Active Noise Control–From Laboratory to Industrial Implementation

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ACTIVE NOISE CONTROL - FROM LABORATORY TO INDUSTRIAL IMPLEMENTATION

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INTRODUCTION

Although there are numerous reports of successful demonstrations of active noise control in laboratory experiments, in computer simulations and in short term tests on real life noise sources, the number and range of long term commercial installations remains relatively small. Perhaps the main reason for this is the seemingly excessive cost of such installations which may easily exceed the actual hardware cost by one or two orders of magnitude. The high cost of the labour component is associated with the high level of engineering expertise and high level of understanding of the principles of active noise control which are required if the installation is to be successful. As active noise control expertise includes acoustics, signal processing, automatic control and electronics, the number of true experts throughout the world who are capable of developing and successfully installing active noise control systems is very limited. This also serves to limit the spread and commercialisation of the technology. Adding to the cost of the labour component is the unique nature of most problems to which active noise control is applied, thus tending to preclude the use of a generic system which can be installed by non-expert technicians. In many cases, the unique nature of the problems involved requires a large injection of development funds just to get to the laboratory demonstration stage.

Unfortunately, an inexpensive, clever, commercial control system which includes a selection of source and sensor transducers to satisfy most problems and software to guide users in the correct choice and location of such transducers does not yet exist. The word “clever” used above to describe the commercial control system which does not yet exist needs some explanation. If a controller is to be useful to a wide range of people, the effort involved in setting it up must be very small. This means that the controller must itself be controlled by a high level expert system or neural network which automatically sets input and output gains to maximise system dynamic range, convergence coefficients to optimise convergence speed and stability trade-offs, control filter type and weight numbers to optimise noise reduction, as well as leakage coefficients and system ID algorithms, filter types and configurations to maximise controller performance and stability. In addition, the controller should also be able to perform as an adaptive feedforward or adaptive feedback controller, be extendable to a large number of channels simply by adding together identical modules and provide advice on the suitability of feedforward control compared to feedback control based on the quality of the available reference signal. Finally the controller/user interface during set-up (which should really be a question/answer session) should be Microsoft Windows based for maximum flexibility. The ability to connect a modem to the controller to allow remote access and interrogation of current performance and the state of transducers and other system components is also an important labour saving device.

Software must also be developed which is user friendly and allows the user to determine optimum control source and error sensor types, configurations and locations based on measured transfer functions between potential control source and error sensor locations as well as a measure of the primary field strength at these locations.

If the field of active noise control is to continue to grow, and if industry is to lose its suspicion of this technology, there needs to be considerably more effort devoted to the development of user friendly commercial systems which allow non-expert or at least semi-expert personnel to successfully install them, in most cases by following a carefully prepared manual. There also needs to be more honesty and less unfounded optimism in statements made in the media about the potential applications of the technology.

Unfortunately, the past history and credibility of active noise control as a viable alternative control measure has been tainted with a minority of commercial companies making premature public claims regarding its application which were far in excess of what was realistic (or what they could deliver). This has resulted in the technology being viewed with suspicion by key manufacturers of particular mass market products which have potential to benefit from the judicious application of active noise control technology. The field has also been tainted by the unscrupulous patenting by some companies of technology which had been published by other unrelated researchers in journal papers or consulting reports years before the filing of the subject patents. This unscrupulous patenting is often followed up with equally unscrupulous threats of legal action against users of the technology. Unfortunately, this activity is still continuing in some cases, and it is very likely that a number the current 800 to 1000 patents relevant to active noise or vibration control can be labelled as invalid.

In the remainder of the paper, some of the issues associated with practical implementation of active noise control and examples of practical problems will be discussed. Also the scope and type of commercial applications which can be sensibly addressed with current or future active control technology will be described, as will applications which have limited potential for success because of limitations imposed by their physical acoustics aspects. Finally some current research directions which are relevant to commercial application of the technology will be discussed.

PHYSICAL MECHANISMS ASSOCIATED WITH ACTIVE CONTROL

To appreciate the physical problems which limit the application of active noise control, it will be useful to first discuss the physical mechanisms which are responsible for the noise reduction achieved by an "anti-noise" source. In applications of active noise control, the cancelling signal is generated electronically and introduced into the system using transducers such as loudspeakers which convert the electronic signal to sound. In many cases, the physical mechanism responsible for the reduction of the unwanted noise is a little more subtle than mere cancellation. In cases where cancellation is the only control mechanism, the noise level may be reduced at some locations, but will be increased at others so that the total energy of the unwanted noise and the cancelling sound is conserved. This type of control is known as "local cancellation". Examples of applications employing this mechanism are active headsets, (where the noise is cancelled at the entrance to the ear canal but increased at other locations) and noise cancelling headrests in aircraft and other transportation vehicles.

Other possible physical mechanisms responsible for successful application of active noise control include a change of the radiation impedance of the unwanted noise source as a result of introducing the "anti-noise" sources, absorption of sound by the "anti-noise" sources or in the case of a confined space such as a duct, reflection of sound by the "anti-noise" sources.

The suppression of the primary source sound radiation by a change in its radiation impedance may be understood on the basis of the following considerations. If it were possible to make the entire control sound field (or almost all of it) 180° out of phase with the original (primary) field, then the sound radiated by the primary source would be effectively "cancelled" leaving one to wonder where all the energy had gone. The answer is that in this case, the control mechanism is not really cancellation; the sound field generated by the control sources has effectively "unloaded" the primary source, changing its radiation impedance so that it radiates much less sound (even though the motion of the physical source such as a vibrating surface may remain unchanged). In this case, the control sources act to suppress the sound power radiated by the primary source. To achieve effective suppression of the primary source output by presenting a purely reactive impedance to it, the control sources must be large enough and located such that they are capable of presenting the required impedance to the primary source. In one dimensional wave guides, such as air conditioning ducts, these constraints are relatively easy to satisfy and the distance between the control and primary sources is not too important. However, in 3-D space, the control source in general will need to be close to the primary source to affect its radiation impedance significantly. It will also need to be of similar size with a similar volume velocity output.

An example of a change in radiation impedance is the active control of a periodic propagating plane wave in a duct originating from a source somewhere in the duct. The active control source generates a sound field which changes the radiation impedance "seen" by the original sound source, thus reducing its sound output. An electrical analogy to this situation is a power point in a wall which may be considered to be a source of electrical energy. If a light is plugged into it, the power point will supply the power for which the light is designed which may be 60W. On the other hand, if a radiator is plugged into the power point, 1500W may be produced (depending on the rating of the radiator). Thus it is seen that the power produced by the power point is dependent on the load impedance that it "sees". A similar argument holds for the radiation impedance of an acoustic source, where the radiation impedance can be altered by the introduction of another sound source.

If sound propagating in a duct is random or transient in nature, then it seems clear that the control mechanism is NOT a change in radiation impedance of the original source by the "anti-noise" sources. This is because the noise to be cancelled is not periodic; thus the required sound field at the location of the undesirable sound source to produce a change in radiation impedance will not be predictable in advance. In this case, the "anti-noise" source either absorbs energy or reflects it back from whence it came where it is eventually dissipated. When the "anti-noise" loudspeaker absorbs sound it will still need electrical power to drive it to the correct displacement for energy absorption as the acoustical energy it absorbs is insufficient to overcome the source mechanical impedance. Thus, in practice, the electrical power requirement to drive a loudspeaker control source is not noticeably different when the speaker is absorbing sound energy than when it is radiating it.

An interesting side issue is that at frequencies which correspond to acoustic resonances in the medium filling a duct, the active control mechanism involving changing the source radiation impedance can still be effective for a random noise source because the reverberation associated with the resonance effectively makes the random noise periodic at this frequency.

For the mechanism involving absorption by the control sources, the primary sound field energy is used to assist in driving the control source (for example the speaker cone if the control source is a loudspeaker). However, the acoustical efficiency of loudspeakers and other artificial noise generators is so poor, that electrical energy is still needed to drive the source with sufficient amplitude and at the correct phase to enable it to absorb energy from the sound field. Except for plane wave sound propagation in ducts, this mechanism is likely to result only in areas of reduced noise close to the control source.

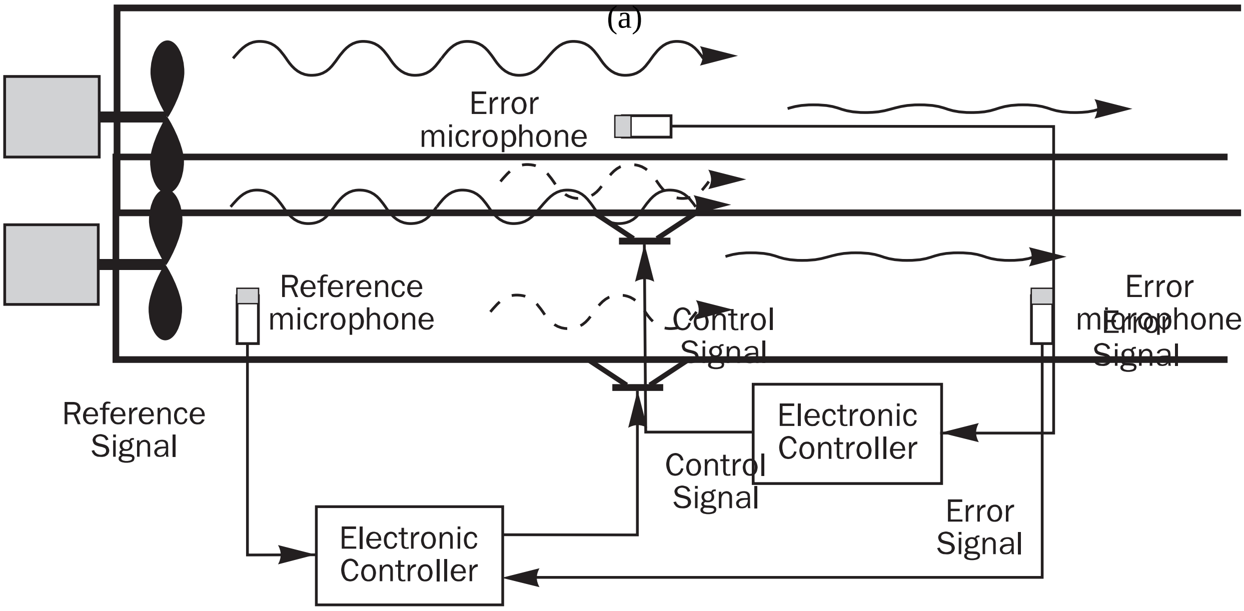
FEEDBACK vs FEEDFORWARD CONTROL

To enable a sensible appreciation of the concepts to be discussed later in this paper, it is useful to begin with a brief overview of single channel adaptive feedforward and adaptive feedback control applied to sound reduction. Non-adaptive systems are generally impractical for most industrial applications (except perhaps for active ear muffs) because of the variability in the physical system being controlled. The simplest example to consider for the illustration of the principles of feedforward and feedback control is the active control of plane wave sound propagation in a duct.

Referring to figure 1(a) (feedback control), the error microphone picks up the incoming signal which is processed to derive a suitable control signal for the control source such that the error signal is minimised. Thus it is clear that this type of controller will function best when the predictability of the incoming signal is good. Thus resonances in structural and acoustical systems can be controlled, and noise which has a high normalised autocorrelation for any time delay greater than the delay through the control system (which includes the control source and error sensor) is also controllable. Tonal noise is characterised by a high autocorrelation and thus is well suited to feedback control.

Adaptive feedback controllers are usually implemented digitally using a DSP chip, as implementation using analog circuitry is not really practical. On the other hand, non-adaptive feedback controllers are usually implemented in analog circuitry to minimise the delay through the controller, thus increasing the maximum autocorrelation value of the noise to be controlled and hence the maximum achievable performance. A disadvantage of feedback controllers is their inherent instability at higher frequencies where the phase response is not easily controlled. This can cause serious acoustic noise problems in the presence of high frequency noise or if the physical system being controlled changes too much from the design condition (for non-adaptive feedback control) or too rapidly between states (for adaptive feedback control). An example may be the unstable oscillation (or screeching) in an active headset with analog control as it is adjusted on the head of the wearer or if the wearer enters an environment characterised mainly by impulsive noise or high frequency noise. The instability problems of these types of controller are usually minimised by keeping the controller gain within reasonable bounds, (which has the effect of limiting the controller noise reduction performance) and using low pass filters to attenuate incoming high frequency signals which cannot be controlled. To maximise robustness (or minimise instability problems) it is essential that the microphone be placed as close as possible to the control source which will have the effect of minimising system delays and thus maximising the autocorrelation of the noise at time delays greater than the control system time delay. The disadvantage of placing the error sensor close to the control source is that because of near field effects, the sound pressure at large distances from the error microphone may not be significantly reduced. This is not a problem, of course for active ear muffs because of the close proximity of the ear drum to the error sensor.

(a)



(b)

Figure 1 Basic active noise control system.
(a) Feedback system
(b) Feedforward system

Referring to figure 1(b) (feedforward control) (usually a microphone) samples the incoming signal which is filtered by the electronic controller to produce the output signal to drive the control source (loudspeaker in this case). The controller effectiveness is measured by the error sensor which provides a signal for the control algorithm to use in adjusting the controller output so that the error is minimised. The signal processing time of the controller must be less than the time for the acoustic signal to propagate from the reference sensor to the control source for broadband noise control, but for tonal noise control, there is no maximum permitted processing time as the signal is repetitive. The cancellation path is the electro-acoustic path from the loudspeaker input to the error microphone output. The transfer function of this path must be taken into account in most controller algorithms and thus it must be measured for every installed system. Indeed, it is essential in most practical commercial systems that some means is implemented in the controller to measure this transfer function regularly while the controller is operating as it can change quite quickly in some cases. This will be discussed in more detail in the next section.

Because of their inherent stability, feedforward controllers are generally preferred over feedback controllers when a reference signal which is correlated with the error signal is available. One exception is the active ear muff case for which it has been found that an adaptive feedback controller seems to cope better than an adaptive feedforward controller to head movement of the wearer [1]. One disadvantage of feedforward controllers which is not shared by feedback controllers is the often encountered problem of feedback of the control source output to the reference sensor via an acoustic path. Unless this is compensated for in the controller adaptation algorithm and control filter, instability is likely to result. Appropriate algorithms and filters are discussed in the next section. One way of avoiding the problem is to use a non-acoustic reference sensor such as a tachometer on the machine causing the noise, as shown in figure 2.

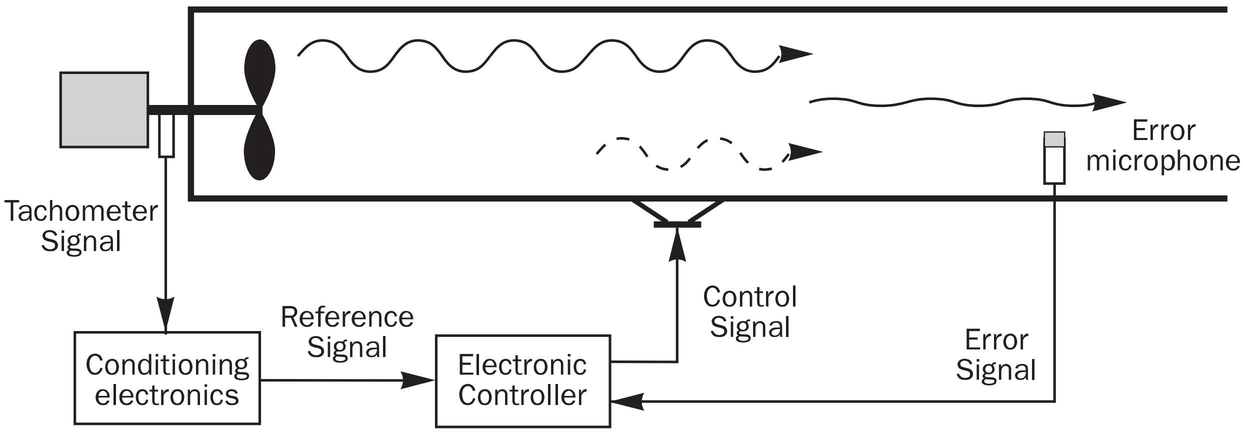


Figure 2 Basic active noise control system with tachometer reference signal.

However this is only useful if it is desired to only control the harmonics of the machine rotational frequency. All other noise detected by the error sensor will be uncorrelated with the reference signal and thus will not affect the control system output (unless a specially developed non-linear filter and algorithm is used - see next section).

In many applications, it is necessary to use multi-channel systems as the sound field is too complex for one channel to be effective. An example may be the active control of interior noise in a car. It will not contribute much to the aims of this paper to discuss multi-channel systems here. The interested reader is referred to the discussions in the three books which cover the subject [2,3,4].

ADAPTATION ALGORITHMS AND CONTROL FILTER STRUCTURES

The components of a single channel feedforward active noise control system for controlling noise propagating in a duct, without on-line identification of the cancellation path from the control source (loudspeaker) input to the error sensor (microphone) output are illustrated in figure 3 and with on-line system identification in figure 4. The type of on-line cancellation path identification shown in the figure is one of the types commonly used in industrial installations.

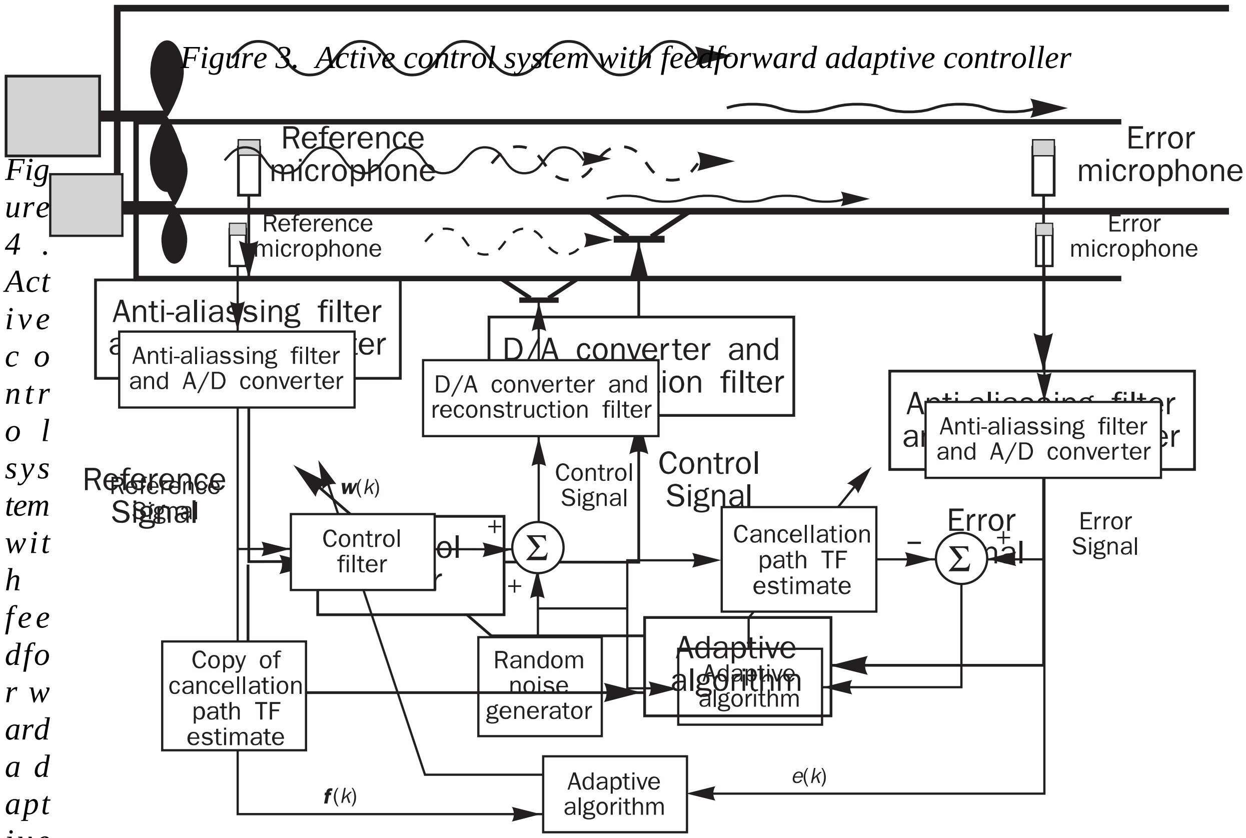


Figure 3. Active control system with feedforward adaptive controller

Figure 4. Active control system with feedforward adaptive controller and on-line system ID

Other types of on-line cancellation path identification are discussed in [2 & 3].

If the control filter is an FIR filter (see following paragraphs for filter architectures), the algorithm for updating the filter at the kth sample is

$$\boldsymbol{w}(k+1) = \boldsymbol{w}(k) - 2\mu e(k)\boldsymbol{f}(k)$$

where μ is the convergence coefficient, $\boldsymbol{w}(k) = [w_0(k)\ w_1(k)\ w_2(k)\ \cdots\ w_{L-1}(k)]^T$ represents the filter weights (of number, L) and $\boldsymbol{f}(k) = [f(k)\ f(k-1)\ \cdots\ f(k-L+1)]^T$ is the reference signal after filtering with the cancellation path estimate (see figure 4). Multi-channel versions of this type of controller (multiple control sources and error sensors which all interact) are discussed in detail in refs [2-4]. In practice, it is necessary to include a leakage factor α which prevents the control filter from overloading due to digital quantisation errors. In this case, the control filter weight update algorithm for sample time $k + 1$ is

$$\boldsymbol{w}(k+1) = (1 - \alpha\mu)\boldsymbol{w}(k) - 2\mu e(k)\boldsymbol{f}(k)$$

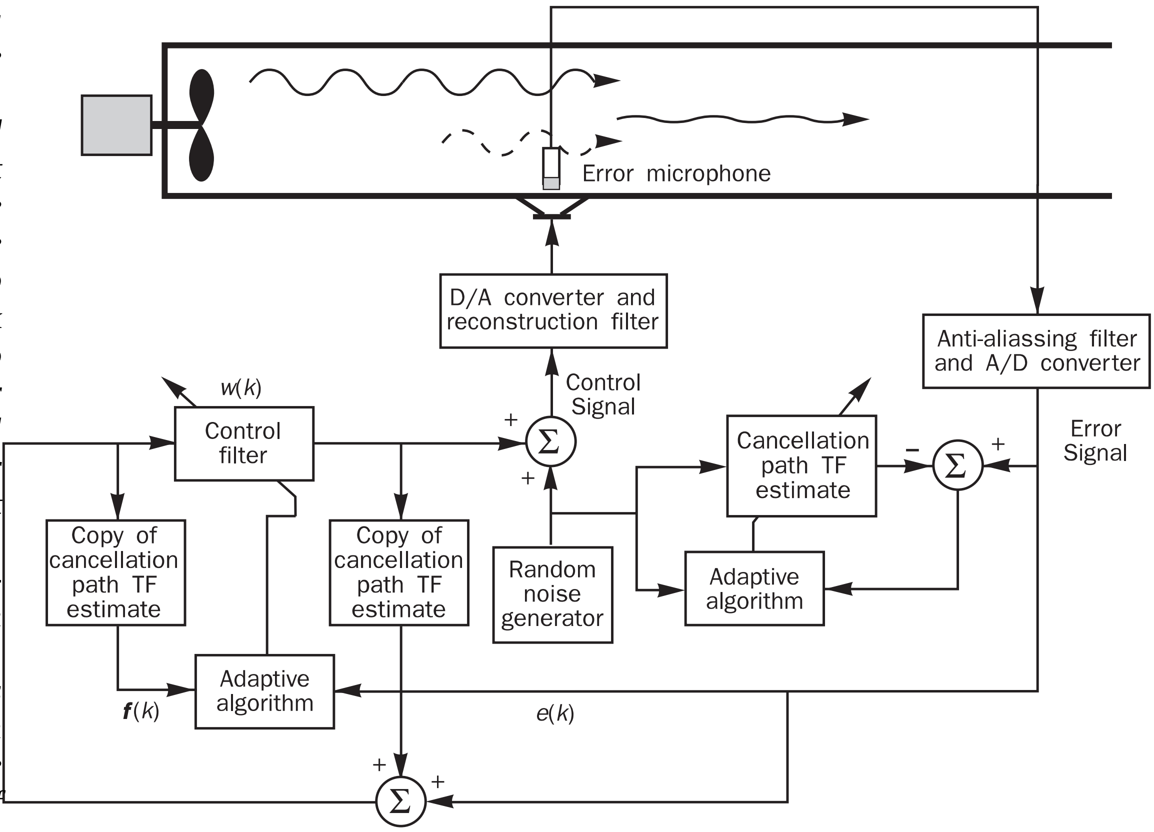
The leakage factor has the effect of including the control output as part of the cost function to be minimised and has a stabilising effect on the algorithm.

An adaptive feedback controller with on-line system ID is illustrated in Figure 5. Note that no reference signal is required. In fact, a pseudo-reference signal is synthesised from the error signal. The FIR (control) filter weight update equation is written as

$$\boldsymbol{w}(k+1) = \boldsymbol{w}(k) - \mu e(k)\boldsymbol{f}(k-1)$$

A multi-channel version of this algorithm is discussed in reference [3].

Figure 5 Adaptive feedback controller with on-line system identification



The control filter may take a number of forms, each of which requires a separate algorithm. The most common form in use is the finite impulse response (FIR) filter for which the algorithms outlined above may be used. An FIR filter may be represented as in figure 6.

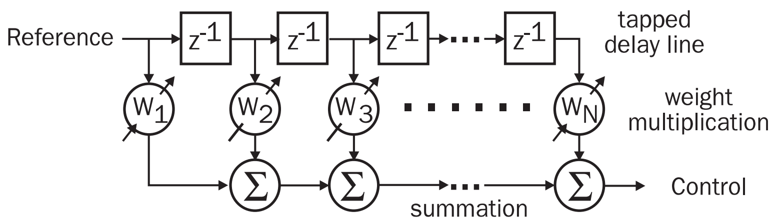
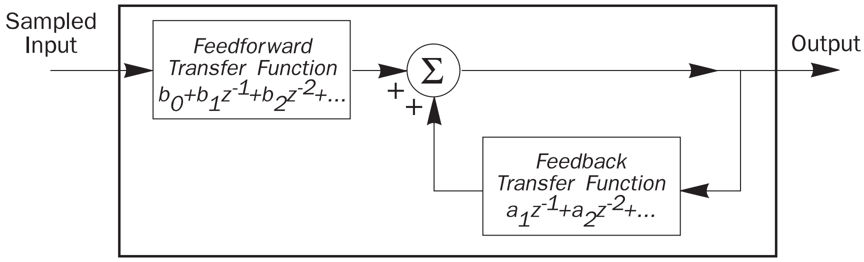


Figure 6 FIR filter architecture

In some cases, especially when there are resonances in the system to be controlled or if there is acoustic feedback from the control source to the reference sensor, the FIR filter is not the best choice and the infinite impulse response (IIR) filter is often chosen for its ability to directly model the poles in the system resulting from the above mentioned effects. Such a filter is made up of two FIR filters and has the architecture illustrated in figure 7. The most suitable algorithm for updating the weights of this filter is discussed in [2,3&4] and is a little more complicated than the algorithms outlined above. The advantage of the IIR filter (in addition to its ability to accurately model the acoustic feedback path and system resonances) is that it can model complex systems with much fewer weight coefficients than required by an FIR filter, thus reducing computational load. This advantage comes at the cost of inherent instability, slower convergence and the possibility of convergence to a local minimum in the error surface instead of a global minimum.



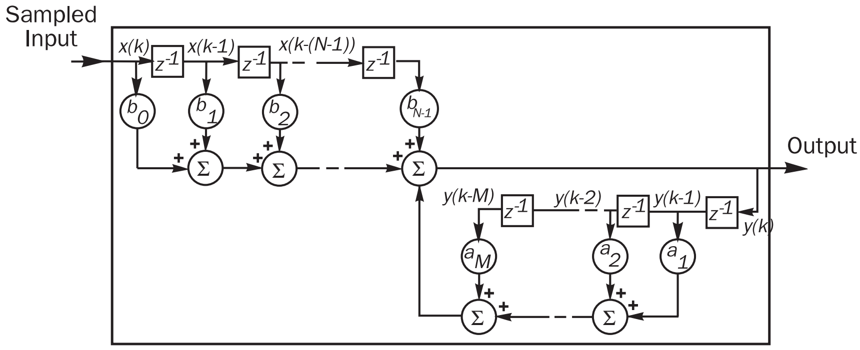


Figure 7 IIR filter architecture

In some cases, particularly when there are non-linear acoustic control sources or non-linear systems to be controlled, neither of the IIR or FIR filter architectures are suitable. One possibility which addresses the non-linear problem is to use a non-linear filter and update it using a genetic algorithm. Such a system is illustrated in figure 8 and an example filter structure is illustrated in figure 9. In figure 9, x is the input signal, W_i are the filter weight coefficients and x^4 and x^5 are the input signal raised to the 4th and 5th powers respectively.

The main advantage (in addition to it being able to address non-linear filter structures) of the genetic algorithm is that is inherently stable and it requires no knowledge of cancellation transfer function which means that the on-line system ID can be eliminated. As the genetic algorithm and filter structure are non-linear, they are capable of generation frequencies not present in the reference signal and which may be used to cancel frequency components arising from non-linear distortion in control source output. The one big disadvantage of the genetic algorithm is its slowness in converging. The genetic algorithm applied to control filter weight adaptation is based on the processes of natural selection and is described in detail elsewhere [5,6], where its effectiveness and good performance is illustrated.

PERFORMANCE HIERARCHY

Active noise control system performance is governed by a strict hierarchy of factors. Referring to Figure 10, the absolute maximum levels of (global) attenuation which are possible with a given active noise control system are limited by the placement of the control sources. This means that no matter how good the electronics are, an active control system will not function properly if the control source placement is not satisfactory. Even for the simple single channel case involving the control of plane waves in a duct, there are preferred locations for the control source which will give better results.

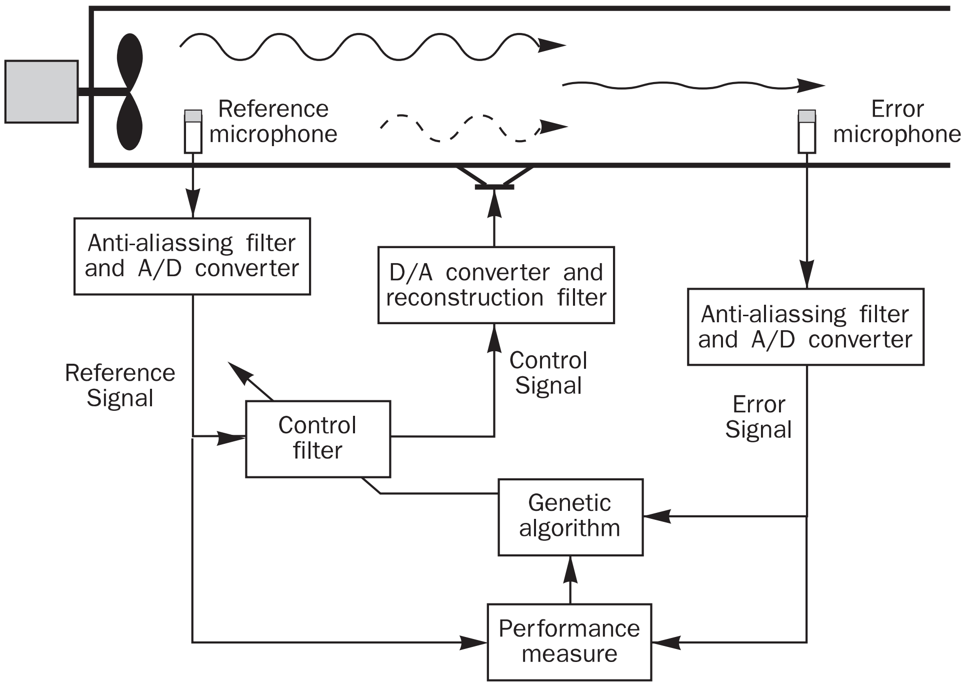


Figure 8   Active noise control system in a duct with a genetic algorithm

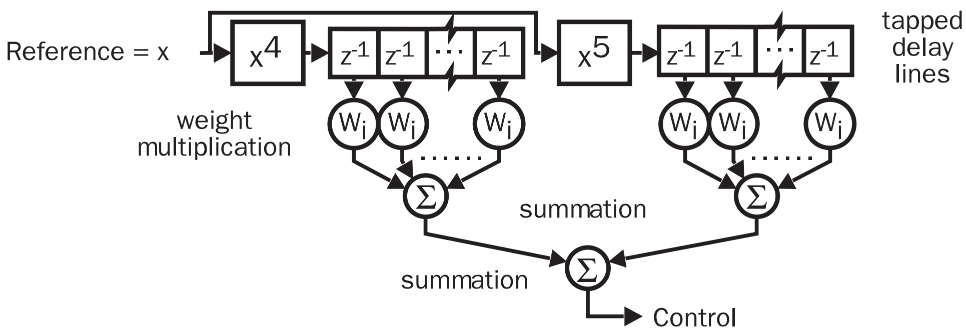


Figure 9 Non-linear polynomial filter example

The second factor limiting control system performance is error sensor placement. Often the optimum error sensor placement is dependent upon the control source placement, which can complicate matters. However, the error sensors must be placed so that they can effectively sense all parts of both the primary and control signals. This means that if the aim is to reduce noise levels in a reverberant space, the error sensors must be capable of adequately sensing all acoustic modes driven by the primary and control sources. Similarly, if the aim is to reduce road noise inside of automobiles, the error sensors must be capable of sensing all acoustic modes excited in the vehicle cabin.

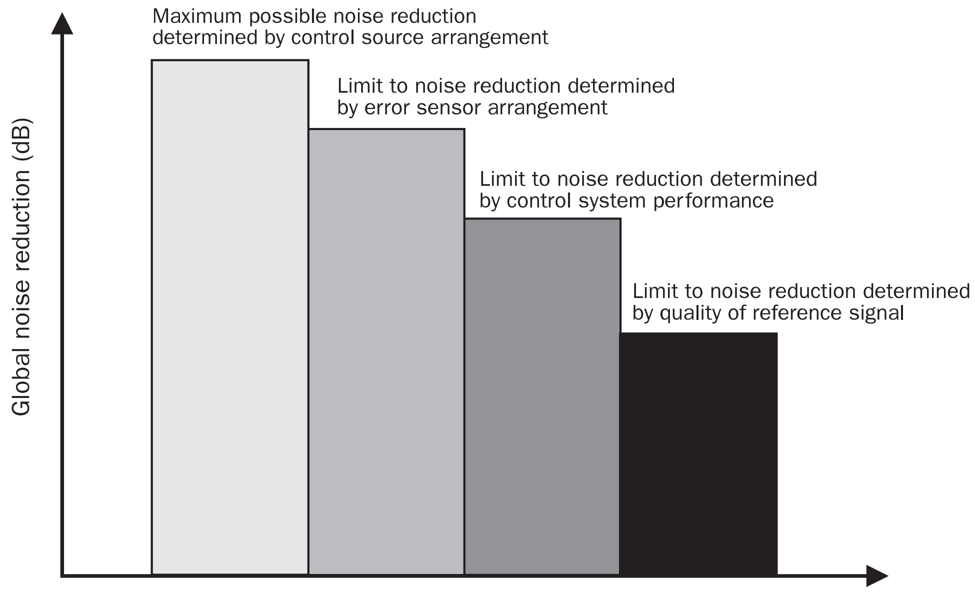


Figure 10 Performance hierarchy of an active noise control system

The third performance factor, which limits how close the actual levels of attenuation are to those physically possible, is the performance of the electronic control system which is related to its dynamic range as well as the algorithms it uses.

Finally, the performance of the electronic control system is limited by the quality of the reference signal provided to it. If the reference signal contains noise not present in the error signal, then the system performance will be degraded.

Each of the four issues mentioned above will be discussed in detail in the next section.

PRACTICAL IMPLEMENTATION ISSUES

Characteristics of noise to be controlled. When implementing an active noise control system, perhaps one of the first things to know is the characteristics of the noise to be controlled. Is it mainly tonal, is it broadband, is it a mixture of both or is it impulsive? Is a reference signal available? If an uncontaminated reference signal (that is, one which only contains the noise components to be controlled) is available, then feedforward control is usually the best choice.

In some cases, if both tonal and broadband noise is to be controlled, a hybrid feedforward/feedback system (see ref [3], p. 207-212) may be the best option as the feedback part will concentrate on the tonal noise and leave the feedforward part to concentrate on reducing the random noise component. If feedback control is considered, it is necessary to know the maximum normalised auto-correlation of the signal in the frequency range of interest. If it is less than 0.8, it is unlikely that feedback control will be worthwhile.

Optimum locations and numbers of control sources and error sensors. Control source placement, while being the ultimate limiting factor in system performance, is far from straightforward. To obtain global attenuation, the basic requirement is that the control source arrangement be able to duplicate the unwanted sound field with some fidelity. For single control source systems, such as the control of plane waves in a duct, the optimal control source location is where the required displacement amplitude (of the control loudspeaker) is a minimum. This location can be determined theoretically or experimentally by trial and error. When many control sources are necessary, the optimal locations can be determined by a numerical procedure involving finite element analysis and genetic algorithm optimisation [5,7]. The optimisation procedure attempts to minimise the squared difference between the unwanted sound field and the control-source generated sound field. For simple systems, (e.g., a simply supported panel radiating noise) the theoretical modelling may be based on an exact analysis using classical mechanics. It should be noted that the modelling does not have to be purely analytical or numerical. Experimental data, namely transfer function measurements between a grid of sensing locations and the primary, and control, acoustic sources, can be used to model active noise control performance. This form of data is ideally suited for use in commercial statistics packages, in multiple regression routines, which perform exactly the same manipulation as required for active control system modelling [8].

There are a few generalisations which can be made concerning control source placement. First, obtaining significant levels of global sound attenuation in free space is often more difficult that obtaining global attenuation in an enclosed space. In free space, being able to duplicate the unwanted acoustic radiation pattern generally dictates that the control sources (usually loudspeakers) be placed in close proximity to the source of the unwanted noise. For example, in the most ideal case, if the source of the unwanted disturbance radiates uniformly (a monopole), and the control source is a second monopole, then to achieve a reduction of 10dB in radiated power the two acoustic sources must be no greater than one tenth of an acoustic wavelength apart. This is where the commonly stated requirement of low frequencies for active control comes into play. If the excitation frequency is 100Hz, and the primary source size is small compared to a wavelength of sound (less than $\lambda/5$), then if the control source is within approximately 30 cm of the source of the unwanted noise, 10dB of global sound attenuation is possible (provided that the control source can mirror the radiation characteristics of the source of the unwanted noise). However, if the excitation frequency is 400Hz, normally not considered "high", the control source must be within a few centimetres of the unwanted noise source. This is not usually practical. Note that this result is for the best case, where the control source arrangement is capable of duplicating the spatial characteristics of the primary noise source. This will invariably not be the case for noise sources larger than about one fifth of a wavelength. In this case, better results will be obtained by using more than one independent control source, and the larger the physical size of the unwanted noise source, the greater will be the number of control sources required.

The question may now arise, what if global control is not required, and local control is adequate; that is, control over a limited spatial angle or in a limited number of specific locations? What is the size of the "zone of quiet" which can be generated around an error sensor for a given control source arrangement? The result follows a similar pattern to that described above. Essentially, one can expect no attenuation at distances greater than one half wavelength away from the sensor, and, realistically, very little at distances greater than one tenth of a wavelength. Note that this is in a perfect world, where all the limiting factors in the performance hierarchy are optimised. Some researchers have managed to achieve minimisation of the sound field at a location away from the error sensor. Although the "zone of quiet" is no larger for this latter case, it is easier to locate the error sensor in a convenient position, away from the head of the listener. Carme, et. al. [9] have used dual microphones and loudspeakers to increase substantially the size of the "zone of quiet" as well as locate it away from the error microphones. Trials in a turbo-prop aircraft [9] with the loudspeakers and microphones mounted in a headrest have produced noise reductions of between 16 and 20dB for the first 3 propeller harmonics (at approximately 71Hz, 142Hz and 223Hz respectively).

To summarise, for free-space radiation problems, if the unwanted noise source is large and/or has a complicated radiation pattern (requiring many control sources to duplicate), and/or is a source of anything other than low frequency acoustic radiation, construction of an active control system which provides significant global sound attenuation will be difficult. It is also important to note that unless the noise being generated is tonal, it is often difficult to obtain a reference signal, representing the impending unwanted disturbance in sufficient time to generate the required control signal. This is not so much of a problem for periodic noise (such as electrical transformer hum) because in most cases, it may be assumed that the reference signal does not vary much from one cycle to the next.

In semi-enclosed spaces, such as in a duct, active control works well up to when there are a few higher order modes cut on, and not so well beyond this. The best case is when only the plane wave mode is propagating, at low frequencies. The majority of (successful) active control installations have been for this very problem. In fact, in ducts containing higher order modes cut-on in the frequency range to be controlled, current practice is to use, where possible, axial splitters to divide the duct section into segments small enough to ensure only plane wave propagation. However, even for plane waves propagating in a duct, finding the optimum source location

One problem with current control source optimisation procedures is that they usually only optimise at a single frequency. There are many industrial applications where this is inadequate and even in cases where a single frequency tone only is to be controlled, the frequency or wavelength is likely to shift with process changes or temperature changes. In these cases, it may be prudent to explore optimum control source locations at a number of frequencies in the range to be controlled. The conclusion may then be that a number of control sources may be required to maximise the noise reduction in cases for which only one source should be sufficient. An example is the control of a tonal noise in an industrial exhaust stack. The wavelength of the noise changes by about due to process temperature changes and fan speed changes. Although a single control source is all that is required to control a single frequency noise in a duct in which no higher modes are cut-on, there are source locations which would require excessive source output. These locations vary as the wavelength varies and to cover all possibilities, it may be necessary to use up to three control sources spaced axially along the duct at intervals of one sixth of a wavelength. The effect of control source location on tha maximum achievable sound power reduction in a duct is illustrated in figure 11 for a constant pressure primary source and a non-infinite impedance in the plane of the primary source. For an infinite impedance in the plane of the primary source, an infinite amount of control is achievable regardless of source location. Note that the downstream duct impedance has no effect on the optimum control source location. Figure 12 is more interesting in that it shows how much volume velocity is required from the control source in relation to the primary source. It is clear that source location has a large influence on how hard it must be driven to achieve optimum control.

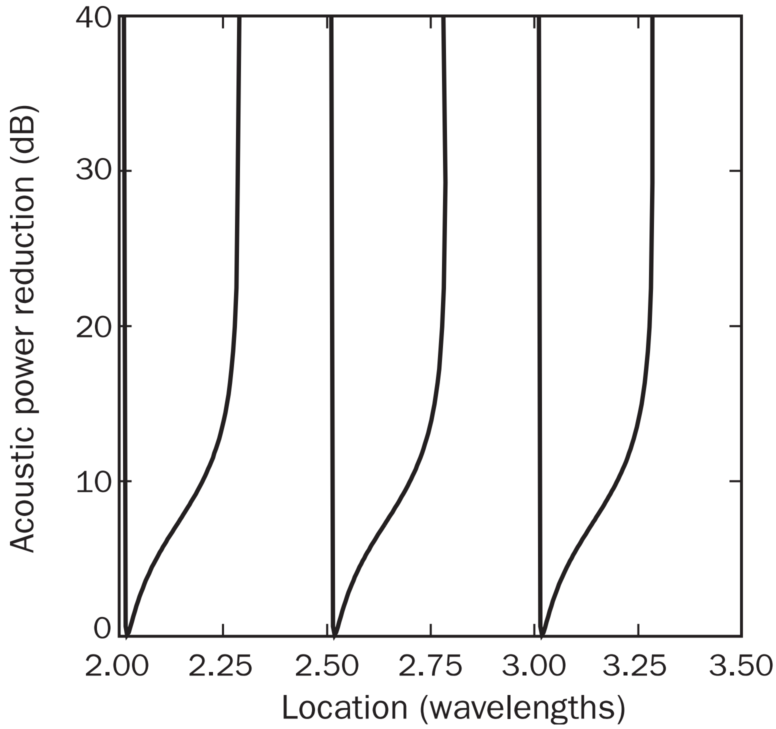


Figure 11 Total acoustic power reduction as a function of primary/control source separation for a constant pressure, non-rigid primary source [2].

For active control of enclosed sound fields, such as in an automobile cabin, attenuation is normally predicted as a function of some measure of the presence or otherwise in the space of acoustic "modes". The most useful measure is modal overlap which is a measure of the density of modes (number of modes per Hz) and the damping associated with each one (Bies and Hansen, 1995). Active control has the potential (with correct control source placement) to work well when the modal overlap is low, where the unwanted noise in predominantly contained in a few lightly damped modes (Elliott, 1989). Generally, it is very poor otherwise. This is (but) one of the problems facing the groups targeting active control of road noise in automobiles, especially luxury models with damped acoustic spaces. With the degree of modal overlap present across most of the frequency range, global control is largely impossible for anything but frequencies below about 150Hz. Local control, "zones of quiet", in enclosed spaces with high modal overlap follow essentially the same rules as free space; as a general noise control technique, beyond headset implementations, it is poor, although some success has been achieved in propeller-driven aircraft, for passengers resting their head against seat headrests, at frequencies up to about 350Hz [9].

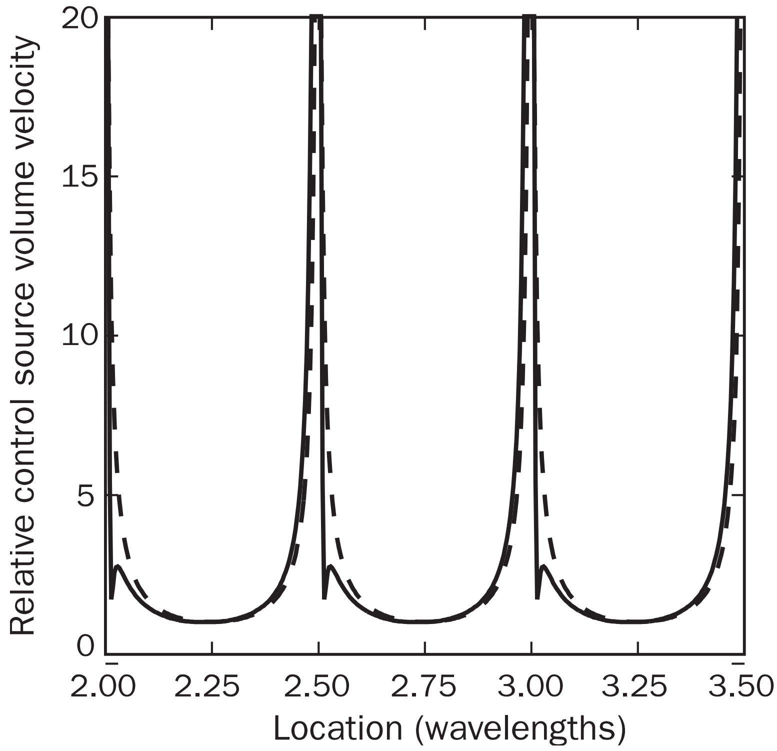


Figure 12 Relative control source volume velocity as a function of primary/control source separation for optimal control and constant pressure primary source [2].
_______ non-rigid termination at primary source
--------- rigid termination at primary source

Use of active absorbers [10] is sometimes contemplated as a means of reducing very low frequency reverberant sound fields at frequencies below which traditional treatment with porous acoustic material (such as fibreglass or rockwool) is ineffective or impractical. The advantage of the active absorbers over passive systems such as Helmholtz resonators is that they can adapt to changes in environmental conditions which result in a change of wavelength in even constant frequency sound fields and the active systems can also absorb over a much wider frequency range.

Active Noise Control Sources. Since active noise control became a practical reality, commercial installations have suffered from insufficient sound source robustness. The robustness requirements necessary to ensure the survival of the sound source obviously differ from one installation to another and clearly active control sources in air conditioning ducts will have different requirements to those in an industrial air handling system in which the environment is very hostile. The sources needed to generate the anti-noise must be capable of producing noise levels similar to those produced by the unwanted noise source. Typical control sources include loudspeakers, horn drivers or vibration actuators (piezoelectric patches, piezoelectric stacks, magnetostrictive actuators, electrodynamic shakers, electromagnetic actuators, hydraulic shakers and pneumatic shakers). The vibration actuators are used sometimes to control the vibration of surfaces which are radiating the unwanted sound.

Current loudspeaker technology is such that loudspeakers used for active noise control will have a life of many years provided they are kept clean and cool. In dirty, hostile industrial environments where the noise levels to be controlled are high, satisfying these requirements is a challenging exercise in mechanical design. To aid the ANC system designer, loudspeakers are available with aluminum or fibreglass cones and with protection of the coil area from contaminants. Most loudspeakers will not function for extended periods in ambient temperature environments above about 50°C. Another problem is associated with the use of small backing enclosures for the purpose of maximising the low frequency output of the loudspeakers. This practice often results in overheated and eventually burnt out speakers unless they are driven at a small fraction of their rated capacity or unless adequate external cooling is provided.

A loudspeaker enclosure design which satisfied the requirements of a cool, clean loudspeaker, even though it was being used to control noise radiated from an 80m high stack containing wet, corrosive sludge with temperatures varying from 100°C to 180°C is illustrated in figure 13 [11]. As shown in the figure, a copper tube containing chilled water was wrapped around each speaker coil and in addition refrigerated air was used to purge the space between the speaker cone and the stack as well as the speaker enclosure. It was found that it was essential for the refrigerated purging air to be present, even when the ANC system was switched off. When this did not happen on one occasion, the loudspeaker suspension became soft and sagged, resulting in the coil touching the magnet. When the cooling air was turned on again, the suspension regained its original stiffness but remained distorted. Thus when the loudspeaker was energised, the coil rubbed on the magnet and after a short time, the coil insulation rubbed off causing a short circuit and a failed speaker. Another requirement which is sometimes overlooked is the provision of a connecting tube to equalise the static pressure on the front and rear faces of the speaker cone. Failure to do this will greatly increase the distortion of the output and reduce the available cone motion, especially in cases where there is a reasonably high vacuum in the duct on which the speaker is mounted as a result of air flow.

In many cases, the noise to be controlled is at the upper limit of capability for continuous loudspeaker operation which results in a serious problem of loudspeaker non-linearity. The non-linearity is heard as higher harmonics of the noise being cancelled and this can negate the subjective benefit of active control in practical installations. The non-linearity problem can be minimised by keeping cone excursions small which may be achieved by only driving loudspeakers at a fraction of their capacity or by clever design of the speaker enclosure.In these cases some effort needs to be devoted to the design of the speaker enclosure to maximise the output in the frequency range of interest. Blondell and Elliott [12] have investigated the use of ported speaker enclosures to minimise speaker non-linearities while at the same time providing a large volume velocity output. In cases where the loudspeaker has to be attached to a duct, the duct acoustics affect the output and the design of [12] is inappropriate. A better design in this case is a double cavity, fully enclosed speaker with a single port as illustrated in figure 13 A design such as this has been used by the author as an active control source to reduce a 30 - 40Hz random noise problem in an air conditioning system in a high rise building.

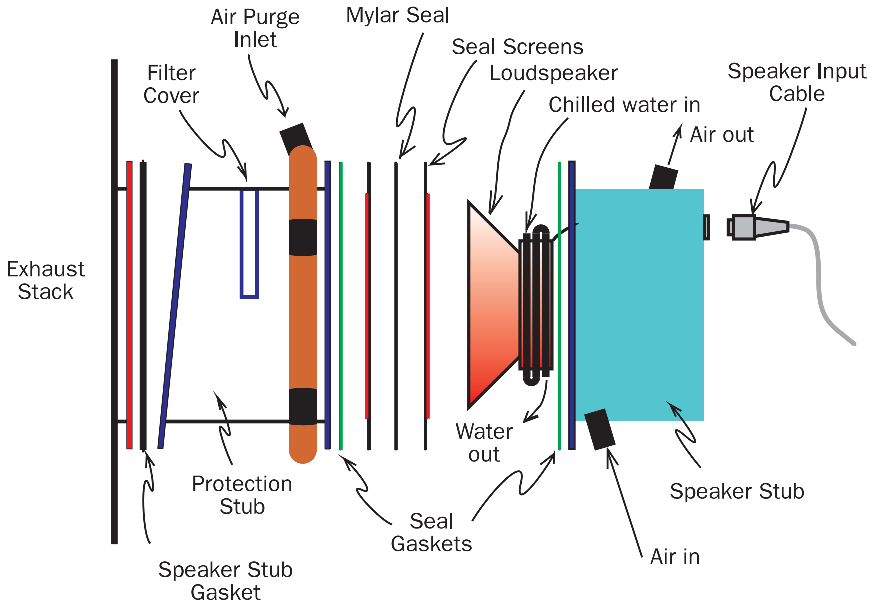


Figure 13 Industrial loudspeaker enclosure with a single port

In terms of the usual active noise control requirement for a large volume displacement over a broad frequency range, it is difficult to beat a conventional loudspeaker. However, there are special applications for which a conventional loudspeaker may not be the best noise source choice. One special application is the control of tonal noise radiated outdoors, such as that radiated by a large electric power transformer. In an effort to minimise the number of control sources required, large panels (1m × 2m) have been used. The panels are tuned to have an acoustically efficient mode resonating at 100Hz and another at 200Hz (or 120Hz and 240Hz respectively, in the USA). The panels are driven by piezo-electric patch actuators. If the actuators are too close to a vibration node, then the force required to excite the panel sufficiently will be too great, whereas if the actuators are placed too close to a vibration antinode, the large amplitude may result in cracked actuators. Thus a compromise is necessary. In addition, it is wise to curve the panel in one dimension so that it looks like part of a cylinder. This enables the extensional motion of the piezo-electric crystal to couple better with the normal motion of the panel to which it is attached. The panel should also be mounted over a closed cavity to prevent sound from the back interfering with that radiated from the from, thus maximising the radiation efficiency at the design frequencies.

Loudspeaker

In the design of the panels to achieve resonance frequencies of 100Hz and 200Hz, it is important that there is some means of compensation for the effect of manufacturing errors. To avoid a large error in the generation of the curved panel, it is important that the panel radius of curvature be at least equal to the width of the cavity over which the panel is mounted. If the panel resonances are designed to be slightly higher than required, then they can be tuned by attaching small masses in appropriate locations.

Figure 14 Low frequency, high volume velocity, speaker enclosure design.

In many cases associated with the control of sound radiated from vibrating structures, it is often more effective to use vibration actuators attached to the structure to reduce the radiated sound. Possible types of actuators are discussed in detail by Hansen and Snyder [2]. Many of the actuators have non-linear characteristics and care must be taken either to not drive them too hard or to use a nonlinear control filter structure and associated algorithm [6].

Error sensors. The most common error sensors used in active control applications are microphones. In most cases inexpensive electrets are adequate. These may be purchased with power supply for tens of dollars or without cable or power supply for $1. Thus multiple error microphones (for which the outputs are summed) may be used economically to improve system reliability. The author has used these microphones in very dirty environments using the holder incorporating an air purge system as illustrated in figure 15. The additional noise at the microphone location resulting from the air purge is dependent on the speed of the air flow over the microphone. For the case considered here it was about 85-90dB which was sufficiently below the duct noise that it was not a problem.

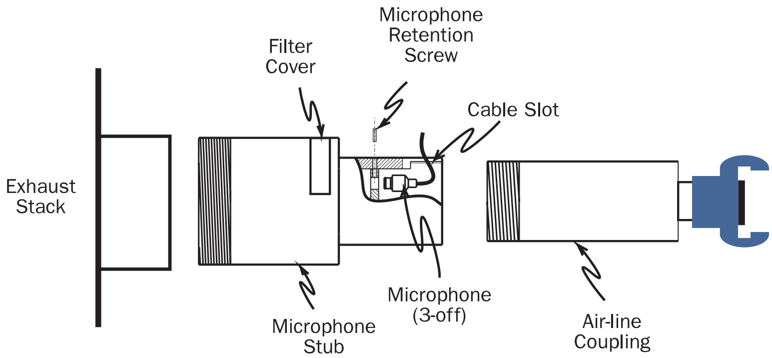


Figure 15 Microphone holder with air purge

Noise levels varied from 112dB to 138dB. Even when the microphone became covered in sludge (due to a malfunction in the air purge supply), the microphones all continued to operate. If noise control is required over a specified frequency range, then better results are obtained if the error signal is filtered (with either an analog or digital filter) prior to being used in the control algorithm. In many cases aerodynamic pressure fluctuations caused by fluid flow and travelling with the speed of the flow contaminate the microphone signals and reduce the achievable noise reduction. This problem can be ameliorated by either using turbulence filters (Hansen and Snyder, 1997) if the flow is uncontaminated or the microphone may be located in a small side branch as illustrated in figure 15. In extreme contamination cases it may be mounted in a side branch and protected by a thin mylar membrane or protected with an air purge system as shown in the figure. Note that the turbulence filter has the added advantage of making the microphone directional, which is particularly useful in reducing acoustic feedback to the reference microphone.

Reference sensors. These are often microphones, especially when random noise is to be controlled. The same comments apply as for error sensors, but for the control of random noise, band pass filtering the reference signal could have a detrimental effect on system performance as a result of the group delay through filter.

Reference signal quality. There are a number of reference signal characteristics which can limit the performance of an active noise control system and unfortunately, industrial environments often result in some of these characteristics appearing.

First, the presence of noise in the reference signal which is uncorrelated with the uncontrolled error signal will degrade the system performance for standard gradient descent (e.g. FXLMS) algorithms. The minimum possible ratio of controlled to uncontrolled error pressure squared at any given frequency, ω, is equal to $1 - \gamma^2(\omega)$, where $\gamma(\omega)$ is the coherence between the reference signal and uncontrolled error signal at frequency ω.

Second, if a microphone is used to provide the system reference signal in an active noise control system, then it is possible that some of the control signal will find its way back to the reference sensor. This feedback of the control signal to the reference signal will affect the system performance. If the feedback is relatively small, the control filter type must be changed from an FIR to an IIR filter. If the feedback is large, then the control system may become unstable even with an IIR filter. It is important to minimise feedback of the control signal to the reference signal where possible. Thus, it is usually preferable to use a tachometer signal rather than a microphone to provide the reference signal to the controller. The tachometer output is converted to a series of sine waves corresponding in frequency and amplitude to the frequencies and relative amplitudes of the harmonics to be controlled. Of course, this is only useful if the noise to be reduced is one or more harmonics of the rotational speed of the equipment producing the noise. If this is not so, then a microphone located near the noise source or an accelerometer mounted on the noise source (if it is a vibrating structure) must be used. When microphones are used as reference sensors the influence of the control sources on the reference signal may be minimised by using directional microphone arrangements.

Third, the majority of active noise control systems operate in the time domain. This means that the system operates on the "shape" of the reference signal, rather than explicitly manipulating its frequency components. The control system operates so as to remove that part of the error signal which is correlated with the reference signal. This means that the control system will attempt to remove the dominant frequency component of the reference signal which is also present in the error signal, even if it is not the target of the implementation. For example, if the target of an active noise control implementation is a 30Hz signal, and a 20Hz signal (which need not be controlled in this particular application) is also present in both the reference and error signals, then the controller will attempt to minimise both the 20Hz and 30Hz components. If the amplitude of the 20Hz component of the reference and error signals is 10dB above that of the 30Hz component, then the controller will attempt to attenuate the 20Hz component by 10dB before it explicitly attenuates the 30Hz component. The control system can be viewed as having a "flattening" affect on the spectrum, where the frequency components in the error signal which are also present in the reference signal will be "pushed" down to approximately equal amplitudes. The control system can also be viewed as having a limited amount of "energy"; the attenuation per frequency component is reduced as the number of frequency components requiring attenuation is increased. For example, assuming that the control system can attenuate both the 20Hz and 30Hz signal components, and if the amplitude of the 20Hz component of the signals is 10dB above the 30Hz component, then the 20Hz component may be attenuated 15dB, while the 30Hz component is attenuated 5dB (the final spectrum is "flattened"). If the 20Hz component is removed, the 30Hz component may be attenuated 15dB. However, if the controller cannot attenuate the 20Hz component, perhaps because the transducers were designed to operate at 30Hz and cannot provide adequate output levels for attenuation at 20Hz, and the 20Hz component of the signals is dominant, the control system will go unstable as it tries to achieve the impossible task of attenuating the dominant signal component. The lesson here is, be sure that the target frequencies are dominant in the reference and error signals, not just present. In an optimal situation, the relative amplitudes of the reference signal spectrum would reflect the amount of control desired at each frequency.

Group delay. For the control of non-periodic noise, the group delay through the path from the reference signal input to the controller output must be less than the time it takes for the acoustic disturbance to travel from the reference sensor to the control source. If this is not the case, the system will be non-causal and the control system performance will be degraded. Of course, this is not true if the noise to be controlled is periodic.

The group delay through the A/D and D/A converters and associated anti-aliasing filters is dependent on the converter type. For successive approximation converters, which can cost hundreds of dollars for 16 bit accuracy (the appropriate accuracy for an active noise control system), the group delay is very small and much less than the delay through the associated anti-aliasing filter. The delay through an anti-aliasing filter can be calculated from the following relationship (assuming a filter cut-off frequency of one third of the sample rate),

$$\tau \approx 3MT/16 \quad \text{seconds}$$

where T is the sample period (reciprocal of sample rate in Hz) and M is the number of poles in the anti-aliasing filter (typically between 4 and 8). Note that the delay through the reconstruction filter on the controller output is calculated using the same relationship. An additional delay of approximately one sample period, T, is usually allowed for the processing time of the control filter.

For sigma-delta converters which cost tens of dollars for 16 bits, the group delay is approximately 30 samples (including the delay through the in-built anti-aliasing filter) and thus is dependent on the sample rate. For the latter converter type, to minimise group delay, it is necessary to sample the incoming signal at the highest possible rate, then pass it through low pass digital filters (the group delay of which may be calculated using the equation above) and down-sample in software to the optimum rate for the particular frequency range which the active noise control system is to address. Note that for good results, the sample rate of the data used in the control algorithm and control filters should be between 4 and 100 times the frequency of the signal to be controlled (with the optimum being approximately 10 times). Adaptation of the controller at the lower frequency end of the above scale ($100f_s$) tends to be very slow, while at the other end of the scale, the performance deteriorates rapidly. Typical maximum sample rates of low cost sigma delta converters are 40,000Hz to 50,000Hz.

EXAMPLES OF APPLICATIONS OF ANC WHICH ARE PRACTICAL

Active control of sound propagation in ducts. Active control of noise propagating in ducts is well suited for the control of low frequency noise where the attenuation which can be achieved using conventional passive silencers may be inadequate. Elements of active systems are usually small and can be mounted in the duct wall, thus minimising air flow pressure losses. Disadvantages of active attenuators are associated with their cost (although this is rapidly decreasing), the need for regular maintenance (speaker replacement every three to five years), the requirement for custom installation and testing by experts, the reduction in performance at frequencies above the first higher order mode cut on frequency and the fact that they often only function well in relatively long (over 3m) sections of duct. Although most installed systems are only intended to control plane wave propagation, the installation of one commercial system which also controls one higher order mode has also been reported [13].

Nevertheless this application of active noise control is the oldest and is now the most commercially successful with numerous systems installed in industry in the USA. Typical results achieved are 15-20dB over two octaves of random noise and 20 to 30dB for tonal noise. Typical frequencies which are controlled range from 40Hz to 400Hz.

Applications of duct noise control include: reduction of noise in air conditioning ducts; reduction of noise in industrial blower systems; and reduction of vehicle exhaust noise. Although vehicle exhaust active mufflers have been demonstrated which perform as well acoustically as their passive counter parts (but without the undesirable pressure drop), they have yet to be used commercially on a widespread basis because of their expense and unreliability (due to the harsh environment). Recent developments involving the use of a radically new type of noise source [14] have revolutionised the possibility of practical active control of engine exhaust noise. The device consists of a rotatable flap placed in the exhaust flow and angled up from the exhaust pipe axis by about 12° as illustrated in figure 16. The active control system is used to oscillate the flap back and forth ±3-5°. As might be expected this device consumes very little power and is reported as being extremely effective.

Figure 16 Flapper valve ANC source

Active control of sound radiation from vibrating structures. Generally it is desired to reduce the total sound power radiated by a vibrating structure rather than the sound pressure at one or two locations. Thus the error sensors must be of sufficient number and arrangement so that they can observe the total radiated sound power. A trade off will almost invariably have to be made between the practical number of error microphones and the need to accurately measure the radiated sound power so that the maximum reduction theoretically achievable with the control source arrangement can be realised.

To control the sound radiated by a vibrating structure, either vibration sources attached to the structure or acoustic sources located in the acoustic medium surrounding the structure, or a combination of both may be used. Similarly, error sensors may be either structural vibration sensors which sense only the vibration modes contributing most to the radiated sound or they may be one or more microphones placed strategically in the acoustic medium surrounding the structure.

Two physical control mechanisms are usually associated with the use of vibration actuators to reduce sound radiated by vibrating surfaces. One is referred to as "modal control" and involves the reduction in the response of the structural modes contributing to the sound radiation. The second mechanism is referred to as "modal rearrangement" and involves rearrangement of the relative phases of the modes contributing to the sound radiation in such a way that the overall radiation efficiency is reduced without necessarily involving an overall reduction of the modal vibration amplitudes.

When acoustic control sources are used, the mechanism responsible for a reduction in radiated sound power was found to be a change in radiation impedance "seen" by the panel as a result of the presence of the acoustic sources. Thus the acoustic sources act to "unload" the panel. Clearly, it is not possible then for a single small acoustic source to provide a significant reduction in the power radiated by a large structure such as an electrical transformer, as such a source could not acoustically unload the transformer. However this does not preclude the control source from providing local areas of sound cancellation at the expense of other areas of increased sound level. Electrical transformer noise control is one well known application of the active control of sound from vibrating surfaces which has been addressed by a number of researchers in recent times. However, working systems are not yet commercially available.

Active headsets and ear muffs. It is well known that conventional passive hearing protectors are not very effective in protecting the wearer from low frequency noise, and that communication using standard headsets in noisy areas is extremely difficult. Both active headsets and active hearing protectors enhance hearing protection at low frequencies (Usually below 1500Hz). Active hearing protectors differ from active headsets in that the former include passive elements to further attenuate high frequency sound (above 250Hz), and the latter allow radio communication to be heard clearly.

The active control of noise at the ear using an active headset or hearing protectors are is a similar problem to that of active control of plane wave noise propagating in ducts, in that the problem is one-dimensional. The one-dimensionality of the problem enables good results to be achieved with a single channel control system.

The need for increased performance of passive hearing protectors in a number of applications is well established, especially in the low frequency range where the performance is particularly poor. In addition to the needs in noisy industries such as sheet metal and forging, better hearing protectors are needed for occupants of tracked military vehicles and military aircraft.

There are a number of active hearing protectors commercially available. Most of them have problems functioning in high level or impulsive noise environments as a result of the difficulty in producing a high level cancelling signal with the small loudspeaker traditionally used. In many cases in the presence of high level noise, the loudspeaker in the headset will be over driven which can also cause the control system to become unstable. The problem with impulsive noise is associated with the feedback nature of the active control systems used in headsets which means that they can oscillate unstably if subjected to a high frequency input. Current research is addressing these problems and also investigating the effectiveness of digital feedforward and adaptive feedback systems [15,16]. It is already clear that active hearing protection for low frequency noise is viable and it is here to stay.

Sound transmission into enclosed spaces. When sound is transmitted from outside an enclosed space to inside, such as propeller noise into an aircraft fuselage, the outside disturbance first sets the enclosing structure into motion. The structural vibration modes then couple with the interior acoustic modes, resulting in an energy transfer from the structure into the acoustic space. For structures which are at least of "moderate" size, which constitute the vast majority of enclosed spaces targeted for active noise control, and where the acoustic medium is not particularly dense, such as air, the response of the structural/acoustic system can be considered in terms of the structural *in vacuo* mode shapes, the acoustic cavity rigid-walled mode shapes, and the modal coupling between the two. Not all structural modes will excite all acoustic modes; in fact, quite the opposite. For modal coupling to occur, the product of the structural and acoustic mode shape functions at the structural/acoustic boundary (the wall), integrated over the entire contacting area, must be a non-zero number. For this type of coupled system, the total response can be considered in two regimes; structure-controlled, where the majority of the total system energy is in the shell, and cavity-controlled, where the majority of the total system energy is in the acoustic space.

The basic components of an active noise control system to control noise transmitted into an enclosure, are shown in Figure 17, where the aim is usually to minimise the acoustic potential energy (or mean squared sound pressure integrated over the enclosure volume). The mean square sound pressure averaged over the enclosure volume is usually approximated with a number of strategically located microphones. The optimum microphone locations are at the locations of the greatest difference in sound pressure between the primary and optimally controlled sound fields (assuming that potential energy was being accurately measured.

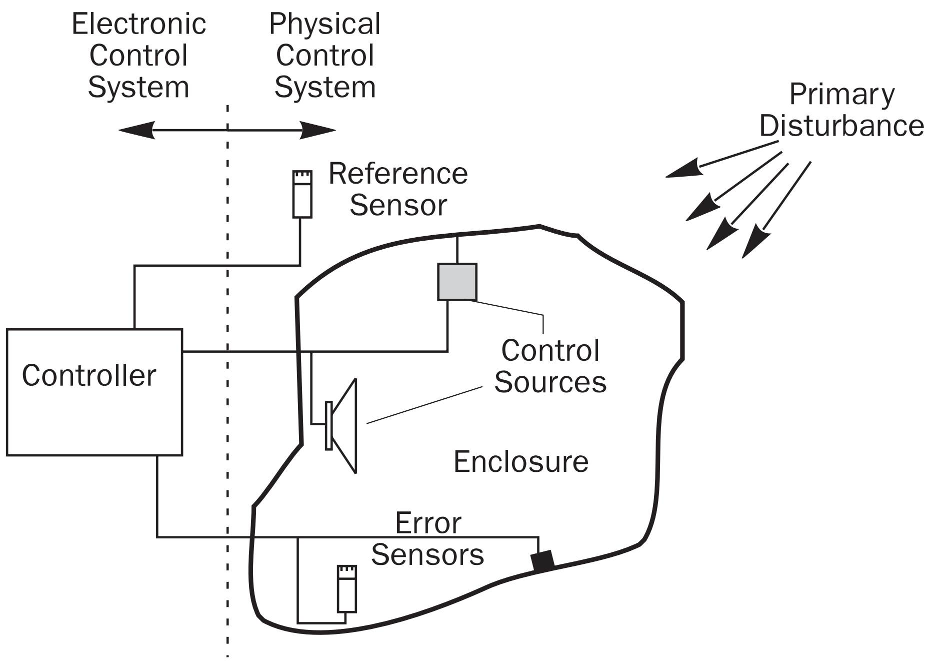


Figure 17 Active noise control system for reducing sound transmitted into an enclosed space.

Noise levels transmitted into an enclosed space may also be controlled using vibration sources driving the enclosure boundary structure. Vibration control sources achieve global sound control by altering the velocity distribution of the structure. This can have two different effects, corresponding to either modal control or modal rearrangement of the structural modes.

Practical applications of active control of noise transmitted into enclosures include: global reduction of low frequency tonal noise in propeller aircraft (e.g. the SAAB 340 or the DC-9); local reduction of broadband noise in large aircraft using active headrests [9]; global reduction of low frequency road noise in cars [17]; and reduction of low frequency sound transmission through double panel walls [18]. Reductions achieved range from 6 to 20dB, depending on the application.

Active vibration isolation. It is necessary in some cases to vibration isolate mechanical equipment from support structures to prevent the transmission of vibratory energy to these structures and the subsequent noise radiation. As periodic vibratory energy is the most common problem, the following discussion will be restricted to it. Custom active vibration isolation systems have been developed for some special problems and these usually involve a multi-channel feedforward control system driving control actuators which are placed in parallel or in series with an existing passive isolator. Control actuators may be piezo-electric stacks, magnetostrictive rods, electrodynamic shakers, hydraulic shakers or electromagnetic drivers (as in loudspeakers).

Although the general principles of active vibration isolation are similar to those discussed in the previous sections for active noise control, there are some added complexities. For example, equipment support systems usually exert moments and horizontal forces as well as vertical forces on the support structure and these generate longitudinal and shear waves as well as bending waves. Although bending waves are responsible for any appreciable sound radiation, longitudinal and shear wave energy can be transformed to bending waves at structural discontinuities. Thus it is important that all wave types are controlled. For this reason it is necessary for an active isolation system containing multiple actuators to be controlled by a multi channel controller and not by individual single channel controllers for each actuator. This allows the actuators to be adjusted together to minimise some cost function which may be the total vibratory power flow through the isolators to the supporting structure or the mean square vibratory energy at a number of locations on the support structure or even the radiated sound measured at a number of error microphone locations.

Applications of active vibration isolation for noise control include: reduction of engine noise transmitted into passenger cars in the low to mid-frequency range; and reduction of noise generated by Naval ships.

EXAMPLES OF APPLICATIONS WHICH ARE IMPRACTICAL FOR PHYSICAL ACOUSTICS REASONS

Global reduction of broadband or high frequency tonal noise in large aircraft. This is not possible because in the case of broadband noise, a suitable reference signal is not available and for the case of high frequency tonal noise, the sound field is too complex to be cancelled with a reasonable number (and acceptable locations) of control sources and error sensors. However, it is possible to achieve local areas of cancellation of low frequency random noise (below 350Hz) in the vicinity of seat headrests by using an independent single channel feedback control system in each headrest. It may also be possible in the future to use individual feedback systems driving actuators fixed to the fuselage to control boundary layer noise in aircraft [19].

Reduction of broadband or transient noise outdoors. This form of active control is physically impossible because of the complexity of the sound field involved and the lack of time for the controller to adapt the control source output to provide the required cancelling field.

Global reduction of tonal noise in a large space such as a factory which contains many noise sources. Active control in a room works best if the room is excited at one or more resonances. The amount of control achievable with a manageable number of control sources and error sensors is a reduction in noise level to the level which would have existed if the original noise source had excited the room at a frequency lying between two resonance frequencies. Thus for sound at frequencies between room resonances, the amount of control achievable is small. In a large space in the audio frequency range, the acoustic resonance frequencies are so close together that the difference between the peaks and troughs is very small, thus physically limiting the effect of active control, even at room resonance frequencies, to an insignificant amount, regardless of the capability of the controller, unless it is possible to use a sufficient number of control sources near each primary source so that all primary sources are acoustically "unloaded".

Global reduction of broadband noise in a large factory. The lack of availability of a causal reference signal generally makes this application impossible for feedforward control and the lack of correlation in most random industrial noise fields makes the application impractical for feedback control.

Reduction of traffic or aircraft noise transmitted into a building. As this noise may be considered random and is generally transmitted into the building by way of many different paths, its control is impractical. Even if a good causal reference signal were obtainable, the large density of acoustic resonances and their large damping mean that the resonance peaks would not be much greater than the troughs and the amount of control achievable with a perfect feedforward or feedback controller would be very limited for this reason, even if the incoming noise were tonal.

COMMERCIAL CONTROL SYSTEM REQUIREMENTS

One of the great impediments to the successful commercialisation of active noise control on a large scale is that the tools which are commercially available are grossly inadequate. Here, low cost, high volume, mass market applications are not discussed as these require the development of low cost custom controller hardware and software which is best left to companies which are capable of developing the required software and hardware themselves. The relatively slow development in this area may be attributed partly to the potential for litigation arising from perceived patent infringement (regardless of whether it is justified or not) and partly to a basic lack of trust on the part of potential customers as a result of a minority of active noise control companies in the past not delivering promised results and exaggerating the potential of active noise control in the national and international media. However, existing active noise control companies are gradually making inroads into this market.

Rather the recommendations outlined below are directed at "one-off" applications for which the labour cost involved in achieving a satisfactory solution is often much greater than the hardware cost.

There are a large number of examples in industry where extraordinarily expensive passive control measures or no measures at all have been implemented because the use of active noise control is seen as too difficult and not sufficiently reliable. What is needed to increase the scale of active noise control activity for these unique applications is a system that can be used successfully by any competent engineer with a very basic understanding of acoustics. The control system aspects should be transparent to the system user. Even though the technology to do so exists today, the development of an ideal control system will require a prolonged and dedicated effort from a number of talented people working together. Here, some of the more important requirements of such a system are discussed. The discussion will be divided into three parts: controller hardware; software for driving the controller hardware to produce the desired active noise control signals; and software for analysing the system to be controlled to provide advice on the optimum locations of control sources, reference sensor and error sensors from data on possible locations provided by the user.

Controller hardware. Two types of system should be available: low cost fixed point system for the control of periodic noise; and a higher cost floating point system for the control of random noise. Although the hardware cost of the fixed point systems is much less (thus making them attractive for mass market implementations), the required software is much more complex. For example, quantisation problems are larger for the fixed point systems so convergence coefficients cannot be too small and care has to be taken in handling situations in which the control filter weights have a large spread in magnitude (a particular problem with IIR filters). Software floating point or block floating point calculations can ease this problem with an associated loss in processing speed and increase in group delay through the controller (not a problem for periodic noise).

Software selectable anti-aliasing filters and signal shaping filters should be available so that the controller performance can be optimised for various user defined frequency ranges or tonal harmonics.

The hardware should be modular in construction so that the basic unit is a single reference, control output and error channel and the latter two could be multiplied to 100 channels simply by using more densely populated printed circuit boards and more of them. Allowance should also be made for multiple reference signals. The interconnection between boards, once they are installed in their box should be transparent to the user.

The maximum input and output voltages should be variable and depend on user supplied information regarding the type of microphones and maximum allowable input voltage for the control sources and associated amplifiers.

The controller manufacturer should also be able to supply a range of loudspeakers and microphones which are compatible with the controller and suitable for active noise control applications. Application notes should also be available which provide designs for protecting speakers and microphones in harsh environments. In addition tachometers and associated conditioning electronics necessary to provide an appropriate reference signal should be available from the controller manufacturer. The conditioning electronics should be sufficiently flexible that the user can select the number of harmonics and relative amplitudes so that the controller performance can be optimised.

The control system should be "stand-alone and sufficiently rugged to use in an industrial environment. However, all set-up and monitoring of control, reference and error signals should be done using a PC and a special windows interface. There should also be a capability for a modem connection to allow interrogation and monitoring of the system over the telephone. Of course it is essential that the control system be able to continually monitor its own state of operation as well as the state (faulty or not) of all sources and sensors to which it is attached. There should be provision for an alarm indicator to show where a particular fault may exist. The alarm functions should be able to be easily connected to a PLC or the main computer control centre of a particular industrial facility.

The control system manufacturer should also provide application notes, indicating types of problems which can be controlled and other problems which should not be attempted due to physical acoustics constraints.

Controller software. The software for driving the controller hardware must be flexible; it must be able to optimally control a given system of reference and error sensors and control sources over a user specified frequency range automatically and it must allow manual intervention by the user. Where the frequency range over which control is desired extends over one decade, the controller must automatically divide the frequency range into manageable parts and use a different control filter for each part. In this case, the controller will automatically sample the input signals at the highest rate required (100 times the highest frequency to be controlled) and use down-sampling and digital anti-aliasing filters to adjust the sampling rate for the optimum for the lower frequency ranges, bearing in mind that the desired sampling rate range is between 8 and 100 times the frequency to be controlled. The outputs from the various control filters are then summed prior to being output to the control source.

The software must also be capable of automatically selecting for any particular problem, the number of control channels (by identifying which channels have control sources or error sensors attached), optimum sampling rate, control filter and algorithm types and length (up to 1000 taps), convergence coefficient (size and fixed or variable), leakage coefficient (control source effort weighting), output and input gains and on-line system ID type and appropriate parameters. A Microsoft Windows interface for the user is also essential. The automatic option should be capable of producing a working control system with minimal input from the user.

The software should also be capable of determining from the quality of the supplied reference signal whether a feedforward, feedback or hybrid feedforward/feedback controller will provide the best performance. The performance measure (tonal noise and wide band reduction goals) will be provided by the user. It is expected that the software would direct the user regarding the desirability of a tachometer reference signal or other non-acoustic sensor may be preferable for the reference sensor. The advice would be based on the level of acoustic feedback from the control source(s) to the reference sensor and the level of noise in the reference signal which is uncorrelated with the error signal(s).

Manual intervention by the user must be available on two levels. The first will allow the user to adjust simple things such as number of channels, which error sensors are to affect which control sources, sample rate, control filter and algorithm types, control filter lengths, input gain, output gain, convergence coefficient (choice of size and fixed or variable), leakage coefficient, system ID algorithm and associated parameters, and signals to be monitored in real time using the windows interface.

The second level of manual intervention is at the level which allows users to load their own control algorithms and filters (written in "C") into the controller and have them added to the windows menu. This will require the development of a translator from "C" to assembler code and software which will allow the user to include the new algorithms and filters in the controller menu used for manual selection of controller parameters. Software which users may wish to add may include neural network filters, new system ID algorithms and genetic filter adaptation algorithms.

Acoustic system software. This software is needed to assist the user to optimise the physical layout of the reference sensors, control sources and error sensors. Based on the frequency range over which control is desired, the software should also be capable of advising on the minimum required physical separation between the reference sensor(s) and control source(s). The user input data would be measured narrow band transfer functions between each possible control source location and each possible error sensor location over the frequency range to be controlled. Of course, these measurements would be semi-automated and require a minimum of effort from the user. Where a large number of possible locations exist, the software should use frequency range information entered by the user to provide advice on recommended separations between adjacent locations prior to the measurements. In addition, measurements of the sound field to be controlled at potential error sensor locations over the frequency range of interest would also be required. The technology for doing this already exists [7,8]; a user friendly software package is all that remains to be developed.

CURRENT RESEARCH DIRECTIONS

In addition to moving active noise control technology to where it is today, research in this area has created a resurgence in the interest and understanding of the nature of the interaction of acoustic fields with sound sources and the nature of the interaction between vibrating structures and their radiated sound fields.

Current research in active noise control may be divided into two main categories; research directed at improving generic control system performance and research directed at applying existing control system technology to a particular application. Only the former type of research will be considered here.

Sound sources. Some manufacturers (e.g. Motran Industries in the USA) are spending effort in improving the ruggedness and maximum continuous power rating of loudspeakers so they will be more useful for industrial applications of active noise control. Others are developing alternative acoustic sources such as tuned curved panels for control of electrical transformer tonal noise, and flapper valves for exhaust noise control [14]. The latter source is illustrated in figure 16 and is an innovative way around the difficulty of using loudspeakers in such a hostile environment.

A further example of a different type of control source is a tuned resonator for which the resonance frequency is tuned by the output of an active noise controller. The control could take the form of a moveable piston which changes the volume and hence resonance frequency of the resonator. A different type of tuned absorber uses a loudspeaker, microphone and feedback control system [20].

For sound radiated from vibrating surfaces, a considerable effort has in the past been spent on the development of suitable actuators. Current research is directed at the development of shaped piezo-electric actuators which will only control the part of the surface vibration responsible for sound radiation and will have a minimal effect on the vibration components not responsible for sound radiation. One of the major problems with this work is obtaining shaped actuators with sufficient force generation capacity. Another form of control source which is the subject of current research activity is the tunable vibration absorber which relies on the active control system to tune the stiffness (and sometimes the damping) of a variable stiffness vibration absorber so that the tonal sound field radiated by the vibrating structure is minimised.

Error sensors. The elusive search for the ultimate "smart" error sensor has dominated recent research in this area. Considerable progress has been made in a number of specific areas and this progress will form the basis of the following discussion.

It is well known that minimising sound levels at one or more error sensors is the traditional task of an active noise control system. It is also generally hoped that the sound field will be reduced in other locations as well and perhaps even on a global basis, although this is a forlorn hope unless the acoustic control sources are close enough to the primary source to effectively unload it. One problem with the approach just described is that the error microphones are located at the sound field minima and this prevents the listener from occupying the quietest area. Recent research has been directed at the idea of projecting the sound field minima to locations away from the error microphones for application to active headrests in aircraft and considerable success has already been reported [9,21]. There is clearly potential to develop these ideas to other applications.

Another area of interest has been in the development of power sensors which involves the combination of a number of far field microphone signals in such a way that the result is proportional to the sound power of the source generating the sound field [22]. This technique has the ability to greatly reduce the number of error sensors needed to minimise the sound power radiated by a vibrating body or surface. The control sources used so far with this method have been vibration actuators attached to the vibrating body.

Sound power can also be determined by using sound intensity measurements at locations close to a vibrating surface. If the sound intensity is minimised at a sufficient number of points around a noise radiator, then the sound power will be minimised. The problem with the use of sound intensity as an error signal is that it is uncorrelated with the reference signal (as it is derived from the product of two signals having the same frequency spectrum as the reference signal with the result of a signal with a d.c. component and a component with a spectrum in which the original frequencies have been doubled). This makes the use of a sound intensity signal difficult as an error signal for controlling random noise. However, for the control of tonal noise, it is possible to use the sound intensity signal as an error signal by low pass filtering it to remove the double frequency component and then multiplying the d.c result with the reference signal. If this is used as an error signal in the controller incorporating an FIR filter and FXLMS algorithm, then the sound intensity will be minimised as shown in recent work [23]. Where there are multiple tones (such as generated by an electrical transformer), it is necessary to band pass filter the signals prior to using them to calculate sound intensity. Thus, if there are three tones of interest, there will be three separate intensity signals and a three channel controller will be required for each error signal; one channel for each tone. It remains to be seen whether or not a modified form of this technique will work for random noise signals.

The great advantage of sound intensity over sound pressure as an error signal for outdoor sound sources is that the ANC system can be more compact and is likely to require significantly fewer near field intensity error sensors than far field pressure sensors in the far field. The use if intensity error sensors also gets around the controller instability problems resulting from the large number of channel and large acoustic delays in the cancellation path.

A different form of energy sensor which has been developed recently [24] as an effective sensor for enclosed sound fields is the pressure and velocity sensor which allows the total energy (potential plus kinetic) to be used as a cost function for control. Although the potential energy and the kinetic energy can vary greatly as a function of position in an enclosed space, the total energy is fairly constant and herein lies the advantage of the pressure/velocity sensor over the traditional pressure sensor. With the pressure/velocity sensor, the total energy (and hence the potential energy) can be minimised with just one measurement, whereas if potential energy is to be minimised by using only pressure sensors, a large number of measurement points is required. This greatly simplifies the complexity of the controller needed to reduce the sound levels in an enclosed space.

Another possible approach to the problem of controlling sound radiated by outdoor sound sources such as electric power transformers is to use a transform method which allows the far pressure field to be determined from a suitable transform of near field measurements. One problem could be the potential variability in the transfer functions from the near to the far field due to environmental variations such as wind and temperature gradients and atmospheric turbulence. The possible use of some form of spatially averaged acoustical holography or wavenumber transforms remains to be investigated.

Another sensing field that has received a large amount of recent attention is the use of structural vibration sensors which have been shaped such that they provide a signal proportional to the radiated sound power; that is, they sense "radiation modes" and not just structural vibration modes [25-29]. Each sensor senses just one radiation mode which is orthogonal with all other radiation modes in terms of acoustic radiation; that is, if the amplitude of one radiation mode is reduced, the radiated sound power is guaranteed to be reduced. The analogy with structural modes is that the latter are orthogonal with respect to structural vibration so that if the vibration amplitude of one structural mode is reduced, the overall structural vibration will be reduced. Radiation modes are made up of varying proportions of normal structural vibration modes and the relative proportions are usually a weak function of frequency. This means that if a number of point sensors are used, their relative weightings to generate an error signal would be a function of frequency and if a continuous sensor such as flexible piezo-electric film (PVDF) bonded to the structure were used, the required shape would be frequency dependent. The weakness of the dependence on frequency is often exploited to generate a practical sensor with a fixed shape that can act over a reasonable frequency range. Recent work [30] has shown that using a number of point sensors rather than a distributed sensor to produce an error signal proportional to a radiation mode is not really practical in structures such as aircraft fuselages owing to the very large number required as a result of "spillover", a process whereby the individual sensor responses may be dominated by higher order mode contributions which are not part of the radiation mode being controlled.

Current work is focussing on 2-dimensional sensors for plates [29] and use of shaped sensors to control sound radiation into an enclosed cylindrical space with application to active control of aircraft interior noise [30]. It is of interest to discuss some of the new results arising from the latter study. Figure 18 shows some numerical results achieved using two radiation mode sensors to control interior noise in a cylinder, 3m in length and 0.9m in diameter, excited by a single point force and controlled by a single control force located remote from the primary force. Both forces were located randomly. The results in figure 18 compare the interior potential energy achieved by using as cost functions: the interior potential energy; the structural kinetic energy; and the amplitudes of the two dominant radiation modes, where it is assumed that the radiation mode sensor shape is optimised at each frequency. As expected, minimisation of the structural kinetic energy is ineffective, but minimising the amplitudes of the two radiation modes is as effective as minimising the interior potential energy. It is also clear that the greatest noise reduction occurs at the resonance frequencies of the acoustic modes (57Hz, 115Hz, 173Hz, 220Hz, 228Hz, 233Hz, 244Hz, 250Hz and 251Hz). In figure 19, similar results to figure 18 are shown, but in this case the distributed sensor shape is fixed for all frequencies at the optimum value for 57Hz and 220Hz respectively. It can be seen that the sensor with a shape fixed at 57Hz works well up to about 150Hz and the 220Hz sensor works well above about 70Hz.

Finally results are shown in figure 20 for the first radiation mode sensor shape fixed at the optimum for 117Hz and the second fixed at the optimum shape for 220Hz, with a cross-over network set up so that only the first mode is minimised at low frequencies and the second at high frequencies which effectively requires a single channel controller. As can be seen from the figure, the results obtained are nearly as good as those obtained by minimising the interior potential energy which would require many microphones distributed throughout the cavity and a many channel control system. One interesting discovery made during this research was that the optimum shape for the structural radiation mode sensor, derived using singular value decomposition of the radiation coupling of all structural modes with the interior acoustic modes, was very close to the mode shape of the interior acoustic field to be controlled. In fact, it was found that when the sensor shapes were made the same as the interior acoustic field shape, very similar results were obtained. This finding has enormous impact on the level of effort required to find the optimal sensor shapes for complex enclosures enclosed by structures with high modal densities.

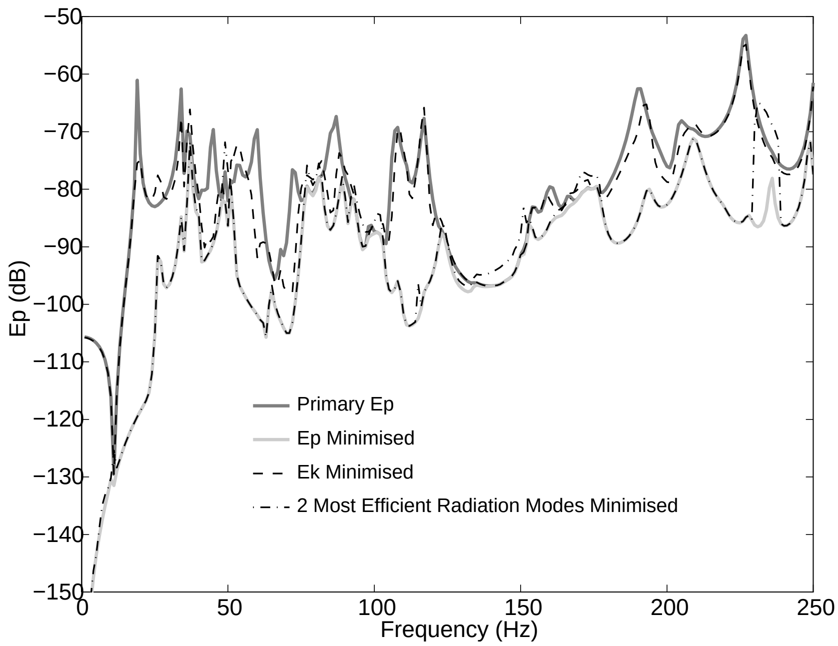


Figure 18 The effect of various error criteria on the achievable reduction of potential

energy in a cylindrical enclosure excited by a point force

Optimisation of control source and error sensor locations. It has already been explained that optimum location of the control sources and error sensors is an extremely important part of optimising control system performance. It seems that current research is focussed on the use of genetic algorithms for this purpose [5,7,31,32]. However, the use of multiple regression as an optimisation technique [2,8] should not be forgotten, especially the important knowledge that the optimal error sensor locations are at the locations of the largest difference between the primary and optimally controlled sound fields.

Control filters and algorithms. Work is continuing on the development of non-linear control filters appropriate filter weight update algorithms to allow compensation for non-linear control sources and non-linear systems being controlled [5,6]. The effect of the non-linearities is the generation of frequencies in the controller output which are not present in the reference signal. Current work is directed towards algorithms which show faster convergence characteristics than the existing algorithms.

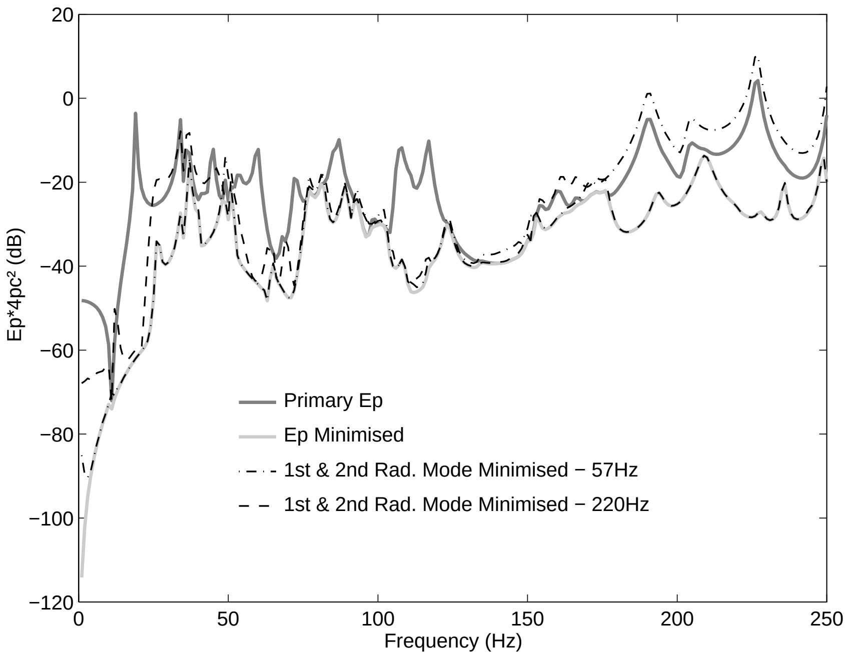


Figure 19 The effect of fixing the shapes of the first and second radiation mode sensors at the optimum shape for 57Hz and 220Hz respectively

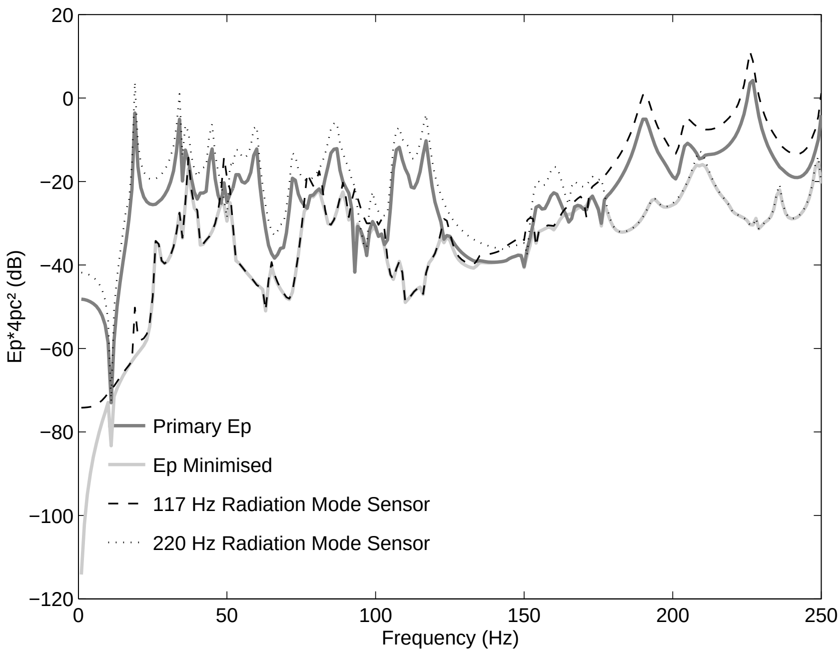


Figure 20 The effect of using a single radiation mode error signal; the first radiation mode for low frequencies (shape fixed at the optimum for 117Hz) and the second radiation mode for high frequencies (shape fixed at the optimum for 220Hz)

Another area of research involves the use of block processing and associated algorithms in time and frequency domains. It has been shown [33] that processing blocks of error signal data rather than individual samples can improve system stability and convergence speed of time domain controllers. Frequency domain controllers, by their very nature involve processing of blocks of data. Frequency domain control is advantageous from the point of view that it allows the use of frequency dependent convergence coefficients which in turn allows certain components of a multi-tonal noise to be controlled more than others. For periodic noise, a similar result could be achieved in the time domain by suitable band pass digital filtering of the reference signal; however for random noise, the increase in group delay associated with the latter technique may be unacceptable. Current research with frequency domain control is concentrating on the development of algorithms with faster convergence rates [34]. Other means of increasing the convergence speed of the filtered-x LMS algorithm for multi-channel systems involves the use of fuzzy logic to optimise the convergence coefficient at each filter weight iteration [35].

More research is needed on systems and associated control algorithms which combine feedforward and feedback control [3] so that the resulting system will exhibit the best possible performance. One particular application involves the reduction of a random noise containing a number of tones. In a well designed feedforward/feedback system, the feedback system would concentrate on attenuation of the highly correlated tonal noise, leaving the feedforward system to concentrate on the random noise components.

It is well known that IIR filters are necessary in systems exhibiting acoustic feedback from the control source to the reference sensor or in systems containing a number of resonances. However there are associated problems with the control algorithm in these cases converging slowly and often converging to a sub-optimal solution. Current research is directed at overcoming these problems [36].

In active control systems involving the single or multi channel FX-LMS algorithm, there is a need to know the transfer function of the cancellation path(s) (path(s) from the control source input(s) to the error sensor output(s)). In most practical systems, the transfer functions change significantly with time, so there is a strong need to determine them "on-line" while the control system is performing its task, without degrading the system noise reduction performance. Although various techniques exist for approximately achieving the required purpose [3], they all suffer from some form of limitation from lack of universal reliability to limiting the achievable system noise reduction performance. Thus, some current research effort is directed towards finding better algorithms and techniques for performing the required on-line system identification [37].

Recently there has been a renewed effort directed towards research on improving the performance of adaptive feedback control [14,38]. Work has also been directed at the use of feedback control to reduce sound radiated by randomly excited vibrating surfaces with a view to reducing turbulent boundary layer noise transmitted into aircraft cabins [19]. Feedback controllers optimised to maximise the amount of power absorbed from an acoustic system have also been the subject of renewed interest in recent times [39].

In the future, it is clear that there will be increasing interest in the development of "smart" structures containing distributed actuators and sensors. There is considerable current research effort devoted to the development of suitable strategies for controlling the noise radiated by such structures. One such strategy involves the use of on-off control [40].

Hybrid active/passive silencers. These silencers for duct systems are of special interest in that they are an example of the effective combination of active (feedback) and passive silencing techniques in the same device [41]. The concept of combining active and passive techniques in vehicle suspensions and vibration isolation systems has been used for some time now and it is clear that future research is likely to extend the concept to other devices and systems for active noise control.

CONCLUSIONS

It is clear that active noise control has a very strong future in industry. However what is not clear and difficult to predict is the time frame which will characterise forward progress. Many outrageous claims for future the technology have been made and it is clear that practical considerations as well as physical acoustics principles will prove some of these to be unfounded. However, at the same time there is a growing accumulation of successful commercial installations as well as some novel laboratory demonstrations which are gradually giving active noise control the credibility that it deserves.

It is difficult to believe that the transition of active noise control technology from laboratory to industry will be very rapid. It could be 10 or 20 years or even longer before we see active noise control used routinely in consumer products, vehicles and in industry. The time scale will be heavily dependent on the ability of active noise control system manufacturers to take up the challenge and produce hardware and software which meets the specifications outlined in this article and allows easy modification and addition as the results of new research comes to light.

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REFERENCES

1. "Use of adaptive filters in active ear defenders," C. Bao, and J. Pan, *Proceedings of Internoise* '96, pp. 1175-1178 (1996)

2. C.H. Hansen and S.D. Snyder, *Active control of noise and vibration* (E&FN Spon, London, 1997)

3. S.M. Kuo and D.R. Morgan, *Active noise control systems* (John Wiley, New York, 1996)

4. P.A. Nelson and S.J. Elliott, *Active control of Sound*, (Academic Press, London, 1992)

5. "Application of genetic algorithms to active noise and vibration control," C.H. Hansen, M.T. Simpson and C.T. Wangler, *Proceedings of the 4th International Congress on Sound and Vibration*, St Petersburg, Russia, pp.371-388 (1996)

6. "Genetic algorithm adaptation of non-linear filter structures for active sound and vibration control," C.T. Wangler and C.H. Hansen, *Proceedings of ICAASP '94*, p.505-508 (1994).

7. "Use of genetic algorithms to optimise actuator placement for active control of interior noise in a cylinder with floor structure," M.T. Simpson, & C.H. Hansen, *Noise Control Engineering Journal*, **44**, 169-184 (1996).

8. "Using multiple regression to optimise active noise control system design," S.D. Snyder & C.H. Hansen *Journal of Sound & Vibration*, **148**, 537-542 (1991).

9. "Active noise reduction in turbo-prop aircraft," C. Carme, V. Delemotte and A. Montassier, *Proceedings of the 4th International Congress on Sound and Vibration*, pp. 1925-1934 (1996).

10. "The state of the art of active-passive noise control," R.J. Bernhard, *Proceedings of Noise-Con '94*, pp. 421-428 (1994)

11. "Practical implementation of an active noise control system in a hot exhaust stack," C.H. Hansen, C.Q. Howard, K.A. Burgemeister and B.S. Cazzolato, *Proceedings of The Annual Conference of The Australian Acoustical Society*, (1996)

12. "Tuned loudspeaker enclosures for active noise control," L.A. Blondel and S.J. Elliott, *Proceedings of Internoise '96*, pp. 1105-1108 (1996)

13. "Active HVAC noise control systems provide acoustical comfort," H.K. Pelton, S. Wise, and W.S. Sims, *Sound and Vibration* **July**, 14-18 (1994)

14. "Industrial applications of pulsed flow," S. Renault, P. Micheau, J. Tartarin and M. Besombes, *Proceedings of Internoise '96*, pp. 1061-1066 (1996)

15. "Use of adaptive filters in active ear defenders", C. Bao and J. Pan, *Proceedings of Internoise '96*, pp. 1175-1178 (1996)

16. "Application of adaptive feedforward active noise control to hearing protectors," J. Pan et. al., *Proceedings of Active '95*, pp. 1319-1328 (1995)

17. "Active control of road noise inside automobiles," R.J. Bernhard, *Proceedings of Active '95*, pp. 21-32 (1995)

18. "????paper on double walls," J. Pan, *Proceedings of the Annual General Meeting of the Australian Acoustical Society* (1996)

19. "Feedback control of sound radiation from a plate excited by a turbulent boundary layer," D.R. Thomas and P.A. Nelson, *J. Acoust. Soc. Am.*, **98**, 2651-2662 (1995)

20. "Noise control in resonant sound fields using active absorbers," P. Darlington and M.R. Avis, *Proceedings of Internoise '96*, pp.1121-1126 (1996)

21. "A virtual microphone arrangement in a practical active headrest," J Garcia-Bonito, S.J. Elliott and C.C. Boucher, *Proceedings of Internoise '96*, pp. 1115-1120 (1996)

22. "Performance enhancement of structural/acoustic active control systems via acoustic error signal decomposition," S.D. Snyder and K. Burgemeister, *Proceedings of Internoise '96*, pp. 1141-1146 (1996)

23. "??Intensity sensing," C.Q. Howard, S.D. Snyder and C.H. Hansen, Submitted for publication in *The Journal of Sound and Vibration*.

24. "Global attenuation of broadband noise fields using energy density control," Y.C. Park and S.D. Sommerfeldt, *J. Acoust. Soc. Am.*, **101**, 350-359 (1997)

25. "Shaped vibration sensors for feedforward control of structural radiation," S.D. Snyder, N. Tanaka and C.H. Hansen, *Proceedings of the Second Conference on Recent Advances in Active Control of Sound and Vibration*, pp. 177-188 (1993)

26. "Radiation modes and the active control of sound power," S.J. Elliott and M.E. Johnson, *J. Acoust. Soc. Am.*, **94**, 2194-2204 (1993)

27. "On the exterior acoustic radiation modes of structures," K.A. Cunefare and M.N. Currey, *J. Acoust. Soc. Am.*, **96**, 2302-2312 (1994)

28. "Calculating total acoustic power output using modal radiation efficiencies," S.D. Snyder and N. Tanaka, *J. Acoust. Soc. Am.*, **97**, 1702-1709 (1995)

29. "Distributed parameter modal filtering using smart sensors," N. Tanaka, S.D. Snyder and C.H. Hansen, *Journal of Vibration and Acoustics*, **118**, 630-640 (1996)

30. "Active control of sound transmission using structural error sensing," B.S. Cazzolato and C.H. Hansen, submitted for publication in the *J. Acoust. Soc. Am.* (1997)

31. "Optimal placement of secondary sources for active noise control using a genetic algorithm," S. Pottie and D. Botteldooren, *Proceedings of Internoise '96*, pp. 1101-1104 (1996)

32. "A genetic algorithm for the simultaneous optimisation of the sensor and actuator positions for an active noise and/or vibration control system," D.A. Manolas, T. Gialamas and D.T. Tsahalis, *Proceedings of Internoise '96*, pp. 1187-1191 (1996)

33. "Multi-channel normalised X-block algorithm for active noise and vibration control," Q. Shen and M.R. Jolly, *Proceedings of Internoise '96,* pp.1219-1222 (1996)

34. "Time and frequency domain X-block least-mean-square algorithms for active noise control," Q. Shen and A.S. Spanias, *Noise Control Engineering Journal*, **44**, 281-293 (1996)

35. "A multi-channel fuzzy LMS algorithm for active noise control," H-D Nam, K-T Kim and W-S Cheung, *Proceedings of Internoise '96*, pp. 1137-1140 (1996)

36. "Application of fast adaptive IIR filter algorithms for active noise control in a kitchen hood," R. Schirmacher, *Proceedings of Internoise '96*, pp. 1193-1198 (1996)

37. "A study of online plant modelling methods for active control of sound and vibration," S. Laugesen, *Proceedings of Internoise '96*, pp. 1109-1114 (1996)

38. "A fast adaptive feedback controller for active noise control," T. Auspitzer, B. Rafaely and S.J. Elliott, *Proceedings of Internoise '96*, pp. 1021-1024 (1996)

39. "Acoustical prediction," P.A. Nelson, *Proceedings of Internoise '96*, pp. 11-50 (1996)

40. "on-off control", C.R. Fuller????

41. "Effective noise reduction with hybrid silencers," J. Kruger and P. Leistner, *Proceedings of Internoise '96*, pp. 1097-1100 (1996)