Down with NOISE

Practical control systems for combatting audible noise show up in aerospace, general aviation, and military roles

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ANNOYING NOISE IN THE passenger cabins of propeller aircraft, the rumble in air-conditioning systems, and the sounds disrupting headset communication are being reduced these days by active noise control, thanks to advances in digital signal processing. The technique relies on the principle of destructive interference between two sound fields; one field is generated by the original or primary sound source, the other by a secondary sound source set up to interfere with, and cancel, that unwanted primary sound. The primary source may be an engine and the secondary source, a loudspeaker with an electronically controlled output.

Destructive interference is at its most efficient when the two sound fields can be accurately aligned in space over an acoustic wavelength. It works best on low-frequency sounds, whose acoustic wavelengths are large compared to the zone in which the noise is cancelled. In contrast, traditional passive techniques, which employ heavy barriers to block the transmission of sound and acoustic materials to absorb sound energy, are more effective at higher frequencies, when the acoustic wavelength is large compared to the thickness of an absorber. But a sound wave with a frequency of 100 Hz, typical of engine noise, has a wavelength of 3.4 meters or so in air under normal conditions. Many low-frequency acoustic noise problems are therefore difficult to control passively, yet may be amenable to active control. The active approach can thus complement the traditional passive control methods.

Recent developments with inexpensive and powerful digital signal-processing (DSP) chips have brought active control techniques within the realm of practicality. At present, they show up most often in aerospace applications, where the weight and space requirements of passive techniques often preclude their use in controlling low-frequency sound. The active control of propeller noise in the passenger cabins of aircraft is a widespread example. Further, earphones with built-in active noise control are available for general aviation pilots [Fig. 1]. Future developments, such as control of the noise inside cars, will require an understanding of both the physical principles of sound cancellation and the technology of producing a reliable, robust control system at a reasonable cost.

Scientific principles

What are the physical mechanisms of active noise control? Consider a pressure waveform as sensed by a microphone positioned so that its output is influenced by both a primary source of sound and a controllable secondary source—a loudspeaker. For simplicity, the primary source may be assumed to be tonal, so that its pressure waveform at the microphone is sinusoidal [red line in Fig. 2]. The amplitude and phase of the secondary source is driven by a sine wave generator at the same frequency as the primary source but with the phase shifted by 180 degrees [blue line in Fig. 2]. If both sources are on at the same time, their acoustic pressures cancel, since sound propagation is very nearly a linear process at all but the highest sound-pressure levels (up to about 140 dB), and the principle of superposition applies—sound waveforms are additive. How much active sound control at a single microphone position like this influences the sound field at other points in space depends upon the separation between the sources and upon the acoustic environment, for example, whether the pressure wave propagates freely in air or is enclosed within a confined space.

The sound field in an enclosure, whether a room or an automotive or aircraft passenger cabin, is typically created by standing, rather than propagating, waves, and depends on the superposition of a number of acoustic modes. A mode is characterized by the number of wavelengths that fits along one dimension of the enclosure. An enclosure of length of about 2 meters and

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height and width of 1 meter each would represent the passenger cabin of a small automobile, in which case the so-called first longitudinal mode, which has one half-wavelength, along the enclosure's length, has a natural frequency of about 85 Hz [see "Acoustic modes in an enclosure," p. 56].

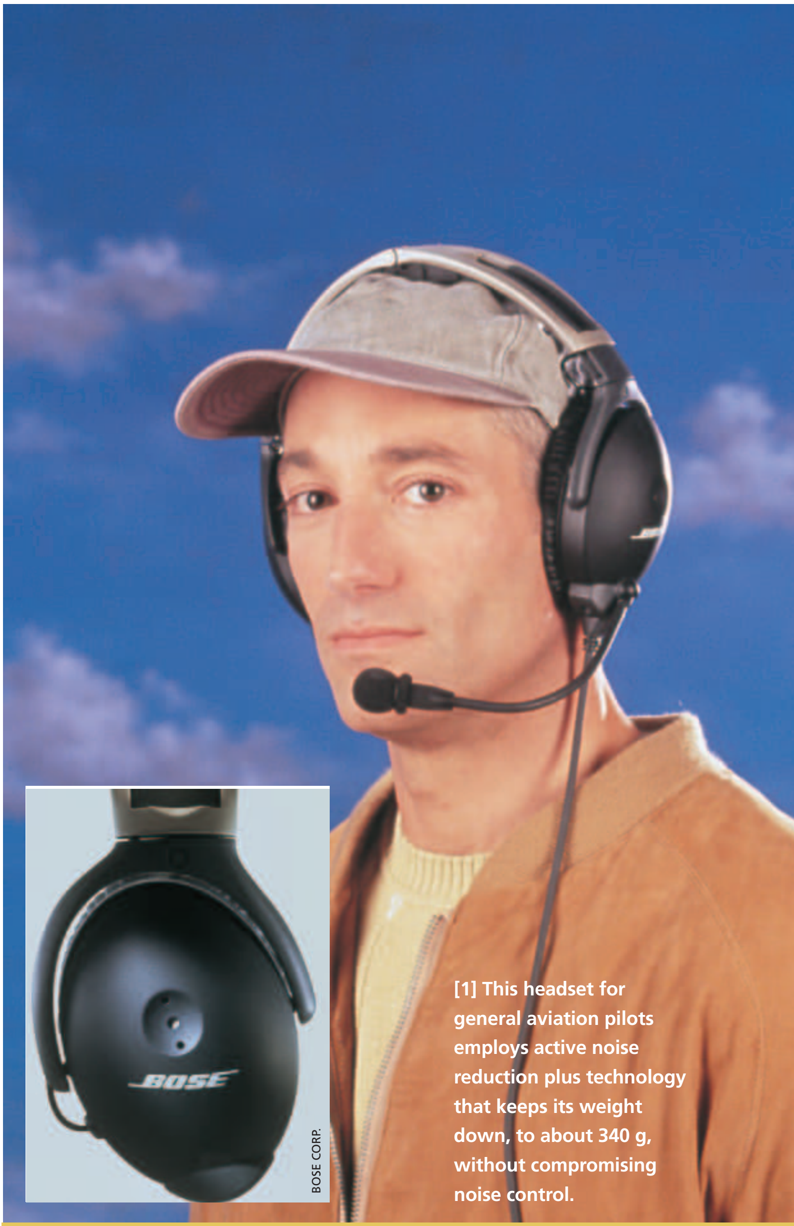
With a primary source in one corner of an enclosure, a loudspeaker placed in the opposite corner can act as a controllable secondary source. It is driven at the same excitation frequencies as the primary source, but its amplitude and phase are adjusted for each excitation frequency to minimize the acoustic energy inside the enclosure. In theory, measuring the acoustic energy should require an infinite number of microphones, since the energy is proportional to the volume integral of the mean-square pressure throughout the enclosure. In practice, only about 10 well-spaced microphones are needed, as the sum of their squared outputs is a reasonable approximation to the true acoustic energy in the frequency range of interest.

In other words, a practical control system designed to minimize the sum of the squared pressures measured with these microphones will perform much like one that keeps the acoustic energy as low as possible. If the single secondary source is adjusted to minimize the acoustic energy at each individual excitation frequency, large reductions in very low-frequency sound throughout the enclosure can result. This is particularly so for excitation frequencies close to the natural frequency, about 85 Hz, of this enclosure's first longitudinal acoustic mode. The reason: the secondary source can control this mode without significantly exciting other modes, whose natural frequencies are well away from 85 Hz.

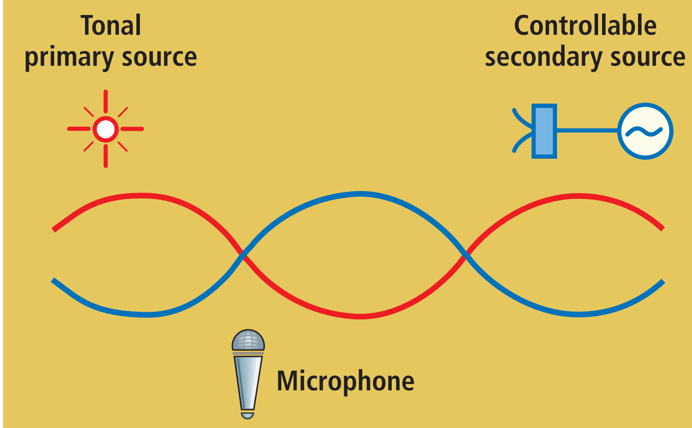
A dilemma

At an excitation frequency of about 170 Hz, however, the single secondary source is hardly able to reduce the acoustic energy in the enclosure at all. The difficulty arises because the first three acoustic modes, which correspond to fitting a whole wavelength in the longitudinal direction or a half-wavelength in either the vertical or transverse directions, all have natural frequencies of about 170 Hz. The secondary source cannot reduce the amplitude of any one of these modes without increasing the amplitude of at least one of the other two. To control all three acoustic modes separately calls for multiple secondary sources whose amplitudes and phases can be individually adjusted. For example, seven such sources achieve some reduction in the enclosure's acoustic energy at 170 Hz. But the reduction—of about 5 dB—is modest, considering the number of loudspeakers used. And even these reductions disappear if the excitation frequency is increased very much further, above about 250 Hz [again, see "Acoustic modes in an enclosure," p. 56].

The upper frequency limit of control, at about 200 Hz in this example, is very abrupt. What happens is that the number of acoustic modes contributing significantly to the total



[1] This headset for general aviation pilots employs active noise reduction plus technology that keeps its weight down, to about 340 g, without compromising noise control.



[2] The sound of a pure sinusoidal tone from a primary source [red line] can be canceled at a given point by a sound from a controllable secondary source. For full cancellation at the microphone, the sound from the secondary source must be 180 degrees out of phase with that of the primary source, but with the same amplitude and frequency, so as to interfere with it destructively [blue line].

energy at any one frequency (the modal overlap) increases dramatically with excitation frequency for a three-dimensional enclosure. To be specific, the modal overlap increases approximately in proportion to the cube of the excitation frequency. Thus, to double the upper frequency limit of active control, eight times the number of loudspeakers would be needed to control all of the acoustic modes that would then be excited. For example, to accomplish active noise control at all frequencies up to, say, 1 kHz in a 2-by1-by-1 meter enclosure with an acoustic damping typical of a vehicle interior would require about 200 loudspeakers.

Clearly such a global strategy of active noise control for enclosures will always be limited to relatively low frequencies. Sometimes, however, the sound in an enclosure is due to vibrations generated by a relatively well-defined mechanical excitation from, say, an engine mount. Here, active vibration control at the source of vibrations—the mount itself—may well enable global control of the sound field to much higher frequencies than may be achieved with loudspeakers in the enclosure.

At excitation frequencies above about 250 Hz in the enclosure described in the sidebar below, so many modes contribute to the response that the sound field is said to be diffuse and is best described statistically. In this frequency region, active control of sound may still be possible locally, that is, with speakers positioned throughout the enclosure to cancel out sound pressure sensed by nearby microphones. But the separation between the loudspeaker and microphone in each pair must be small compared with an acoustic wavelength; then the loudspeaker will not have to be driven very hard to achieve control of the local pressure, and will not have a very great effect on the sound field in the rest of the enclosure.

This kind of control system had been suggested as early as the 1950s for the reduction of sound present at a seated person's head in automobiles and aircraft. Relatively recently, researchers have shown that canceling the sound pressure at the microphone position in a diffuse sound field creates a "zone of quiet" (where the pressure level is decreased by at least 10 dB on average) that is spherical and has a diameter of about one-tenth of an acoustic wavelength. This is a useful distance for a seat-based system at an excitation frequency of 100 Hz (340 mm) but probably not for one of 1 kHz (34 mm). One application in which the microphone can be placed very close to the ear is in a headset for use by general aviation pilots [opening photo].

Two basic techniques

The basic techniques for the implementation of a closed-loop active control system are adaptive feedforward control, and feedback control. Both techniques have begun to be analyzed within a unified theoretical framework, so that a clear understanding of their comparative advantages and disadvantages is now emerging. Much can be learned about the limitations of the two control approaches from measurements of the response of a typical acoustic plant—in other words, how a microphone in the enclosure responds to a loudspeaker using either tonal excitation (frequency response) or a short pulse (impulse response).

Suppose a plant is composed of a loudspeaker and a microphone placed at opposite ends of a passenger cabin of a small automobile having internal dimensions of about 2 by 1 by 1 meter. This plant's frequency response does not exhibit any sharp resonances as the acoustic response is heavily damped by the absorptive seats and facings.

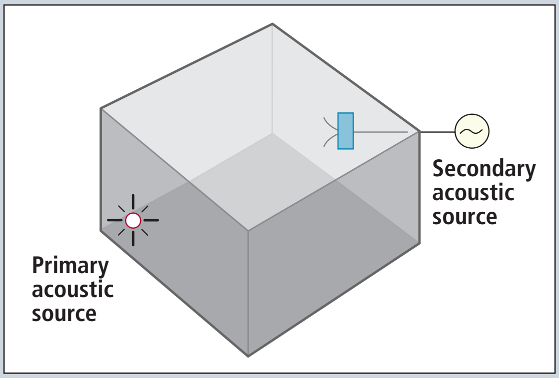
A number of dips do occur, all the same, at about 160, 240, and 460 Hz, where the acoustic modes naturally cancel each other out [Fig 3]. The overall peak in the frequency response, at about 100 Hz, is due to the loudspeaker's response. The phase response is also fairly smooth, apart from the discontinuities between –180 and +180 degrees; but it shows a characteristically increasing lag with frequency, consistent with an overall delay of about 5 ms. This delay is mainly due to the propagation time of the sound wave from the loudspeaker to the microphone, and it shows up in the time delay before the first peak in the impulse response.

Acoustic modes in an enclosure

The sound field inside an enclosure can be viewed as the superposition of the effects of a number of acoustic modes. Each mode has a characteristic shape and natural frequency.

For an enclosure with a simple geometry, such as the rectangular box depicted here [below right], the mode shapes are a series of cosine functions in space. A zeroth-order mode, with a natural frequency of 0 Hz, corresponds to the uniform compression of the air throughout the enclosure.

The first longitudinal mode, which has a natural frequency of about 85 Hz in a 2-meter-long rectangular enclosure, has a half-wavelength cosine distribution along the enclosure's longest dimension and a uniform pressure distribution in its other two dimensions. The second longitudinal mode has a a whole-wavelength cosine distribution along the largest dimensions of the same enclosure, with a natural frequency of about 170 Hz. With a simple primary acoustic source in one corner of an enclosure, the acoustic energy in the enclosure varies with excitation frequency [top curve, graph opposite]



Strictly speaking, acoustic energy here means the total acoustic potential energy. It is proportional to the volume integral of the mean square pressure throughout the enclosure and thus provides a global measure of the sound level within it.

It can be shown that the acoustic energy in the enclosure is proportional to the sum of the squared amplitudes of the acoustic modes. Besides the peak due to the zeroth-order mode in the illustration, there are also a peak at the frequency of the first longitudinal resonance and a smaller peak at the frequency of the second . The higher the excitation frequency, the less pronounced the resonance peaks.

Part of the reason for this is the damping ratio of the acoustic modes, which is proportional to the ratio of their bandwidth to their natural frequency. In many enclosures, this damping ratio is quite large and more or less constant with frequency, so that as the natural frequency goes up, so does the modal bandwidth. For the simulation used to generate the illustration, the modal damping ratio was assumed to be 10 percent, typical of a car interior.

Note that the width and height of a typical automotive interior are about half its length. Consequently, those acoustic modes for which half a wavelength is fitted into either width or height have about the same natural frequency as the second longitudinal resonance, or 170 Hz. The peak depicted at about this frequency is

The impulse response is also very well damped, decaying to almost zero after about 70 ms. Because it does not last very long, it is efficiently modeled by finite impulse response (FIR) filters in active noise control applications. This is in contrast to the infinite impulse response (IIR) models that are more common in describing the vibration response of structures. Such structures as a panel in an aircraft may well be very lightly damped, and they exhibit plant responses of a relatively low order, because their modal overlap generally increases only linearly with frequency.

Uncertainty, as well

The plant response in active noise control applications is not only subject to delay and damping, but also influenced by a large amount of what control engineers would call uncertainty. The movement of passengers within, say, the cabin of a vehicle can cause variations of 3 dB in amplitude and 45 degrees in phase.

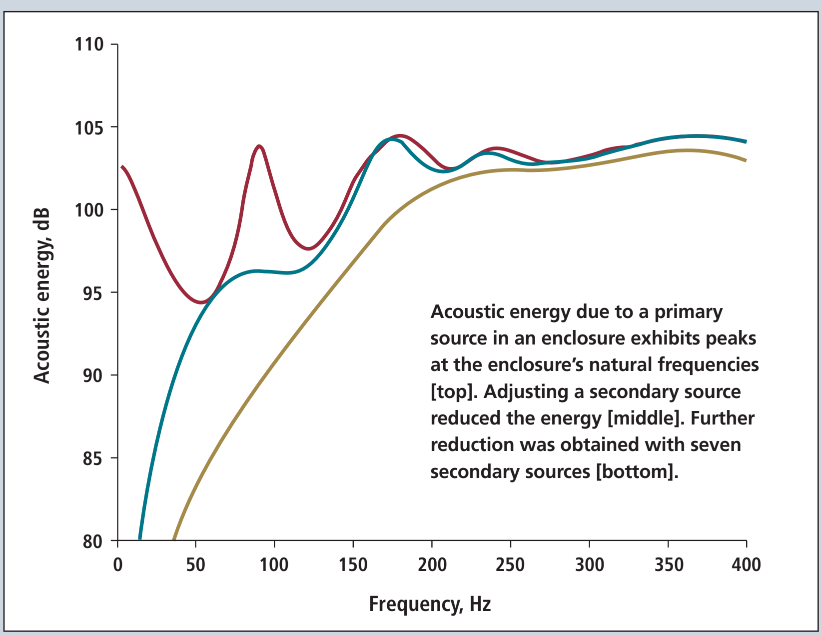
In many control systems, plant response changes of this nature can be measured by injecting a test signal into the plant and identifying the plant's response. But suppose the changes in the plant response occur within less than a second, as happens with the movement of passengers and in many other uses of active noise control. Then, for accurate identification of the plant response the test signal would need to be fed into the loudspeaker at levels as high as the noise being controlled. Unfortunately, these test signals will be experienced as added noise by the car's passengers and, if the level is louder than the noise being canceled in the first place, all the benefit of the active noise control system is lost.

It is thus seldom possible to track the variations in the plant response in many active control systems, and any practical control system will therefore have to be robust to these changes. That means that the stability of the control system will have to be assured in the face of any realistic change in the plant response. Ideally, also, the control system's performance should not be too badly affected.

The sound being canceled often changes its characteristics over time. Changing road speed in a car may create a nonstationary (varying) noise spectrum, and ideally, to maintain good performance, the controller should be adapted to optimize its response with each new noise spectrum. A good active noise control system must thus obey the following two design principles: it must be robust to (remain stable with) rapid changes in the plant's response, yet, at the same time, be adaptive to changes in the spectrum of the primary noise source.

Feedforward control

These design principles will first be illustrated for a feedforward control system fash-



Acoustic energy due to a primary source in an enclosure exhibits peaks at the enclosure's natural frequencies [top]. Adjusting a secondary source reduced the energy [middle]. Further reduction was obtained with seven secondary sources [bottom].

thus due to the combination of three acoustic modes, with similar natural frequencies. Above 200 Hz, the number of acoustic modes excited at any one frequency (the modal overlap) increases in proportion to the cube of the natural frequency. This is the other reason why the resonance peaks depicted become less pronounced as the excitation frequency increases. *—S.J.E.*

Defining terms

Acoustic modes: independent spatial distributions of sound pressure responses in an enclosure. (In a rectangular enclosure, the modes vary as cosines in all three directions.)

Antialiasing filter: an analog filter used at the input of an analog-to-digital converter to prevent aliasing in a digital, sampled-time, control system. (A reconstruction filter is one used at the output of a digital-to-analog converter.)

Excitation frequency: the frequency of sound as emitted at the source.

Gain and phase margin: the additional gain or phase shift that can be introduced into a feedback control system without its becoming unstable.

Modal damping ratio: the ratio of the damping in an acoustic mode to the critical damping. (The damping ratio is inversely proportional to the Q factor, which is widely used in electrical circuits to describe the sharpness of a resonance curve of, say, voltage versus frequency.)

Modal overlap: the number of modes whose natural frequencies fall within the bandwidth of any other mode. (A mode's bandwidth is the frequency range over which its response is within 3 dB below its response at its natural frequency.)

Nonstationary disturbance: a disturbance whose spectrum changes with time.

Plant uncertainty: variations in plant response that occur during normal operation of the control system.

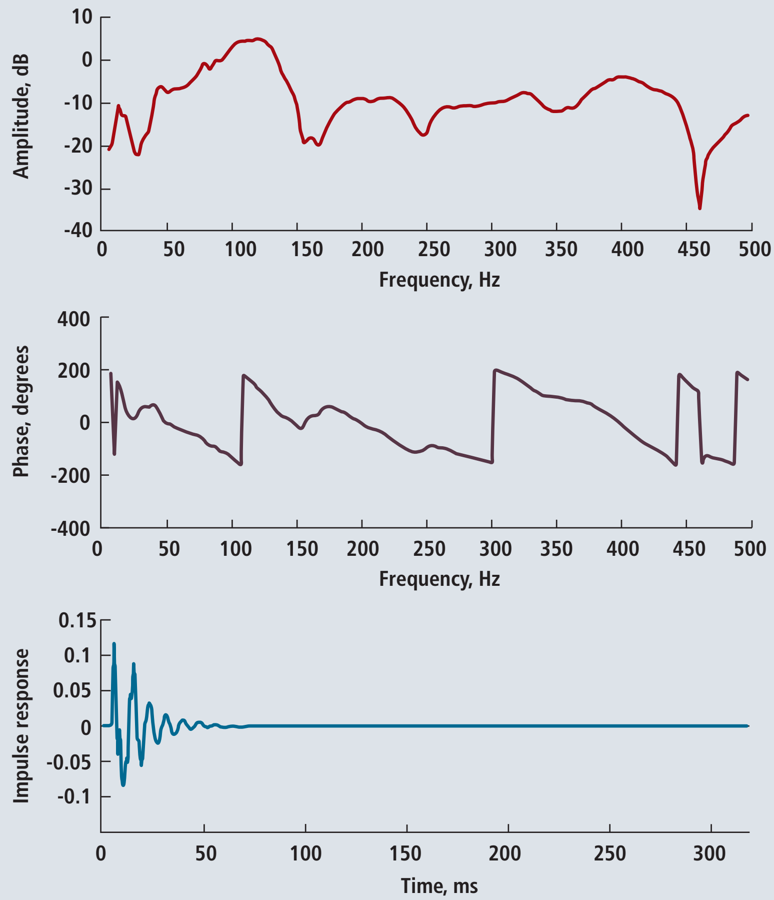
Plant: a physical system under control. In active noise control, it includes such transducers as loudspeakers and microphones and the acoustic environment—an enclosure, for example.

Robust control: a control system that is stable despite a specified range of changes in the plant response and whose performance is relatively unaffected by such changes.

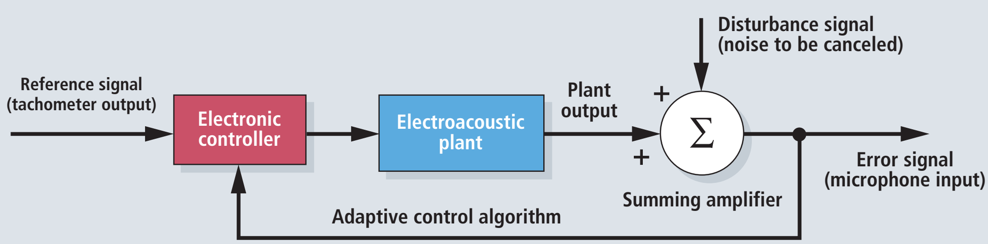
Sound field: a region containing sound waves.

Sound pressure level: a logarithmic measure of the mean square acoustic pressure expressed in decibels, with a reference pressure of 20 μPa rms. (Normal conversation at 1 meter has a sound pressure level of about 60 dB, a vacuum cleaner about 80 dB, and large industrial machines 100–120 dB, or close to the threshold of pain.)

One-third–octave spectrum: a graph of the sound power contained in each 1/3 octave frequency band of a spectrum.



[3] The interior of a small automobile has a loudspeaker and a microphone placed in opposite corners. The amplitude of the frequency response of this acoustic plant suddenly dips at certain frequencies—events attributable to natural cancellations among acoustic modes. Phase lag can be seen to increase with frequency, apart from discontinuities between –180 and +180 degrees. At the same time, the plant exhibits an overall delay of about 5 ms and a well-damped impulse response.



[4] This simplified feed-forward active noise control system employs an adaptive control algorithm in order to accommodate the changing amplitude and phase of the noise from the propeller—the disturbance signal. A loudspeaker in the electroacoustic plant [not shown] emits the noise-canceling signal.

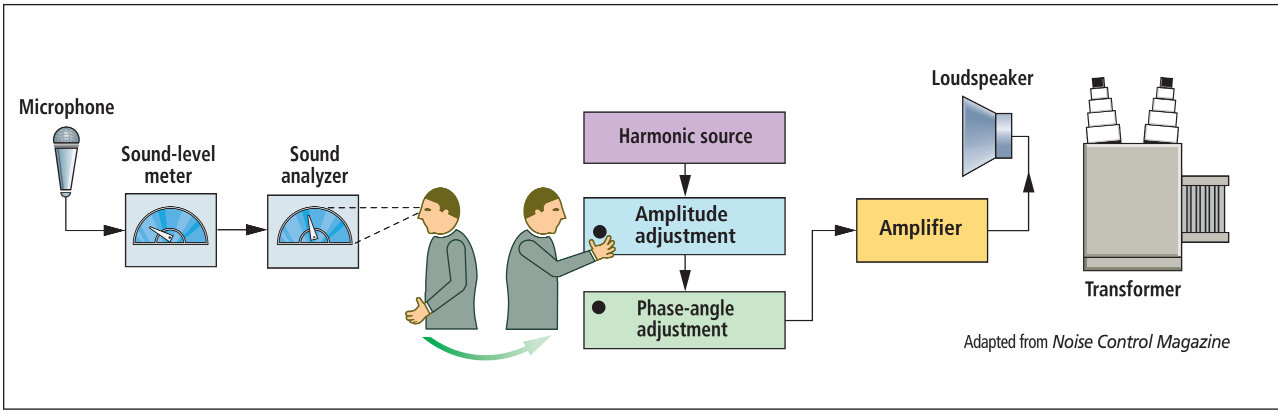
The system minimizes the mean square error signal measured by a microphone. A tachometer signal taken from the engine serves as a reference signal.

ioned to control the propeller noise in a passenger aircraft [Fig 4]. The primary noise source is the aircraft's engine, and a signal derived from the engine is used as a reference. From this reference the electronic controller in turn derives the noise-canceling signal to be fed to the secondary source—the loudspeaker, which is here an implicit part of the electroacoustic plant. For such a harmonic primary source as a propeller aircraft engine, the reference signal may well be a simple sinusoid and so may be derived from a tachometer signal taken off the engine.

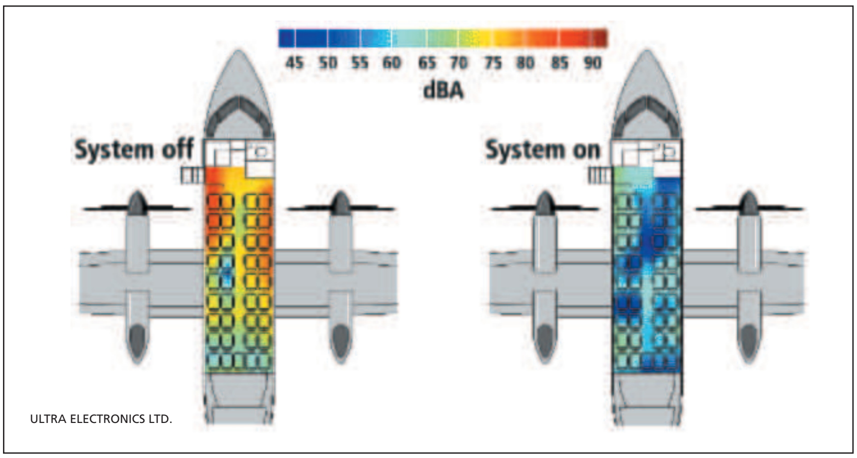
The noise to be canceled—properly termed the disturbance—is added to the plant's output signal to create the error signal, which the control system is, of course, trying to reduce. The error signal is picked up by a microphone (which is also implicit in the plant response) and fed back to the controller, which employs an adaptive control algorithm to control the loudspeaker's noise-canceling signal.

The human ear responds mainly to the mean square value of the pressure it registers. So the quantity that most active control systems are designed to minimize is the mean square value of this error signal.

The need to make such a feedforward control system adaptive, so that it could cope with changes in the characteristics of the noise to be canceled, was well understood decades ago by William Conover, then a researcher with General Electric Co. In 1956 he devised a manually adaptive feedforward system for the control of the hum in power distribution transformers [Fig. 5]. The magnetostriction in the transformer tends to make it hum at even harmonics of the line frequency. In this case, the harmonic reference signals were derived from a full-wave rectified version of the line voltage, band-pass–filtered to obtain its even harmonics. The amplitudes and phases of these reference signals were adapted manually by



[5] Manual adjustment of amplitude and phase by a utility system operator, for example, was incorporated into a feedforward system designed by William Conover in 1956 for the cancellation of transformer noise.



[6] A noticeable noise reduction of about 7 dBA on average was measured in the passenger cabin of a propeller aircraft fitted with an active noise control system. Structural shakers attached to the fuselage replaced loudspeakers inside the cabin as secondary (noise cancellation) sources.

Conover. These days adaptive digital filters are used to adjust the amplitudes and phases of multiple reference signals, driving multiple secondary loudspeakers to control the sum of the squared pressures at multiple microphones. But the principles are exactly the same as those shown in Fig. 5.

Adaptive feedforward controllers have been used to control the predominantly tonal low-frequency engine noise inside automobiles and cabin noise in propeller aircraft. The passenger cabins of many propeller aircraft are now fitted with active control systems that control four harmonics of the blade-passing frequency (the frequency at which the propeller blade passes the fuselage). A system made by Ultra Electronics, Cambridge, England, for a 50-seat Saab 2000 aircraft generally uses 37 loudspeakers and 72 microphones. Other systems made by Ultra and Lord Corp., Cary, N.C., for the smaller Beechcraft King Air turboprop aircraft mostly use eight loudspeakers and 16 microphones.

The microphones and loudspeakers are usually mounted behind the internal wall panels on the aircraft, so that the passengers may not even be aware that the active noise control system is in operation. Loudspeakers with rare-earth magnets are used to reduce their weight. They generate high sound pressure levels (about 90 dB) at low frequencies so as to match the propeller noise. This output is not heard since it is being used to cancel the propeller noise. Any high-frequency components due to loudspeaker distortion, however, will be clearly audible and high quality components must be used to keep this from occurring.

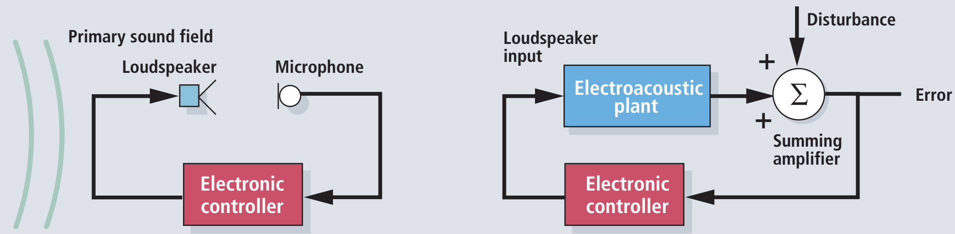
Even if the low-frequency noise is reduced with an active control system using loudspeakers, the propellers can cause the cabin to vibrate in a very disturbing manner. Another system made by Ultra Electronics and used in the deHavilland Dash 8 propeller aircraft relies not on loudspeakers but on structural shakers attached to the fuselage as secondary sources of sound. This system reduces the vibration of the fuselage and prevents vibration as well as sound from being transmitted into the cabin. The overall reduction of 7 dBA that has been measured (A refers to a frequency weighting that represents the sensitivity of the human ear) would be very difficult to achieve with conventional, passive noise control methods without considerably increasing the aircraft weight [Fig. 6].

Although active noise control has been highly successful in attenuating low-frequency tonal noise, not all noise sources that it would be nice to control have such a convenient external reference signal. To illustrate, random noise in an aircraft due to air turbulence has no single observable source for such a reference signal. Feedback is needed to control sound fields of this kind.

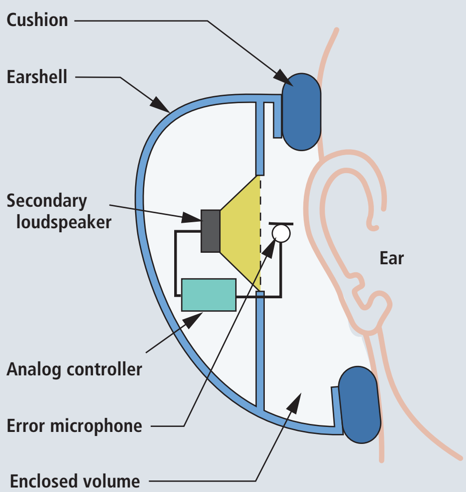
Feedback control

A simple active noise control system using feedback includes a microphone, an electronic controller, and a loudspeaker. The microphone's output is fed directly back to the loudspeaker through the electronic controller [Fig 7, left]. The controller must be so designed as to provide negative feedback (which attenuates the disturbance) over the frequency range of interest, rather than positive feedback (which amplifies the noise). As before, the response between the loudspeaker input and microphone output corresponds to that of the electroacoustic plant under control [Fig 7, right]. The disturbance signal is again equal to the microphone's output due to the primary source, in the absence of any active control.

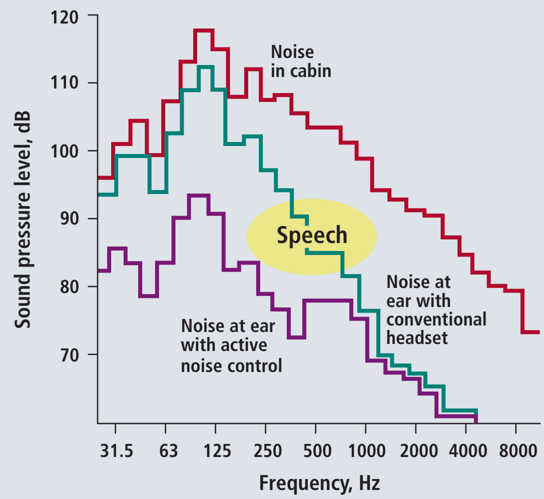
The design of the controller for such feed-



[7] An active noise control system using feedback includes an electronic controller, loudspeaker and microphone [far left]. An equivalent block diagram [left] shows a closed feedback loop, designed to minimize the error between the disturbance (the noise to be canceled) and the output of the loudspeaker (the secondary, or noise-canceling source).



[8] A headset with active noise control based on feedback includes a loudspeaker, a microphone and an (analog) controller. This last sits against the listener's ear, inside the earshell.



[9] Noise levels in the cabin of a military vehicle may reach 120 dB, close to the pain threshold. A conventional headset (without active noise control) attenuates the sound at frequencies upward of 125 Hz by 10–20 dB. A headset with feedback-based active noise control, however, reduces the sound level at low frequencies—below 300 Hz—by 15 dB or so.

SOURCE: P. WHEELER, ROYAL AERONAUTICAL SOCIETY

back systems is discussed in many standard textbooks. The most intuitively appealing methods of design are based on plots of open-loop frequency response, incorporating the response of both plant and controller. Either Nyquist or Bode diagrams can be used to design simple analog controllers having a guaranteed gain and phase margin and therefore a robust stability in the face of the kinds of anticipated variation in the plant response described above.

The bandwidth over which attenuation of the disturbance can be achieved is fundamentally limited, however, by the delay in the plant. Consequently, the bandwidth, in hertz, is proportional to the reciprocal of the delay, in seconds. The phase shift associated with this delay inevitably changes the sign of the feedback at higher frequencies and thus turns a negative feedback system into one with positive feedback. So apart from any requirements on the sound field, there is a need to place the microphone close to the loudspeaker in order to reduce the acoustic propagation delay in this situation.

Active headsets

One widely used application of feedback in active noise control is in so-called active headsets. Consumers use them to reduce background noise while they are listening to music. Representative products here are made by NCT Group, Stamford, Conn. (NCT stands for noise cancellation technologies), Sony Corp., Tokyo, and Sennheiser electronic GmbH & Co. KG, Wedemark, Germany.

Another category helps general aviation pilots, by reducing background noise while they communicate with other pilots or with ground control. Active control systems were originally used in the cockpit of military aircraft to overcome the inherent lack of low-frequency performance in headsets [Fig 8]. In an active headset, the microphone can be placed within 1 cm or so of the loudspeaker, thus reducing the propagation delay and enabling the feedback controller to attenuate frequencies of up to about 1 kHz. As the microphone is also within a few centimeters of the entrance to the ear canal, the active headset provides a favorable acoustic environment in which attenuation of the sound pressure at the microphone result in similar attenuation at the ear.

The noise level in a military vehicle can come close to 120 dB, or almost the threshold of pain at about 100 Hz [Fig. 9]. Speech becomes almost unintelligible, and loss of hearing a possible consequence. A conventional headset reduces the noise levels at the ear for high frequencies, above 1 kHz, but it has less effect below about 500 Hz—all frequencies of importance in spoken communication. The feedback active control system, commonly referred to as an ANR (active noise reduction) system in this application, can attenuate the noise at the ear by about a further 10 dB and do so up to about 500 Hz [bottom curve of Fig 9].

This lowering of the overall noise level makes speech a much more reliable form of communication and considerably lessens the fatigue the noise inflicts on the headset wearer. Headsets of this kind for military use are made by Racal Acoustics Ltd., Harrow, United Kingdom; Lectret Precision Ltd., in Singapore; and Bose Corp., Framingham, Mass.

The feedback controllers for active headsets are generally analog devices. Designed to reduce delay in the controller, they employ classical techniques—using a Bode plot, for example—to shape the open-loop characteristics. This is possible because the plant uncertainty, being due mainly to movement of the headset on the head, can in this case be quantified, and the spectrum of the disturbance is known and stationary inside such a military vehicle.

The need for a more analytic design procedure has led to an alternative interpretation of the feedback controller. This approach may also pave the way toward making active control systems using feedback about as adaptive to changes in the disturbance spectrum, as the feedforward controllers discussed above.

One scheme is known as internal model control (IMC) to the process control community, where it has been most widely developed. In this arrangement, an estimate of the disturbance is used to drive a control filter that feeds the secondary source and so completes the feedback loop. Not only does the IMC architecture afford a very useful insight into the performance limitations of an active feedback controller, but it also provides a method of designing optimal controllers for a given environment.

The IMC controller has one potential disadvantage, however. To provide the flexibility needed to implement an accurate internal plant model and an optimal control filter, digital filters based on digital signal-processing (DSP) systems must generally be used. The drawbacks are that the sampling process associated with such filters introduces an added delay into the feedback loop. Further delay is introduced by the analog antialiasing and reconstruction filters usually necessary to prevent high-frequency aliased components of the low-frequency disturbance from being annoyingly broadcast from the loudspeaker.

Since the total loop delay is increased, the control bandwidth is inevitably reduced by the use of a digital controller. To a certain extent, this reduction in control bandwidth can be overcome by sampling at a higher rate, but this requires greater processing power from the DSP. Combined analog and digital systems could be used to overcome these problems.

What lies ahead

As the performance of DSP devices increases and their prices fall, the prospect of controlling noise by using active control will continue to be an attractive one. It is not true, however, that with ever-increasing DSP power, active control will solve all noise problems. Recall that there are fundamental physical limitations on most active noise control systems, which make them impractical above a few hundred hertz.

Remember, too, that the control of tonal noise is much simpler than the control of random noise. Many low-frequency tonal noise problems are everyday irritants and in principle many are amenable to active control. One of the problems at the moment is the cost of the complete system, which includes not only that of the DSP device but also that of the secondary loudspeakers, microphones, and associated interfaces.

An application whose cost matters to manufacturers a great deal is the active control of noise in cars. Notwithstanding many successful demonstrations of such systems for both low-frequency engine and road noise, fully active control systems are currently fitted to very few production vehicles. The key to future development here may be integration. Loudspeakers and associated amplifiers, and maybe even the DSP requirements of an active noise control system, could be shared with the vehicles' audio systems. Microphones are also now being fitted into vehicles, for hands-free telephone dialing, for example, and once again these could be shared with an active control system.

In the longer term, more attention will be paid to the control of random noise using feedback systems. Current work is focused on whether a system like the pilot's active headset could be implemented as an active headrest arrangement for passengers in commercial aircraft.

Future emphasis will also fall on increasing the upper frequency range of control, particularly in applications such as the control of engine and gearbox noise in commercial airliners and helicopters. Probably the most feasible approach here will be to concentrate on actively controlling the vibration of these components as near the source of noise as possible. The ultimate aim would be to actively control the source mechanism itself. Research is being conducted in this area for gearbox and aeroengine noise control, but the commercial realization of such systems is probably many years away.

To probe further

The signal-processing algorithms alluded to in the article are reviewed in greater detail in "Active Noise Control," by this article's author and P. A. Nelson, which appeared in the October 1993 issue of *IEEE Signal Processing Magazine.*

Feedback control approaches are reviewed in the December 1995 *IEEE Control Systems Magazine* in "Active Control of Sound and Vibration," by C. R. Fuller and A. H. von Flotow.

Several textbooks have been written about the physical aspects of active noise and vibration control, including *Active Control of Sound* by P. A. Nelson and S. J. Elliott (Academic Press, San Diego, Calif., 1992) and *Active Control of Vibration* by C. R. Fuller, S. J. Elliott, and P. A. Nelson (Academic Press, 1996).

S. M. Kuo and D. R. Morgan describe the algorithms and digital signal-processing implementation of many active control systems in their book *Active Noise Control Systems* (John Wiley & Sons, New York, 1996). The same topics are discussed in the comprehensive text *Active Control of Noise and Vibration* by C. H. Hansen and S. D. Snyder (E. FN. Spon, 1997).

An account of the synthesis of signal processing and automatic control using state space methods is provided by R. L. Clark, W. R. Saunders, and G. P. Gibbs in *Adaptive Structures, Dynamics and Control* (John Wiley & Sons, 1998). A more complete synthesis using the approaches outlined in this article is provided by the present author in the book *Signal Processing for Active Control*, to be published by Academic Press in 1999.

World Wide Web addresses of companies that are relevant to the article are: ultraquiet.com/nvs, bombardier.com, lordcorp.com, bose.com, racal-acoustics.co.uk, nct-active.com, and swiftnet.com.sg/lectret.

Further information about research in active noise control may also be found on the Web at isvr.soton.ac.uk/active/, val.me.vt.edu, and mecheng.adelaide.edu.au.

A list of other Internet resources is provided at hh.se/staff/wolfgang/orpheus/anclinks.

About the author

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