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TECHNOLOGICAL CHANGE AND ACTIVE NOISE CONTROL

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1. INTRODUCTION

Technological change results from the triumph of a new technology over an old technology. The growing advantages of the emerging technology gradually cause the strengths of the old methods to become weaknesses. This emergence can be difficult to anticipate since the new technology is often initially inferior to the older, more mature technology. In addition, the potential for improvement of the new technology is frequently not immediately evident and can be of critical importance. The development of active noise control offers an excellent example of several of these concepts. In this paper, the characteristics of passive silencers will be compared to those of active silencers. The relative strengths and weaknesses of each will be discussed. It will be shown that for many applications the most important issues are related to the specific strengths of current passive silencing technology. The future of active noise control will be determined by the rate of improvement in active noise control technology and the degree to which the current strengths of passive silencing are transformed into weaknesses.

2. PRINCIPLES OF CHANGE

The process of change from one paradigm to another or from one technology to another has many common characteristics, regardless of the specific change being considered. Birch and Rasmussen [1] have described this process in a way that is particularly useful and may be summarized as the following:

1) Old ways must reach a crisis
2) New ways must emerge
3) New ways rapidly improve
4) Strengths of the old ways become weaknesses
5) Weaknesses of the new ways become strengths
6) Ultimately, rapid change occurs.

What does this mean for technological change? It means that the inadequacies of the old technology become more clearly recognized as the performance requirements increase and new alternatives become available. These new alternatives, although originally impractical, may have the potential for rapid improvement [2]. This results in the transformation of the former strengths of the old technology into weaknesses and the former weaknesses of the new technology into strengths.

This process is illustrated in Fig. 1. Perceived needs and requirements show a gradual increase over time. The current technology offers certain strengths that meet most of the critical requirements. However, eventually a crisis develops due to the gap between the perceived needs and the capabilities of the old technology. This gap will be tolerated until a new option emerges in the form of a new technology. Initially, this new technology has significant weaknesses compared to the old technology. However, if the rate of improvement of the new technology is sufficiently rapid, a crossover will occur, at which point the former strengths of the old technology are now perceived as weaknesses compared to the strengths of the new alternative technology [3].

An example of this process is illustrated in Fig. 2 for the case of the shift from large, powerful battleships to small, missile-carrying ships. In the past, battleships were valued due to their two primary strengths, which were their resistance to damage and large guns. Small ships were perceived to be weak due to their fragile nature and small guns. However, two trends changed this perspective. An increased perceived value was placed on speed and evasiveness while a new technology emerged in the form of missiles. Ultimately, these trends resulted in a reversal of the strength-weakness

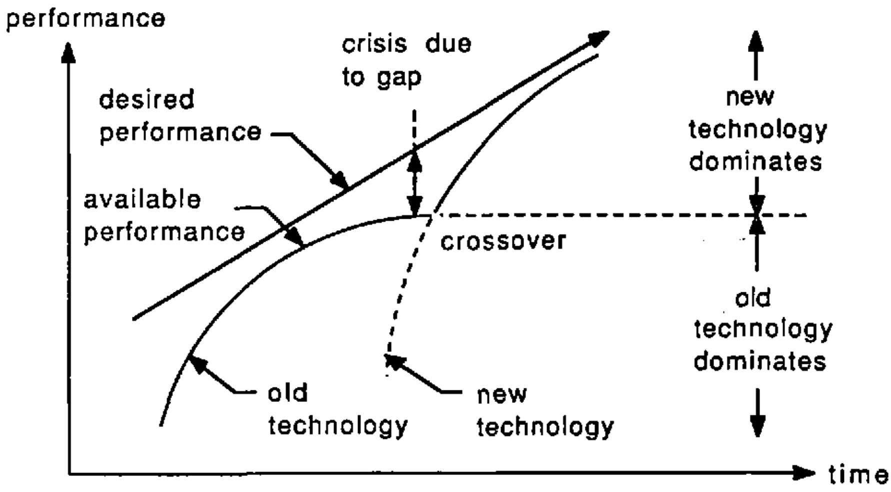

Fig. 1 - Technological performance as a function of time illustrating the crossover that occurs when the weaknesses of a new technology become strengths and the strengths of an old technology become weaknesses.

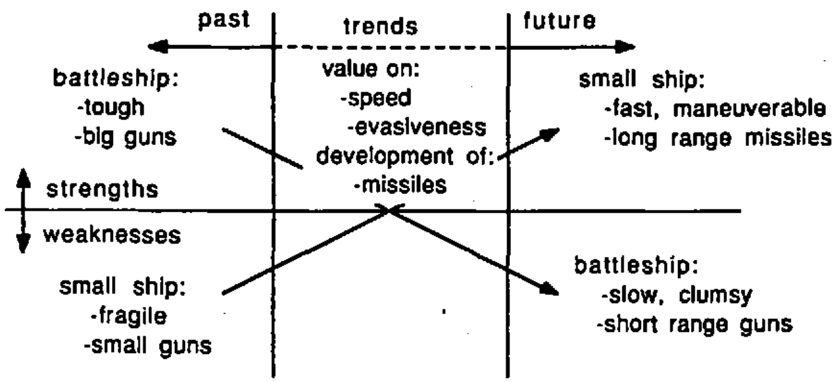

Fig. 2 - Crossover of technologies depends on the recognition of new needs and the improvement of new technology.

relationship. Today, small ships are valued due to their small size and long-range, powerful missiles while battleships are generally considered vulnerable due to their large size, slow speed, and short range guns.

3. PASSIVE SILENCERS

Passive silencers, a mature technology, may be either primarily reactive or resistive [4]. Reactive silencers use the impedance changes caused by a combination of baffles and tubes to silence the undesired sound. They are most commonly used as mufflers on internal combustion engines and in silencers for such devices as rotary blowers. Resistive silencers use the energy loss caused by sound propagation in a duct lined with sound absorbing material to provide the silencing. These types of lined duct or parallel baffle passive silencers are most commonly used in silencers for duct-borne fan noise.

4. ACTIVE SILENCERS

Active silencers, an emerging technology, use an electronically driven loudspeaker to generate a sound cancelling waveform. Major advances in adaptive digital signal processing have enabled the development of powerful new approaches for applying this basic concept. Recently, a complete practical system for active noise control has been described [5,6]. A block diagram of the basic system is shown in Fig. 3. The undesired noise is measured by an input microphone, processed by an IIR adaptive filter, and cancelled by a sound wave generated by a loudspeaker [7,8]. The loudspeaker and error plant characteristics are continually modelled by an independent adaptive filter using a random noise source [9,10]. The transfer function associated with this filter is then copied to the IIR adaptive filter to ensure proper convergence.

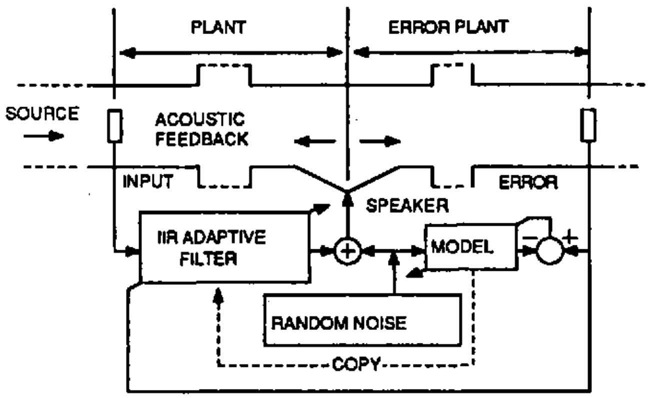

Fig. 3 - Fully adaptive active noise control system [5-10].

5. TECHNOLOGICAL TRENDS

5.1 Overview:

A chart similar to that in Fig. 2 has been prepared in Fig. 4 to illustrate present and possible future trends in active and passive noise control. In the past, the perceived primary strengths of passive silencers have been high attenuation over a broad frequency range, reliable operation, and relatively low purchase cost. Active silencers have not been widely used due to their perceived weaknesses of only low frequency attenuation, unproven reliability, and higher purchase costs.

However, there are several trends under way that may change these perceptions for some applications in the future. First, there is an increased recognition that it may be more important to use silencing approaches that provide a smooth, balanced spectrum, programmable operation, and low system cost. Second, active noise control is an emerging technology that is showing rapid improvement in acoustical performance, overall reliability, and initial cost.

These trends may lead to a reversal in the relative strengths and weaknesses of passive and active silencers as shown for the future in Fig. 4. Active silencers may come to be valued for their strengths in providing a smooth,

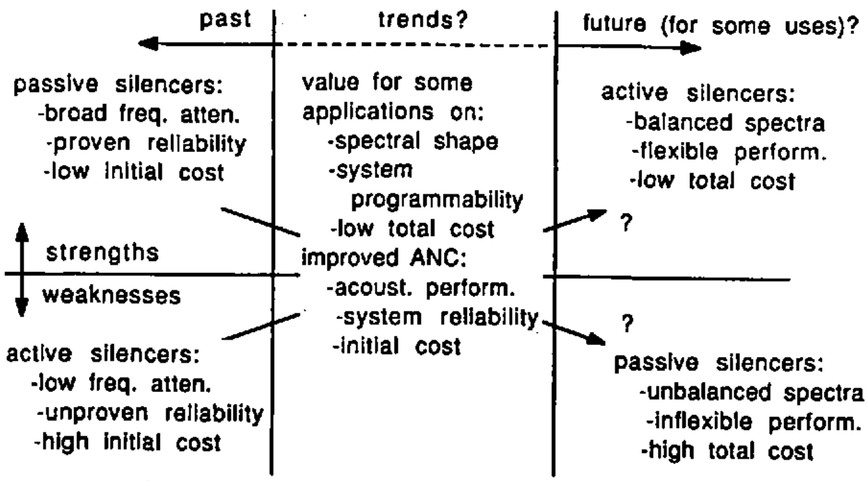

Fig. 4 - Crossover of passive and active silencer technology depends on the direction of specific trends for each application.

balanced spectrum, easily programmed flexibility, and low overall system cost. The tendency of passive silencers to produce an unbalanced spectrum, be restricted to a fixed structure, and to have higher operating costs may come to be regarded as significant weaknesses.

5.2 Attenuation Characteristics:
Passive silencers are valued for their high attenuation over a broad frequency range. Active silencers have been limited to low frequency attenuation. However, active silencers have the unique ability to focus their attenuation on peaks in the noise spectrum, resulting in a smoother, more properly shaped spectrum.

Spectral shaping is an important factor in the reduction of the annoyance of noise [11,12]. Passive silencers simply reduce the entire spectrum, often overattenuating the high frequencies. This inherently creates an unbalanced spectrum, leading to the need for the addition of electronic masking noise. Active silencing has the ability to change the shape of the noise spectrum. Peaks associated with pure tones or narrow bands of noise are attenuated more than the overall background noise. This automatically concentrates silencer effectiveness where it is needed, producing a more pleasing, balanced spectrum. The result more closely approximates the shape of the recommended RC or NCB noise criteria [11,12].

This characteristic is shown in Fig. 5. In this laboratory evaluation, the unsilenced spectrum (measured upstream of the silencing system) contains considerable low frequency content including several prominent tones. A large passive-resistive silencer with an annular center-body was used to obtain the passive only results. Although substantial attenuation was obtained, the result continues to be dominated by low frequency noise, including several tones. With the addition of the active silencer, the spectrum becomes much more balanced due to the reduction of the tonal peaks [13]. In this case, the hybrid system is particularly effective, and active noise control effectively adds to the overall silencing capability rather than replacing the passive unit [14]. To obtain a similar result with only passive silencing would typically require the addition of masking noise at the mid and high frequencies to balance the spectrum.

A similar result is shown in Fig. 6 for a field installation of only active silencing in a building ventilation system. The offices were dominated by low frequency noise below 125 Hz sufficient to excite vibrations in the structure. The use of active silencing shaped the spectra to a much more pleasing result that, although still relatively loud, was a vast improvement over the original problem [15].

5.3 Durability/flexibility:
Although passive systems have provided reliable operation and the long-term reliability of active systems has been unproven, current trends are suggesting this relationship may change. As the durability of active systems improves with further development, the focus may shift to lifetime flexibility. Active systems are inherently easy to maintain, upgrade, and re-use. The ability to reprogram an active silencer to utilize new technology or to meet a new silencing requirement may lead to the concept of "computer-aided silencing." The value of "flexible silencing" systems may become more widely recognized as a parallel development to the flexible, computer-aided systems now in use in other areas.

5.4 System cost:
Today, passive silencers often have an initial cost that is less than that of an active silencer.

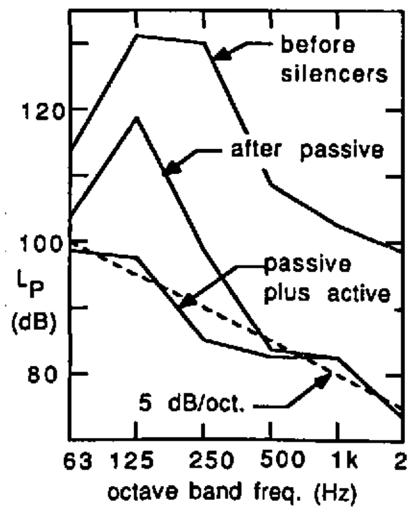

Fig. 5 - Noise reduction of passive and active silencer in laboratory [13].

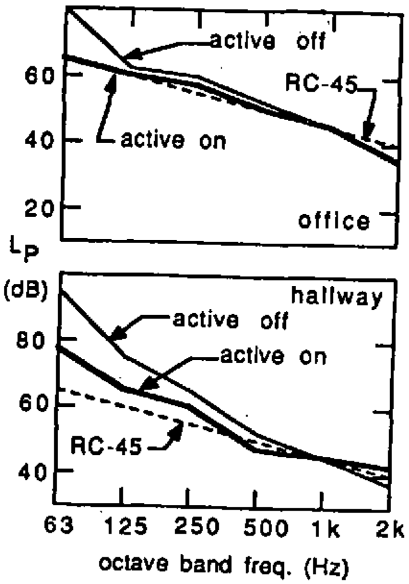

Fig. 6 - Noise reduction of active silencer on office building ventilation system [15].

Interestingly, however, even today this is not always the case, particularly for larger industrial applications or complicated retrofit problems. In addition, the purchase costs of active systems are rapidly decreasing as the electronics are re-packaged, hardware is refined, and production volumes increase. Perhaps equally significant are the low operating costs associated with an active system due to the very low flow restriction produced by an active silencer. Thus, total system costs, including purchase, installation, and operation, may come to be more favorable for active noise control.

6. SUMMARY

The change to active noise control will depend on how fast active systems acquire improved acoustical performance, proven reliability, and lower cost as well as on how much the value of balanced spectra, programmable hardware, and low system cost is recognized. These trends may cause the relative strengths and weaknesses of passive and active noise control to be reversed for some applications. The process of change will be enhanced by the development of improved electronic hardware, computer software, signal processing algorithms, acoustical hardware (including sources), and application knowledge. The relative rate of increase of performance requirements for silencers in specific applications and the performance improvements of active silencers for these applications will determine the future use of active noise control technology.

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RESONANCE SUPPRESSION IN THE ACTIVE CONTROL OF LOW FREQUENCY RANDOM SOUND FIELDS

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1. INTRODUCTION

The sound field in an enclosure can be described as a sum of modal contributions, and if the source has a stationary random white noise signal then all the modal resonances will be excited. A secondary source, whose output is a function of the primary source, may be introduced to act as a feedforward active control system. An optimal controller for this secondary source can be derived from a knowledge of the acoustic modal frequency response functions of the enclosure. In this study, a numerical simulation has been used to model the sound field in a shallow cuboid enclosure excited by a low-pass filtered white noise source, and to examine the potential for feedforward active control to suppress the acoustic resonances. An outline only of the theory is presented here, with some of the interesting results. A fuller account of the derivation and method of numerical solution is given in references [1] and [2].

2. THEORY

If an enclosure has a simple geometry and the source position is known, then the spatial variation in the sound pressure throughout the enclosure is readily predicted using a modal model. The block diagram of Figure 1 illustrates how the sound field in an enclosure can be expressed as an infinite sum of modal contributions, where the $\Psi_n(\mathbf{x})$ are the mode shape functions and the $a_n(t)$ are the modal amplitudes. In the case of a lightly damped rectangular enclosure, the modal frequency response is that of a simple second order system. The modes can be treated as independent linear systems operating in parallel, and it is convenient initially to consider the behaviour of just one mode, n, and to work in terms of the modal amplitude $a_n(t)$. In the case of a single piston source of area S_p and uniform normal velocity, in an enclosure of area S, we may define a modal shape function for that source as :

$$\psi_{np} = \left(\frac{1}{S_p}\right) \int_{S_p} \psi_n(y)\, dS \qquad (1)$$

We now introduce a secondary piston source into the enclosure at position S_s, and thus the two sound fields are superposed and will interfere. The modal shape function for the secondary source is defined for position S_s by analogy with equation (1).

It will now be assumed that the secondary source strength is related to the primary source strength by a filter whose impulse response function is h(t).

The block diagram for a given mode is then as shown in Figure 2. Since the system can be considered to be linear and time-invariant, the order of the blocks may be rearranged.

The problem is now to determine the filter, $h_o(t)$, which is in some sense optimal. An appropriate cost function, which has been used in the earlier related work of reference [3], is the total acoustic potential energy in the enclosure. This is given by

$$E_p = \frac{1}{2\rho_0 c_0^2} \int_V E[p^2(x,t)]dV \qquad (2)$$

where E[...] denotes the expectation operator, which can be interpreted as the time averaging operation for the class of signals dealt with here.

It will be instructive to observe the effect of the active control strategy on the frequency distribution of the energy, so we define an acoustic potential energy spectral density function by

$$E_p(\omega) = \frac{1}{2\rho_0 c_0^2} \int_V \mathcal{F}\{E[p(x,t)p(x,t+\tau)]\} \qquad (3)$$

Where such a multimodal system is excited by a harmonic source, the optimal controller h(t) can be determined by frequency domain analysis, which was the approach taken in references [3] and [4]. However, where the input varies randomly with time, the controller must be causal - the secondary source cannot respond to changes in the primary source output before those changes have occurred. This constraint can be applied to the present analysis, provided that this is carried out in the time domain.

The problem can be dealt with using an extension of the classical optimal filter theory of Wiener. It has been shown in reference [2] that the optimal causal filter must satisfy a Wiener-Hopf integral equation of the form :

$$\sum_{n=0}^{\infty} \psi_{np}\psi_{ns}R_{ssn}(\tau_1) + \int_0^{\infty} h_o(\tau_2) \sum_{n=0}^{\infty}\psi_{ns}^2 R_{ssn}(\tau_1 - \tau_2)d\tau_2 = 0 \qquad (4)$$
$$\tau_1 > 0$$

Where the primary source strength fluctuation $q_p(t)$ is white noise, the $R_{ssn}(t)$ term may be interpreted as the autocorrelation of the output from a shaping filter whose input is white noise. The shaping filter in this study corresponds to the n'th order modal response function, which is known for a given enclosure geometry.

3. NUMERICAL SOLUTION FOR THE OPTIMAL CONTROLLER AND SIMULATION OF THE EFFECT ON THE ENCLOSURE POTENTIAL ENERGY

In this simulation, a numerical approach was taken to solve the Wiener-Hopf integral equation (4) as applied to a shallow cuboid space shown in Figure 3. The sound sources were assumed to be pistons of uniform normal velocity, whose positions were as shown in Figure 3. Four different positions for the secondary source were considered, so as to compare the performance of the active control system.

Equations (3) and (4) were reformulated in discrete time, so as to form expressions for the impulse response function of the optimal causal filter, $h_o(t)$, and for the resulting energy spectral density $E_p(w)$.

With a true white noise input, an infinite number of modes would be excited equally, and for the enclosure studied here a very large number of modes would be required before the summation terms in equation (4) converged. A more efficient approach is to look at the response of the same system to a white noise input which has been low-pass filtered so as to include just a few modes, to seek an understanding of the system behaviour. This is a reasonable model for real applications : a real sound source is effectively band-limited as it cannot have components at infinitely high frequencies.

As illustrated in Figure 4, a low pass filter is introduced into each branch of the block diagram, just before the modal frequency response filters, so that we may consider an equivalent system with a pure white input and a bank of composite shaping filters.

Note that the zero order mode (n=0) was modelled as a low-pass filter, whose coefficients were chosen (somewhat arbitrarily) to give a frequency response which was consistent with the other low-order modes. This may be interpreted as a piston moving into the enclosure at a constant slow velocity, where a leak is present so that there is a balance between the pressure inside and the flow rate through the hole.

4. COMPUTER IMPLEMENTATION, SIMULATION AND RESULTS

The numerical formulation, solution and simulation were performed on a VAX 11/750 computer at ISVR, Southampton, using a series of Fortran 77 programmes and DATS signal processing modules. Details of the operations carried out are given in reference [5]. The low-pass filter on the primary input was a second order Butterworth filter with a 1200 radians/second cutoff frequency. The first eight non-zero modes were included, the sampling frequency was 600 Hz and the filter had 512 points. The system damping ratio was taken to be 0.01.

The impulse response function of the filter $h_o(t)$ found for the four secondary source positions respectively are given in Figure 5. In each case, it quickly reaches a large negative value, then oscillates and dies away. The vertical lines show the appropriate propagation times calculated between the secondary and primary source positions, and in each of the four cases this lies close to the delayed negative peak in $h_o(t)$. This shows that an important element of the signal from the secondary source is an inverted and delayed version of the primary signal. This is consistent with the result found analytically by Nelson et al (reference [2]) for the special case of a one-dimensional duct. The secondary source may be considered to be acting as a power absorber.

In Figure 6, the dashed line is a frequency plot of the potential energy in the enclosure, calculated without active control, E_{pp}. The residual energy spectrum, when the appropriate optimal controller is applied for the secondary source at position S2, is shown by the unbroken line, E_{po}.

The potential energy is reduced to 31% of its initial level. Some resonant peaks are removed, and the energy level in the troughs between the resonances is increased : the energy has been redistributed away from those peaks, with a mechanism similar to damping. It is noticeable that only the peaks of the first, fourth and fifth modes have been reduced, while the degenerate second/third and the sixth and seventh modes are unaffected. Reference to the nodal plane diagram for this enclosure (Figure 7) suggests the mechanism for this : source S2 lies close to the nodal lines for the second/third, sixth and seventh modes and thus would not be expected to influence these resonances.

The residual energy spectra calculated for secondary sources at four different positions in the enclosure are shown in Figure 8. Comparing these with Figure 7 shows that in each case a controller which lies close to a nodal line cannot effectively control that mode. The reduction in total potential energy calculated for each secondary source position are given in the Table below.

Total Energy Remaining in the Enclosure
(expressed as a percentage of the original energy, E_{pp})

secondary source position	residual energy E_{po}/E_{pp}
S1	0.73 %
S2	31.0 %
S3	42.0 %
S4	24.7 %

Source position S1 is the most successful across the spectrum because it lies very close to the primary source and thus can produce a similar sound field to superpose on the primary field. The next most successful sound source position, S4, lies close to a corner, from where it is able to excite most of the modes of the enclosure and control most of the resonances.

6. CONCLUSIONS

The optimal causal controller can be derived for a feedforward active noise control scheme, using the Wiener-Hopf method. A numerical approach has been used to model an enclosure a with low frequency noise input. The mechanism of resonance suppression is basically a damping effect - the secondary source acts as an absorber. The effectiveness of the strategy depends on the position of the secondary source with respect to the acoustic nodal pattern in the enclosure.

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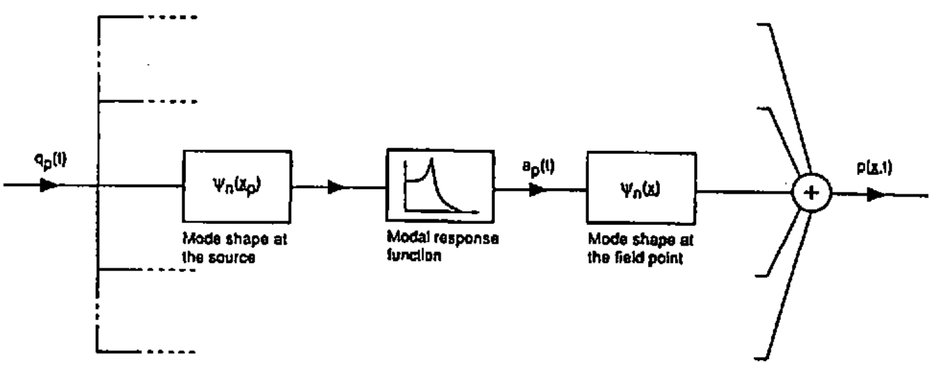

Figure 1 THE REPRESENTATION OF A PRESSURE FIELD AS A SUM OF MODES.

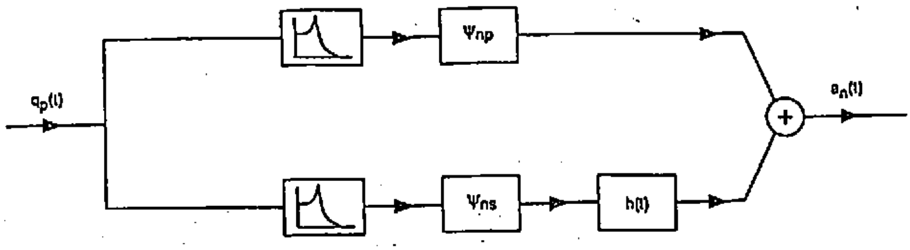

Figure 2 THE RESPONSE OF THE nth PRESSURE MODE

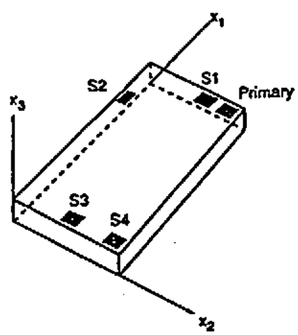

Figure 3 SCHEMATIC DIAGRAM OF THE ENCLOSURE MODELLED IN THE COMPUTER SIMULATIONS.

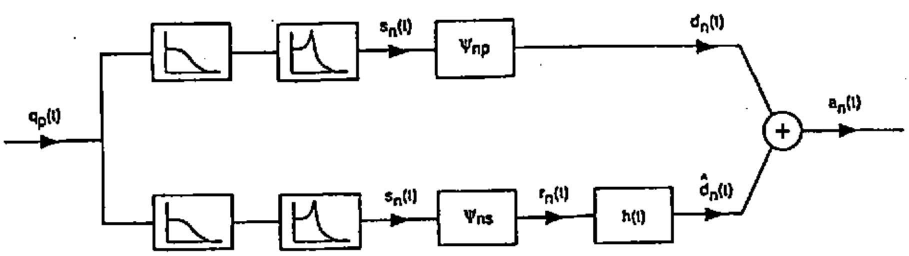

Figure 4 THE RESPONSE OF THE nth PRESSURE MODE FOR A LOW-PASS FILTERED INPUT.

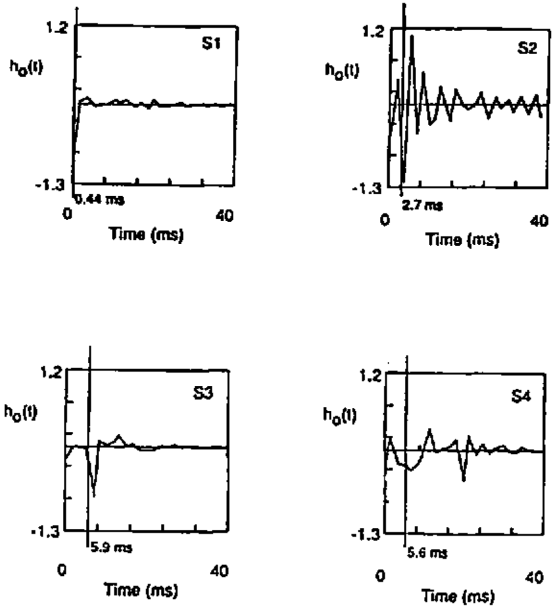

Figure 5 THE OPTIMAL CONTROLLER IMPULSE RESPONSES FOR THE DIFFERENT SOURCE POSITIONS S1 - S4 AND A DAMPING RATIO OF 0.01.

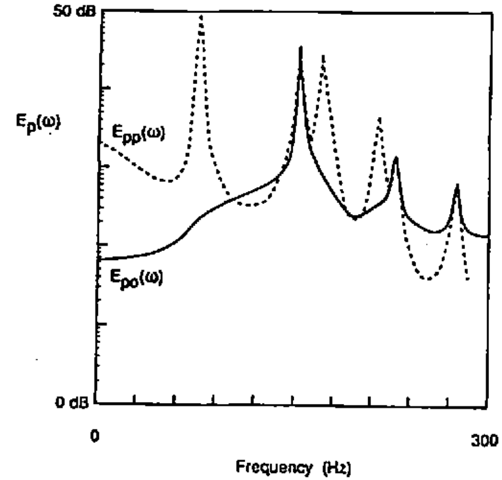

Figure 6 THE RESIDUAL ENERGY WITH ACTIVE CONTROL USING SOURCE S2 AND A DAMPING RATIO OF 0.01. THE UNCONTROLLED CASE IS SHOWN DASHED.

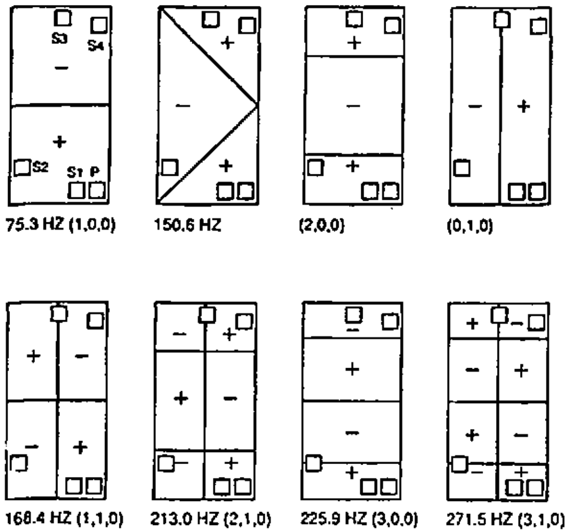

Figure 7 THE DISTRIBUTION OF NODAL PLANES FOR THE FIRST SIX RESONANCES OF THE ENCLOSURE, WHEN DRIVEN BY THE PRIMARY SOURCE ONLY.

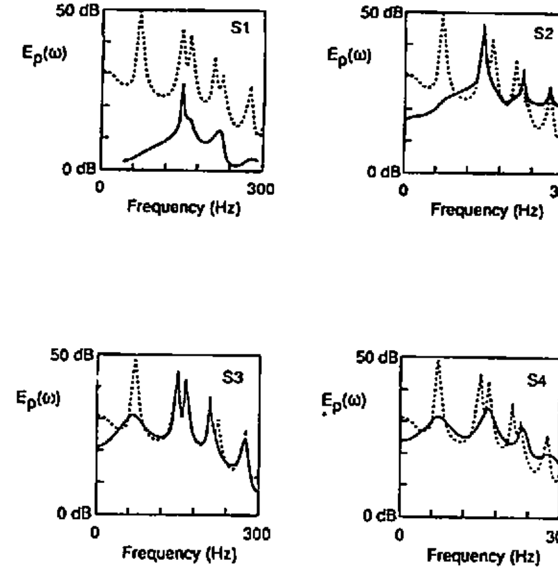

Figure 8 THE RESIDUAL ENERGY FOR THE DIFFERENT SOURCE POSITIONS S1-S4 AND A DAMPING RATIO OF 0.01. THE UNTREATED CASE IS SHOWN DASHED.