# SENSORS AND ACTUATORS FOR ACTIVE NOISE CONTROL SYSTEMS

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Successful implementation of an active noise and vibration control system requires an effective control system and a good understanding of the physics of the problem to be controlled. However, there can be no successful implementation without appropriate transducers for transforming acoustic signals into voltages for the electronic control system and for transforming voltages output from the control system into sound. Although a considerable amount of research has been devoted to control algorithms and optimal physical arrangements for sensors and control actuators, little has been written about the requirements for the sensors and actuators themselves. Here a number of practical actuator and sensor implementations are discussed for both fully active and semi active noise control systems.

## INTRODUCTION

There are numerous practical applications where active noise control has the potential to provide significant benefits. Algorithms [1-11] are available for many types of noise problem, ranging from multi-tonal to broadband and a considerable amount of work has been done by various researchers on extending the controllable bandwidth from 1.5 octaves to three or more [12]. However, in many of these cases, the implementation of a successful active noise control system is hindered by the lack of availability of robust, generic and low cost hardware. This hardware includes sensors and actuators as well as the control system itself. Requirements for a user friendly control system have been discussed elsewhere [13] so here the discussion will be restricted to control approaches and associated sensors and actuators. The discussion will also include vibration actuators and sensors as these are sometimes used to control sound radiation.

There are two main categories of active noise control: fully active and semi-active. Although both fully active and semi-active systems may use the same kind of sensors, they invariably differ in the kinds of actuator or control source that they use. Also the sensors and actuators that are most suitable for a particular application depend on the control strategy that is chosen. For example if the noise to be controlled is generated by a vibrating surface, it may be better to use vibration actuators to control the vibration of the surface rather than use loudspeakers adjacent to the surface.

A fully active system uses actuators to directly generate a cancelling or suppressing noise or vibration signal. An example of a feedforward fully active system to control noise propagating down a duct would consist of a microphone (referred to as a reference sensor) in the duct to measure the noise that is to be controlled, a loudspeaker (referred to as the control source), mounted in the wall of a duct downstream of the microphone, to introduce the cancelling noise and a second microphone (referred to as the error sensor), mounted in the wall of the duct, to measure the residual noise sufficiently far downstream from the loudspeaker, with the control system using the signals from the two microphones to generate the required signal to minimise the noise at the second (error) microphone. The required distance between the control source and error sensor is controlled by the delays in the loudspeaker response and the control system itself as well as possible near field effects in the vicinity of the loudspeaker. At room temperature, most systems will cope well with a distance of between 1 and 2 m. A feedback system would not require the reference microphone, and in this case the error microphone would have to be moved as closely as possible to the loudspeaker control source to minimise instability problems. Feedback and feedforward control systems are discussed in more detail elsewhere [14].

A semi-active system uses actuators to modify the dynamics of a system or to tune a passive noise-suppressing device so that the system produces less noise. Using our duct example, a Helmholtz resonator may be a good choice for suppressing tonal noise. However, if the frequency of the tonal noise varies (for example, it may be associated with a variable speed engine and the duct may be an exhaust pipe), the Helmholtz resonator would need to be adjusted to keep in synchronisation with the varying engine speed. The control system to automatically keep the Helmholtz resonator optimised as the engine speed changes is referred to as a semi-active system. Its input would be a signal that is proportional to the acoustic power propagating down the duct and its output would drive a motor that adjusted either the resonator volume or neck length to minimise the input signal.

## 1. ACTUATORS FOR FULLY ACTIVE SYSTEMS

Although many people have tried, it is difficult to beat loudspeakers as an acoustic control source in active noise control systems. Current development is aimed at maximising the life and maximising the physical protection of these devices in dirty industrial environments. In the past, loudspeakers have been manufactured with stainless steel and plastic diaphragms but these have been very expensive options and not as efficient as paper cone speakers.

The ANVC group at The University of Adelaide has spent a considerable effort experimenting with a number of methods to prevent the build up of contaminants on the speaker cone in instances where they are mounted on industrial exhaust stacks ranging from a cement plant [15] to a powdered milk processing plant. The final design, which is effective for stack temperatures of up to 90°C, is illustrated below.

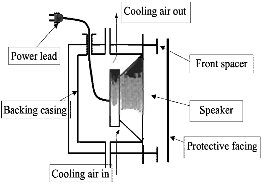


**Figure 1.** Schematic of the speaker unit

The speaker coil must be cooled either by compressed air or by wrapping a copper pipe around the coil and passing cooling water through it. It is also important that if compressed air is used, the air pressure is kept low to avoid damage to the loudspeaker. Note that there is a pressure equalisation passage between the rear and front of the speaker cone. This is essential as it prevents temperature differentials between the two sides of the cone from jamming the cone against one of its "stops". If the temperature in the exhaust stack is higher than 90°C, it may be necessary to direct some of the air flow over the front face of the loudspeaker by drilling holes in appropriate places in the spacer and outer casing. The above speaker enclosure design was arrived at after trying numerous protective coatings (including epoxy paint as well as automotive gasket spray covered with silver paint spray) on the speaker cone itself. Although the protective coatings withstood the normal operational environment in the milk powder processing exhaust stack, they were unable to withstand the daily aggressive steam cleaning with a highly alkaline product. To overcome this problem, a protective membrane as shown in the figure was used. First a 1.5 mm thick, rubber-like material known as "vyon" and used extensively in the dairy industry, was tried. This reduced the loudspeaker output by 20dB at the 180 Hz frequency at which control was needed and this meant that the maximum available amplitude of the controlling sound in the duct was too small. Although the transmission loss of the vyon was small, its high damping properties loaded the loudspeaker and suppressed its capacity to generate noise. Second a 0.1 mm thick printed circuit board material was tried and this resulted in a 10 dB reduction of speaker output capability, which was still too much. Finally a 0.1 mm thick sheet of mylar was used with a resulting 3 dB reduction in the speaker output, which was acceptable. A picture of the mylar installed in a spray dryer exhaust stack is shown below.

**Figure 2**. A speaker unit, mounted in the exhaust stack (view from inside the exhaust stack). Note the spout at the bottom to drain any residual liquid after cleaning.

Another type of actuator that is commonly used to control the vibration of structures that are radiating sound is the piezo electric crystal. When excited by an electrical signal, they expand and contract proportionally to the voltage across the two largest faces, thus causing a cyclic strain on the structure to which they are attached. These crystals commonly range in thickness from 0.25 mm to 6 mm and are bonded to the structure whose vibration is to be controlled. The maximum excitation voltages range from 100 volts for the thin crystals to 2000 volts for the 6 mm thick crystals. Clearly the latter voltage is too dangerous for most applications.

Perhaps the most useful and versatile actuator for controlling structurally radiated sound is the inertial actuator. This type of actuator essentially consists of a coil surrounding a permanent magnet. When the coil is energised with an alternating current, the permanent magnet moves back and forth. The permanent magnet is held in place in a structure, the stiffness of which determines the resonance frequency of the shaker. The difference between an inertial actuator and an electrodynamic shaker is that in the former case, the heavy permanent magnet moves while the coils remains stationary, while in the latter case the opposite is true. In practice, this means that an inertial shaker can be directly attached to the structure it needs to excite and in contrast to the electrodynamic shaker, it needs no other support. If tonal noise is to be controlled, maximum shaker output is obtained if the excitation frequency is close to the resonance frequency of the inertial shaker, which is controlled by the mass of the permanent magnet and the stiffness of its suspension.. Thus it is sometimes desirable to be able to easily adjust the resonance frequency of the shaker by adjusting the suspension stiffness. The ANVC group at the

University of Adelaide is currently working on a device that does just that and is able to be controlled by an adaptive filter in such a way that the suspension stiffness is automatically and continuously adjustable. In practice, the shaker resonance frequency must be slightly different to the frequency of sound radiation being controlled or the active noise control system may suffer from stability problems. An inertial actuator controlling sound radiation from an irregular enclosure shell is illustrated in Figure 3.

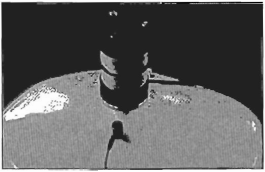

**Figure 3.** Inertial actuator mounted on an irregular enclosure.

## 2. HARMONIC DISTORTION

Harmonic distortion is the process whereby a transducer generates mechanical excitation at frequencies that are multiples of the electrical excitation signal frequency. In active noise control systems, this can result in a tonal noise sounding worse after the application of active control, because even though the fundamental tone may be considerably reduced in amplitude, the amplitude of higher order harmonics has been increased and the ability of the ear to hear the higher harmonics better, compounds the problem (as most problems attacked using active noise control are low frequency in nature). One way to minimise harmonic distortion is to drive the transducer at power levels that are less than 10% of the rated maximum and this applies to loudspeakers as well as piezo-electric crystal actuators. Keeping the input power below 10% of the rated maximum has the added advantage of extending transducer life so that in the case of loudspeakers, five years of continuous operation would not be unusual.

## 3. DISTRIBUTED ACTUATORS

Recent work [16] has reported on investigations using large numbers of actuators on structures to control sound transmission through structures. There is currently speculation supported by anecdotal evidence that if a sufficiently large number of control sources are used in a feedforward active noise control system, it is not too important where they are placed in terms of achieving a global reduction in some cost function. That is, it is not necessary to go through an optimisation process to optimally place the control sources and error sensors. In an attempt to further simplify the control process, Fuller and Carneal [17] suggested using hierarchical bio control in which a small number of signals are sent from an advanced, centralized controller and are then distributed by local simple rules to multiple control actuators.

Some recent results comparing the effectiveness of multiple single channel control systems versus a single multi-channel control system attached to the same sensors and actuators are shown in Figure 4 [18]. Actuators to achieve this result consisted of tiny (10 mm diameter and 20 mm long), low-cost inertial actuators, which provide a spring stiffness using magnetic repulsion forces. In the figure are shown results for vibration control of a cantilevered beam using a foam damper, inertial actuators (unactivated) in the foam, activated inertial actuators driven by single channel controllers and activated inertial actuators driven by a multi-channel controller. It can be seen that for this simple example involving broadband control, the multi-channel controller achieves much better results than multiple single-channel controllers, supporting the view that it may be worth investing in the development of fast multi-channel controllers capable of handling many channels simultaneously using a multi-channel algorithm.

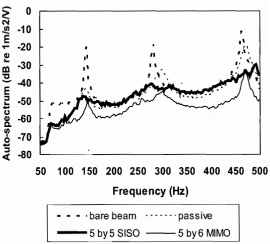


**Figure 4.** Acceleration response of a bare cantilever beam, a beam with a thick layer of foam containing five embedded, non-driven inertial actuators, with the actuators driven using five single channel controllers and then driven with a 5-out.6-in fully coupled (MIMO) controller.

## 4. SEMI-ACTIVE ACTUATORS

Given the difficulty in implementing fully active noise control systems in industrial environments, there has been considerable interest in the development of semi-active systems that can optimise the characteristics of a passive noise control system. Clearly these systems are appropriate for controlling single or multi-tone noise, where the wavelength associated with the excitation frequency is likely to vary with time. The wavelength change could be caused by an excitation frequency change as a result of changes in speed of the rotating machine generating the noise or it could be caused by changes in the temperature of the environment through which the sound is travelling.

When a Helmholtz resonator is used to control a frequency varying tonal noise propagating along a duct, the resonance frequency of the resonator can be varied by varying the resonator volume, neck length or neck diameter. In practice, it is easiest to vary the resonator volume and many ideas have been patented for doing this [19-22]. Although a number of schemes have been patented for varying the neck length [23] or cross sectional area [24, 25] or both [26], these are generally more difficult to implement in practice. The volume variations can be achieved by controlling a d.c. motor or stepper motor which drives a lead screw attached to a plunger or similar device. Neck length variations can be achieved by using a sleeve inside the neck and a motor to extend the sleeve into the resonator volume or into the existing neck [23]. As the sleeve extends into the resonator volume, the effective neck length becomes longer and the resonator volume reduces slightly.

Note that for best results, the Helmholtz resonator needs to be mounted at a location in the duct where there is a sound pressure maximum. This implies that the range of speed variation or frequency variation in the tone to be controlled should not be more than about 20% on either side of the centre frequency.

If a semi-active Helmholtz resonator is to be commercialised, it would be desirable to market is as a self contained unit. That means that it may not be desirable to minimise the signal from a particular sensor. It may be necessary to maximise the signal from a sensor in the resonator or even drive the ratio of two signals or the phase between two signals to a pre-determined value that would correspond to minimum sound power transmission down the duct. This latter approach is referred to as model reference control [27].

Tonal noise radiating from the open end of a duct can be controlled by varying the duct length. It is well known that when a source located in a duct, generates tonal sound at the resonance frequency of the duct, the sound radiated from the end of the duct can vary as the duct temperature changes. This is because the changes in temperature cause a change in wavelength of the sound propagating along the duct and this causes the difference between the excitation frequency and the closest duct longitudinal resonance frequency to vary. As the excitation frequency approaches a longitudinal resonance frequency, the level of radiated sound increases. For a typical industrial exhaust stack, the variation in sound power radiated from the end of the stack can be between 10 and 15 dB. Thus it is possible to minimise the tonal sound radiated from the end of a duct by controlling the duct length. This may be achieved by attaching an adjustable sleeve to the outside of the duct and moving it up and down with a stepper motor. This has been demonstrated and proven to be effective on a laboratory scale test rig (see Figure 5) [28].

The motor used to drive the sleeve was driven by a PLC under the control of a very simple algorithm. Initially, the sleeve is driven to the bottom of its travel. It is then advanced slowly upwards to the other extent of its travel and the location of minimum rms acoustic pressure is recorded. In practice, a narrow band filter is used to ensure that only the tonal noise is considered in the rms pressure signal. The sleeve is then driven to the minimum location. Periodically the sleeve is moved up or down to track any changes in the location of minimum acoustic pressure. However, for a vertical industrial exhaust stack it would be much more practical to adjust the level of water in the sump at the bottom of the stack as shown in Figure 6 [29].

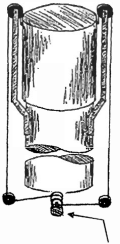


**Figure 5**. Exhaust stack with an adjustable sleeve to minimise the noise radiated from the top of the stack.

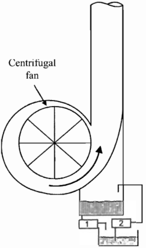


**Figure 6.** Exhaust stack showing a means of adjusting the effective length by pumping water into and out of a sump. Blocks 1 and 2 represent water pumps.

## 5. ACOUSTIC SENSORS FOR ACTIVE NOISE CONTROL

It is well known that using acoustic pressure sensors to provide the error signal for an active noise control system has some limitations, including the generation of a very large noise reduction within 1/10 of a wavelength of the sensor but not much reduction at other locations (unless the active control noise source is close enough to the primary source to physically affect its sound radiation ability). When microphones are used as error sensors, the pressure gradient in the vicinity of the error sensor in the controlled sound field can be quite large, which is a subjective problem – listeners just need to move their head a small amount and the noise level will vary by a large amount. This problem is exacerbated if the number of error sensors is equal to or less than the number of control sources. Ideally, there should be twice as many error sensors as control sources. Another way of reducing the pressure gradient problem is to measure it and include it in the objective function that is being minimised by the control system. Sensors that sense both the acoustic pressure and its gradient are referred to as energy density sensors. One such commercially available sensor is illustrated in Figure 7. A description of its use as a 3D sound intensity probe may be found in [30].

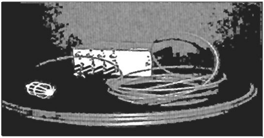

**Figure 7.** Optical, energy density sensor manufactured by Phone-Or Ltd (Israel).

## 6. VIRTUAL SENSING

A significant disadvantage of using physical sensors at the location of the sound field minimum is that the greatest noise reduction is achievable right at the sensor, which makes it difficult for the person, at whom control is aimed, to have their ear in the same location. Attempts to solve this problem have given rise to a whole new discipline. Although a number of variations have been published, there are three basic methods for implementing a virtual sensor. The first involves putting an actual physical sensor at the virtual location where it is desired to minimise the sound field, prior to full implementation of the control system. Transfer functions between the temporary microphone at the virtual location and the permanent physical microphones at some distance from the virtual location are then measured for both the primary sound field and the controlling sound field. The physical microphone is then removed from the virtual location and the control system started. During controller operation, the transfer functions measured during initialisation are used to adjust the control algorithm so it can use error signals from the actual physical microphones to minimise the sound field at the virtual location. Of course the preceding description can be extended to apply to many more than one virtual location.

The second means of implementing a virtual sensor consists of measuring the pressure gradient at the remote permanent microphones and then extrapolating this gradient quadratically or linearly to estimate the sound field at the virtual location.

The third method again involves placing a temporary physical microphone at the virtual location and then uses an adaptive algorithm to adjust the contributions from each of a number of remote microphones so that the resulting combined signal matches the signal from the virtual location when exposed to the primary sound field.

The above three methods and a number of variations are described in more detail in [13], [31] and [32] and the theoretical limitations to the accuracy of virtual sensing in a random sound field is discussed by Petersen et al [33].

## 7. VIBRATION SENSORS

In many cases where it is necessary to develop a compact active noise control system to reduce sound radiation from a vibrating structure, it is not practical to insert acoustic sensors in the surrounding sound field. In these cases it is desirable to be able to use vibration sensors on the structure to control the sound radiation. This is not as simple as it may at first seem, because reducing the overall vibration level in a planar structure may not reduce the sound radiation. This is because although normal vibration modes on a structure are orthogonal in terms of structural vibration, they are not orthogonal in terms of sound radiation. This means that reducing the vibration level of one or a number of modes will not necessarily reduce the overall sound radiation. There are two ways of overcoming this problem. The first [34] involves developing a sensing system that transforms the modes that are sensed so that they are orthogonal in terms of sound radiation but not in terms of structural vibration, so that reducing any one output of the sensing system will automatically reduce the radiated sound. Sensing systems would typically use accelerometers or piezo-electric patches as vibration sensing elements.

The second way of overcoming the sensing problem is to use model reference control [27]. In this case, the sensing system is initialised using physical microphones located such that when their signals are minimised, the radiated sound field is minimised. This is achieved using a feedforward adaptive control system. The outputs from the accelerometers mounted on the vibrating structure, corresponding to the minimum sound field, are then recorded and during operation of the control system, the physical microphones are removed and their inputs to the control system are replaced with the accelerometer signals. A new control algorithm is then used that attempts to drive the accelerometer signals to those that were recorded during initialisation when the microphone signals were used to minimise the radiated sound field.

## 8. CONCLUSIONS

Sensors and actuators are important components of any active noise control system and their cost often inhibits commercial applications of the technology. It has been shown that there are a number of sensor and actuator choices and even the potential to develop very low cost devices. However, it seems that mass market applications of active noise control will have to be found before the low cost possibilities will be fully developed.

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