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THE PERFORMANCE OF ACTIVE CONTROL OF SOUND AND VIBRATION IN A FULLY-COUPLED STRUCTURAL-ACOUSTIC SYSTEM USING DIFFERENT REFERENCE SENSORS

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Abstract

The increasing demand for automotive refinement and improved audio quality in cars motivates the need for better techniques to reduce random road noise. One approach that shows promise for substantial reduction of low frequency road noise is active control, which can be integrated into a car audio system for commercial applications. This paper investigates the effectiveness of various configurations of reference sensors to detect the primary disturbance in a feedforward active control system. The results from a model problem are presented based on a full coupling analysis between the vibration of the car panels excited by multiple uncorrelated sources and the interior acoustic field. Different numbers and positions of force sensors, microphones and accelerometers are used as reference sensors to evaluate the performance of active control. Unconstrained frequency domain optimisation is implemented in minimising the mean square errors. With the correct locations, only a small number of microphones or accelerometers acting as reference sensors are needed to give the best overall performance despite many uncorrelated primary disturbance sources being present. It is also shown that microphones give a slightly better performance compared to accelerometers.

INTRODUCTION

Manufacturers in the automotive industry have recognised that reducing the interior noise of vehicles promises better quality in their products and therefore will increase sales. The road noise inside vehicles arises from the irregularities of road profiles and changes of vehicle speed, which then generates the non-stationary vibrations of tyres and wheels. This random broadband noise is then transfered into the car interior via the structural path.

Passive technologies have been used to reduce road noise inside vehicles. The problem was that this technique contributes to weight increase, which is not good to maximise fuel efficiency. In 1936, P. Lueg [2], introduced the concept of active sound cancellation to reduce low frequency sound in a duct. Since active control offers the possibility of controlling noise with little weight penalty, R&D in active noise control (ANC) of road noise inside vehicles has been the subject of interests since 1990s [4] - [8]. However, not many systems have been widely implemented in commercial applications, especially in reducing the effect of road noise inside moving vehicles. This is because the manufacturing cost is too expensive due to the fact that feedforward control strategy is usually used whereby many reference sensors (typically accelerometers which can be expensive for commercial implementation) are needed in order to have enough information of the primary noise disturbance. This information is crucial in order to cancel the noise using the secondary actuator to get the optimum noise control.

ACTIVE CONTROL OF STOCHASTICALLY-EXCITED LOADED SYSTEM WITH DIFFERENT REFERENCE SENSORS

The work presented here considers a theoretical investigation into the active control of sound transmission into a structural-acoustic coupled system modelled by a rectangular acoustic enclosure coupled with a flexible structural panel driven by multiple random forces **f** as the primary source and an acoustic monopole q_s as the secondary sound source, as illustrated in Fig.(1) and its associated block diagram as shown in Fig.(2) in order to control the field.

In a fully-coupled analysis, the kinetic energy of the panel not only depends on the primary force excitation, but also from the back reaction of the pressure in the enclosure [3]. The total complex amplitude of acoustic modal pressure vector $\mathbf{a}_{c_t}$ can be written as

$$\mathbf{a}_{c_t} = \mathbf{a}_{c_p} + \mathbf{B}_c\mathbf{q}_s \tag{1}$$

where

$$\mathbf{a}_{c_p} = \mathbf{Z}[\mathbf{I} + \mathbf{YZ}]^{-1}\mathbf{b}_u \tag{2}$$

$$\mathbf{B}_c = [\mathbf{I} + \mathbf{ZY}]^{-1}\mathbf{B}_u \tag{3}$$

where $\mathbf{a}_{c_p}$ is the coupled acoustic mode amplitude without active control, $\mathbf{B}_c$ is the matrix of coupled coupling coefficient, **Y** is the structural modal mobility matrix of the panel and **Z** is the acoustic modal impedance matrix of the enclosure [3]. In order to consider different reference signals as the input to the system, the secondary acoustic source can be represented as

$$\mathbf{q}_s = \mathbf{Wx} \tag{4}$$

where **W** is the matrix of control filters to be designed. Therefore, Eq.(1) can now be written as

$$\mathbf{a}_{c_t} = \mathbf{a}_{c_p} + \mathbf{B}_c\mathbf{Wx} \tag{5}$$

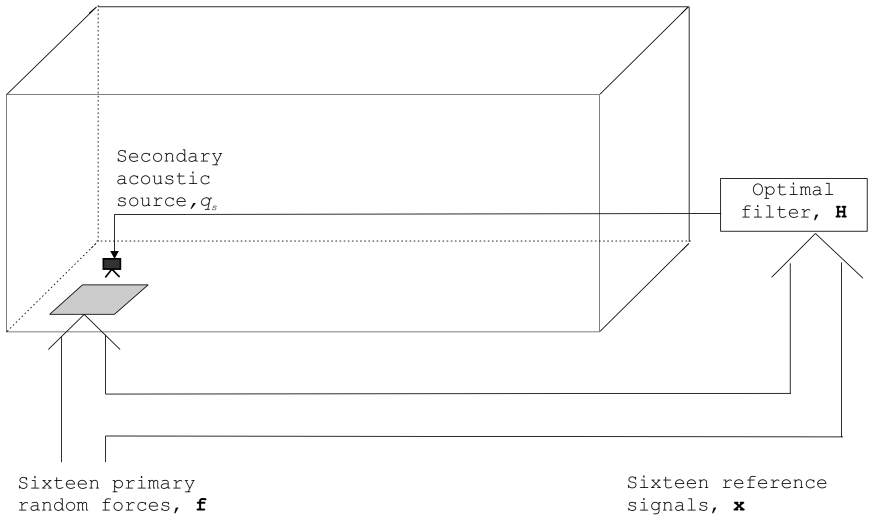

Figure 1: *Multiple random forces excite the flexible panel which is coupled to an acoustic enclosure controlled by a secondary acoustic source* q_s*. The reference signals,* **x**, *are obtained by direct observation of the primary forces, as shown, or from accelerometers on the panel or microphones in the enclosure.*

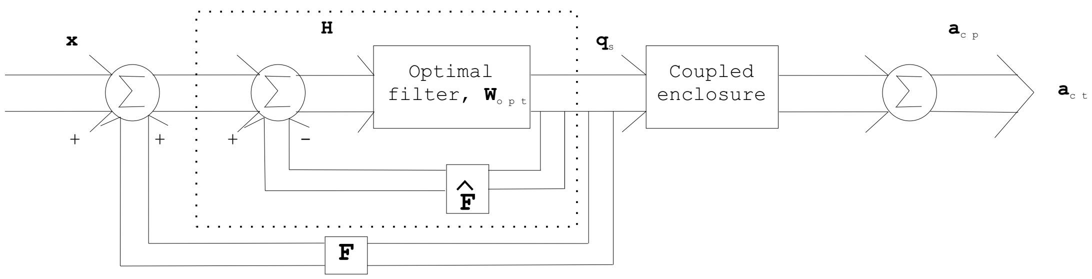

Figure 2: *A block diagram shows the internal model control (IMC) arrangement, assuming perfect feedback cancellation in the feedforward system. The reference signals,* **x**, *can also be* **u** *or* **p**, *depends on velocity vibration of the flexible panel measured by accelerometers or acoustic pressure of the enclosure measured by microphones, respectively.*

where **x** is the reference signals.

The potential energy of the stochastically-excited loaded system can be calculated from

$$\text{trace}\left[\mathbf{S}_{a_{c_t}a_{c_t}}\right] = \text{trace}\left[E\left[\mathbf{a}_{c_t}\mathbf{a}_{c_t}^{\mathrm{H}}\right]\right] \tag{6}$$

where E denotes expectation operator, $\mathbf{a}_{c_t}$ is given by Eq.(5) and

$$\mathbf{a}_{c_p} = \mathbf{D}_c\mathbf{f} \ , \tag{7}$$

where $\mathbf{D}_c = \mathbf{Z}[\mathbf{I} + \mathbf{YZ}]^{-1}(1/M_s)\mathbf{B}\phi_m(\mathbf{y}_p)$ [3]. Therefore, expanding the outer vector product of Eq.(6), we get

$$\text{trace}\left[\mathbf{S}_{a_{ct}a_{ct}}\right] = \text{trace}\left[E\left[\mathbf{B}_c\mathbf{W}\mathbf{S}_{xx}\mathbf{W}^{\mathrm{H}}\mathbf{B}_c^{\mathrm{H}} + \mathbf{B}_c\mathbf{W}\mathbf{S}_{x,a_{cp}}^{\mathrm{H}} + \mathbf{S}_{x,a_{cp}}\mathbf{W}^{H}\mathbf{B}_c^{\mathrm{H}} + \mathbf{S}_{a_{cp}a_{cp}}\right]\right] \tag{8}$$

where

$$\mathbf{S}_{xx} = E[\mathbf{x}\mathbf{x}^{\mathrm{H}}], \qquad k \text{ x } k \text{ matrix} \tag{9}$$

$$\mathbf{S}_{x,a_{cp}} = E[\mathbf{a}_{cp}\mathbf{x}^{\mathrm{H}}], \qquad N \text{ x } k \text{ matrix} \tag{10}$$

$$\mathbf{S}_{a_{cp}a_{cp}} = E[\mathbf{a}_{cp}\mathbf{a}_{cp}^{\mathrm{H}}], \qquad N \text{ x } N \text{ matrix} \tag{11}$$

The optimal filter, $\mathbf{W}_{opt}$, which minimises Eq.(6) is obtained by using least mean squares method as follows

$$\mathbf{W}_{opt} = -\left[\mathbf{B}_c^{\mathrm{H}}\mathbf{B}_c\right]^{-1}\mathbf{B}_c^{\mathrm{H}}\mathbf{S}_{x,a_{cp}}\mathbf{S}_{xx}^{-1} \tag{12}$$

Therefore, by substituting Eq.(12) into Eq.(8), the minimised acoustic potential energy can be written as [1]

$$\mathbf{E}_{p_{min}} = \text{trace}\left[\mathbf{B}_c\mathbf{W}_{opt}\mathbf{S}_{xx}\mathbf{W}_{opt}^{\mathrm{H}}\mathbf{B}_c^{\mathrm{H}} + \mathbf{B}_c\mathbf{W}_{opt}\mathbf{S}_{x,a_{cp}}^{\mathrm{H}} + \mathbf{S}_{x,a_{cp}}\mathbf{W}_{opt}^{H}\mathbf{B}_c^{\mathrm{H}} + \mathbf{S}_{a_{cp}a_{cp}}\right] \tag{13}$$

With direct observation of the primary forces, the method has already been described as above. When using accelerometers to detect the structural vibration velocity of the coupled panel-enclosure or microphones to detect the acoustic pressures inside the coupled panel-enclosure, simplified equations were used to recalculate the optimal least squares filter and the minimised acoustic potential energy, as defined in Eq.(12) and Eq.(13), respectively.

Calculation of overall attenuation

The overall (or average) attenuation of the acoustic potential energy in dB for all frequencies between 1 Hz to 500 Hz, is calculated as follows

$$\text{Overall attenuation in dB} = -10\ \text{log10}\left[\frac{\int_1^{500} PE_{aftercontrol}}{\int_1^{500} PE_{beforecontrol}}\right] \tag{14}$$

SIMULATION RESULTS

For these simulations, it is assumed that sixteen uncorrelated random forces are exciting the flexible panel simultaneously and up to sixteen reference signals were used to determine the performance. These reference signals could be obtained from direct observation of the primary forces, from accelerometers on the panel or from microphones inside the enclosure adjacent to the panel. The result can be seen in Fig.(3) where we can see that despite the fact that there are sixteen random forces acting as primary disturbances, as long as the accelerometers or microphones were carefully positioned on the panel, then only four reference sensors are needed to obtain the best attenuation in potential energy (about 4.3 dB), with the microphone having slightly better performance compared to using accelerometers.

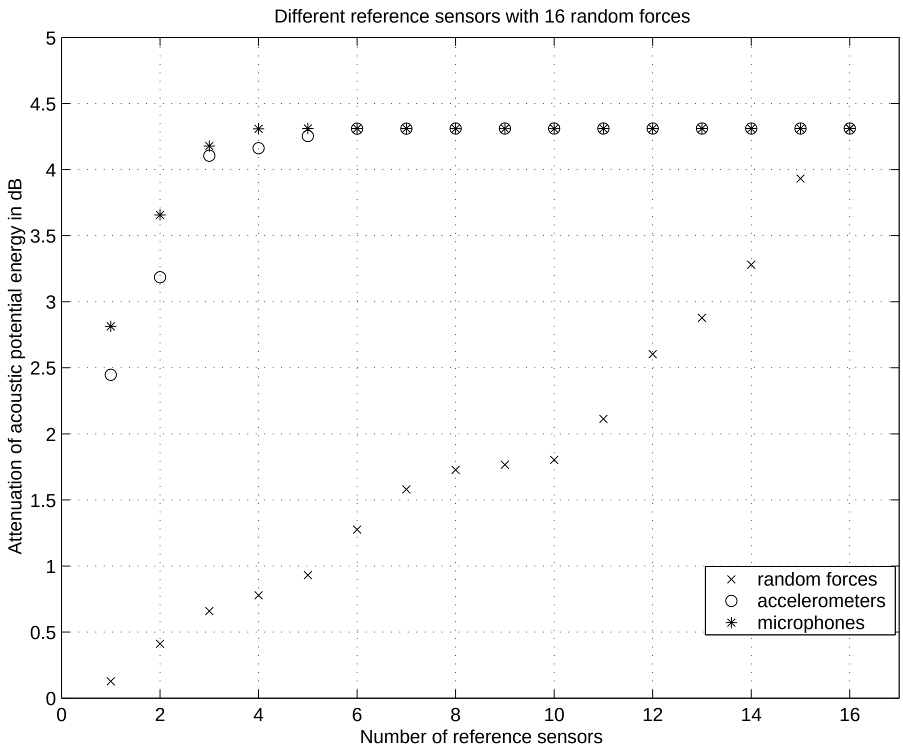

Figure 3: *The average attenuation in acoustic potential energy at all frequencies when one to sixteen random forces were used as reference signals* (x), *accelerometers* (o) *and microphones* (*) *are used as reference sensors.*

Other simulations have been carried out with half of the number of primary disturbance as well as the reference sensors in different sensor position configuration, and it was again found that only about four accelerometers or microphones are required, regardless of the number of independent forces acting on the panel. The required number of reference sensors is thus dependent on the number of plate and enclosure modes significantly excited, rather than the type of plate excitation.

CONCLUSIONS AND FURTHER WORK

The active control of random sound transmission into a fully-coupled structural-acoustic system has been considered. The simulation results demonstrate that a slightly better reduction in potential energy is obtained when microphones are used as the reference sensors. With the correct positioning, accelerometers can also show similar performance as of microphones, although slightly lower overall attenuation. It is anticipate that future work will continue to investigate the effect of multiple random inputs as the primary source into the coupled system in the time domain optimisation in order to fully modelled the real interior road noise in a car.

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