ACTIVE NOISE CONTROL AT UWA
– A Brief Review of the Acoustical Understanding and Practical Application of ANC Systems

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The design, analysis and realization of Active Noise Control (ANC) systems have been challenges to acoustical and control communities over the last two decades. It is largely due to the effort in the acoustical community and to advances in digital signal processing technologies that significant progress has been made in this field. As part of this international quest in advancing ANC technology, researchers at the University of Western Australia (UWA) have focused on (1) applying an understanding of acoustical systems to the design of ANC systems and (2) the development of practical ANC systems for the Western Australian mining, shipbuilding and building industries. This paper presents a brief review of the contribution to these two areas by the UWA team including the results obtained from several practical applications.

INTRODUCTION

Active noise control is a field of research and application concerned with attenuating unwanted noise using active devices and arrangements. The system that implements the active noise control usually consists of an acoustical system, a controller, sensors and actuators with associated interface electronics that provide interaction between the acoustical system and controller. The design, analysis and realization of ANC systems have been challenges to acoustical and control communities over the last two decades because of the complicated nature of acoustical systems (distributed parameters, effect of boundary, vibro-acoustical coupling, broadness of the frequency and dynamic ranges of the acoustical signal to be attenuated) and lack of general control theory feasible for various requirements in practical noise control. It is largely due to the effort in the acoustical community, where good understanding of the physical system was brought into the design of ANC systems, and to advances in digital signal processing technologies that significant progress has been made in this field.

In this brief paper, we use several practical examples, which have been studied at UWA, to illustrate two issues (utilization of the understanding of acoustics for effective ANC, and development of practical ANC systems) in the recent development of ANC. The importance of these two issues is also discussed by outlining some relevant key achievements in an international context.

1. UNDERSTANDING ACOUSTICS FOR EFFECTIVE ANC

1.1 Feedback ear defender (FED)

The acoustic system within a FED (Figure 1) is the sound pressure P_c in the ear-cup which is due to the transmission of external sound P_o and radiation from the vibrating diaphragm of the loudspeaker installed in the ear-cup. H_a is the frequency response function of the sound pressure in the cavity with respect to the sound pressure generated by the speaker diaphragm and includes absorption and leakage. The total sound pressure is measured by a microphone (*M*) and actively attenuated by the loudspeaker (*L*) suitably excited by an electronic compensator (*C*) which takes the output of the microphone as its input. As a result of the feedback control, the ratio of P_c and uncontrolled sound pressure P_u can be expressed as

$$\frac{P_c}{P_u} = \frac{1}{1 - H_c H_m H_l H_a}, \tag{1}$$

where H_c, H_m and H_l are the frequency responses of the compensator, microphone, and loudspeaker respectively.

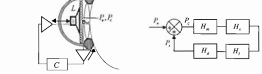


Figure 1 (a) Schematic and (b) block diagram of a feedback noise control ear defender.

It becomes clear that an effective reduction of the ratio can be achieved by designing a large gain of the compensated open-loop frequency response function $H_{co} = H_C H_M H_l H_a$ and a small gain ($|H_{co}| \ll 1$) at the phase cross over frequency where the phase of H_{co} is equal to zero [1]. Although a compensator with biquadratic filter characteristics and an optimal design in selecting the filter parameters [2-3] is capable of increasing the gain near one frequency while reducing it at others, the uncompensated open-loop frequency response ($H_o = H_m H_l H_a$) of the system can significantly affect the performance of the control system. For example, an H_o with broad and uniform magnitude and phase response allows a high noise reduction level within a broader frequency band when optimal feedback is introduced. However noise reduction can only be achieved in a limited frequency range by optimal feedback if the acoustical system has a fast phase decay rate with frequency. Therefore it is necessary to select H_a, H_m and H_l so that the magnitude and phase of H_o are suitable for the effective noise control. It can be shown [1] that

$$H_o = H_M H_L H_A = \frac{E_m}{E_g} = \frac{H_M \left(\frac{Bl}{A_L}\right) Z_{\inf}}{Z_e \left[Z_L + Z_{\inf} + Z_{inb} + \left(\frac{Bl}{A_L}\right)^2 \frac{j\omega}{Z_e} \right]} \qquad (2)$$

where $Z_{\inf}$, Z_{inb}, Z_L and Z_e are respectively acoustical impedances of the front and back cavities of the ear-cup, acoustical impedance due to air leakage and electrical impedance of the loudspeaker. $\frac{Bl}{A_L}$ is a parameter associated with the magnetic field strength, coil length and effective diaphragm area of the loudspeaker. Equation (2) indicates that the characteristics of H_o are dependent on many system parameters. Figure 2 shows the change of the magnitude and phase curves of H_o for a typical active ear defender when the mass of the speaker diaphragm is used as a varying parameter [3]. Obviously, the open-loop response with smaller loudspeaker mass will give rise to a broader frequency range of noise reduction than that with a larger loudspeaker mass. It is therefore necessary to carefully design the uncompensated open-loop response of an active ear defender before designing an optimal controller for best performance.

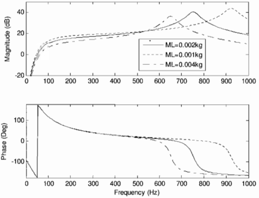
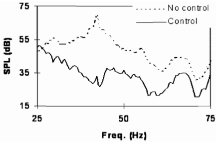


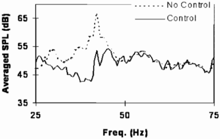
Figure 2 Magnitude and phase plot of the uncompensated open-loop frequency response function H_o when the mass of the speaker diaphragm is used as a varying parameter.

1.2 Feedback control of noise in an office

Feedback control was used to attenuate the modal response of low frequency noise in an office due to random excitations [4]. In this application, the distance r between the control



(a)



(b)

Figure 3 Uncontrolled (dotted) and controlled (solid) sound pressure level (a) at the error microphone location and (b) by spatial average when $r = 13cm$.

loudspeaker and error microphone was a crucial parameter in producing a global noise reduction in the office. The control was implemented in a $4{\times}2.9{\times}3m^3$ office and the uncontrolled sound field in the office was generated by a white noise source. The controller design is based on the constrained optimization of the compensator coefficients, the method similar to the compensator design for active ear defenders [1]. Figure 3 shows the experimental results (for a near field microphone placement at $r = 13cm$ from the control source) of the uncontrolled (dotted curved) and controlled sound pressure levels respectively (a) at the error sensor location and (b) by spatial average of sound pressure throughout the room. For this experiment, a large amount of noise reduction is achieved at the error sensor location, but little reduction is achieved globally. On the other hand if the sensor is placed far from the control loudspeaker ($r = 170cm$), the frequency range for global noise reduction becomes narrower and an increase in sound pressure is observed outside of the range as depicted in Figure 4. This is because of the increase in phase decay by the longer travel distance of the radiated sound, resulting in the phase crossover frequency being brought within the range. Finally when the error microphone is located adequately far away from the near field ($r = 28cm$), significant local and global noise reduction were achieved within an adequate broad frequency range (Figure 5).

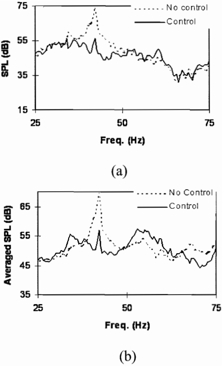


Figure 4 Uncontrolled (dotted) and controlled (solid) sound pressure level (a) at the error microphone location and (b) by spatial average when $r = 170cm$.

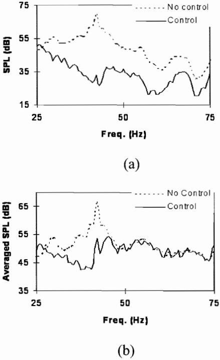


Figure 5 Uncontrolled (dotted) and controlled (solid) sound pressure level (a) at the error microphone location and (b) by spatial average when $r = 28cm$

The above experimental results indicate that there exists an optimal distance between the control loudspeaker and error microphone. If the actual distance is less than this optimal distance, a direct field dominates the control field, and only local attenuation at the error sensor location is possible. Further than the optimal distance, a large phase delay in the open-loop frequency response causes the phase crossover frequency to be too close to the frequency range of interest and therefore poor control performance is achieved.

1.3 Active control of sound transmission through double panel partitions

The acoustical system for active control of sound transmission through a double panel partition includes the sound fields in the receiving room and air cavity between two panels, and vibration in the two panels (Figure 6). As the acoustical system has an increased number of sub-systems, the option for the placement of a finite number of actuators increases even for the same aim of the control: minimization of the total acoustical potential energy in the room. Three arrangements of control actuator were considered [5, 6] and they are respectively (1) to directly attenuate the sound pressure field in the room using one point acoustic control source in the room; (2) to control the sound radiation into the room using a vibration control source on plate 2; and (3) to block the noise transmission path by inserting one point acoustic control source between the double walls. The condition for effective reduction of the sound energy in the room is that the primary and secondary pressure modal coefficient vectors in the room are proportional. For case (1), this condition is described as

$$[P_{N_2}^{(p)}] \propto [G_{N_2}^{(c)}(r_2^{(c)})] \qquad (3)$$

where $[P_{N_2}^{(p)}]$ is the modal coefficient vector of the primary sound pressure in the room (with mode shape vector, it gives rise to the sound pressure in the room $p_2^{(p)} = [\Phi_{N_2}][P_{N_2}^{(p)}]$) and $[G_{N_2}^{(c)}(r_2^{(c)})]$ is the modal coefficient vector of the point source Green function in the room (which gives rise to the sound pressure in the room $p_2^{(s)} = [\Phi_{N_2}][G_{N_2}^{(c)}(r_2^{(c)})]V_{r_2}^{(c)}$). To satisfy Equation (3) requires similar excitation of all modes from the primary and control sources. Successful attenuation of low-frequency potential energy in a room by placing a point control source near the primary point source is an example where Equation (3) is approximately satisfied. In the control of sound transmission, the elements in $[G_{N_2}^{(c)}(r_2^{(c)})]$ depend on the control source location $(r_2^{(c)})$ while those in $[P_{N_2}^{(p)}]$ are determined by the coupling between the room and plate modes. Use of a single control source is difficult in simultaneously adjusting several modes. As a result Equation (3) can only be satisfied at those frequencies where the primary sound field is dominated by a single room mode and the modal overlap in the room is low. For this case, if the control source is located so as to excite the dominating mode only, Equation (3) will be approximately satisfied and large reduction of potential energy at this frequency is possible.

For case (2), the condition for sufficient global noise reduction in the room is

$$[Z_A^{(2)}][V_{M_2}^{(p)}] \propto [Z_A^{(2)}][G_{M_2}^{(c)}(\sigma_2^{(c)})] \qquad (4)$$

where $[V_{M_2}^{(p)}]$ is the modal coefficient vector of the vibration velocity of panel 2 due to the incident sound and contributes to the primary sound pressure through the modal acoustic transfer impedance matrix $[Z_A^{(2)}]$ from panel 2 to the room ($[P_{N_2}^{(p)}] = -[Z_A^{(2)}][V_{M_2}^{(p)}]$). $[G_{M_2}^{(c)}(\sigma_2^{(c)})]$ is the modal coefficient vector of the point force Green's function of panel 2. Together with the pressure generated by the control point force at $\sigma_2^{(c)}$ of the panel, they give rise to the modal coefficient vector of the secondary vibration in panel 2 ($[V_{M_2}^{(c)}] = [G_{M_2}^{(c)}(\sigma_2^{(c)})]P_{\sigma_2}^{(c)}$). The condition in Equation (4) can be satisfied by two physical mechanisms. In the first place, the control arrangement may let $[G_{M_2}^{(c)}(\sigma_2^{(c)}] \propto [V_{M_2}^{(p)}]$ and the optimal control force is used to simply suppress the total modal amplitudes in plate 2. As a result, the source of sound radiation into the room is significantly reduced, and therefore the resultant total sound pressure. In the second place, the condition can be satisfied by adjusting the total velocity vector of panel 2 ($[V_{M_2}] = [V_{M_2}^{(c)}] + [V_{M_2}^{(p)}]$) to be orthogonal with the row vectors in $[Z_A^{(2)}]$ (or with those row vectors corresponding to the dominating pressure components in the total sound pressure vector $[P_{N_2}]$). For this case, the total plate velocity is not necessarily attenuated. The magnitude and phase of each mode in panel 2 are rearranged such that the superimposed contribution of all the elements in $[V_{M_2}]$ to the sound pressure components in the room is significantly reduced.

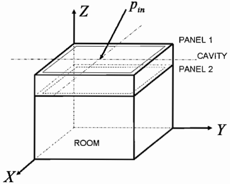


Figure 6 Typical acoustical system for active control of sound transmission into a room through a double panel partition.

For case (3), the condition for sufficient global noise reduction in the room is

$$[Z_A^{(2)}][Y_P^{(2)}][Z_A^{(1)}][Y_P^{(1)}][P_{M_1}^{(in)}] \propto [Z_A^{(2)}][Y_P^{(2)}][G_{N_1}^{(p)}(r_1^{(c)}] \qquad (5)$$

where $[Y_P^{(2)}]$ and $[Y_P^{(1)}]$ are the modal transfer mobility matrices respectively from the sound field in the cavity to the vibration velocity in panel 2 and from the external sound field to the velocity in panel 1, $[Z_A^{(1)}]$ is the modal transfer impedance matrix from panel 1 to the cavity, and $[G_{N_1}^{(p)}(r_1^{(c)}]$ is the modal coefficient vector of the point source Green function in the cavity.

Equation (5) indicates three possible mechanisms involved in this control arrangement, one of which is suppression of the cavity modal response ($[Z_A^{(1)}][Y_P^{(1)}][P_{M_1}^{(in)}] \propto [G_{N_1}^{(p)}(r_1^{(c)}]$). The other two mechanisms are (1) the direct rearrangement of the cavity sound pressure components to minimize the amplitude of the dominating radiating modes in panel 2; (2) the indirect rearrangement of the modal components in panel 2 by adjusting the cavity pressure components; such that the superimposed sound radiation into the room is reduced. These two modal rearrangement mechanisms may be accompanied by an increase of sound pressure in the cavity and vibration in panel 2.

1.4 Discussion

One of the important roles acousticians have been playing in the development of active noise control systems is to use the acoustical features of the systems to improve the control performance. Many examples in relation to this can be found in the books by Nelson and Elliott [7], and Hansen and Snyder [8]. A large number of papers on how the physical understanding is used to improve the effectiveness of ANC systems can also be found in the several Proceedings of Active Noise and Vibration Conference since 1991. In particular, the reader is referred to the work by Fuller [9] and Clark [10] on how the physical understanding of sound and structural interaction can assist the design of ANC systems for attenuating structural sound radiation; by Elliott [11] on how the understanding of the open loop features of local control systems is related to the effective decentralized control of the dynamic response in distributed systems, and by Nelson [12] on how the basic physics of the sound field actually dictate the design of virtual acoustic imaging systems.

2. DEVELOPMENT OF PRACTICAL ANC SYSTEMS

Apart from contributing to the understanding of acoustical systems in order to improve the design and performance of ANC systems, research at UWA also focused on the development of practical ANC systems. This section briefly presents the results obtained from real applications such as ventilation ducts, heavy mining vehicles and high-speed boats. When a reference signal that is highly coherent with the error signal is available or can be derived, feedforward control can be employed. The examples presented below make use of a multi-channel adaptive feedforward controller with online system identification based on a novel algorithm developed at UWA [13] to reduce tonal noise.

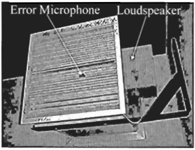
2.1 Active control of fan noise

The noise associated with axial flow fans used in ventilation is typically characterised by tonal frequency components superimposed on broadband random noise. The tonal components are associated with the blade passing frequency (BPF) of the fan and its harmonics depending on the fan rotational speed. The random noise is usually associated with flow noise and turbulence. In most cases, the tonal peaks are significantly higher than the random noise. Apart from being the main contributors to the overall noise level, they are a source of annoyance. Hence, it is important to have adequate noise attenuation of fans especially inside buildings.

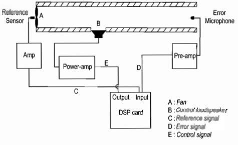
On the ground floor of the civil and mechanical engineering building at the University of Western Australia is a small computer room that houses the mainframe computer network, which consists of several racks of network hubs and computers [14]. These equipments are left on permanently and an exhaust fan is mounted inside the room to discharge the hot air into the corridor. The discharge fan is a backswept, 5-bladed, 6-pole, 300mm diameter axial flow fan delivering an air flow of 0.35m^3/s. Power spectral measurements of the fan noise show that it contains low level broadband noise and high level discrete tonal noise as depicted in Figure 8. The uncontrolled spectrum clearly depicts the tonal components associated with the blade passing frequency of 82 Hz and its first three harmonics around 164 Hz, 246 Hz and 328 Hz for a 5-blade fan rotating at 16.4 revolutions per second. Since the tonal peaks are at least between 15 to 30 dB above the background noise level they are clearly audible and also contribute significantly to the overall noise level of the fan. The noise control solution consisted of a combination of passive and active noise control to attenuate both the broadband and discrete components of the fan noise respectively.

For the passive noise control of the higher frequency broadband noise ($f > 800$Hz), a short square duct of dimensions 0.45m x 0.45m x 0.9m was constructed of 2mm thick galvanised panel and placed over the outlet side of the fan (Figure 7 (a)). The inside walls of the duct are lined with 1.5cm thick wool blanket to provide sound attenuation. The duct provided a sound attenuation of approximately 10dB for broadband frequencies above 1kHz while the level of the BPF and harmonics remained unchanged. However, the A-weighted overall sound pressure level measurements taken at these positions indicated a decrease of about 2dB(A) only. The small reduction can be attributed to the fact that the overall sound pressure level is predominantly determined by the high-level discrete frequency noise of the BPF and harmonics, which are hardly reduced even after the installation of the duct. Hence active control was applied to reduce these tonal components.

A schematic of the control system used for the fan noise is shown in Figure 7(b). In the fan ANC system the error sensor is a miniature electret microphone, the reference sensor is an optical switch tachometer and the control actuator is a 150 mm diameter closed-box moving-coil loudspeaker system. The controller is a digital adaptive controller implemented on a Digital Signal Processor (DSP) hardware platform. It incorporates an online system identification scheme using the additive random noise technique to model the system. This allows any changes in the secondary path transfer function due to aging of components, changes in flow characteristics and temperature to be tracked in real time, thus optimising noise



(a)



(b)

Figure 7. (a) Photograph and (b) schematic of fan ANC setup in a duct.

reduction at all times. The measured rpm of the fan blades is post processed electronically to give rise to sine voltage signals correlated to the blade passing frequency and its harmonics. The advantage of using an infra-red optical switch as a reference sensor is that it produces marginally better control compared with a microphone reference sensor. Moreover, long term stability is guaranteed as the possibility of acoustic feedback between the reference and error sensor is eliminated [14].

The control software is programmed into an EPROM that runs the entire ANC system. If an error occurs at any time an automatic hardware reset is initiated and the ANC system restarted. The electronics used for the fan noise include sensors interface, DSP and power amplifier to drive the control loudspeaker. These are housed inside a standard polycarbonate wall-mounted box.

The noise spectra measured at the error microphone before and after control are shown in Figure 8. The controlled spectrum shows that the discrete frequencies have been reduced to background noise level. Reductions of more than 30 dB at 82 Hz, 25 dB at 164 Hz, 15 dB at 246 Hz and 15 dB at 328 Hz have been achieved. Such results would be practically very difficult, bulky and expensive to obtain by using passive noise control methods alone. Another benefit of ANC apart from its effectiveness to control low frequency noise is its compactness and minor modification required to the existing fan setup.

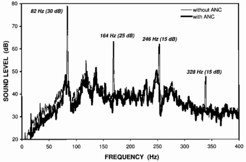


Figure 8 Noise spectrum at error microphone before and after control.

The combined passive/active system produced an overall noise attenuation of about 4 to 5dB(A). Since its implementation in June 2001 the control system has been operating continuously without any problem while maintaining optimal noise reduction. The DSP controller is designed with an automatic restart including secondary path identification and control in case of temporary power failure. Sufficient safety factor has been used in the selection of the dynamic ranges of the control loudspeaker and electronic devices to handle any unexpected high level transient signals in the control loop. Whilst providing a solution to the noise problem inside the building, this system also serves as a teaching tool for students and a demonstration of active noise control to industry.

2.2 Active control of noise in a high-speed boat

High-speed boats are often made of lightweight aluminium and as a result low frequency noise and vibration can be transmitted to occupants. Active noise control has been applied to reduce the low frequency tonal noise inside a 3.4 m wide by 2.5 m deep by 2 m high wheelhouse in an 18 m long boat fitted with two diesel engines. Although the ANC system used in this case was similar to that for the fan noise it however consisted of 2 accelerometers as reference sensors located on the aft wall of the wheelhouse, 4 error microphones positioned near three seats to achieve local control and 4 loudspeakers fitted appropriately [15]. This multi-channel ANC system is necessary to achieve noise reduction around passenger heads inside the large enclosure.

Figure 9 shows the A-weighted noise spectrum when the two nearly synchronized engines are running at 2135 rpm during cruising condition. The tonal components around 48, 71, 94 and 141 Hz are due to the generator, impellers blade passing frequencies and their harmonics. In such an application considerable effort was required to ensure that the coherence between the reference signals and the error signals was high over the frequency range of interest. This was done by carefully selecting the locations of the reference accelerometers such that they give a good representation of the cause of the noise inside the wheelhouse. Figure 9 also shows the noise spectrum after ANC and indicates that good noise reduction can be obtained for the tonal components. Reductions of 6 dB at 48 Hz, 11 dB at 71 Hz, 7 dB at 94 Hz and 12 dB at 141 Hz have been measured. As a result the overall noise reduction that could be achieved was between 2 to 3 dB(A). This type of result would have been difficult and costly to achieve by using passive means alone without a considerable increase in weight. The current ANC system can be extended to deal with higher frequency components (above 200 Hz), but to maintain an adequate size of the 'quiet zones', it will be necessary to increase the number of error microphones and control loudspeakers. Moreover, the DSP employed will also have to be more powerful.

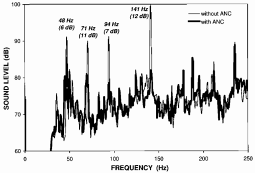


Figure 9 A-weighted noise spectrum at skipper seat before and after ANC.

2.3 Active control of noise in a truck cabin

Low frequency noise is recognised as a main cause of annoyance and fatigue among drivers of heavy mining vehicles [16]. Active noise control can be used to reduce the low frequency tonal noise inside such vehicles and increase occupant comfort while reducing the overall noise level when passive control is ineffective. A multi-channel system comprising of 2 error sensors, 2 control loudspeakers and 3 reference sensors as shown in Figure 10 was used to achieve local control around the driver head in a 105 tons mining truck. The three reference sensors are necessary to ensure a high coherence between the reference signals and error microphones under normal operating conditions.

When the truck is stationary or cruising at constant speed the noise spectrum is dominated by tonal components associated with the engine and onboard auxiliary equipment as depicted in Figure 11. The ANC result obtained for a stationary vehicle is shown in Figure 11(a). Reductions of 24 dB at 33 Hz, 15 dB at 96 Hz, 25 dB at 191 Hz and 12 dB at 228 Hz have been measured. For this case the corresponding reduction in overall noise level is about 4 dB(A) which is clearly significant. When the truck is cruising at around 50 km/h the result shown in Figure11(b) is obtained. Again tonal components are present and ANC is effective at reducing these. Reductions of 13 dB at 28 Hz, 18 dB at 58 Hz, 10 dB at 101 Hz and 15 dB at 168 Hz have been measured. These significant tonal noise attenuations combined with some passive treatment under the cabin floor [15] resulted to an overall noise reduction of between 3 to 4 dB(A). The results also show that when the truck is moving tonal components at higher frequencies are present and contribute more to the overall dB(A) level than the low frequency tonal components. It is possible to improve the frequency range of noise reduction around the driver head with additional error and control channels and more processing power.

Both ANC systems described in subsections 2.2 and 2.3 have been prototyped and tested through field trials with acceptable noise reduction. The UWA team and its industry collaborators are seeking opportunities for the commercial realization of the prototypes.

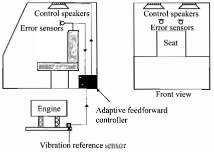
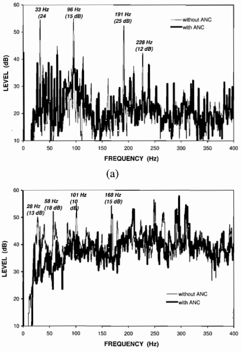


Figure 10 Schematic of feedforward control strategy inside truck cabin



(a)

(b)

Figure 11 A-weighted level when (a) stationary, (b) cruising using 3 reference signals

2.4 Discussion

The world has witnessed nearly two decades of effort in laboratory research and industrial application of ANC technology. We are now faced with frequent questions like "when and where can we see ANC products as practical installation?" Indeed successful examples of such installation may play the role of flagship in encouraging the arrival of the "fourth wave" of wide utilization of ANC in industry. Examples of such applications are limited mainly to the control of sound propagation in air handling ducts, gas turbine exhausts or diesel engine exhausts, the control of sound pressure in a small cavity inside of active ear muffs, and the reduction of tonal noise in propeller driven aircraft using active engine mounts and vibration actuators mounted on the fuselage rings [17]. Some mass produced products for reducing booming noise by Nissan and low frequency road noise by Honda were also reported [18]. It is expected that such practical application of ANC will continue and bring an increasing number of real life and successful installations to satisfy the ever increasing demand of modern noise control.

3. CONCLUSIONS

In this paper a brief review of the contribution of research at UWA in the field of active noise control has been presented. In particular, there has been a focus on understanding the fundamental properties of acoustical systems in order to improve the ANC design and on the development of practical ANC systems to reduce low frequency noise problems in industry. In the first instance three examples show the importance of a thorough understanding of acoustical systems in determining the performance and limitations of ANC methods. In the second instance practical ANC has been demonstrated by three real application examples. The results show that ANC is effective in reducing low frequency noise where passive control becomes impractical. The success of these systems relies on a good understanding of the acoustics, proper design of hardware and software as well as a good knowledge of control methods.

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