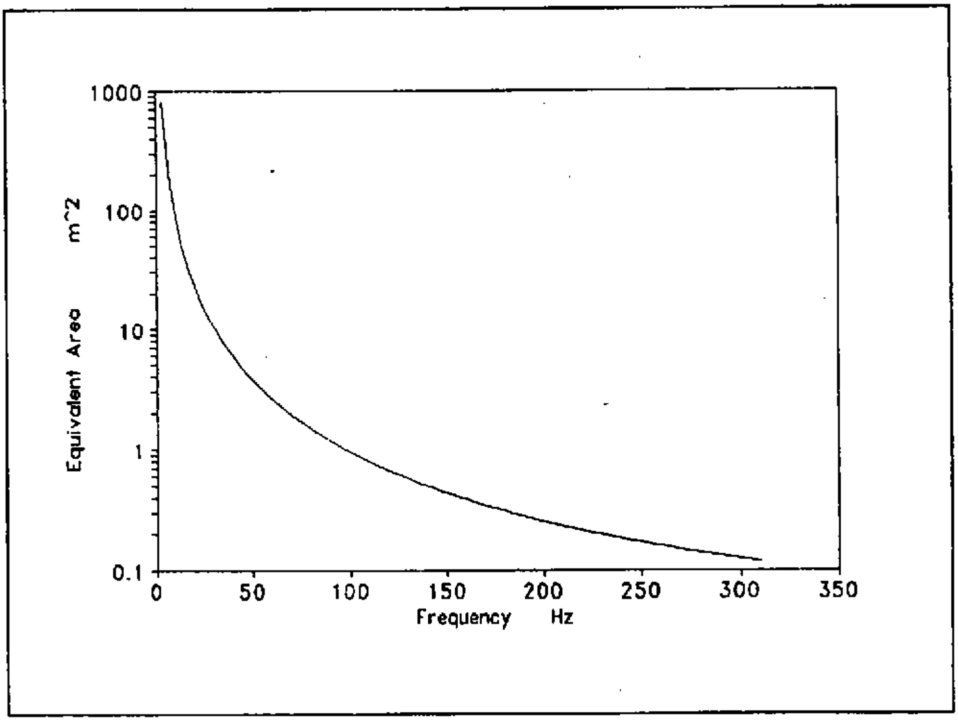

Figure 3 Equivalent Absorbing Area of the Optimal Circular Active Absorber on a Plane Wall

A SUB-OPTIMAL ACOUSTIC ABSORBER

A robust control system which forces the ratio of absorber velocity to total pressure to a real scalar has been developed at the University of Salford and its operation in a number of "one dimensional" applications reported [3,5]. Given the availability of this controller, it is instructive to consider the operation of an absorber system addressing the three dimensional application depicted as Figure 2 using this sub-optimal control strategy.

The sub-optimal real scalar controller is represented by:

$$I_R=\frac{U}{P_{total}} \tag{19}$$

and the resulting power at the surface of the absorber is:

$$W=\frac{\pi a^2}{2}\Re e\,[P^*_{total}\ I_R P_{total}]$$

which can be rearranged to give :

$$W=I_R\frac{\pi a^2}{2}\,|P^2_{total}| \tag{20}$$

The magnitude squared of P_{total} can be calculated by substituting for velocity in equation 9:

$$|P_{total}|^2=\frac{4\,|P_{incident}|^2}{(1-\rho_0 c\ I_R\ \cos^2\theta_a)^2+(\rho_0 c\ I_R\ \cos\theta_a\ \sin\theta_a)^2} \tag{21}$$

The power absorbed is controlled by the value of I_R and the maximum power will be absorbed when I_R takes value:

$$I_{R,opt}=-\frac{1}{\rho_0 c\ \cos\theta_a} \tag{22}$$

The efficiency of the sub-optimally controlled absorber can be assessed by comparing it with the optimally controlled absorber. The ratio of power absorbed by the absorber in the system depicted in Figure 2 controlled by the maximum power value of the sub-optimal real controller to that absorbed when the device is optimally controlled is given by :

$$\frac{W_{R,max.power}}{W_{opt}}=\frac{2\cos\theta_a}{1+\cos\theta_a} \tag{23}$$

The efficiency of the sub-optimal real controller is seen to depend upon frequency and the size of the absorber. As an example, the power ratio is plotted for an absorber radius of 0.08m as a function of frequency as Figure 4.

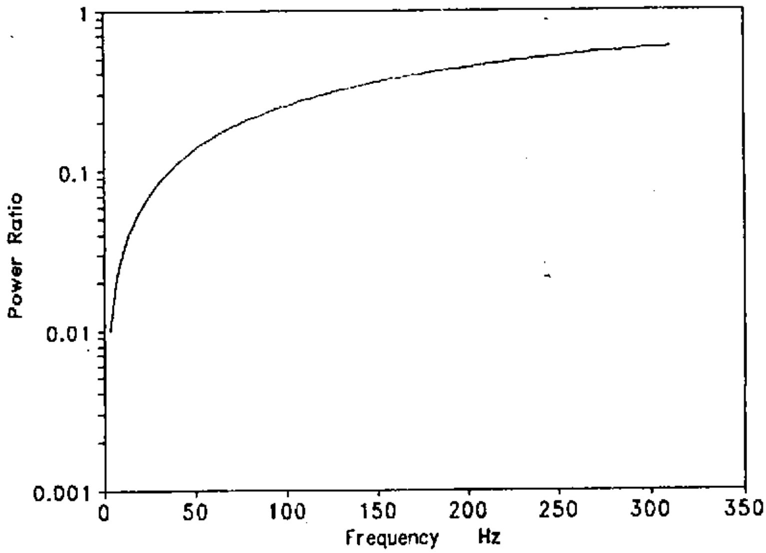

Figure 4 Ratio of Power Absorber by Sub-Optimally Controlled Absorber to that Absorbed by the ideal Optimal Absorber

Although the sub-optimal absorber is not as effective as the optimal system at low frequencies, it is still capable of absorbing such high levels of acoustic power as to materially influence the very low frequency acoustics of a room, as is demonstrated by the following example.

PERFORMANCE OF A PRACTICAL ABSORBER

The performance of a practical realisation of the sub-optimally controlled absorber is illustrated by the following data measured in a small reverberation chamber. The chamber has approximately parallel end walls, with a longest dimension such that the first half-wave resonance occurs at a frequency of approximately 44 Hz. A single loudspeaker was placed in the centre of one end wall and used to excite the mode with constant velocity. A second loudspeaker, of 8 inch nominal diameter, mounted in a closed cabinet, was placed at the centre of the opposite wall and controlled by computer to absorb power.

Direct measures of the quality factor of the mode were made with various values of the real controller I_R. These quality factor measurements were translated into the more familiar reverberation times plotted as Figure 5.

The single 8 inch loudspeaker is seen to be capable of almost halving the reverberation time of the 44 Hz mode, even with the sub-optimal control system.

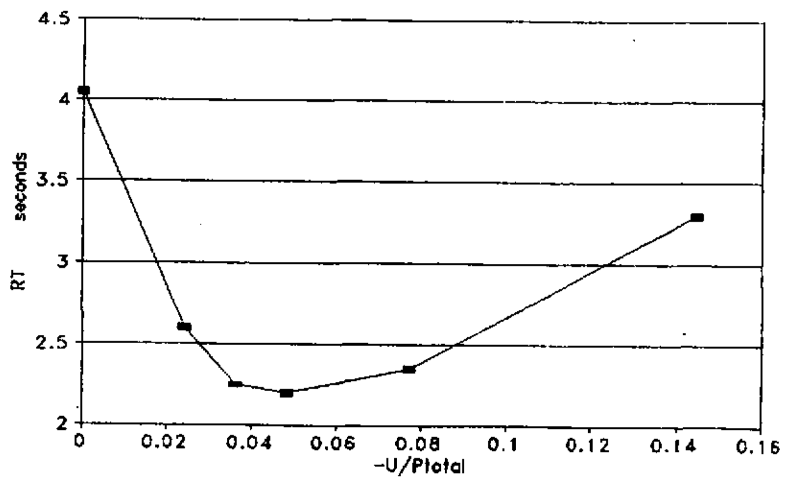

Figure 5 Reverberation Time of the First (44 Hz) Mode of a Room as a function of the Sub-Optimal control Law of an 8" Active Absorber

The greatest reduction in reverberation time in Figure 5 occurs for a value of I_R slightly more negative than the optimum value of

$$I_{R,opt}\big|_{ka=0.0645} = -0.03744$$

This is believed to be due to experimental error, as the power absorbed is not very sensitive to errors in I_R around the optimum value and the quality factors are practically difficult to measure.

In addition to adding the damping required to reduce the reverberation time of a mode, the active absorber causes a significant reduction of the acoustic potential in a mode's reverberant field. Although this effect occurs when the absorber is adjusted to absorb maximum power, greater reductions in averaged pressure levels across a space can be achieved when a single active "absorber" is configured to implement other more explicitly reactive impedances. As these distinctions between active *absorption* and active *cancellation* are rather blurred it is not considered helpful to document sound pressure level reductions in the present paper.

HARDWARE REQUIREMENTS

The prototype active absorber hardware used in the experiments reported in this paper was built around a conventional 8 inch low frequency long throw electrodynamic loudspeaker. The pressure and velocity were transduced using electret devices with subsequent proprietary signal conditioning electronics. The absorber was driven by a conventional audio power amplifier, with input derived from a DSP system resident in a personal computer.

A stand-alone active absorber has been constructed which uses a very high performance floating point DSP chip at a total component prime cost of less than £1000. It is anticipated that a simpler system could be constructed, using a lower performance processor and lower resolution data converters at a prime cost of under £750.

CONCLUSIONS

The performance of an ideal circular absorber in abstracting energy from acoustic plane waves normally incident upon a reflecting wall has been analyzed. An absorber based upon a simple controller has shown to offer a useful fraction of the performance of an optimally adjusted system and its application demonstrated in the context of the control of the first mode of a reverberant room. A relatively cheap system has been proven to be capable of significantly reducing the reverberation time of a space at frequencies where passive treatments offer little or no benefit.

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