isvr Institute of Sound and Vibration Research

# A Review of Active Noise and Vibration Control in Road Vehicles

**S.J. Elliott**

ISVR Technical Memorandum No 981

December 2008

University of Southampton

# SCIENTIFIC PUBLICATIONS BY THE ISVR

***Technical Reports*** are published to promote timely dissemination of research results by ISVR personnel. This medium permits more detailed presentation than is usually acceptable for scientific journals. Responsibility for both the content and any opinions expressed rests entirely with the author(s).

***Technical Memoranda*** are produced to enable the early or preliminary release of information by ISVR personnel where such release is deemed to the appropriate. Information contained in these memoranda may be incomplete, or form part of a continuing programme; this should be borne in mind when using or quoting from these documents.

***Contract Reports*** are produced to record the results of scientific work carried out for sponsors, under contract. The ISVR treats these reports as confidential to sponsors and does not make them available for general circulation. Individual sponsors may, however, authorize subsequent release of the material.

UNIVERSITY OF SOUTHAMPTON

INSTITUTE OF SOUND AND VIBRATION RESEARCH

SIGNAL PROCESSING AND CONTROL GROUP

# A Review of Active Noise and Vibration Control in Road Vehicles

by

**S.J. Elliott**

ISVR Technical Memorandum No. 981

December 2008

Authorised for issue by
Professor R. Allen
Group Chairman

## CONTENTS

## Abstract

Active control works by destructive interference between the original sound or vibration field in the vehicle and that generated by a controllable, secondary, source. Physical limitations generally confine its usefulness to low frequencies and so that it complements conventional passive control methods. The development of powerful processors at an affordable cost and the increasing trend towards integration in vehicles has allowed the commercial implementation of active control systems by several manufacturers, mainly for the reduction of low frequency engine noise. As vehicles become lighter to achieve fuel efficiency targets, it is expected that active control will play an important part in maintaining an acceptable NVH environment, in terms of sound quality as well as overall level.

# 1. INTRODUCTION

Active control involves driving a number of actuators to create a sound or vibration signal out of phase with that generated by the vehicle, thus attenuating it by destructive interference. Its successful application requires that there is both a good spatial and a good temporal matching between the field due to the actuators, or secondary sources, and that due to the vehicle. The requirement for spatial matching gives rise to clear limits on the upper frequency of active noise control, due to the physical requirement that the acoustic wavelength must be small compared with the zone of control. The requirement for temporal matching requires a signal processing system that can adapt to changes in the vehicle speed and load. Both the physical limitations and the signal processing control strategies will be described in this report, together with a description of some of the practical systems that have found their way into production at the time of writing. Active noise and vibration control can provide a useful alternative to passive noise and vibration control, particularly at low frequencies and on vehicles with particular problems. Although active control has been experimentally demonstrated in vehicles for over 20 years, it is only recently that the levels of integration within the vehicle's electronic systems have allowed the cost to become affordable. Active control may now allow a reduction in the weight of conventional passive methods of low frequency noise control, helping the push towards lighter, more fuel efficient, vehicles.

## 2. PHYSICAL PRINCIPLES OF ACTIVE CONTROL

Active vibration control in vehicles often involves the isolation of a particular transmission path, particularly of engine orders through engine mounts. Here the spatial matching problem requires that the mechanical actuator drives in the same direction as the dominant transmission path from the source and at more or less the same point. For this reason most systems use combined active/passive mounts, in which the actuator is integrated into a conventional hydro mount. Although piezoelectric and magnetostrictive actuators have been considered for this application, the most successful actuator type appears to be electromagnetic. Cost considerations generally limit the number of actuators that can be used, since these are generally the most expensive parts of automotive active vibration control systems, and so the successful use of active control is limited to vehicles with a few dominant transmission paths. The introduction of active mounts can significantly ease the conventional problems in trading off high static stiffness with low dynamic stiffness.

An idealised arrangement to illustrate the function of an active mount is shown in Figure 1.

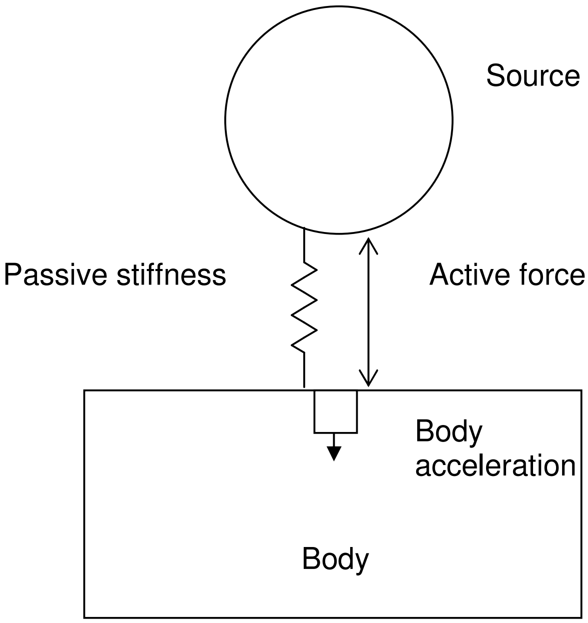


Figure 1. Idealised active vibration system consisting of a single active mount, which includes a passive stiffness $S_M$ and an actuator generating an active force $f_S$, connecting a vibration source such as an engine to the vehicle's body.

The force transmitted through the mount, assuming it behaves reasonably linearly, can be written at a particular frequency as

$$f_T = f_S + S_M (x_S - x_B), \qquad [1]$$

where $f_S$ is the secondary force generated by the actuator, $x_S$ is the displacement of the source above the mount, $x_B$ is the displacement of the body below the mount and $S_M$ is the mount stiffness. The body is assumed to only be driven by this total force, so that if $F_B$ is the flexibility of the body below the mount, then

$$x_B = F_B f_T = F_B f_S + F_B S_M (x_S - x_B), \qquad [2]$$

so that solving the equation for the displacement of the body gives

$$x_B = \frac{F_B (f_S + S_M x_S)}{(1 + F_B S_M)}. \qquad [3]$$

If $x_B$ is measured using an accelerometer on the body, for example, and, at a particular engine order, an adaptive controller is used to adjust the secondary force, $f_S$, so that the output of this accelerometer is cancelled, then it can be seen by setting equation [3] to zero that $f_S$ under these optimum conditions must be equal to

$$f_{S\,(optimum)} = -S_M x_S. \qquad [4]$$

Notice that the required force does not depend on the flexibility of the body, since this has been brought to rest. Also, notice that the total force in equation [1] has also been set to zero with this control force, and that not only does this total force act down on the body but it also acts up on the source structure. Thus, if the total force becomes zero, the source structure is effectively floating at this control frequency and $x_S$ becomes the free displacement of the source. Equation [4] thus provides a useful way of estimating the force requirements of an active mount, depending as it does on only the mount stiffness and the free source displacement at the frequency of interest. There are some advantages to measuring the total transmitted force for use as the control signal, rather than the body acceleration, since it is related to the secondary force in a more straightforward way than the latter. The measurement of the total transmitted force, using a load cell for example, can, however, be more complicated than measuring the acceleration.

The requirement for spatial matching is more complicated for active noise control, since it is the acoustic field inside the vehicle which must be controlled and this is generally excited by the distributed vibration of the whole body, excited by multiple sources. In order to illustrate the frequency limitations of active noise control inside a vehicle, we assume that the soundfield is tonal and use the complex pressure at a single frequency, $\omega$, which may be described in modal form as

$$p(\mathbf{x},\omega) = \sum_{n=0}^{\infty} a_n(\omega)\Psi_n(\mathbf{x}), \quad [5]$$

where **x** is the position vector, $a_n(\omega)$ is the amplitude of the *n*-th acoustic mode and $\Psi_n(\mathbf{x})$ is its mode shape.

Although in principle an infinite number of modes must be used to describe the sound in the enclosure, the sound field can always be approximated to arbitrary accuracy with a finite modal series. In the low frequency range, where active control is most effective, the modal description is a very efficient representation of the sound field since relatively few modes need be considered.. Conventional passive noise control techniques also do not work very well in this low frequency region, unless very massive barriers or bulky absorbers are used, and so active control conveniently complements the effect of passive noise control techniques and can provide significant weight and space savings at low frequencies.

Two active control problems will be briefly considered: global control and local control. The objective of a global control system is to reduce the sound throughout the enclosure by adjusting the amplitudes and phases of a number of secondary sources, which are typically loudspeakers. The fundamental limits of such a strategy can be assessed by calculating the reductions which are possible in the mean square pressure integrated over the whole volume, which is proportional to the acoustic potential energy in the enclosure that may be written as

$$E_p(\omega) = \frac{1}{4\rho_0 c_0^2} \int_V \left| p(\mathbf{x},\omega) \right|^2 dV. \quad [6]$$

Assuming that the mode shapes are orthonormal, $E_p(\omega)$ is equal to the sum of the modulus squared mode amplitudes (Nelson and Elliott, 1992). Since the secondary sources linearly couple into each mode amplitude, $E_p(\omega)$ is a quadratic function of the complex secondary source strengths, and this function has a unique global minimum. This minimum value of $E_p(\omega)$ provides a measure of the best performance that can be obtained in a global control system for a given distribution of secondary sources and a given excitation frequency. Figure 2(b), for example, shows the result of a number of such calculations for the levels of $E_p(\omega)$ at various excitation frequencies in a computer model of an enclosure of dimensions $1.9\times1.1\times1.0$ m as shown in Fig. 2(a), which are approximately the conditions inside a small car (Elliott, 2001). The solid line in Fig. 2(b) shows the way that the energy due to a primary source in one corner of the enclosure varies with excitation frequency, the dashed line the energy after it has been minimised using a single secondary loudspeaker in the opposite corner and the dot-dashed line after minimisation using seven secondary loudspeakers placed at all the corners of the enclosure away from the primary. The first longitudinal resonance, at about 80 Hz, is significantly attenuated by the action of a single secondary loudspeaker, but almost no reduction is achieved in the energy at about 160 Hz, close to which three acoustic modes have their natural frequencies. This is to be expected, since in general a single loudspeaker can only control a single mode. Even with seven secondary loudspeakers, however, a reduction in energy of only about 5 dB is achieved at this excitation frequency and this reduction becomes less than 1 dB at about 250 Hz.

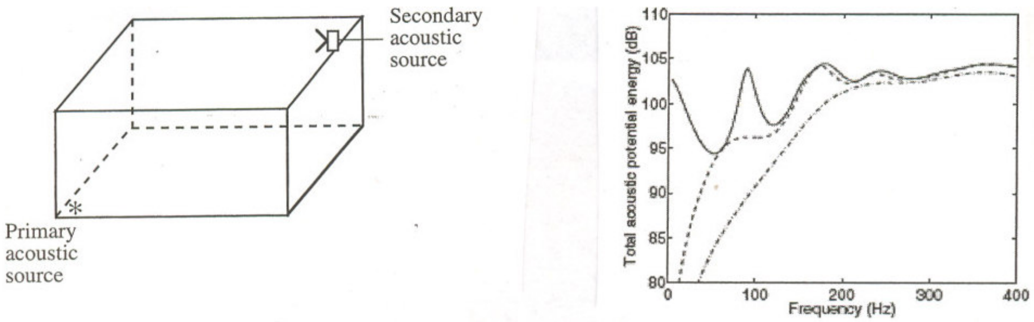


Figure 2. An active noise control system in which a loudspeaker is used to globally control the sound in an enclosure of about the size of a small car interior (a). The total acoustic potential energy in the enclosure as a function of excitation frequency, when driven by the primary source alone, solid line, when the energy is minimised using a single secondary source (dashed line) and when the energy is minimised using seven secondary sources (dot-dashed line).

The number of secondary sources required to achieve active control is approximately equal to the number of significantly excited acoustic modes, which can be estimated from the modal overlap, *i.e.,* the average number of modes with natural frequencies within the half power bandwidth of a single mode (Elliott, 2001). The modal overlap in this enclosure is about seven at 250 Hz, which explains the limited performance with seven loudspeakers, but at higher frequencies the acoustic modal overlap in an enclosure rises as the cube of the excitation frequency. This sharp rise with frequency in the required number of secondary sources provides a very clear upper frequency limit to global control with a system of reasonable complexity.

An alternative strategy to global control would be to only control the sound at specific locations in an enclosure, such as close to the ears of passengers in a vehicle. Such a local control strategy was originally suggested by Olsen and May (1953) who describe an active headrest using a feedback control system from a microphone to a closely-spaced loudspeaker acting as the secondary source. The acoustic performance of such a system depends on the detailed geometric arrangement of the headset and the position of the passenger's head, but some physical insight can be gained by considering simplified models. Figure 3, for example, shows a cross-section through the zone of quiet, within which the sound has been attenuated by at least 10 dB, generated when a diffuse primary soundfield is cancelled at the point $x = L$ by an acoustic monopole at the origin (Elliott, 2001). The two graphs correspond to an excitation frequency for which $L$ is much smaller than the acoustic wavelength, in which case a "shell" of quiet is generated around the secondary source, and to an excitation frequency for which $L$ is of the order of the acoustic wavelength, in which case the zone of quiet is spherical with a diameter which is about one tenth of an acoustic wavelength (Elliott *et al.*, 1988). The rule of thumb that the spatial extent of a local active control system is about one tenth of a wavelength has proved to be very useful in the initial stages of many practical designs.

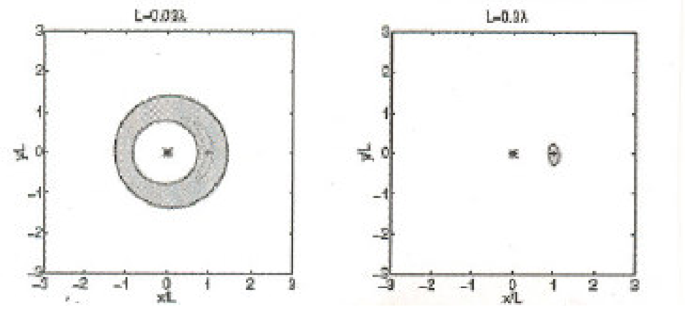


Figure 3. The zone of quiet, within which a diffuse primary field is attenuated by more than 10 dB for a local control system in which a monopole acoustic source at the origin is arranged to cancel the pressure at $x = L$ for two different excitation frequencies for which $L$ is much larger than the acoustic wavelength (left) and $L$ is of the order of the acoustic wavelength (right).

# 3. CONTROL STRATEGIES

The effective attenuation of a signal using active control requires a high degree of temporal matching between the waveform due to the vehicle alone and that due to the active control system. This is most easily illustrated with a sinusoidal signal, for which a 10 dB attenuation requires that the two signals must be matched within about 2 dB in amplitude and within about $20^{o}$ in phase. In practice, this temporal matching is achieved by using a control system with a sensor to monitor the residual difference between the two signals. This "error sensor" may be a microphone, or microphone array, for an active noise control system inside a vehicle, or an accelerometer on the chassis next to an active mount for an active vibration control system.

There are two different ways in which the control system can use the signal from this error sensor: either using feedforward control or feedback control. Idealised, single channel feedforward and feedback controllers are illustrated in Fig. 4. In addition to the signal from the error sensor, which is illustrated as a microphone in Fig. 4, a feedforward control system also requires an independent "reference" signal, generally denoted *x*, that is correlated with that being controlled. In an engine noise controller, for example, it is the engine orders which are being attenuated and a reference signal at the frequency of these engine orders is required, which will track these orders as the engine speed changes. Originally, such a reference signal was derived by analogue processing of the ignition signal, but is now commonly synthesised from knowledge of the engine speed supplied over a CAN bus, for example. The signal driving the secondary loudspeaker is obtained by adjusting the amplitude and phase of this tonal reference signal.

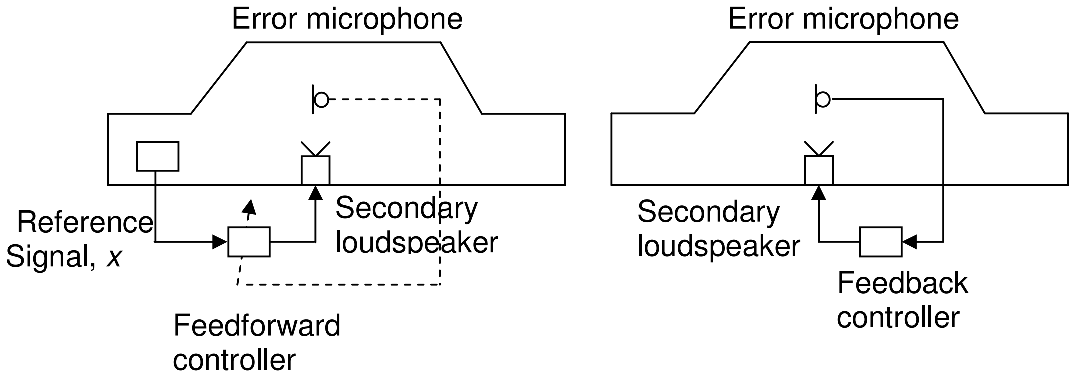


Figure 4. Illustration of single channel active noise control systems using a feedforward (left) and a feedback (right) control strategy. Note that the feedforward control signal requires an additional reference signal, *x*, and uses the error signal to adapt the controller.

It is not generally effective to operate a feedforward controller with a fixed response, since it is then not responsive to changes in the error signal or the environment, i.e. it is "open loop". The error signal is thus used to adapt the response, so that it is able to change with engine speed and load for example, and the system then becomes "closed loop". The need to adapt the controller response means that most practical implementations use digital electronics, operating in sampled versions of the reference and error signals and producing a sampled signal to drive the actuator, which all need to be filtered to remove components above half the sampling frequency. A variety of algorithms have been used to adapt the controller, but most are based on an adaptive filtering method called the "LMS" (or least mean square) algorithm which was originally developed in the 1960s for applications such as echo cancellation on telephone lines (Widrow and Stearns, 1985). The modified algorithm required to get the controller to converge reliably in active control applications, is called the filtered reference LMS or "filtered *x* LMS" (Widrow and Stearns, 1985; Elliott, 2001). In practice, multiple loudspeakers are driven to minimise the sum of the mean square responses from a number of microphones, which requires a multichannel generalisation of the filtered *x* LMS algorithm (Elliott *et al.*, 1987).

The feedback system, illustrated in Fig. 4(b), by contrast is generally implemented with a fixed controller and so can be efficiently built using analogue electronics. The feedback system has several other advantages, such as not requiring a reference signal, but also has a number of significant disadvantages. One disadvantage is that the control is not selective, *i.e.* any signal will be attenuated, not just those correlated with the reference signal. Another disadvantage is that the error sensor, which is the microphone in Fig. 4(b), must be placed close to the secondary loudspeaker.

The fundamental trade-off in the design of any feedback controller is that between performance and stability (Franklin *et al.*, 1994; Elliott, 2001). The feedback system can become unstable when the loop gain, which includes the response between the actuator and sensor and that of the controller, has a phase shift of $180^{o}$. The feedback then changes from being negative, which leads to an attenuation of the error signal, to being positive, which leads to an enhancement of the error signal. All practical systems will have a loop gain with such a phase shift at high frequencies, due to the acoustic propagation delay from the loudspeaker to the microphone, and thus inevitably have enhancement at some frequencies. If, however, the loop gain is greater than unity at the frequency where this phase shift is $180^{o}$, then the feedback system will become unstable. Some mitigation of this condition can be achieved using analogue 'compensator' circuits, but sooner or later this unstable condition will arise as the feedback gain is increased, which will then limit the performance at lower frequencies. A phase shift of $180^{o}$ is reached at a frequency for which the distance between the loudspeaker and microphone, *d*, is half an acoustic wavelength. If $c_0$ is the speed of sound, the upper frequency of operation of a feedback control system, $f_{(max)}$, will be significantly below this frequency, which is given by $c_0/2d$, so that

$$f_{(\max)} << \frac{c_0}{2d}, \qquad [7]$$

and, in practice, the upper frequency is about one tenth of $c_0/2d$. One can thus see that for an active headphone, for which feedback controllers are widely used and in which $d$ is about 1 cm, then the maximum frequency will be about 1.7 kHz, since $c_0$ is 340 metres per second. For an active control system in a car, however, where the microphone is close to the driver's head but the loudspeaker is fitted in the door or the dashboard, then $d$ will be about 50 cm and so the upper frequency will be about 34 Hz. The feedback system is thus of limited use unless either very low frequency noise is to be controlled or the loudspeaker can be brought significantly closer to the microphone, by mounting it in the headrest, for example. Laboratory versions of such headrest controllers have been demonstrated (Elliott 2001), with performance up to about 300 Hz. At this frequency, however, the size of the zone of quiet, predicted to be one tenth of an acoustic wavelength following the discussion in Section 2, is about 10 cm, so that significant active control would only be experienced if the listener's head was fairly stationary.

## 4. COMMERCIAL SYSTEMS

Early automotive active noise systems were feedforward arrangements for tonal engine noise control (Elliott *et al.*, 1988), developed as part of a research programme that also focused on the active control of interior noise in propeller aircraft (Nelson and Elliott, 1993). The physical arrangement of a typical feedforward active noise control system, developed in collaboration with Lotus Engineering, is shown in Fig. 5, in which four of the six loudspeakers, in the dashboard and front doors, were adjusted at the engine firing frequency and its harmonics, to control the sum of the mean square pressures at eight microphones, mounted in the head lining. The A-weighted sound pressure levels at the engine firing frequency (second order for this 1.1 litre 4-cylinder car) are shown in Fig. 6, measured at the front seat positions using monitoring microphones separate from the error microphones used by the control system. Reductions of about 10 dB are measured in the front seats above about 3,000 rpm (a firing frequency of 100 Hz), which gave an improvement in the overall dB(A) level of 4 to 5 dB(A). Reductions at lower speeds were measured in the rear due to the suppression of the first longitudinal acoustic mode, that has a nodal line near the front passengers' heads. Nissan first put such a system into production on a Bluebird vehicle in 1992 (Hasegawa *et al*., 1992), in a system which used loudspeakers, amplifiers and processors separate from the audio system, and so was relatively expensive.

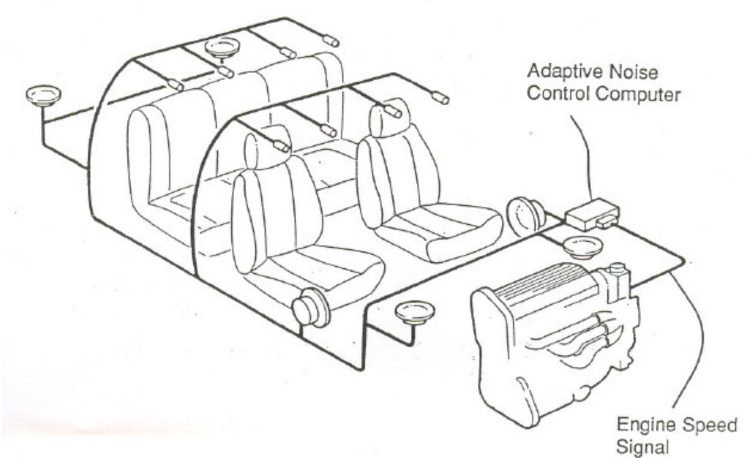


Figure 5. The components of an active noise control system for engine noise, showing the feedforward controller deriving a reference signal from the engine and driving four loudspeakers at a number of engine orders, adjusted in amplitude and phase to minimise the sum of the mean square responses at eight microphones, positioned in the roof lining.

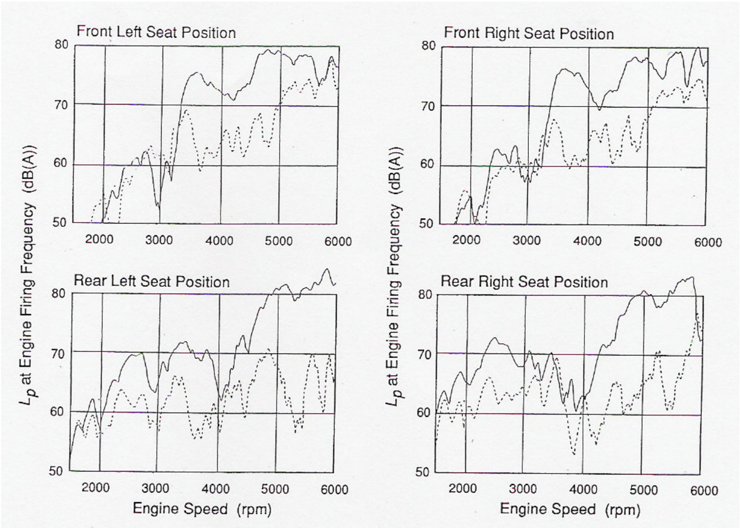


Figure 6. A-weighted engine second order levels at the four seat positions as a function of engine speed as a small, 4 cylinder car is accelerated along a track, without (dashed) and with (solid) the active control system shown in Figure 5 operations.

Demonstration systems were also developed for the active control of random road noise using feedforward techniques, with reference signals derived from accelerometers on the vehicle suspension and body (Sutton *et al.*, 1994; Bernhard, 1995; Dehandschutter and Sas, 1998; Mackay and Kenchington, 2004). In order to obtain reasonable levels of active control, however, it was found that about six such reference signals were required. This is because the tyre vibration is relatively uncorrelated in its various degrees of freedom, and reference signals have to be used for all the significantly contributing sources (Sutton *et al.*, 1994). Typical locations for the accelerometers used to generate the reference signals are shown in Fig. 7.

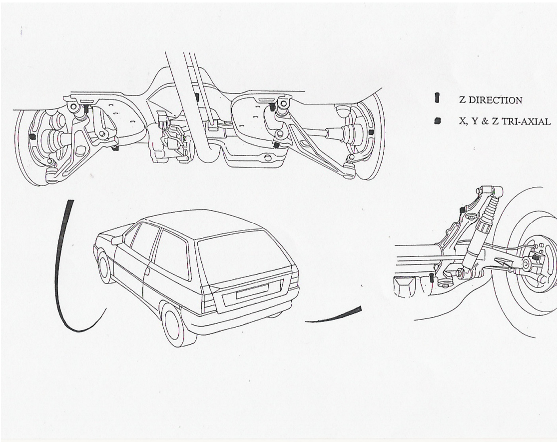


Figure 7. Typical locations of the accelerometers used to obtain the six reference signals in a feedforward active control system for road noise.

The A-weighted spectrum of the pressure at the driver's ear using a real-time feedforward system is shown in Fig. 8 (Sutton *et al.*, 1994), which shows that from 100 Hz to 200 Hz reductions of up to 10 dB are measured. The additional expense of these accelerometers has prevented the mass production of such feedforward active control systems for road noise. In 2000, however, Honda demonstrated a mainly feedback system to control a 40 Hz boom in the front of their Accord wagon car (Sano *et al.*, 2001). A fixed feedforward system was then used to prevent the noise in the rear of the car being amplified. This excitation was relatively narrow band and the microphone was placed fairly close to the loudspeakers, compared with the wavelength at 40 Hz, so that good performance was obtained in suppressing this resonant boom.

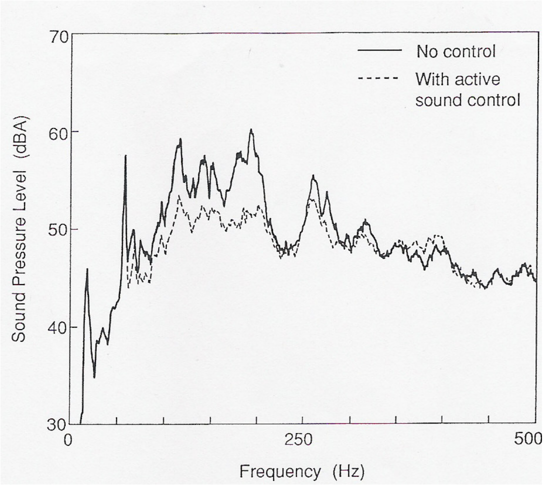


Figure 8. Spectrum of the A-weighted sound pressure level in a small car using a real-time active control system for road noise, measured at the drive's ear. Vehicle speed was 60 km/h over asphalt and coarse chippings.

Figure 9 shows the configuration of the active control system, from Sano *et al*. (2001), and Fig. 10 shows the measured spectrum at a front seat, illustrating the suppression of the narrowband boom at 40 Hz by about 10 dB. An important aspect of reducing the cost of this system, so that it could be used in mass production, was the integration of the loudspeakers with the audio system, although at that time a separate active control unit was used from the audio system, since a number of different audio head units were offered on this vehicle. It is the full integration of the active control system with the audio system that would make this technology affordable on many vehicles.

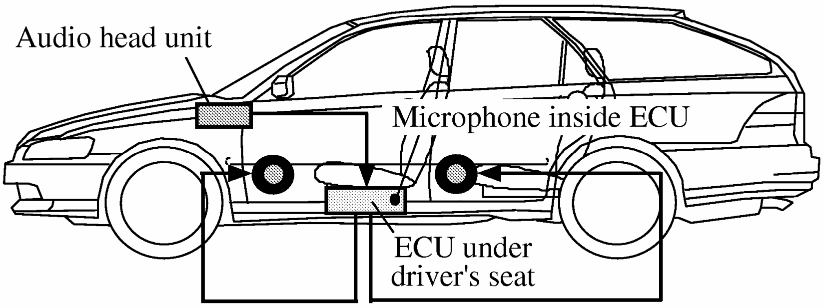


Figure 9. Configuration of the active control system in the Honda Accord (Sano *et al.*, 1993).

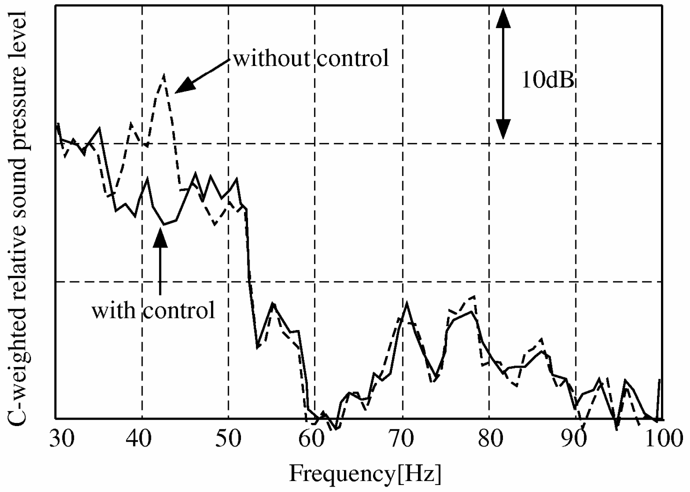


Figure 10. Spectrum of the *C*-weighted pressure in the front seat of a Honda Accord (Sano *et al.*, 1993).

Since 2000, active engine noise systems have been introduced on vehicles by Honda, for variable cylinder management systems (Inoue *et al*. 2004) and hybrid vehicles (Honda, 2005), and by Toyota for hybrid vehicles (Toyota, 2008), to improve the sound quality inside the vehicle. The synthesis of external noise on hybrid vehicles has also been suggested in order for them to be safer for pedestrians (Lotus, 2008).

Systems for the active control of vibration in vehicles have tended to concentrate on active engine mounts, which may also reduce internal noise. Nissan, for example, used two electromagnetic engine mounts to reduce the vibration in a 4-cylinder direct injection diesel in 1998 (as discussed by Sano *et al.*, 2002), which used feedforward control operating from 20 Hz to 130 Hz and a load cell on the body to provide an error signal. Figure 11 shows the main components of such an active engine mount, which integrates an electromagnetic actuator capable of generating about 50 N into a conventional hydromount. The reduction in internal noise when such mounts are used in a diesel vehicle is illustrated in Fig. 12. Honda also has introduced active engine mounts on vehicles with variable cylinder management in addition to engine noise systems to keep 6 cylinder noise and vibration quality during 3 cylinder operation (Matsuoka *et al.*, 2004). More recently, Jaguar have used Avon electromagnetic engine mounts to control the vibration on the diesel version of their XJ vehicle (Avon, 2005) again largely for sound quality reasons.

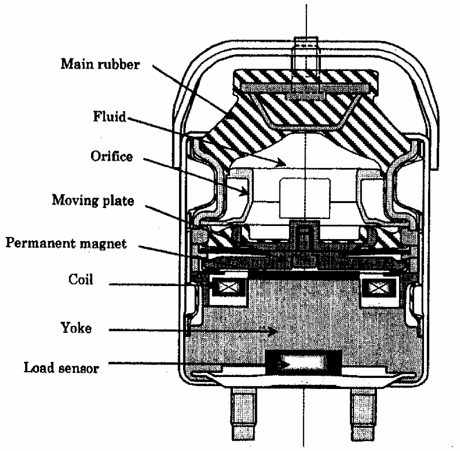


Figure 11. Components of an active engine mount (Sano *et al.*, 2002).

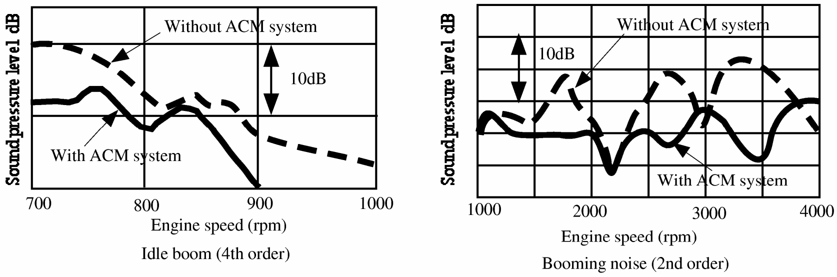


Figure 12. Measured spectrum of the sound in a diesel vehicle fitted with active engine mounts (Sano *et al.*, 2002).

## 5. FUTURE TRENDS

An important overall trend in the automotive industry is clearly the design of more fuel efficient vehicles. One of the main ways in which improvements in fuel efficiency can be achieved is by reducing weight. Reducing the weight of the body panels inevitably degrades their ability to attenuate low frequency noise, and so increases noise levels within the vehicle in this frequency range. Passive methods of noise control using absorption are very effective from mid-frequencies upwards, but generally rely on tuned systems to provide significant low frequency control. These tuned systems are only suitable for certain frequencies and themselves add weight to the vehicle. The drive towards more fuel efficient vehicles thus provides a real opportunity for active systems to be used to control the low frequency noise and vibration without significantly increasing weight.

Another trend is the active control of the overall sound quality in a vehicle, rather than just sound level. This may become important with the use of hybrid vehicles and in vehicles with variable cylinder management, since the changes in the quality of the sound inside these vehicles as the power source changes character can be disconcerting to the driver. The active control of sound quality generally involves a control system that drives the microphone signals inside the car towards a target, or command, signal, rather than just minimising it. This has been termed "noise equalisation" (Ji and Kuo, 1993), "sound synthesis" (McDonald *et al*. 1994), "active design" (Scheuren *et al.*, 2002) and "sound profiling" (Rees and Elliott, 2006), and can include the use of psychoacoustic models (Rees and Elliott, 2004). Recent trends also include the use of active control systems to provide a smoothly changing sound profile with engine speed, but with an emphasis on sporty sound during acceleration, to make the vehicle "fun to drive" (Kabayashi *et al.*, 2008). Further development along these lines is also possible, by providing an acoustic environment inside the vehicle that encourages the owner to drive in a more fuel-efficient way, for example. There has been some resistance to this trend towards active control of sound quality in some parts of the automotive industry, who see such electronic sound control as 'cheating' compared to mechanical re-design. As more virtual systems are introduced in vehicles, however, with active braking, stability and steering, and with a younger generation of customer, more used to audio manipulation, these objections are likely to die away.

Another important development is in the integration of electronic vehicle systems. Early active control systems were completely stand-alone, which meant that amplifiers, loudspeakers and processors were all duplicated in the audio and active control systems. This duplication has continued, to some extent because of the different responsibilities of the vehicle manufacturer and audio supplier, and the requirement that a given vehicle may be fitted with a number of different audio systems. It is a paradox that a customer can be sold a very expensive audio system in a vehicle for which the low frequency vehicle noise impairs their enjoyment of the audio system, and yet the additional cost of an active control system would be small compared with that of the audio system if it was fully integrated.

It is clear that active control can have most impact on light weight vehicles and that in these vehicles the active noise control system could be effectively integrated with the audio system, reducing the cost considerably. The loudspeakers may need to be upgraded for low frequency active control use, but this is still considerably less expensive than having two systems with duplicate drivers, wiring and amplifiers. Many audio systems now also contain considerable digital processing power, for CD and MP3 players, radio tuning and audio effects. This processing power is no longer expensive and can also be used to implement active control algorithms. The microphones and associated wiring remain an additional cost for active control systems on some vehicles, although others already have microphones fitted as standard, for hands-free telephone operation, for example. A fully integrated system could then be achieved if these microphones could be positioned so that they effectively measured the noise field in the car as well as the driver’s voice, and their signal could be integrated into the audio system.

## 6. SOURCES OF FURTHER INFORMATION

The physical basis of active noise control is explained in the textbook *Active Control of Sound* by P.A. Nelson and S.J. Elliott (1992). The principles of feedforward and feedback control, together with a more detailed discussion of the control algorithms used and an introduction to hardware and optimum placement of actuators and sensors, are described in *Signal Processing for Active Control* by S.J. Elliott (2001).

A good review of the development of automotive active noise and vibration control is provided by Sano *et al.* (2002) and an interesting discussion of the integration of active and passive control to reduce cost is provided by Su (2002).

Further information on individual systems can be found on the following websites:

http://www.isvr.soton.ac.uk/ACTIVE/Introduction.htm

http://www.mecheng.adelaide.edu.au/avc/projects/

http://www.mech.kuleuven.be/mod/other/topic_03_08

http://www.grouplotus.com/mediacentre_pressreleases/view/406

Movies

http://www.thefutureschannel.com/dockets/critical_thinking/bose/index.php

http://cnettv.cnet.com/?type=externalVideoId&value=6214373

http://uk.youtube.com/watch?v=zOtTX_NPPBs

## 7. REFERENCES


Avon-Rubber (2005) *Vibramount brochure.*

Dehandschutter, W. and Sas, P. (1998) Active control of structure-borne road noise using vibration actuators. *Journal of Vibration and Acoustics,* **120**, 517-523.

Bernhard, R.J. (1995) Active control of road noise inside automobiles. *Proc. ACTIVE 95*, 21-32.
Elliott, S.J. (2001) *Signal Processing for Active Control.*

Elliott, S.J., Stothers, I.M., Nelson, P.A., McDonald, A.M., Quinn, D.C. and Saunders, T. (1988) 'The active control of engine noise inside cars', *Proc. Internoise 88*, 987-990

Hasegawa, S., Tabata, T. and Kinoshita, T. (1992) The development of an active noise control system for automobiles. *Society Automotive Eng., Tech. Paper*, 922 086.

Honda (2005) *Honda Accord hybrid sedan interior*. Press release.

Inoue, T., Takahashi, A. Sano, H., Onishi, M. and Nakamura, Y. (2004) NV countermeasure technology for a cylinder-on-demand engine – Development of active booming noise control system applying adaptive notch filter. *Proc. SAE*, 976.

Ji, M.T. and Kuo, S.M. (1993) Adaptive active noise equaliser. *Proc. ICASSP-93* **1** 189-192.

Kobayashi, Y., Inoue, T., Sano, H., Takahashi, A. and Sakamoto, K. (2008) Active sound control in automobiles. *Proc. Inter-Noise 2008*, 112.

Lotus (2008) www.greencarcongress.com/2008/08/Lotus-engineer.htlm).

Mackay, A.C. and Kenchington, S. (2004) Active control of noise and vibration – A review of automotive applications. *Proc. ACTIVE 2004*, Williamsburg.

Matsuoka, H., Mikasa, T. and Nemoto, H. (2004), *SAE paper* 04Annual-714. NV countermeasure technology for a cylinder-on-demand engine – development of active control engine mount.

McDonald, A.M. *et al.* (1994) Sound synthesiser in a vehicle. *US Patent* 5381902

Nelson, P.A. and Elliott, S.J. (2001) *Active Control of Sound*, Academic Press.
Olsen, H.F. and May, E.G. (1953) Electronic sound absorber, *J. Acoustical Society of America*, **25**, 1130-1136.

Rees, L.E. and Elliott, S.J. (2006) Adaptive algorithms for sound profiling. *IEEE Trans. Speech and Audio Processing*

Rees, L.E. and Elliott, S.J. (2004) Psychoacoustic modelling for active sound profiling in automobiles. *Proc. UK Inst. of Acoustics* **26**(2).

Sano, H., Inoue, T., Takahashi, A., Terai, K. and Nakamura, Y. (2001) Active control system for low-frequency road noise combined with an audio system. *IEEE Trans. Speech and Audio Processing* **9**, 755-763.

Sano, H., Yamashita, T. and Nakamura, M. (2002) Recent applications of active noise and vibration control in automobiles. *Proc. ACTIVE 2002*, ISVR, Southampton, 29-42.

Scheuren, J., Shirmacher, R. and Hobelsberger, J. (2002) Active design of automotive engine sound. *Proc. Internoise 2002*, paper **N629**.

Su, J. Hannsen (2002) A  proposal for affordable applications of active control technology. *Proc Internoise 2002*, Paper N109;

Sutton, T.J., Elliott, S.J. and McDonald, A.M. (1994) Active control of road noise inside vehicles. *Noise Control Engineering Journal* **42**, 137-147.

Toyota (2008) *Toyota cuts 'muffled noise' in Crown Hybrid*. Tech-on, June 23.