STRUCTURAL-ACOUSTIC COUPLING AND PSYCHO-PHYSICAL EFFECTS IN THE ACTIVE CONTROL OF NOISE IN VEHICLES

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1 INTRODUCTION

The application of active noise control to vehicles has been investigated for over 20 years[1] and, consequently, a wide variety of systems have been proposed to control both road[2] and engine noise[1]. Until recently such systems were not sufficiently integrated into the vehicle's electronic systems to provide a cost effective noise control solution, however, more recent systems[3] have overcome this restriction. There has also been an increasing desire to reduce the fuel consumption of vehicles by making them lighter, which inevitably increases low frequency noise and the need for a lightweight solution such as active control.

The presented work proposes an active noise control system consisting of a feedforward element to control tonal engine noise and a modal feedback element to control road noise. The proposed hybrid system employs a single set of error microphones for the two control elements, a reference signal for the feedforward element is directly available from the ignition circuit and the car audio loudspeakers may be employed as secondary sources. The proposed system is, therefore, largely integrable to the vehicle's standard electronic system and may provide an affordable solution to low frequency noise control of both engine and road noise. However, due to the significant influence of structural-acoustic coupling upon the low frequency sound field within a car's passenger compartment[4] and the reliance of the proposed modal feedback system upon the acoustic mode shapes of the car's cabin, it is important to investigate the effect of structural-acoustic coupling upon the system's performance.

The two active noise control strategies are first introduced and the performance of the two control strategies is then simulated using an elemental model of structural-acoustic coupling in an enclosure for the cases where it is either fully or weakly coupled. The results of these simulations are used to highlight the effect of structural-acoustic coupling on the proposed control methods.

The psychoacoustic effects of active noise control are also considered. The human perception of sound is not the same as a simple measurement of the frequency spectrum, so the audible effects of attenuating low frequencies with active control are not always clear. Sound quality in cars is generally improved through the use of active control, mainly through the reduction in loudness[5, 6]. Active noise controllers can also take the frequency sensitivity profile of the ear into account by minimising a weighted spectrum[7]. In order to further understand the perception of active noise control, an analysis of the changes in perceived loudness, as opposed to just sound pressure, is presented here.

2 ACTIVE NOISE CONTROL STRATEGIES

2.1 Global Feedforward Control

Global control of enclosed sound fields has been extensively researched and a comprehensive review is provided by Nelson and Elliott[8]. In the context of noise control within vehicles global feedforward control has been used to control both engine[1] and road noise[2]. In order to control road noise it is necessary to employ a number of reference sensors, such as accelerometers, to provide the feedforward system with a reference signal; this results in an expensive system that is unsuitable for an integrable solution. Conversely, in order to control engine noise a reference signal may be obtained from either the ignition circuit, a tachometer or the Controller Area Network bus; this is achieved relatively cheaply and, therefore, provides a suitable control method.

The proposed global feedforward control strategy attempts to minimise the sum of squared pressures at a set of error sensors using a set of secondary sources that are driven by the reference signal via an adaptive filter, as shown in Figure 1 for a single secondary source. For a set of error sensors positioned in the corners of a rectangular enclosure this is approximately equivalent to minimising the total acoustic potential energy.

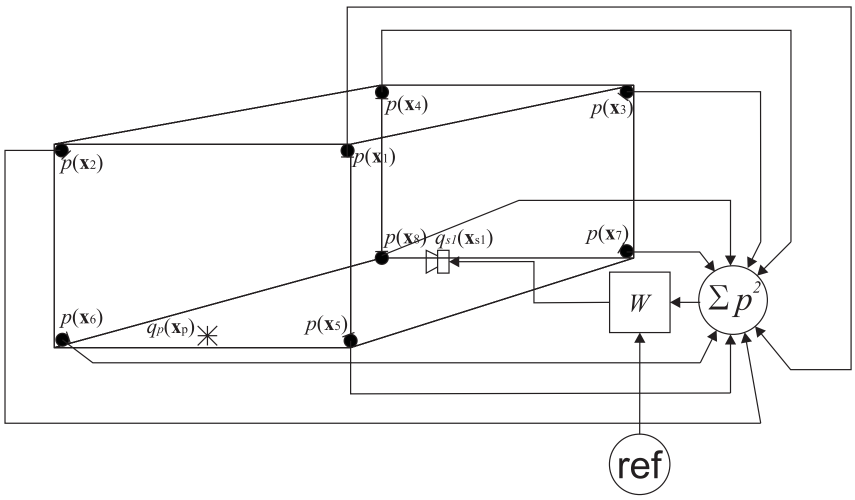

Figure 1: Global feedforward control strategy.

The cost function that global feedforward control aims to minimise is,

$$J_p = \frac{V}{4\rho_0 c_0^2 L_E} \boldsymbol{p}_e^H \boldsymbol{p}_e \quad (1)$$

where V is the enclosure volume, ρ_0 is the air density, c_0 is the speed of sound, L_e is the number of error sensors and $\boldsymbol{p}_e$ is the column vector of error sensor pressures. The error sensor pressures result from the summation of the primary and secondary sources such that $\boldsymbol{p}_e$ can be expressed as,

$$\boldsymbol{p}_e = \boldsymbol{Z}_p \boldsymbol{q}_p + \boldsymbol{Z}_s \boldsymbol{q}_s \quad (2)$$

where $\boldsymbol{Z}_p$ is the $(L_e \times 1)$ vector of transfer impedances between the error sensors and the primary source, $\boldsymbol{Z}_s$ is the $(L_e \times M)$ matrix of transfer impedances between the error sensors and the M secondary sources, $\boldsymbol{q}_p$ is the vector of primary source strengths, although only a single element is used here, and $\boldsymbol{q}_s$ is the column vector of secondary source strengths. The cost function J_p can thus be expressed in Hermitian quadratic form by substituting equation (2) into equation (1) and rearranging:

$$J_p = \frac{V}{4\rho_0 c_0^2 L_E}\left[\boldsymbol{q}_s^H \boldsymbol{Z}_s^H \boldsymbol{Z}_s \boldsymbol{q}_s + \boldsymbol{q}_s^H \boldsymbol{Z}_s^H \boldsymbol{Z}_p \boldsymbol{q}_p + \boldsymbol{q}_p^H \boldsymbol{Z}_p^H \boldsymbol{Z}_s \boldsymbol{q}_s + \boldsymbol{q}_p^H \boldsymbol{Z}_p^H \boldsymbol{Z}_p \boldsymbol{q}_p\right] \quad (3)$$

The vector of optimal secondary source strengths that minimises the sum of the squared error sensor pressures is then given by[8],

$$\boldsymbol{q}_{s0} = -[\boldsymbol{Z}_s^H \boldsymbol{Z}_s]^{-1} \boldsymbol{Z}_s^H \boldsymbol{Z}_p \boldsymbol{q}_p \quad (4)$$

2.2 Modal Feedback Control

The use of feedback control offers a potential alternative to feedforward control for road noise as it does not require additional reference sensors and as such may reduce the cost of implementation. Although research into feedback control in car cabins is limited, based on its possible advantage Sano *et al*[3] have implemented a feedback control system that reduces the drumming noise at the front seats by around 10 dB, whilst avoiding enhancements at rear seats. In order to improve upon this performance and achieve global control of road noise an alternative feedback control strategy is proposed herein based on the system presented by Clark and Gibbs[9].

The feedback system presented by Clark and Gibbs employs a set of collocated transducer pairs that are spatially weighted in order to control specific acoustic modes. However, in order to employ the same error microphones for the feedforward and feedback control elements it is necessary that the sensors and actuators are not collocated, as in the feedforward control system this would create small zones of localised control around the transducer pairs. The proposed system therefore differs from that previously presented in its use of non-collocated sensors and actuators.

The principle of modal feedback control is to sum the pressures at a number of error sensors in order to maximise the composite, modal error signal at a specific mode and thus maximise the control of that particular mode when the modal error signal is reproduced by the secondary source. The polarity with which the error signals are summed is determined by their position relative to the nodal lines of the acoustic mode to be controlled. For the source-sensor system shown in Figure 2, in order to control the first longitudinal acoustic mode the polarity of the four error sensors in the rear corners of the enclosure is inverted before summation with the outputs of the four error sensors in the front of the enclosure.

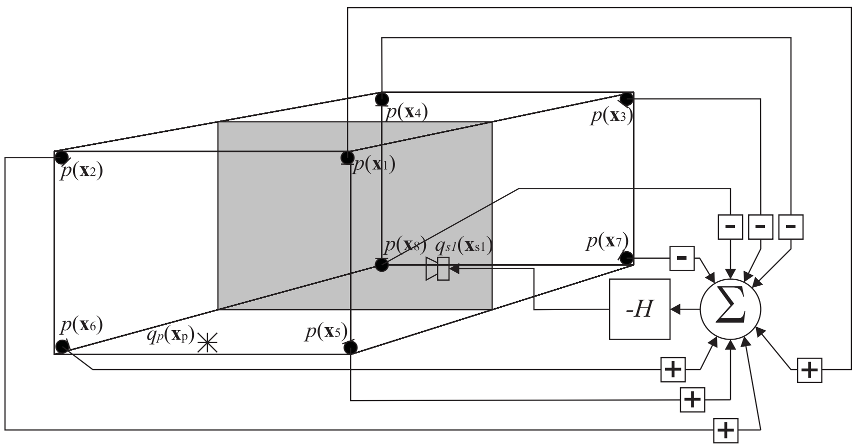

Figure 2: Modal feedback control system.

For the system presented in Figure 2 the composite error signal, p_c can be formulated as,

$$p_c = \boldsymbol{\varphi}_L \boldsymbol{p}_e \quad (5)$$

where $\boldsymbol{\varphi}_L$ is a $(1 \times L_e)$ vector of polarity inversions. By the action of the negative feedback the secondary source's volume velocity is given by,

$$q_s = -H p_c Y \quad (6)$$

where H is the feedback gain and Y is the volume velocity produced by the loudspeaker per unit current input and thus describes the loudspeaker response. A single degree of freedom model of the loudspeaker dynamics will be used within this paper where Y is given by,

$$Y = \frac{j\omega B l A_d}{S\left(1 + j2\zeta_L \frac{\omega}{\omega_L} - \left(\frac{\omega}{\omega_L}\right)^2\right)} \quad (7)$$

where ω is the angular frequency, Bl is the transduction coefficient, A_d is the area of the diaphragm, S is the stiffness, ζ_L is the damping ratio, and ω_L is the loudspeaker's natural frequency. Using equations (2), (5) and (6) it can be shown that the composite error signal is given by,

$$p_c = \frac{\boldsymbol{\varphi}_L \boldsymbol{Z}_p \boldsymbol{q}_p}{1 + \boldsymbol{\varphi}_L \boldsymbol{Z}_s H Y} \quad (8)$$

where the open-loop response is $G = \boldsymbol{\varphi}_L \boldsymbol{Z}_s H Y$.

From the description of the modal feedback control system it is clear that its performance is dependent upon the modal properties of the acoustic enclosure. It is widely reported that structural-acoustic coupling has a significant effect upon the modal properties of a small enclosure such as a car cabin[4, 10] and, therefore, it is important to validate the modal feedback system in an enclosure with structural-acoustic coupling.

3 THE EFFECT OF COUPLING ON ACTIVE NOISE CONTROL

The effect of structural-acoustic coupling upon the active noise control strategies presented above will be investigated using the elemental model of structural-acoustic coupling derived by Cheer and Elliott[11]. For both global feedforward and modal feedback control the acoustic potential energy produced by a single primary monopole source with a volume velocity of 10^{-5} m^3s^{-1} will be controlled using a single secondary source, the sources and eight error sensors will be positioned in both cases as shown in Figure 1 and Figure 2. The properties of the modelled enclosure are detailed in Table 1.

Table 1: Enclosure properties

Property	Value
Enclosure length, L_1	2.4 m
Enclosure width, L_2	1.2 m
Enclosure height, L_3	1.1 m
Acoustic damping ratio, ζ_n	0.1
Young's Modulus, *E*	5×10^9 Nm^{-2}
Panel thickness, *h*	12 mm
Poisson's ratio, *v*	0.3
Panel density, ρ_s	465 kgm^{-3}
Structural damping ratio, ζ_k	0.05

3.1 Simulations

3.1.1 Global Feedforward Control

The optimum secondary source volume velocity to minimise the sum of the squared pressures has been calculated using equation (4) with the acoustic transfer impedance calculated according to the model of structural-acoustic coupling for the weakly-coupled (rigid walled) and fully-coupled (non-rigid walled) enclosures. Figure 3 shows the acoustic potential energy before control, E_p, and after control, E_{p0}, for the rigid and non-rigid walled enclosures. From this plot it can be seen that, although the uncontrolled responses (sold lines) are significantly different, the responses after control (dashed lines) are almost identical.

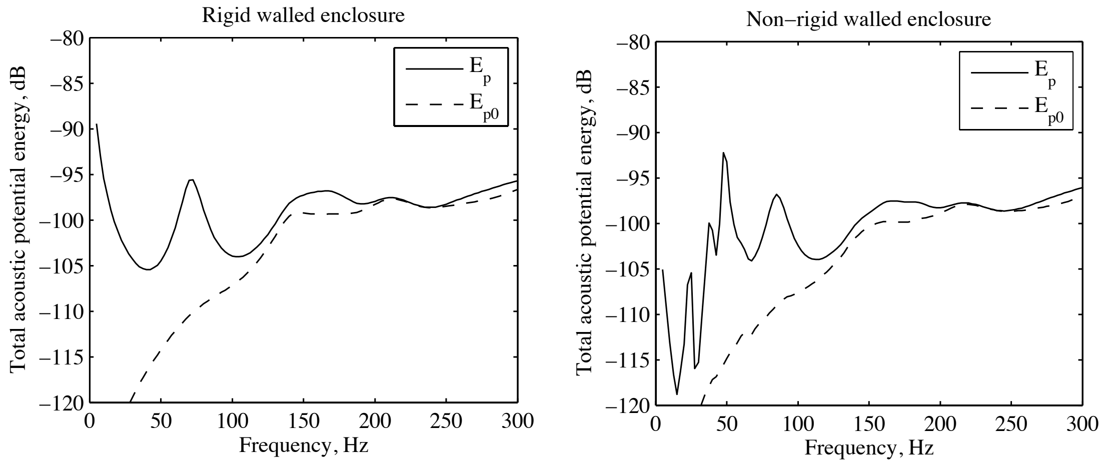

Figure 3: Total acoustic potential energy in the rigid and non-rigid enclosures when driven by a primary source alone (solid lines) and when the sum of the eight squared error sensor pressures has been minimised using a single secondary source positioned as shown in Figure 1 (dashed lines).

3.1.2 Modal Feedback Control

The performance of the modal feedback control strategy described in Section 2.2 has been simulated once again using the acoustic transfer impedances according to the elemental model. The gain of the feedback control system, *H*, has been set to ensure that the maximum enhancement of the error signal given by equation (8) is 6 dB. Figure 4 shows the change in the total acoustic potential energy as a result of the modal feedback control strategy for the rigid and non-rigid enclosures. From this plot it can be seen that in both the rigid and non-rigid cases the feedback control strategy achieves a significant reduction around the first longitudinal mode despite the shift in its resonance frequency introduced by structural-acoustic coupling. Although the minimum of the potential energy after control is identical in both cases, the reduction at the 85 Hz target mode in the non-rigid system is 3 dB worse than that achieved at the 71 Hz mode in the rigid walled system. Additionally, enhancements in the potential energy at higher frequencies are around 1 dB greater for the non-rigid walled system. Therefore, although the effects of structural-acoustic coupling reduce the achievable control, the energy reduction is still comparable to that achieved previously[3].

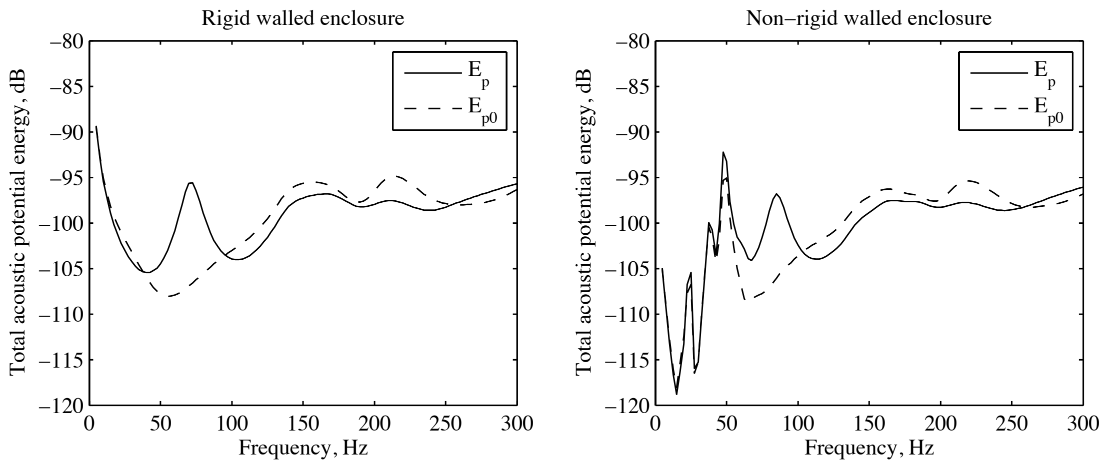

Figure 4: Total acoustic potential energy in the rigid and non-rigid enclosures when driven by a primary source alone (solid lines) and when the modal feedback control strategy using a single secondary source positioned as shown in Figure 2 has been employed with a maximum error signal enhancement of 6 dB (dashed lines).

4 PSYCHOACOUSTICS OF ACTIVE NOISE CONTROL

Active control is more effective at low frequencies than higher frequencies; feedback control not only attenuates low frequencies but also enhances higher frequencies due to spillover. In both control regimes, the reduction of low frequencies means the higher frequencies can become more prominent. It is well known that the human ear does not hear all frequencies equally, with the ear being less sensitive to low frequencies in particular. In addition, psychoacoustic masking means that some combinations of frequency spectrum components might not be audible, or may become audible after noise control removes other frequencies. The masking of high frequency sounds by low frequency tones is particularly effective. The perceived reduction in loudness might therefore not be as great as the reduction in total acoustic potential energy might suggest.

4.1 Perceived Loudness

Measures of sound level such as total acoustic potential energy or sound pressure level are convenient to use when calculating or measuring the performance of active noise control systems, but these objective measures do not necessarily predict the subjective performance. The perceived loudness of a sound is not directly related to the sound pressure level, although as a rule of thumb, the sound pressure level needs to reduce by 10 dB to halve the loudness. One factor affecting loudness is psychoacoustic masking, where low-frequency sounds can make higher-frequency sounds less audible. This can be important if the reduction in low frequencies by an active control system unmasks higher frequencies, making them audible.

A model of loudness developed by Zwicker and Fastl[12] was implemented to investigate the perceived effectiveness of active noise control. The loudness model converts the frequency spectrum into critical bands, the frequency bandwidth of the ear. The signal within each critical band is converted into an excitation profile which represents the response of the basilar membrane to the sound in that critical band. Each excitation curve is then translated into a specific loudness, N', which is a measure of loudness per critical band. The total loudness, N, is the integral of N' over all 24 critical bands. Loudness, in sones, can be converted into loudness level, L_N, in phons.

The total acoustic potential energy as a result of the modal feedback system in Figure 4 shows a reduction at about 80 Hz and some enhancement at higher frequencies. A very simplified two-tone representation of the spectral effect of such an active control system is shown in Figure 5. With active control, an 80 Hz tone is attenuated by 10 dB and a 200 Hz tone is enhanced by 5 dB. The lower part of Figure 5 shows the specific loudness curve, in which the 80 Hz tone falls in critical band 1 and is reduced by about 1 sone/Bark. The 200 Hz tone is enhanced by 0.6 sone/Bark within critical band 3 and also causes increased loudness at higher frequencies as indicated by the curve that decays away at higher critical bands. The total loudness is defined as the area below the specific loudness curve and in this case the reduction of the area from the 80 Hz tone is counteracted by an increase in area from the 200 Hz tone. Table 2 shows the results of the loudness calculations. Although there is a small reduction in sound pressure level, there is also a slight increase in loudness level.

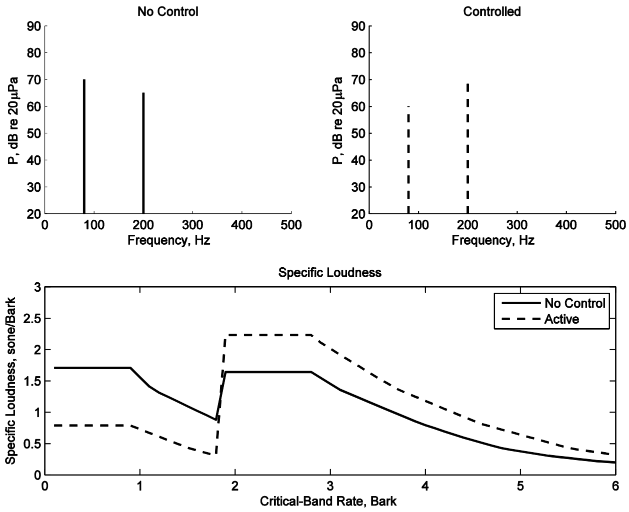

Figure 5: *Top* – line spectrum before (left) and after (right) active noise control. *Bottom* – specific loudness contours before (solid line) and after (dashed line) control

Table 2: The effectiveness of active noise control on sound pressure level and loudness level

	SPL (dB)	L_N (phon)
Not controlled	71.2	67.6
Controlled	70.4	68.5
Change	**-0.8**	**+0.9**

5 CONCLUSIONS

The use of integrated active noise control systems to control unwanted engine and road noise inside vehicles offers a potential method of reducing the weight of the sound package and, therefore, improving fuel efficiency. An active control system with a feedforward element to control engine noise and a feedback element to control road noise has been presented. The feedback element of the system is based on forming a modal error signal from a number of error sensors and thus via negative feedback reducing the acoustic response at a particular mode. Due to the effects of structural-acoustic coupling upon the modal response of vehicles[4] and the particular reliance of the proposed modal feedback system upon the modal response the main focus of this paper is to determine the effect of structural-acoustic coupling upon the proposed active noise control system.

Using an elemental model of structural-acoustic coupling previously presented[8] the performance of the two active noise control systems has been evaluated in both rigid and non-rigid enclosures. From these simulations it has been shown that for feedforward control, despite the change in the uncontrolled responses, the controlled responses are almost identical. For the modal feedback system the reduction in acoustic potential energy at the first longitudinal mode is reduced by around 3 dB for the non-rigid enclosure compared to the rigid walled enclosure. However, the level of control is still useful.

The effect of active control on the perception of noise has also been considered. An illustration based on modal feedback control simulations has shown that the perceived loudness might not reduce as much as expected even if the overall sound pressure level has been reduced.

Future work will validate the performance of the proposed control systems in an actual vehicle as well as investigate the implementation of multi-modal control using the proposed feedback strategy. This work has shown that we can include loudness and other psychoacoustic factors as a measure of the control performance, with the potential for incorporating them into the control system itself.

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