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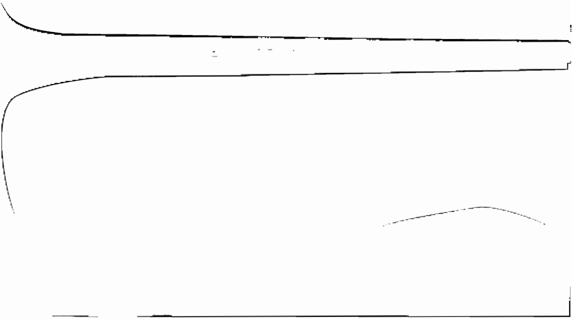

SPECIAL ISSUE: ACTIVE NOISE CONTROL

- *The 'cube of difficulty' in active noise control*
- *Acoustics and practical applications*
- *Sensors and actuators*
- *Active noise control and vibrating structures*

Australian Acoustical Society

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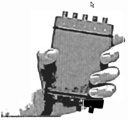

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PAPERS

Cover: Not waving but flexing. Heidi Hereth

From the President

I cannot believe that two years have almost passed since I became President and this will be my sixth and final President message. Irrespective of how many good intentions you may have about what you hope to bring into a role or to achieve during your time in that role, other aspects of your life plus factors beyond your control always seem to hamper your best intents.

I personally do not feel that I have achieved a great deal as there haven't been any significant changes in the past few years. However, this is possibly just a reflection on what is a generally very well run organisation where there does not appear to be significant feedback from our members about the need to change. Much of the time at a National Council level which is represented by two Councillors from each State is spent dealing with more administration-type issues and allocating relatively small, but not insignificant funds for acoustic purposes. It is really at the State level where a great deal more action happens.

As a result of the member survey, the National Council is well aware that this primarily relates to the quality of Acoustics Australia and also the number and quality of technical seminars held. Of course, the success of both these aspects is primarily the willingness of members to contribute both in offering to present or write papers, but also other members to support their peers and attend where ever possible.

I would like to pass on my thanks particularly to Marion Burgess and Joe Wolfe for the seamless transition of Acoustics Australia to a new editorial team. Joe has reminded me that the technical forum section is available for members to provide very short pieces, which do not usually undergo formal peer review.

On another note, Joe also commented that for at least this and the previous special topic editions of the journal, interested customers may order copies of Acoustics Australia via the website. Until I read the draft of this special edition, I used to think active noise control was our clients actually doing something about noise control we recommended rather than just asking consultants to write reports about what the options are!!!

Registration for the Conference "Noise of Progress" is now available online www.conference.co.nz/acoustics06 and I believe the early bird registration will almost be over by the time this publication reaches you. I have been fortunate enough to attend the last five AAS annual conferences and they seem to just keep getting better and better. I think the concept of a joint conference in New Zealand is an excellent idea and an opportunity to share ideas with our closest neighbours. If this conference is a success (which relies on your support) I am sure we will be keen to hold other joint conferences in the future.

Just make sure your passport isn't out of date. (A valuable piece of advice – I recently arrived in England and realised my British passport had expired 4 days earlier.)

Neil Gross

From the Guest Editor

Editorial, special issue on active noise control

Active noise control (ANC) has been suggested as a viable alternative for the control of low frequency noise for many years. When it was first suggested, the electronic technology, control theory and signal processing knowledge needed to implement practical systems were all inadequate. Over the past 20 or so years the technology required to realise active noise control systems has progressed to a point where the implementation of a practical system is feasible and there are many current examples of such implementations. Nevertheless, the industrial use of the technology is still not widespread. It seems that there is still some way to go to transform the technology and developments from the laboratory to industrial environments where there are many things such as transducer failures, power failures and transient noise events that can adversely affect the operation and performance of an ANC system.

In this issue of Acoustics Australia, a snapshot is provided of some of the activities being undertaken in the active noise control area in three Australian Universities. This work is certainly not representative of the entire range of activities being undertaken in active noise control in those three universities, let alone the whole of Australia. However, it is hoped that the papers will provide some insight regarding the issues associated with the practical implementation of active noise control systems.

The first paper by Mike Kidner from the University of Adelaide uses a novel "cube of difficulty" to classify active noise control problems and he shows that those applications in the easy corner of the cube have found their way into industry. However, there are not many actual problems that fit into that classification. Research is still being conducted in the more difficult problems, but there are many problems for which solutions exist but for which industrial implementations have not been realised due to complexity and unreliability. There is huge scope for work to be undertaken to simplify the implementation of active noise control systems for many applications, but unfortunately this work is not being done due to lack of available funds for this type of work: it is too applied to attract research funding and too risky to attract industry funding.

The second paper by Jie Pan and Roshun Paurobally outlines some of the applications that have been implemented on an experimental basis by the University of Western Australia. Their work shows that continuous operation of systems in an industrial setting can be realised and that there is a bright future for commercial active noise control systems for some particular noise problems.

The third paper by Colin Hansen discusses transducers that can be used in practical active noise control applications. Transducers are often the weak link in an industrial application as special precautions have to be taken to avoid contamination from liquids, dust and various chemicals. The transducers also need to be very robust and capable of continuous operation over many years.

The fourth paper by Nicole Kessissoglou discusses the control of the vibration of and sound radiation from basic structures such as beams, plates and shells. Means of optimising the controller performance are discussed for each application and some insights into the physical control mechanisms are provided.

I would like to thank the authors for their excellent contributions and I am hopeful that a future issue of Acoustics Australia will be able to include papers on other aspects of active noise control that were omitted here.

Colin H Hansen

ACTIVE NOISE CONTROL: A REVIEW IN THE CONTEXT OF THE 'CUBE OF DIFFICULTY'.

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Over the past twenty years active control of noise has developed into a mature research field and into a product for some technical companies. This paper reviews the current state of the art in both the research and development fields using the context of a *cube of difficulty*. The cube illustrates how the three physical quantities: frequency bandwidth, spatial extent and signal coherence, contribute to the difficulty of achieving control performance. The literature is reviewed and placed within the cube to reveal patterns in research and areas of further work.

INTRODUCTION

Active control of noise is no longer an esoteric research topic, it has been implemented many times in the *real world* [1] and has become one of the tools available to the noise control engineer. However, its limitations and subtleties are still misunderstood by many. The simple explanation of creating an *anti-sound* field has left many people disillusioned when faced with the difficulties of implementation. The development of a robust and simple active control system suitable for wide application has illuded many companies [2].

The successful implementation of active noise control is effected by three physical concepts; spatial extent, frequency bandwidth and correlation or coherence of signals.

Spatial Extent: This describes the complexity of the control problem in terms of spatial variables. This could be the physical size of the required zone of quiet or the dimensionality of the control problem. Global control of plane waves in a duct, [3] could be considered to have a large spatial extent as it is possible to cancel all the sound downstream of the error sensor by reflecting all the sound energy back along the duct. However the problem is only 1-D and so can be considered of fairly low spatial complexity compared to the case above the cut-on frequency of the duct [4].

Frequency Bandwidth: Control of sound over a large bandwidth is more complex than control of a single tone for a number of reasons. The time interval over which control actions need to be calculated is smaller at high frequencies, the response of the plant is more complex and the spatial variation is higher. Due to the linear relationship between frequency and wavelength, bandwidth and spatial extent are intrinsically linked [5].

Coherence: To control a sound field the controller must have inputs that are coherent with the primary field. This correlation can be limited for several reasons, such as the sound field has low spatial correlation, as in a diffuse field at a single frequency[6]. Note, that in this case the field is correlated temporally, as only a single frequency is present. The input and error sensors may be incoherent due to the sound field being unrelated in time. This is the case for random disturbances when the delay between input and error signals is longer than the correlation delay. Some control problems are difficult because both types of incoherence are present, as is the case with turbulent boundary layer noise [7].

These three parameters are related in figure 1 by the *cube of difficulty*[8]. In the next section of the paper, each face and vertex of the cube of difficulty is discussed in turn with respect to the literature. Some approaches to current problems in active noise control are described in section three. Control algorithms are discussed in the penultimate section.

1.1 THE CUBE OF DIFFICULTY

The cube of difficulty can be used to visualise the physical limitations of active control systems, but it is only a metaphor for the relationship between the parameters. The axis of the cube are spatial extent, frequency bandwidth and incoherence. At one corner is *control of a single frequency at one point in space* at the other apex is *global control of broadband spatially incoherent noise*. These corners can also be referred to as the *easy* and *very difficult* types of control problem.

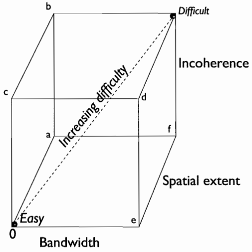

Figure 1: The cube of difficulty for active noise control.

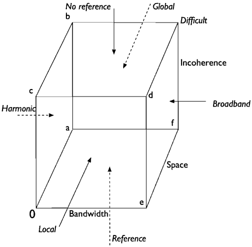

Figure 2: Problems in active control

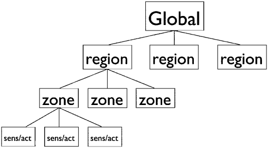

Figure 3: Possible active control system hierarchy based on division of space.

The '*0–c–d–e*' face: Local Control: The front face (*0–c–d–e*) represents local control. The most successful commercial application of active noise control, (noise canceling headsets), lie on this face.

Recent work by Jones [9] has shown that local zones of quiet can be created at a seat location, and that the neighbouring seat can also maintain its own zone of quiet or desired sound field.

The '*a–b–difficult–f*' face: Global Control: Global control is often sort after, and rarely achieved as it requires a significantly more complex control approach, especially at higher frequencies. The development of multi-channel control systems and associated high performance DSP chips has helped solve some of the control problems that lie close to this face. To appreciate the complexity of the problem it should be noted that multichannel control of random sound is not discussed until the final chapter of Nelson and Elliotts *Active Control of Sound*[5].

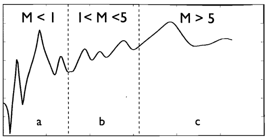

Figure 4: Possible active control system based on division of frequency.

The '*a–b–c–easy*' face: Harmonic Control: Feedforward control of single frequency noise was the focus of the Leugs patent[10], which is considered to be the first active noise control system. Harmonic control is also of greatest application to industry as the majority of noise sources are rotating machines.

The '*Difficult–d–e–f*' face: Broadband Control: These areas of active noise control have been investigated [11, 12] and have seen a huge increase in the number of successful applications due to the increase in computing power.

Noise sources that fall into this category are jet/flow noise, tyre/road noise and impact noise. All of which are difficult targets for active control systems. Reductions of up to 6dB in automotive interiors have been achieved by Park *et al.* [13].

The spatial coherence of broadband fields has been investigated by Chun et al [14] and Rafaely[15].

The '*0–a–f–e*' face: Control of coherent signals: Problems on this face are applicable to feedforward control. The sound field is coherent and so a reference is easily found. Feedforward control is more robust and can have a wider control band than feedback control, however digital hardware is usually required. Control of fan or propeller noise or any sound field where a coherent reference signal such as a tachometer output is available falls onto this face.

The '*b–c–d–Difficult*' face: Control of incoherent signals: If the sound field is incoherent over space/time, feedback control can be used. The error signal is used as the input to the controller, if the delays in the controller are small enough, the control output will be coherent with, and hence able to control the sound field. Feedback control of a large area is complex because multiple channels may be required. It is better suited to local control problems such as control of sound in headsets.

The use of multiple reference signals can help improve the control achieved. Often the sound field is due to many different sources; papers by Tu and Fuller discuss this issue [16, 17]. The problem of finding appropriate reference signals still exists. The use of multiple reference signals can cause instabilities if they are correlated with each other. This might be the case if they are structural or acoustic measurements. The method proposed by Tu pre-processes the reference signals to form a set of orthogonal signals which are then input to the controller.

The 'easy' corner: The lower left corner of the cube has been completed. Examples of local or 1-D control of single frequency noise can now be found in industrial installations and consumer products. It should be noted that on the left face of the cube, *i. e.* the single frequency face, the coherence axis refers to spatial coherence. Any single frequency is temporally coherent, however in a true diffuse field a harmonic signal will be incoherent over space.

Changes in bandwidth: The line from the origin to apex (a), represents single frequency control problems of varying spatial extent. Research by Rafaely [18] has shown that by specifically targeting a zone of quiet significant reductions can be achieved within that zone, as the extent of the zone increases more control effort and secondary sources are required.

Moving from the origin to point *c* is equivalent to increasingly the diffusity of the sound field. At point *c* the sound field only has a single frequency component and is the result of summation of many waves with random phases, as in an ideal diffuse field. The spatial correlation of diffuse fields is discussed by Rafaely [15].

The control bandwidth widens as noise control problems are located further along the *0–e* vertex. Increases in the control bandwidth have been obtained mostly by improved DSP performance. However some work on multi rate filtering has also yielded significant results [19].

Changes in spatial extent: The move from local to global control requires an increase in both the number of secondary sources and error sensors. Some success has been achieved by using virtual sensors [20]. However it is preferable to use good design, such as control of choke points [21] to achieve large areas of noise reduction.

Changes in signal correlation: At point *e* the sound field is coherent and so with a suitable reference and adequate actuators and sensors global control can be achieved. At point *d* a coherent reference no longer exists and so feedback must be used. Travel along this vertex is equivalent to the progression from control of propeller tones[22] to jet noise in aircraft interiors[23].

The *a–b* vertex represents the progression from global control of a harmonic/coherent signal to global control of a harmonic/incoherent signal. Point *a*, represents the global control of the sound field in an anechoic chamber, point *b* represents the global control of the sound field in a reverberation chamber. As most control systems act on a limited volume, we find these problems offset along the *b–c* vertex. An example of control of the sound field in a reverberant enclosure is given by [24]. It is shown that reduction of the global potential energy of the enclosure and maximum attenuation or extent of zones of quiet are not the same solution. This is inherent in active control problems.

Point *d* to the *difficult* corner: Control of spatially and temporally incoherent noise

The control of turbulent boundary layer noise was the focus of a great deal of work throughout the 1990s [7]. The turbulent boundary layer problem occupies space near the vertex (*d–Difficult*). The excitation is incoherent in time and space, and the required zone of quiet has to extend over many seats in the aircraft. This is a very difficult problem and it is accepted that control performance will decrease with coherence. Global, broadband control requires a complex control system and the performance is limited by many factors.

2 CURRENT PROBLEMS

2.1 Divide and conquer

An increasingly common approach is the subdivision of the controller into many smaller uncoupled units. Each unit controls a smaller, simpler problem, preferably one that is on the (*0–c–d–e*) face. Gardonio has applied this to vibration control problems [25, 26] and Mathur *et al* [27] have applied it to noise control problems.

2.1.1 Spatial Separation

Figure 3 shows how a control system could be split based on regions of control. The lowest level in the hierarchy is divided based on transducer coupling to the acoustic space.

This has been investigated by researchers such as Rob Clark[28] who investigated the effect of transducer location on modal observability and the use of spatially distributed sensors to reduce spillover between modes.

The *zone* level is based on the size of the zone of quiet and this level is split based on the interaction between zones of quiet[9].

The *region* level is divided as a function of gross system attributes, such as dividing the control problem within an aircraft into regions for the rear, wing/engine and front sections of the cabin.

The top level would be a scheduling and management layer. This layer could set weightings for individual regions or zones to maximise reduction in certain areas or make decisions based on hardware failure.

2.1.2 Frequency Separation

Figure 4 shows a division of the control problem based on frequency. Papenfuss *et al.* [29] demonstrated this approach for structural problems. The frequency range is split based on the modal overlap. In each region a different control scheme is applied. In the low modal overlap region optimal feedback controllers (H_2, H_∞) were used. In the mid frequency ∞ range where the modal overlap is between 1 and 5 an FX-LMS feedforward controller was used. In the high frequency range where only spatially limited control is possible a simple analog feedback loop was implemented.

The use of modal sensing or weighted transducer arrays is a form of selective sensing of frequency or spatial patterns [30][31] and has been applied to feedback control of structurally radiated sound.

2.2 Towards the upper right hand corner

Active control of sound now has two foci, implementation of established algorithms in *real world* applications and the continued pursuit of solutions to problems from the upper right hand corner of the *cube of difficulty*. As explained above reaching this corner is challenging due to the nature of the physical world. The divide and conquer approach may offer some solutions but these are likely to be highly theoretical and impractical.

3 IMPLEMENTATION

3.1 FX-LMS Feed forward

After its first proposal by Morgan[32] and independently by Widrow[33] the filtered-x LMS algorithm has been the mainstay for many active noise control systems. The reasons for this are: the algorithms ability to track changes on the same timescale as the delay in the error path, robustness to errors in the estimate of the error path, simplicity of implementation.

Reducing the computational load of feedforward algorithms has been the focus of much research, for example Bouchard *et al.* [34].

3.2 Feedback

Tseng *et al.* [35] have shown that by using optimal control techniques such as H_2 or H_∞ better control over the zones of quiet can be obtained. The H_2 approach will attempt to minimise the energy in the system and so is likely to be more robust than the minimisation of pressure at a number of discrete points. The energy density methods [36, 37] show similar results.

3.3 Neural Networks, Fuzzy Logic & Non-linear controllers

In an attempt to broaden the number of problems to which active control can be applied, linear controllers have been substituted by neural networks and fuzzy logic controllers. The neural networks have been used for system identification and the creation of the control outputs[38]. [39].

4 SPILLOVER

The technologies developed for active control of sound have been applied to several problems that are not noise control problems at first glance. Virtual acoustics [40] [41] is one example. The virtual acoustics problem can be written in terms of the minimisation of an error function by combination of primary and secondary fields. Active noise control researchers are now applying their knowledge to this field [42][43][44].

5 LIMITS OF THE CUBE ANALOGY

The cube of difficulty is only one way of describing the way physical properties of systems effect active noise control. As such it cannot encompass the full range of phenomena that influence ANVC design. Non-linear and time varying plants are an example of factors that are not well described by the cube analogy.

6 CONCLUSION

The upper right hand corner of the cube is unobtainable, however the division of the complex problems near this corner into many simple problems is yielding some advance. It should be borne in mind that much of the cube has been successfully tackled, (at least in the laboratory). However the full transition of active noise control technology to the market place is still yet to happen. Development is the key to successfully crossing from laboratories to general use. Only close collaboration between research groups in industry and academia will achieve this. More important, is the identification of a *killer application* in which active noise control can fulfill its potential.

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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

$$\frac{P_c}{P_u} = \frac{1}{1 - H_c H_m H_l H_a}, \qquad (1)$$

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

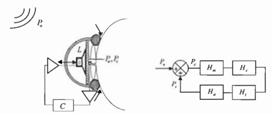

Figure 1 (a) Schematic and (b) block diagram of a feedback noise control ear defender.

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.

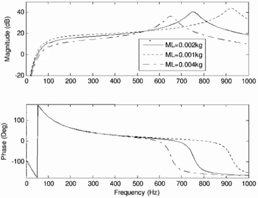

Figure 2 Magnitude and phase plot of the uncompensated open-loop frequency response function H_o when the mass of the speaker diaphragm is used as a varying parameter.

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 loudspeaker and error microphone was a crucial parameter in producing a global noise reduction in the office. The control was implemented in a 4×2.9×3m^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).

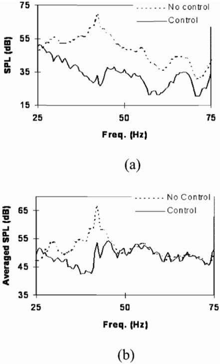

Figure 3 Uncontrolled (dotted) and controlled (solid) sound pressure level (a) at the error microphone location and (b) by spatial average when $r = 13cm$.

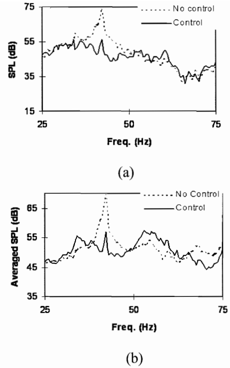

Figure 4 Uncontrolled (dotted) and controlled (solid) sound pressure level (a) at the error microphone location and (b) by spatial average when $r = 170cm$.

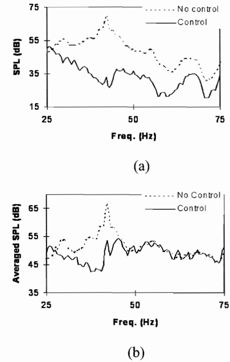

Figure 5 Uncontrolled (dotted) and controlled (solid) sound pressure level (a) at the error microphone location and (b) by spatial average when $r = 28cm$

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

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

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.

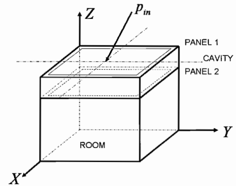

Figure 6 Typical acoustical system for active control of sound transmission into a room through a double panel partition.

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 > 800Hz), 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

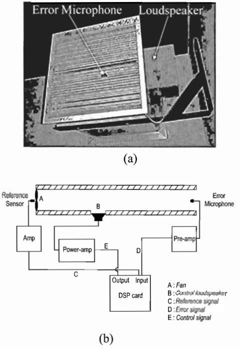

Figure 7. (a) Photograph and (b) schematic of fan ANC setup in a duct.

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.

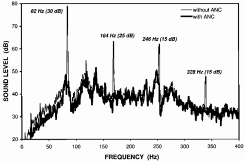

Figure 8 Noise spectrum at error microphone before and after control.

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.

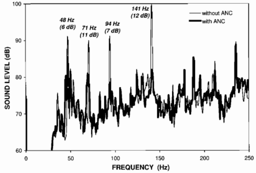

Figure 9 A-weighted noise spectrum at skipper seat before and after ANC.

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.

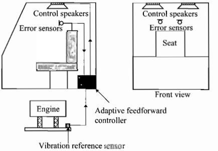

Figure 10 Schematic of feedforward control strategy inside truck cabin

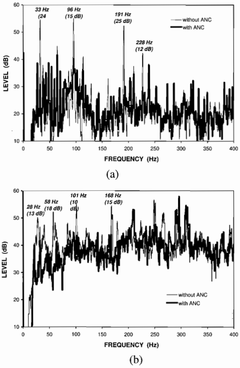

Figure 11 A-weighted level when (a) stationary, (b) cruising using 3 reference signals

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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SENSORS AND ACTUATORS FOR ACTIVE NOISE CONTROL SYSTEMS

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Successful implementation of an active noise and vibration control system requires an effective control system and a good understanding of the physics of the problem to be controlled. However, there can be no successful implementation without appropriate transducers for transforming acoustic signals into voltages for the electronic control system and for transforming voltages output from the control system into sound. Although a considerable amount of research has been devoted to control algorithms and optimal physical arrangements for sensors and control actuators, little has been written about the requirements for the sensors and actuators themselves. Here a number of practical actuator and sensor implementations are discussed for both fully active and semi active noise control systems.

INTRODUCTION

There are numerous practical applications where active noise control has the potential to provide significant benefits. Algorithms [1-11] are available for many types of noise problem, ranging from multi-tonal to broadband and a considerable amount of work has been done by various researchers on extending the controllable bandwidth from 1.5 octaves to three or more [12]. However, in many of these cases, the implementation of a successful active noise control system is hindered by the lack of availability of robust, generic and low cost hardware. This hardware includes sensors and actuators as well as the control system itself. Requirements for a user friendly control system have been discussed elsewhere [13] so here the discussion will be restricted to control approaches and associated sensors and actuators. The discussion will also include vibration actuators and sensors as these are sometimes used to control sound radiation.

There are two main categories of active noise control: fully active and semi-active. Although both fully active and semi-active systems may use the same kind of sensors, they invariably differ in the kinds of actuator or control source that they use. Also the sensors and actuators that are most suitable for a particular application depend on the control strategy that is chosen. For example if the noise to be controlled is generated by a vibrating surface, it may be better to use vibration actuators to control the vibration of the surface rather than use loudspeakers adjacent to the surface.

A fully active system uses actuators to directly generate a cancelling or suppressing noise or vibration signal. An example of a feedforward fully active system to control noise propagating down a duct would consist of a microphone (referred to as a reference sensor) in the duct to measure the noise that is to be controlled, a loudspeaker (referred to as the control source), mounted in the wall of a duct downstream of the microphone, to introduce the cancelling noise and a second microphone (referred to as the error sensor), mounted in the wall of the duct, to measure the residual noise sufficiently far downstream from the loudspeaker, with the control system using the signals from the two microphones to generate the required signal to minimise the noise at the second (error) microphone. The required distance between the control source and error sensor is controlled by the delays in the loudspeaker response and the control system itself as well as possible near field effects in the vicinity of the loudspeaker. At room temperature, most systems will cope well with a distance of between 1 and 2 m. A feedback system would not require the reference microphone, and in this case the error microphone would have to be moved as closely as possible to the loudspeaker control source to minimise instability problems. Feedback and feedforward control systems are discussed in more detail elsewhere [14].

A semi-active system uses actuators to modify the dynamics of a system or to tune a passive noise-suppressing device so that the system produces less noise. Using our duct example, a Helmholtz resonator may be a good choice for suppressing tonal noise. However, if the frequency of the tonal noise varies (for example, it may be associated with a variable speed engine and the duct may be an exhaust pipe), the Helmholtz resonator would need to be adjusted to keep in synchronisation with the varying engine speed. The control system to automatically keep the Helmholtz resonator optimised as the engine speed changes is referred to as a semi-active system. Its input would be a signal that is proportional to the acoustic power propagating down the duct and its output would drive a motor that adjusted either the resonator volume or neck length to minimise the input signal.

1. ACTUATORS FOR FULLY ACTIVE SYSTEMS

Although many people have tried, it is difficult to beat loudspeakers as an acoustic control source in active noise control systems. Current development is aimed at maximising the life and maximising the physical protection of these devices in dirty industrial environments. In the past, loudspeakers have been manufactured with stainless steel and plastic diaphragms but these have been very expensive options and not as efficient as paper cone speakers.

The ANVC group at The University of Adelaide has spent a considerable effort experimenting with a number of methods to prevent the build up of contaminants on the speaker cone in instances where they are mounted on industrial exhaust stacks ranging from a cement plant [15] to a powdered milk processing plant. The final design, which is effective for stack temperatures of up to 90°C, is illustrated below.

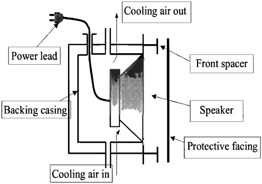

Figure 1. Schematic of the speaker unit

The speaker coil must be cooled either by compressed air or by wrapping a copper pipe around the coil and passing cooling water through it. It is also important that if compressed air is used, the air pressure is kept low to avoid damage to the loudspeaker. Note that there is a pressure equalisation passage between the rear and front of the speaker cone. This is essential as it prevents temperature differentials between the two sides of the cone from jamming the cone against one of its "stops". If the temperature in the exhaust stack is higher than 90°C, it may be necessary to direct some of the air flow over the front face of the loudspeaker by drilling holes in appropriate places in the spacer and outer casing. The above speaker enclosure design was arrived at after trying numerous protective coatings (including epoxy paint as well as automotive gasket spray covered with silver paint spray) on the speaker cone itself. Although the protective coatings withstood the normal operational environment in the milk powder processing exhaust stack, they were unable to withstand the daily aggressive steam cleaning with a highly alkaline product. To overcome this problem, a protective membrane as shown in the figure was used. First a 1.5 mm thick, rubber-like material known as "vyon" and used extensively in the dairy industry, was tried. This reduced the loudspeaker output by 20dB at the 180 Hz frequency at which control was needed and this meant that the maximum available amplitude of the controlling sound in the duct was too small. Although the transmission loss of the vyon was small, its high damping properties loaded the loudspeaker and suppressed its capacity to generate noise. Second a 0.1 mm thick printed circuit board material was tried and this resulted in a 10 dB reduction of speaker output capability, which was still too much. Finally a 0.1 mm thick sheet of mylar was used with a resulting 3 dB reduction in the speaker output, which was acceptable. A picture of the mylar installed in a spray dryer exhaust stack is shown below.

Figure 2. A speaker unit, mounted in the exhaust stack (view from inside the exhaust stack). Note the spout at the bottom to drain any residual liquid after cleaning.

Another type of actuator that is commonly used to control the vibration of structures that are radiating sound is the piezo electric crystal. When excited by an electrical signal, they expand and contract proportionally to the voltage across the two largest faces, thus causing a cyclic strain on the structure to which they are attached. These crystals commonly range in thickness from 0.25 mm to 6 mm and are bonded to the structure whose vibration is to be controlled. The maximum excitation voltages range from 100 volts for the thin crystals to 2000 volts for the 6 mm thick crystals. Clearly the latter voltage is too dangerous for most applications.

Perhaps the most useful and versatile actuator for controlling structurally radiated sound is the inertial actuator. This type of actuator essentially consists of a coil surrounding a permanent magnet. When the coil is energised with an alternating current, the permanent magnet moves back and forth. The permanent magnet is held in place in a structure, the stiffness of which determines the resonance frequency of the shaker. The difference between an inertial actuator and an electrodynamic shaker is that in the former case, the heavy permanent magnet moves while the coils remains stationary, while in the latter case the opposite is true. In practice, this means that an inertial shaker can be directly attached to the structure it needs to excite and in contrast to the electrodynamic shaker, it needs no other support. If tonal noise is to be controlled, maximum shaker output is obtained if the excitation frequency is close to the resonance frequency of the inertial shaker, which is controlled by the mass of the permanent magnet and the stiffness of its suspension.. Thus it is sometimes desirable to be able to easily adjust the resonance frequency of the shaker by adjusting the suspension stiffness. The ANVC group at the

University of Adelaide is currently working on a device that does just that and is able to be controlled by an adaptive filter in such a way that the suspension stiffness is automatically and continuously adjustable. In practice, the shaker resonance frequency must be slightly different to the frequency of sound radiation being controlled or the active noise control system may suffer from stability problems. An inertial actuator controlling sound radiation from an irregular enclosure shell is illustrated in Figure 3.

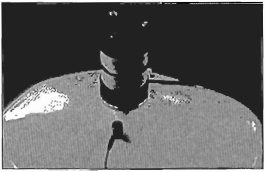

Figure 3. Inertial actuator mounted on an irregular enclosure.

2. HARMONIC DISTORTION

Harmonic distortion is the process whereby a transducer generates mechanical excitation at frequencies that are multiples of the electrical excitation signal frequency. In active noise control systems, this can result in a tonal noise sounding worse after the application of active control, because even though the fundamental tone may be considerably reduced in amplitude, the amplitude of higher order harmonics has been increased and the ability of the ear to hear the higher harmonics better, compounds the problem (as most problems attacked using active noise control are low frequency in nature). One way to minimise harmonic distortion is to drive the transducer at power levels that are less than 10% of the rated maximum and this applies to loudspeakers as well as piezo-electric crystal actuators. Keeping the input power below 10% of the rated maximum has the added advantage of extending transducer life so that in the case of loudspeakers, five years of continuous operation would not be unusual.

3. DISTRIBUTED ACTUATORS

Recent work [16] has reported on investigations using large numbers of actuators on structures to control sound transmission through structures. There is currently speculation supported by anecdotal evidence that if a sufficiently large number of control sources are used in a feedforward active noise control system, it is not too important where they are placed in terms of achieving a global reduction in some cost function. That is, it is not necessary to go through an optimisation process to optimally place the control sources and error sensors. In an attempt to further simplify the control process, Fuller and Carneal [17] suggested using hierarchical bio control in which a small number of signals are sent from an advanced, centralized controller and are then distributed by local simple rules to multiple control actuators.

Some recent results comparing the effectiveness of multiple single channel control systems versus a single multi-channel control system attached to the same sensors and actuators are shown in Figure 4 [18]. Actuators to achieve this result consisted of tiny (10 mm diameter and 20 mm long), low-cost inertial actuators, which provide a spring stiffness using magnetic repulsion forces. In the figure are shown results for vibration control of a cantilevered beam using a foam damper, inertial actuators (unactivated) in the foam, activated inertial actuators driven by single channel controllers and activated inertial actuators driven by a multi-channel controller. It can be seen that for this simple example involving broadband control, the multi-channel controller achieves much better results than multiple single-channel controllers, supporting the view that it may be worth investing in the development of fast multi-channel controllers capable of handling many channels simultaneously using a multi-channel algorithm.

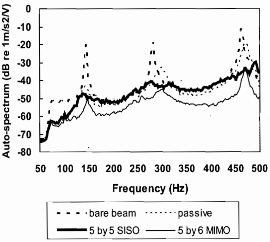

Figure 4. Acceleration response of a bare cantilever beam, a beam with a thick layer of foam containing five embedded, non-driven inertial actuators, with the actuators driven using five single channel controllers and then driven with a 5-out.6-in fully coupled (MIMO) controller.

4. SEMI-ACTIVE ACTUATORS

Given the difficulty in implementing fully active noise control systems in industrial environments, there has been considerable interest in the development of semi-active systems that can optimise the characteristics of a passive noise control system. Clearly these systems are appropriate for controlling single or multi-tone noise, where the wavelength associated with the excitation frequency is likely to vary with time. The wavelength change could be caused by an excitation frequency change as a result of changes in speed of the rotating machine generating the noise or it could be caused by changes in the temperature of the environment through which the sound is travelling.

When a Helmholtz resonator is used to control a frequency varying tonal noise propagating along a duct, the resonance frequency of the resonator can be varied by varying the resonator volume, neck length or neck diameter. In practice, it is easiest to vary the resonator volume and many ideas have been patented for doing this [19-22]. Although a number of schemes have been patented for varying the neck length [23] or cross sectional area [24, 25] or both [26], these are generally more difficult to implement in practice. The volume variations can be achieved by controlling a d.c. motor or stepper motor which drives a lead screw attached to a plunger or similar device. Neck length variations can be achieved by using a sleeve inside the neck and a motor to extend the sleeve into the resonator volume or into the existing neck [23]. As the sleeve extends into the resonator volume, the effective neck length becomes longer and the resonator volume reduces slightly.

Note that for best results, the Helmholtz resonator needs to be mounted at a location in the duct where there is a sound pressure maximum. This implies that the range of speed variation or frequency variation in the tone to be controlled should not be more than about 20% on either side of the centre frequency.

If a semi-active Helmholtz resonator is to be commercialised, it would be desirable to market is as a self contained unit. That means that it may not be desirable to minimise the signal from a particular sensor. It may be necessary to maximise the signal from a sensor in the resonator or even drive the ratio of two signals or the phase between two signals to a pre-determined value that would correspond to minimum sound power transmission down the duct. This latter approach is referred to as model reference control [27].

Tonal noise radiating from the open end of a duct can be controlled by varying the duct length. It is well known that when a source located in a duct, generates tonal sound at the resonance frequency of the duct, the sound radiated from the end of the duct can vary as the duct temperature changes. This is because the changes in temperature cause a change in wavelength of the sound propagating along the duct and this causes the difference between the excitation frequency and the closest duct longitudinal resonance frequency to vary. As the excitation frequency approaches a longitudinal resonance frequency, the level of radiated sound increases. For a typical industrial exhaust stack, the variation in sound power radiated from the end of the stack can be between 10 and 15 dB. Thus it is possible to minimise the tonal sound radiated from the end of a duct by controlling the duct length. This may be achieved by attaching an adjustable sleeve to the outside of the duct and moving it up and down with a stepper motor. This has been demonstrated and proven to be effective on a laboratory scale test rig (see Figure 5) [28].

The motor used to drive the sleeve was driven by a PLC under the control of a very simple algorithm. Initially, the sleeve is driven to the bottom of its travel. It is then advanced slowly upwards to the other extent of its travel and the location of minimum rms acoustic pressure is recorded. In practice, a narrow band filter is used to ensure that only the tonal noise is considered in the rms pressure signal. The sleeve is then driven to the minimum location. Periodically the sleeve is moved up or down to track any changes in the location of minimum acoustic pressure. However, for a vertical industrial exhaust stack it would be much more practical to adjust the level of water in the sump at the bottom of the stack as shown in Figure 6 [29].

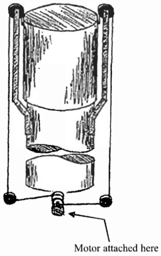

Figure 5. Exhaust stack with an adjustable sleeve to minimise the noise radiated from the top of the stack.

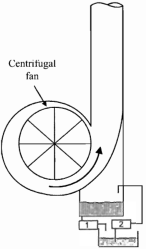

Figure 6. Exhaust stack showing a means of adjusting the effective length by pumping water into and out of a sump. Blocks 1 and 2 represent water pumps.

5. ACOUSTIC SENSORS FOR ACTIVE NOISE CONTROL

It is well known that using acoustic pressure sensors to provide the error signal for an active noise control system has some limitations, including the generation of a very large noise reduction within 1/10 of a wavelength of the sensor but not much reduction at other locations (unless the active control noise source is close enough to the primary source to physically affect its sound radiation ability). When microphones are used as error sensors, the pressure gradient in the vicinity of the error sensor in the controlled sound field can be quite large, which is a subjective problem – listeners just need to move their head a small amount and the noise level will vary by a large amount. This problem is exacerbated if the number of error sensors is equal to or less than the number of control sources. Ideally, there should be twice as many error sensors as control sources. Another way of reducing the pressure gradient problem is to measure it and include it in the objective function that is being minimised by the control system. Sensors that sense both the acoustic pressure and its gradient are referred to as energy density sensors. One such commercially available sensor is illustrated in Figure 7. A description of its use as a 3D sound intensity probe may be found in [30].

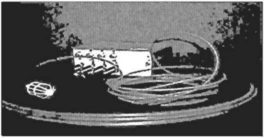

Figure 7. Optical, energy density sensor manufactured by Phone-Or Ltd (Israel).

6. VIRTUAL SENSING

A significant disadvantage of using physical sensors at the location of the sound field minimum is that the greatest noise reduction is achievable right at the sensor, which makes it difficult for the person, at whom control is aimed, to have their ear in the same location. Attempts to solve this problem have given rise to a whole new discipline. Although a number of variations have been published, there are three basic methods for implementing a virtual sensor. The first involves putting an actual physical sensor at the virtual location where it is desired to minimise the sound field, prior to full implementation of the control system. Transfer functions between the temporary microphone at the virtual location and the permanent physical microphones at some distance from the virtual location are then measured for both the primary sound field and the controlling sound field. The physical microphone is then removed from the virtual location and the control system started. During controller operation, the transfer functions measured during initialisation are used to adjust the control algorithm so it can use error signals from the actual physical microphones to minimise the sound field at the virtual location. Of course the preceding description can be extended to apply to many more than one virtual location.

The second means of implementing a virtual sensor consists of measuring the pressure gradient at the remote permanent microphones and then extrapolating this gradient quadratically or linearly to estimate the sound field at the virtual location.

The third method again involves placing a temporary physical microphone at the virtual location and then uses an adaptive algorithm to adjust the contributions from each of a number of remote microphones so that the resulting combined signal matches the signal from the virtual location when exposed to the primary sound field.

The above three methods and a number of variations are described in more detail in [13], [31] and [32] and the theoretical limitations to the accuracy of virtual sensing in a random sound field is discussed by Petersen et al [33].

7. VIBRATION SENSORS

In many cases where it is necessary to develop a compact active noise control system to reduce sound radiation from a vibrating structure, it is not practical to insert acoustic sensors in the surrounding sound field. In these cases it is desirable to be able to use vibration sensors on the structure to control the sound radiation. This is not as simple as it may at first seem, because reducing the overall vibration level in a planar structure may not reduce the sound radiation. This is because although normal vibration modes on a structure are orthogonal in terms of structural vibration, they are not orthogonal in terms of sound radiation. This means that reducing the vibration level of one or a number of modes will not necessarily reduce the overall sound radiation. There are two ways of overcoming this problem. The first [34] involves developing a sensing system that transforms the modes that are sensed so that they are orthogonal in terms of sound radiation but not in terms of structural vibration, so that reducing any one output of the sensing system will automatically reduce the radiated sound. Sensing systems would typically use accelerometers or piezo-electric patches as vibration sensing elements.

The second way of overcoming the sensing problem is to use model reference control [27]. In this case, the sensing system is initialised using physical microphones located such that when their signals are minimised, the radiated sound field is minimised. This is achieved using a feedforward adaptive control system. The outputs from the accelerometers mounted on the vibrating structure, corresponding to the minimum sound field, are then recorded and during operation of the control system, the physical microphones are removed and their inputs to the control system are replaced with the accelerometer signals. A new control algorithm is then used that attempts to drive the accelerometer signals to those that were recorded during initialisation when the microphone signals were used to minimise the radiated sound field.

8. CONCLUSIONS

Sensors and actuators are important components of any active noise control system and their cost often inhibits commercial applications of the technology. It has been shown that there are a number of sensor and actuator choices and even the potential to develop very low cost devices. However, it seems that mass market applications of active noise control will have to be found before the low cost possibilities will be fully developed.

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A REVIEW OF ACTIVE CONTROL APPLIED TO PLATES AND CYLINDERS

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This paper presents adaptive feedforward active control applied to simple structures comprised of beam, plate and cylindrical elements. For each system under consideration, by initially obtaining a good understanding of the physics of the structural and acoustic responses, the active control application can be tuned to improve the control performance. In particular, the use of active structural acoustic control to attenuate the structurally radiated sound fields is investigated.

INTRODUCTION

Sound radiation from distributed vibrating structures is a continuous problem in transportation and other industries. At low frequencies, passive control techniques provide a poor reduction in structural dynamic responses, promoting active control as a more attractive solution. A novel approach to actively attenuate the structurally radiated sound fields is to directly modify the structural response. This is achieved by adding control inputs to the structure. The acoustic cost function is typically based on a global measure such as the radiated sound power, or the local sound pressure. This control technique is known as active structural acoustic control (ASAC).

Previous work on active control of sound radiating from structures has mainly dealt with homogeneous structures such as beams (Burdisso and Fuller, 1992) and plates (Fuller *et al.*, 1991; Wang *et al.*, 1991; Pan *et al.*, 1992). Using actuator and sensor mechanisms associated with smart structures, ASAC strategies have been applied utilising piezoelectric materials as both the structural actuator and error sensor (Clark and Fuller, 1992). This ASAC strategy, where the structure is equipped with a sensor bonded to the surface, can be practical in cases when the use of acoustic sources such as microphones or hydrophones located in the surrounding fluid may not be practical (for example, in the case to actively suppress the externally radiated sound field from a submarine). It is important to note however that when using structural error sensors in an ASAC application, it is necessary to have an efficient design procedure and good *a priori* knowledge of the structure-acoustic coupling, due to the inability of the structural sensors to directly measure the acoustic response.

ASAC techniques have also been employed to attenuate the structure-borne sound fields generated by subsonic wave scattering at a structural discontinuity, which may be a boundary (Guigou and Fuller, 1993), a line discontinuity (Gu and Fuller, 1991) or a beam-stiffened plate (Kessissoglou and Pan, 1998). In these cases, it has been shown that whilst the influence of flexural near-field waves generated due to the presence of a structural discontinuity can be neglected in terms of the dynamic response, they significantly contribute to the far-field structurally radiated sound.

While most ASAC systems are designed using feedforward control techniques, active control of the radiated sound pressure from a simply supported plate has been investigated using feedback control (Meirovitch and Thangjitham, 1990). A good review of ASAC applied to plate systems and cylindrical structures is given by Fuller *et al.* (1996). This paper reviews the use of active vibration control (AVC) and active structural acoustic control (ASAC) based on a conventional adaptive feedforward algorithm, to respectively attenuate the structural and acoustic responses associated with a beam-stiffened plate and a cylinder submerged in a fluid.

OVERVIEW OF ADAPTIVE FEEDFORWARD ACTIVE CONTROL

The objective of feedforward control is to produce a secondary disturbance to a system that cancels the effect of a primary disturbance at the location of an error sensor. Adaptive feedforward active control is effective in situations of tonal noise and when a reference signal correlated to the primary disturbance is available. This signal is passed through an adaptive filter, as shown in Fig. 1, where the output of the adaptive filter is applied to the physical system by secondary sources. The filter coefficients are adapted in such a way that the error signal at one or more critical points is minimised.

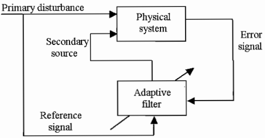


Figure 1. Block diagram of an adaptive feedforward control system (Widrow and Stearns, 1985).

Local control is achieved as there is no guarantee that the response is reduced at any other locations other than the error sensor. Unless the response is dominated by a single mode, there are locations where the total response may be amplified. Using the conventional adaptive feedforward least mean square (LMS) algorithm (Fuller *et al.*, 1996), the optimal control force is obtained in what follows. Let the primary structural response (such as the flexural displacement of a beam or plate) be denoted by $w_p(\mathbf{x})$. This can often be written in terms of the product of the system transfer function G_p and the primary disturbing force F_p, that is, $w_p(\mathbf{x})=F_pG_p(\mathbf{x})$. When a control point force of amplitude F_s applied at a position $\mathbf{x}_s$ on the system, the secondary structural response can be expressed as $w_s(\mathbf{x})=F_sG_s(\mathbf{x})$, where G_s is the secondary transfer function. The total response at some point $\mathbf{x}$ can be obtained by adding the structural responses induced by the primary disturbance and the control force:

$$w_{tot}(\mathbf{x}) = w_p(\mathbf{x}) + w_s(\mathbf{x}) \tag{1}$$

A variety of cost functions can be developed depending on the response to be minimised; these cost functions may be the squared displacement or acceleration, kinetic energy, transmitted power, mean square sound pressure, out-of-plane velocity, etc. Using the squared error sensor output at a location $\mathbf{x}_e$ as the cost function to be minimised, a quadratic function expression in terms of the complex control force amplitude is obtained as:

$$J = w_{tot}(w_{tot})^* = F_s^* A F_s + F_s^* B + F_s B^* + C \tag{2}$$

where the asterisk * denotes the complex conjugate, and $A=G_s^*G_s$, $B=G_s^*G_pF_p$ and $C=F_p^*G_p^*G_pF_p$. In the adaptive feedforward LMS algorithm, the optimal control force that results in the minimisation of the cost function can be obtained by differentiating the cost function with respect to the real and imaginary components of the control force (Fuller *et al.*, 1996). The optimal control force corresponds to the force value when both derivatives are zero, that $(\partial J/\partial F_{s,real})=0$ and $(\partial J/\partial F_{s,imag})=0$. The optimal control force can then be obtained as:

$$F_{s,opt} = F_{s,real} + jF_{s,imag} = -\frac{B}{A}. \tag{3}$$

APPLICATIONS OF AVC AND ASAC

AVC of the wave transmission in a rib-stiffened plate

Beam stiffened plates are commonly found in ship hulls, aircraft and machine casing. The transmission of plate flexural waves through a reinforcing beam is related to the coupling between the plate flexural waves and the flexural and torsional waves in the beam. For an infinite beam-stiffened plate, the maximum flexural wave transmission occurs at the optimal trace wave matching between the flexural waves in the plate and the flexural and torsional waves in the beam. These are described as flexural and torsional *coincidence conditions*, respectively. The coincidence conditions are dependent on the angle of incidence φ of the flexural plane wave W_{in} propagating in the x–y plane in plate 1 and impinging on the beam boundary, as shown in Fig. 2.

The relationship between the wavenumber k and the wavelength λ ($k=2\pi/\lambda$) allows the explanation of the coincidence conditions. The plane wave in plate 1 at a frequency ω has wavelength λ_p, while the wavelength of the

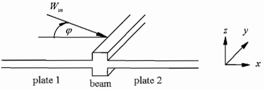


Figure 2. Beam-stiffened plate separated into three subsystems: plate 1, beam and plate 2, showing the flexural plane wave in plate 1 incident at the beam boundary.

flexural wave in the beam at frequency ω is λ_B. λ_p and λ_B are not necessarily equal. If the intercepts of the incident wave on the y–axis equals the natural flexural wavelength of the beam as shown in Fig. 3, then optimal flexural trace wave matching occurs (flexural coincidence). Similarly, if λ_y matches the natural torsional wavelength of the beam ($\lambda_y=\lambda_p/\sin\varphi=\lambda_T$), then optimal trace wave matching occurs between the plate flexural waves and the beam torsional waves (torsional coincidence). It is at these coincidence conditions that the greatest coupling between the plate and beam motion occurs, resulting in the maximum transmission of the flexural wave motion through the reinforcing beam. Since the plate and beam flexural wavenumbers vary with frequency in the same way, the flexural coincidence condition becomes frequency independent and occurs for a single angle of incidence only ($\varphi=\sin^{-1}(k_B/k_p)$). The torsional coincidence condition is dependent on both angle and frequency, that is, the angle at which this coincidence condition occurs increases with the corresponding coincidence frequency ($\varphi=\sin^{-1}(k_T/k_p)$).

The characteristics of the transmission of the plate flexural waves through the reinforcing beam are shown in Fig. 4 for a frequency range up to 2000 Hz and for a relevant range of angles of the incident waves from 0° to 20°. The beam-stiffened plate has material properties of aluminium, with a structural loss factor of 0.001, plate thickness of 1.6 mm, and beams of both width and height of 20mm. The size of the beams has been chosen to greatly exaggerate the coincidence conditions. Figure 4 shows that the flexural coincidence condition occurs for a single incident angle corresponding to $\varphi_B\approx 11.5°$ for this beam-plate model. At torsional coincidence, the angle of incidence increases with frequency.

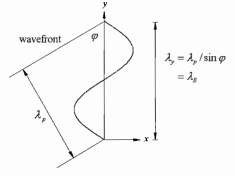


Figure 3. Optimal trace wave matching between the plate and beam flexural waves.

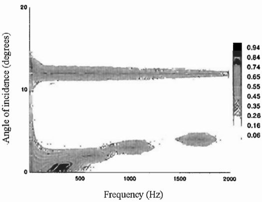


Figure 4. Flexural wave transmission in plate 2 showing flexural and torsional coincidences.

An array of point control forces are applied along the length of the beam to excite flexural motion, and are equally distributed by a distance Δ (Kessissoglou & Pan, 1997) . Similarly, point control moments can be applied to excite torsional motion in the beam. The error sensors are located in the far field of plate 2. Under plane wave propagation, the control forces are arranged to have the same magnitudes and prefixed phases as follows:

$$F_n^s = F_s e^{j\phi_n}, \quad n = -N,...,N \tag{4}$$

For attenuation of the flexural wave transmission due to flexural coincidence, the phases of the point control forces can be arranged to have the same spatial phase variation to that of the primary flexural waves in the beam. The phases of the control forces become:

$$\phi_n = k_B n\Delta, \quad n = -N,...,N \tag{5}$$

where $\Delta=0.3\lambda_B$. It should be noted that this arrangement is similar to the biologically inspired control strategy, where a group of actuators are connected together with certain phase and amplitude relationship, and only one control signal is needed to drive them (Carneal and Fuller, 1995). Under point moment control, the phases of the control moments are arranged to be $\phi_n=k_T n\Delta$ where $\Delta=0.5\lambda_T$. The displacement at the error sensor locations in the far-field of plate 2 is the superposition of the primary transmitted waves and secondary flexural waves generated by the control forces. The cost function to be minimised is the squared total plate flexural displacement at the error sensor location $x_e=10\lambda_p$ and averaged at M discrete locations along the y–direction.

Figure 5 shows the flexural wave attenuation level at the error sensor locations ($x_e=10\lambda_p$) and for 401 discrete locations along the y–direction corresponding to the range $(-2\lambda_p, 2\lambda_p)$. Examination of a relevant range for the incident angle from 0° to 20° at excitation frequencies of 500 Hz and 1000 Hz shows that significant attenuation of around 14 dB has been achieved at the flexural coincidence angle of $\varphi_B \approx 11.5°$. The level of attenuation at both frequencies is the same. Due to the nature of the arranged control force excitation, the radiated secondary

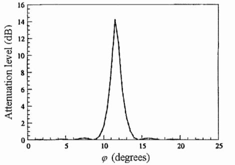


Figure 5. Attenuation levels of the flexural wave transmission at excitation frequencies of 500 Hz (solid line) and 1000 Hz (dashed line) using point control forces.

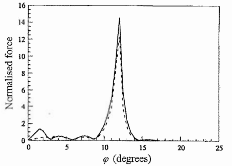


Figure 6. Normalised magnitude of the optimal control force at 500 Hz (solid line) and 1000 Hz (dashed line).

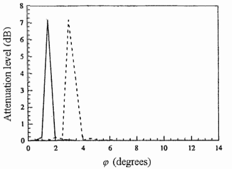


Figure 7. Attenuation levels of the flexural wave transmission at excitation frequencies of 500 Hz (solid line) and 1000 Hz (dashed line) using point control moments.

flexural waves have poor spatial phase correlation with the transmitted primary waves away from the flexural coincidence angle, which results in poor attenuation. However, away from the coincidence condition, the beam itself acts as an effective passive attenuation device. Changing the frequency of excitation has no effect on the attenuation level or the angle at which attenuation is achieved, as the flexural coincidence condition is independent of frequency.

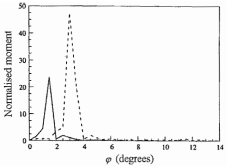


Figure 8. Normalised magnitudes of the optimal control moments at 500 Hz (solid line) and 1000 Hz (dashed line).

Figure 6 shows the corresponding dimensionless magnitude of the optimal control force $F_{t,opt}$, which shows that a large force is required at the flexural coincidence angle. As the frequency increases, the beam is able to vibrate more freely and hence a slightly smaller amplitude of the control forces is required in order to generate secondary vibrational levels in the beam to match the primary flexural energy level in the beam.

Under point moment control, attenuation of the flexural wave transmission is achieved for the same number of discrete locations along the y–direction as for the point control force application, corresponding to the range $(-2\lambda_T, 2\lambda_T)$. Figure 7 shows the attenuation levels of the flexural wave transmission using point control moments at excitation frequencies of 500 Hz and 1000 Hz. The peaks of the attenuation levels occur at the corresponding torsional coincidence angles of 1.5° and 3°, respectively. Unlike force control, as the frequency increases, the magnitudes of the control moments required for the same level of attenuation increases (Fig. 8). This is due to the fact that at higher frequencies, the beam torsional stiffness relative to the plate bending stiffness increases (Goyder and White, 1980).

ASAC of a rib-stiffened plate

Using ASAC, the dynamic response of the ribbed plate can be modified to attenuate the structurally radiated sound field. In this ASAC application, the control forces are again applied to the beam, while the error sensing devices are located in the surrounding fluid. An incident wave propagating in the low frequency range corresponding to frequencies well below critical (where the *in vacuo* phase speed of the flexural waves in the plate c_p is less than the speed of sound in the fluid c_o), is known as a subsonic wave and does not radiate energy into the far-field of the surrounding fluid. When the subsonic flexural wave is incident on the beam discontinuity, the scattering of the structural wave field generates both supersonic and subsonic wave types in the structural response, resulting in structurally radiated sound. The structure is considered to be in air and hence the response is not affected by the fluid loading. For periodic distribution of the sound pressure in the y–direction (along the length of the beam), the sound pressure field is evaluated in a cylindrical coordinate system defined by $x=r\cos\theta$, $z=r\sin\theta$, $y=0$, as shown in Fig. 9.

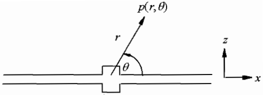


Figure 9. Sound pressure field evaluated in a cylindrical co-ordinate system in the x–z plane.

An expression for the sound pressure field of the ribbed plate under plane wave incidence has been derived using the acoustic wave equation and structure-fluid coupling conditions at the surface of the plate (Junger and Feit, 1985). The expression for the primary sound pressure field is described in terms of both a non-integral component and an integration of the structural wavenumber spectrum. Each component is a function of the scattered structural waves (transmitted and reflected, propagating and evanescent plate flexural waves). The wavenumber spectrum of the integral component can be separated into supersonic and subsonic wavenumber spectrums. The supersonic wavenumber spectrum directly contributes to the far-field radiating sound pressure, whereas the subsonic wavenumber spectrum dominates the near-field acoustic response. The supersonic wavenumber spectrum is further restricted by a limiting range for the incident angle, where φ must be less than the critical incident angle φ_c, which is defined by $\varphi_c=\sin^{-1}(k_o/k_p)$ and k_o is the acoustic wavenumber ($k_o=\omega/c_o$).

The primary radiated sound power is obtained by integrating the acoustic intensity over a semi-cylindrical surface centred at the beam discontinuity, where the acoustic intensity is related to the mean square pressure. At the structural coincidence conditions, the far-field primary sound pressure level is slightly decreased. This is due to the fact that at the coincidences, there is a large amount of flexural wave transmission through the reinforcing beam, thereby resulting in a more balanced distribution of the structural waves along the x–direction (normal to the beam). It has been previously shown that an imbalance in the structural response along the x–direction will generate a greater number of supersonic wavenumber components, which will thereby result in an increase in the far-field radiated sound pressure. The phases of the control forces are pre-fixed such that the forces have a spatial phase variation with each other at the flexural coincidence condition. Although the primary structural response may be either at coincidence or off-coincidence, the secondary structural response is always generated by forces with the phase delay in the beam corresponding to that of the flexural coincidence condition. At flexural coincidence, the total-far-field sound pressure due to the superposition of the primary and secondary sound fields is optimised. At any off-coincidence condition, the superposition of the primary and secondary sound fields will result in the least increase in the supersonic wavenumber components to be attenuated.

Using the control force approach described previously, in which for $2N+1$ forces applied to the beam, the control forces have a pre-fixed phase relationship with each, only the complex amplitude F_s needs to be optimised. For minimisation of the acoustic responses, two cost functions are examined corresponding to (i) the square of the total sound pressure $|p_{tot}(r,\theta)|^2$ at a far-field error sensor location (r_e,θ_e), and (ii) the radiated sound power. An expression for the total sound pressure is obtained by the superposition of the primary and secondary sound fields (Kessissoglou and Pan, 1998). Both cost functions can be expressed as a quadratic function in terms of the control force amplitude. Using the adaptive feedforward LMS algorithm, the optimal control force is obtained.

Using the same material properties and dimensions of the ribbed plate and air to represent the surrounding acoustic field, the flexural and torsional coincidence angles are respectively φ_B=11.5° and φ_T=1.5° (for an excitation frequency of 500Hz). The critical incident angle defined by $\varphi_c=\sin^{-1}(k_o/k_p)$ is 14.5°. Since both coincidence angles are less than critical, they both contribute to the radiation of sound into the far-field. Figure 10 shows that only those incident waves with angles less than the critical incident angle contribute to the radiated sound power, as these angles correspond to the supersonic wavenumber spectrum (Kessissoglou and Pan, 1998). At incident angles greater than critical, there is very little sound power radiated into the far-field as the sound pressure is dominated by the subsonic wavenumber spectrum which only contributes to the acoustic near-field. An interesting feature in Fig. 10 is the effect of the structural coincidences on the far-field radiated sound power. At the structural coincidence conditions corresponding to φ=1.5° and 11.5°, there is a decrease in the radiated sound power. This due to the fact that at a structural coincidence condition, the greatest plate flexural wave transmission through the beam occurs. As a result, the structural energy is more uniformly distributed in the plate and less energy is radiated into the acoustic fluid. As the incident angle approaches critical, the far-field sound power is a maximum as this condition corresponds to the optimal coupling between the structure and the fluid.

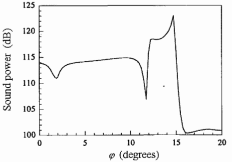


Figure 10. Radiated sound power as a function of the angles of the incident wave.

The radiated primary sound pressure at a far-field dimensionless radius of $k_o r$=10 increases at directivity angles close to the surface of the plate (0°≤θ≤5° and 175°≤θ≤180°). This is due to the fact that even at a far-field radius, directivity angles close to the surface of the plate correspond to the near-field. Therefore, in order to accurately represent the radiated sound power, these 'grazing' angles were not included in the integration of the sound intensity, that is, the sound power was evaluated over the hemispherical range of 5°≤θ≤175° (centred on the beam).

Figure 11 presents the attenuated sound power as a function of the error sensor location θ_e for an incident angle of 11.5° (flexural coincidence). The attenuated sound power resulting from minimisation of the sound power and from minimisation of the far-field sound pressure at each local error sensor location in the range of 5°≤θ ≤175° are compared. For the present control arrangement, using the radiated sound power as the cost function results in attenuation levels of 23dB, 16dB and 36dB for incident angles of 5° (off coincidence), 11.5° (flexural coincidence) and 14.5° (critical angle), respectively. Similar levels of sound power attenuation can be achieved by minimising the local sound pressure using a single error sensor located in the range of 45°≤θ_e≤50° or 130°≤θ_e≤135°. The exact location for the optimal error sensor differs slightly for each incident angle of the incoming structural wave. Examination of the controlled sound pressure levels show that optimising the error sensor location results in a reduction in the radiation efficiency. The ASAC system can now be designed with an appropriate cost function and error sensor location to achieve the best control performance. For a single, optimally located error sensor, it has been shown that global attenuation of the squared sound pressure is achieved at all directivity angles away from the grazing angles (Kessissoglou and Pan, 1998). This reduces the complexity of the control application since using the sound power as the cost function would require an array of error sensors located in a hemisphere centred on the ribbed plate.

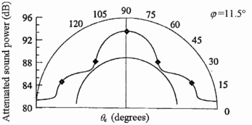


Figure 11. Attenuated sound power as a function of the local error sensor location θ_e, resulting from minimisation of the sound power (solid line) and from minimisation of the far-field sound pressure at each local error sensor location in the range of 5°≤θ_e ≤175.

Active control of a finite cylinder

Active control of the structural and acoustic responses of cylinders has mainly concentrated on actuating and minimising the radial motion of the cylindrical shell. A review of earlier literature on active control to minimize cylinder interior acoustic fields (in the case of an aircraft interior) and exterior acoustic fields (in the case of sound radiation from a submarine or noise from piping systems) is given by Fuller *et al.* (1996). Active control of sound radiation from cylinders using piezoelectric actuators and structural sensors has shown to yield similar performances in attenuating the far-field radiated pressure as error microphones (Maillard and Fuller, 1999). In this section it is demonstrated that active modal control of both the axial and radial motions of a finite cylinder is required to globally attenuate the structurally radiated sound pressure. An idealized model of a submarine hull is considered, which is modelled as a ring-stiffened cylindrical shell with finite rigid end closures, separated by bulkheads into a number of compartments and under axial excitation from the propeller-shafting system (Tso *et al.*, 2003; Dylejko *et al.*, 2005). The fluid loading effects are modelled as an increase in inertia of the shell. Lumped masses are added at each end to represent on-board equipment and to maintain a condition of neutral buoyancy. A schematic of the submarine model is shown in Fig. 12. Excitation of the hull axial modes causes both axial motion of the end closures, u, and radial motion of the shell, w, resulting in a high level of structurally radiated noise. Under axial excitation, it is assumed that only the breathing mode of the cylinder is excited which gives rise to an axisymmetric case. An expression for the radiated sound pressure contributed by axial movement of the end plates and radial motion of the shell was obtained using the Helmholtz integral equation (Junger and Feit, 1985), and by considering the radiating surfaces separately.

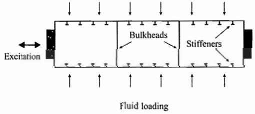


Figure 12. Schematic of the hull under axial excitation.

A secondary axial force F_s was used to excite the pressure hull at $x=L$, as shown in Fig. 13. For active control of the axial response of the end closures, an error sensor was located at each end of the pressure hull, denoted by es^u. For control of the radial response of the hull, one or more rings of error sensors, es^w, were located circumferentially around the hull. It is important to note that due to the Poisson effect which causes coupling between the axial and radial motions of the hull, the use of a control force to generate a secondary axial response will also generate secondary radial motion. Similarly, active control of the radial motion will also result in secondary axial vibration. This must be taken into account in the evaluation of the total radiating sound pressure due to the primary and control forces. The squared total axial or radial displacements due to the primary and control excitations were minimised at the error sensors using the adaptive feedforward LMS algorithm described previously. When both the axial and radial displacements were simultaneously controlled, two control forces were used, one for each of the displacements.

Numerical calculations were performed on a ring stiffened steel cylinder of 6.5 m diameter, 40 mm hull plate thickness, 45 m length, and with two evenly spaced bulkheads of thickness

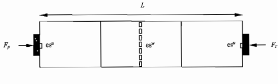


Figure 13. Locations of the control force and error sensors for active control of the axial and radial motions of the finite hull.

40 mm. The cylinder was submerged in water of density 1000 kg/m^3. A neutrally buoyant condition was maintained by using a distributed mass of 1000 kg/m^2, and with lumped masses of 200 tonnes at each end. Internal structural damping was included in the analysis by using a structural loss factor of 0.02. A primary axial force of 1 N is applied to one end of the hull. The first three axial resonances were observed to be approximately 20.5, 42 and 64 Hz, as shown in figure 14. The small peak at approximately 9 Hz is caused by the resonance of the bulkheads. At the first and third axial resonances, the end plates of the cylinder are vibrating out of phase with each other. At the second axial mode, where the end plates of the cylinder are vibrating in phase, the axial response of the end plates accounts for approximately two-thirds of the primary radiated pressure.

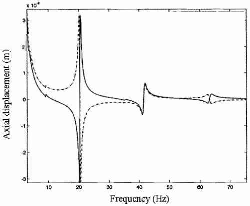
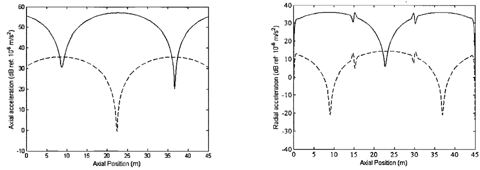


Figure 14. Axial response of the hull at x=0 (solid line) and x=L (dashed line).

Active vibration control was applied at the second axial resonance of 42 Hz. Figures 15(a) and (b) present the primary and controlled responses of the axial and radial motions, respectively, as a function of axial position along the hull (from x=0 to L). Comparison of the primary axial and radial motions shows that when the axial response is a maximum (at the cylinder ends and midway along the hull length for the second axial mode), the radial response is a minimum, and vice versa. Figure 15(b) also shows the localised effect of the bulkheads. In Fig. 15(a), the squared axial displacement was simultaneously minimised at error sensors located at each end of the pressure hull. In Fig. 15(b), the squared radial



(a) (b)

Figure 15. Primary (solid line) and controlled (dashed line) acceleration distributions of the axial (a) and radial (b) responses as a function of axial position along the length of the hull.

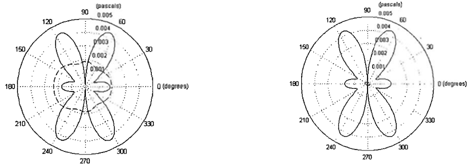
displacement was simultaneously minimised at two rings of error sensors located at anti-nodal axial positions along the hull length. The corresponding magnitudes of the control forces are nearly unity, attributed to the axisymmetric motion of the hull and symmetry of the control application.

Figure 16 shows the primary structurally radiated sound pressure at the second axial resonance. The radiated pressure for control of the radial response only and for control of both the axial and radial responses is given in Figs. 16(a) and (b), respectively. For control of the radial response, the control performance is strongly dependent on the error sensor locations for a given axial resonant frequency. Active control of both the axial and radial hull displacements results in complete cancellation of the radiated sound pressure.

Excitation of the hull at one of its low frequency axial resonant frequencies results in an efficiently radiating structural mode. Due to the coupling between the axial and radial motions of the cylinder, a control actuator for each wave type is required. However, due to active control at a resonant frequency, the use of a single actuator for each wave type is sufficient. For active control at an off-resonant frequency, the modal density is higher and an increase in the number of control actuators is required to improve the control performance. The number of error sensors should either be equal or greater than the number of control inputs used. Increase the number of error sensors can generally improve the attenuation achieved due to the cost function being closer to an estimate of a global property (such as kinetic energy or sound power), and is thereby more robust to changes in the physical system response.

CONCLUSIONS

This paper presents a review of adaptive feedforward active control and its applications to attenuate the structural and acoustic responses associated with beam-stiffened plate and cylindrical structures. In each case, *a priori* knowledge of the dynamics of the physical system has enabled arrangement of the control actuators and sensors to improve the control performance. For active control of the flexural wave transmission through the reinforcing beam of a stiffened plate, global attenuation of the plate vibration can be achieved using an array of point forces or moments to the beam. Mechanisms of the global control are due to the significant reduction in the beam flexural or torsional energy using an array of control forces or moments, respectively. The control arrangement is



(a) (b)

Figure 16. Primary (solid line) and controlled (dashed line) radiated sound pressure for active control of the radial response only (a) and both the axial and radial responses (b).

dependent on a pre-fixed phase relationship between the forces or moments using information on the flexural or torsional coincidences. Global attenuation of the structurally radiated sound fields can be achieved by carefully selecting the location of a single acoustic error sensor. In the case of a finite cylinder with rigid end caps excited at an axial resonance, active control of both the axial and radial motions is required to globally attenuate its acoustic signature. It is important to note that the arrangement of the control actuators and error sensors in the cases presented in this paper and corresponding levels of attenuation achieved, are dependent on coincidence or resonance conditions, at which the angles of the incident structural waves or modes of vibration contributing to the structural and acoustic responses are clearly defined.

ACKNOWLEDGEMENTS

The author would like to acknowledge Prof. Jie Pan from the University of Western Australia with whom the work on beam stiffened plates was conducted, and Drs Yan Tso and Chris Norwood from the Maritime Platforms Division, Defence Science and Technology Organisation, for the collaboration on active control of cylinders.

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Farewell Scott

This is the last edition of Acoustics Australia to be printed by Scott Williams. Scott took over the business from his father Fred and the two of them have printed Acoustics Australia - and before that the Bulletin of the Australian Acoustical Society - since August 1982 (Vol 10 # 2).

Neville Fletcher, Marion Burgess, Joseph Lai (the previous editorial team), Howard Pollard, who was editor in chief in 1982, and the current editorial committee wish Scott all the best for his new career as a teacher.

Future Meetings

Acoustics 2006

1st Joint NZAS and AAS Conference
20-22 November, Christchurch

Registration details are now available for the 2006 Australian and New Zealand Acoustical Society Conference. This conference has the theme "Noise of Progress" and is to be held at the Clearwater Resort, Christchurch, New Zealand November 20-22 2006. Early bird registration rates apply till 1 September 2006

The Keynote Speakers will include Professor Michael Vorlander, University of Aachen who will talk on "Building acoustics: From prediction models to auralisation" and Professor Chris Tindle, University of Auckland on "Sounds interesting: Wavefronts, caustics, whales and reefs". An exciting program of contributed papers is being developed and once the papers have been reviewed, the detailed program will be available on the www.

The social program includes a Conference Dinner at the Christchurch Town Hall. The dinner speakers will be Sir Miles Warren, who has been at the forefront of New Zealand's architectural profession and Professor Harold Marshall, who established Marshall Day Acoustics.

Information from www.conference.co.nz/acoustics06

ASA ASJ Joint Meeting

28 Nov-2 Dec, Hawaii

This 4th joint meeting of the Acoustical Society of America and the Acoustical Society of Japan will be held in Hawaii just preceding Internoise. Abstracts closed 30 June but Early bird registration rates are available until 24 October.

Information from http://asa.aip.org

Internoise 2006

3-6 December, Hawaii

Internoise 06 with the theme "Engineering a Quieter World" will be held at Sheraton Waikiki in Honolulu, Hawaii, USA 3–6 December, 2006. This conference will have all the features that one expects at an Internoise Conference as well as being in the relaxing environment of Honolulu - the congress banquet will be a traditional luau.

Details of the special and structured sessions are available from the webpage. Abstracts are due 15 May with completed papers 18 August. Early bird registration rates are available till 18 August.

Information from www.internoise2006.org

ICSV 14

9-12 July, Cairns

The 14th International conference on Sound and Vibration, ICSV14, incorporating the Annual Conference of the Australian Acoustical Society will be held 9 to 12 July, in Cairns, Queensland.

IIAV is an international non-profit scientific society affiliated with the International Union of Theoretical and Applied Mechanics (IUTAM). IIAV currently has 550 individual members in 55 countries and is supported by 31 national and international scientific societies and organisations. The ICSV14 is part of a sequence of congresses held in the USA (1990, 1992 and 2002), Russia (1993, 1996 and 2004), Canada (1994), Australia (1997), Denmark (1999), Germany (2000), Hong Kong (2001), Sweden (2003), Portugal (2005) and Vienna (2006), each attended by several hundred participants worldwide.

Theoretical and experimental research papers in the fields of acoustics, noise and vibration will be presented. Abstracts are due 1 Dec 2006.

Information from www.icsv14.com

ICA 2007

2-7 September, Madrid

The 19th International Congress on Acoustics is organized by the Spanish Acoustical Society, and the Institute of Acoustics and the collaboration of the Municipality of Madrid, under the auspices of the International Commission for Acoustics, ICA. This 5 day congress comprises plenary sessions, keynote presentations and contributed papers and posters. The ICA is a unique international acoustics congress as it welcomes papers in all areas of acoustics. Abstracts for the ICA 2007 are due by 30 January 2007.

Information from www.ica2007madrid.org

Note that the Australian Acoustical Society is hosting the following ICA congress, ICA 2010, and is providing special travel grants for attendance at ICA 2007 see item elsewhere in this issue and www.acoustics.asn.au for details of the submissions.

The customer is always wrong

Confirming our fear about the acoustics and acoustic policy of some restaurants (Acoustics Australia, 33, p45), the 'Good Living' section of the Sydney Morning Herald reveals that the owner of the 'Icebergs' restaurant at Bondi Beach is likely to increase the volume if diners complain about the music. "It's great sometimes to rattle the dining room and remind the client that they are in your room." A pity, because the food is good and the view is great.

News

Acoustics to the Schools

The AAS is a cognate society of the Australian Institute of Physics (AIP) and at a recent meeting the opportunity to promote a career in acoustics as part of the AIP education program was recognized. Dr Mark Butler leads the AIP education program and is the Head Science Teacher at Gosford High School, NSW. He considered that the opportunities presented by a career in acoustics were little known and was enthusiastic to arrange a talk by an acoustic consultant. Ken Scannel agreed to undertake this task on behalf of the NSW Div of the AAS. Following is a short report from Mark Butler on Ken's talks. Mark is clearly pleased with the presentation and the interest it sparked in the students to see real application of what they are learning in school science. It is likely that this will become a regular activity for Gosford High School and we hope that, building upon this success, the message will be passed around the science teachers and the AAS will be asked to provide more talks in more schools. We hope that now the ball has been set rolling that members of the AAS will willingly take a break in their busy schedule and accept invitations to give a talk to science students in their local schools.

Gosford High School is a government, academically selective school on the Central Coast of NSW with very strong 'science culture'. We have found one of the most effective ways of exposing students to science and engineering in the 'real world' is to bring practising scientists and engineers into the School to speak to students.

In May 2006 we had our first visitor speaking from the Australian Acoustics Society. Ken Scannel, a highly experienced Acoustics consultant, gave us two wonderful talks. The first talk was to fifty senior Physics students and the second was to our Science Extension Group which has members from yr7 to yr12. Ken's enthusiasm for the subject and the large number of interesting demonstrations he used to illustrate his talk had students on the edge of their seats. Many of the demonstrations involved student participation and Ken was inundated with questions from a very engaged and excited audience.

Henry Adams said, "A teacher affects eternity, he never knows where his influence ends." I suspect Ken's short visit to our school may well have opened a few eyes to another application of science and to a possible career that students may have never previously considered. On behalf of the School I would like to thank Ken for his time and the Acoustics Society for supporting this initiative. We look forward to continuing this initiative and having an annual talk on acoustics as part of our school science programs.

Dr Mark Butler, Head Teacher Science, Gosford High School

Listen Hear!

The Economic Impact and Cost of Hearing Loss in Australia

Many of us are aware of relatives or friends with hearing loss, and the significant impact that this has on their ability to communicate and to participate in society, but few of us would be aware that hearing loss represents a real financial cost to Australia of $11.75 billion per annum or 1.4% of GDP according to a new research study by Access Economics. The report, officially delivered at the opening of the Audiology Australia National Conference in Perth, identifies that 1 in 6 Australians is affected by hearing loss, and this number is projected to increase to 1 in every 4 Australians by 2050. Hearing loss is age-related, affecting 3 in every 4 people aged over 70 years.

The Listen Hear! Report, commissioned by the Cooperative Research Centre for Cochlear Implant and Hearing Aid Innovation (CRC HEAR) in partnership with VicDeaf, identifies that productivity loss related directly to hearing impairment accounts for well over half (57%) of the total financial costs – or some $6.7 billion a year. CRC HEAR CEO, Associate Professor Bob Cowan, says the study is the first of its kind to quantify the economic costs and impact on Australia associated with deafness, and will be important for informing policy making and directing health and research resources to the preventive and therapeutic interventions that are most cost effective.

"Deafness suffers from low exposure and its full implications are not immediately obvious. However, this report allows the community to better understand the cost and resource issues associated with hearing loss. The study reports that hearing loss ranks with asthma, diabetes and musculoskeletal diseases in terms of burden of disability, and should be considered as a national health priority. Hearing loss reduces the capacity to communicate, and this in turn impacts on a person's life chances through the reduced opportunity to equitably participate in education, to gain competitive skills and employment and to participate in relationships. While interventions such as hearing aids and cochlear implants can enhance a person's ability to communicate, the majority of people with hearing loss (85%) do not use such devices."

Excessive noise in the work place and social environments is not conducive to good hearing retention. With 36% of hearing loss attributable to excessive noise exposure, all of which is preventable, approaches to better management of noise prevention are needed. "Now that the true costs are known, it's time to act to significantly reduce this impact on Australia's economy." Associate Professor Cowan said. "Research into mechanisms and behavioural approaches that encourage hearing loss prevention, improvements in hearing technology, and more efficient ways to undertake clinical hearing assessments and (re)habilitation particularly in rural and remote areas are all required if we are to address the projected increase in hearing loss in our community"

Report is available from www.audiology.asn.au/pdf/ListenHearFinal.pdf

Wilkinson Murray Changes

Wilkinson Murray opened an office in Hong Kong at the end of April. Barry Murray has relocated to Hong Kong to manage and develop the office. Barry expects Wilkinson Murray to take advantage of the large amount of infrastructure development occurring in the Asian region, particularly related to tunnelling works. Barry can be contacted at barrym@wilkinsonmurray.com.hk.

To continue the development of the Sydney office, John Wassermann became a director on 1 July. John is keen to maintain the excellent reputation in acoustics and also promote his skills in air quality assessment. In line with these changes we have updated our corporate logo, changed our email addresses and also separated Wilkinson Murray (www.wilkinsonmurray.com.au) and SoundScience (www.soundscience.com.au)

Davidson relocation

The NSW Davidson sales office and team have relocating to the Biolab Offices in North Ryde. Contact details: Unit 5, 19 Khartoum Rd, North Ryde NSW 2113, tel: 1-300-SENSOR (736-767), fax: 1-300-736-755, info@davidson.com.au

Academy of Science Grants

The Academy of Science is also calling for expressions of intent for support for travelling fellowships and for research conferences to be held in Australia. The closing date is for these is 30 September

Information: www.science.org.au/awards/research.htm

SUSTAINING MEMBERS

The following are Sustaining Members of the Australian Acoustical Society. Full contact details are available from www.acoustics.asn.au/sql/sustaining.php

ACOUSTIC RESEARCH LABORATORIES

ACRAN

ACU-VIB ELECTRONICS

ADAMSSON ENGINEERING PTY LTD

ASSOCIATION OF AUSTRALIAN ACOUSTICAL CONSULTANTS

BORAL PLASTERBOARD

BRUEL & KJAER AUSTRALIA

CSR BRADFORD INSULATION

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DOOR SEALS OF AUSTRALIA

G P EMBELTON & CO PTY LTD

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PEACE ENGINEERING PTY LTD

SINCLAIR KNIGHT MERZ

SOUNDGUARD PYROTEK

SOUND CONTROL PTY LTD

VIPAC ENGINEERS AND SCIENTISTS LTD

WORKCOVER COMPLIANCE COORDINATION TEAM

FASTS

Government support

The Federation of Australian Scientific and Technology Societies (FASTS) gratefully acknowledges the decision announced in the budget to provide $800,000 over four years. The President of FASTS, Professor Tom Spurling said the funding will provide a significant boost to FASTS' policy and analysis capability. As described in the budget papers, the funding has been provided to support "the Federation's role in policy formulation and raising public awareness in, and promoting the importance of science and technology in addressing important national issues."

Professor Spurling, FASTS president, said the funding sends a clear signal to the science and innovation sectors that the Government values informed and constructive contributions to debates on national policy. "The funding will also enhance FASTS' capability to build non-partisan links between working scientists and Parliamentarians as exemplified by the highly successful, annual 'Science meets Parliament' initiative."

Pathways to Technological Innovation

The House of Reps Science and Innovation Committee has released its report on pathways to technological innovation. The report can be accessed at http://www.aph.gov.au/house/committee/scin/pathways/report.htm. The report makes a number of quite sensible recommendations although its overall position is to accept the basic policy thrust of Backing Australia's Ability and cognate policy settings. The FASTS summary includes recommendations for: Better promotion of Government programs, DEST to establish working party to consider better 'whole of Government' approach and co-ordination, DEST to expand statistical data collection to include combined SET/Business degrees; SET graduates by state and territory; greater detail on SET qualifications and subjects of international students and workforce participation rates, Govt should introduce support mechanisms to assist innovation other than through start-ups, DITR to extend support for later stage commercialisation including marketing and sales strategies

Science Education Directions

On 7th of August, FASTS hosted a meeting to discuss the current state and future directions of science education. The rationale for calling this meeting was that FASTS believe that while working scientists, educators and communicators are well aware there are problems in science education, there is not a shared view on; what the problems are; how they might be addressed; what role(s) should or could working scientists and their organisations play in science education.

Participants at the meeting included representatives of groups with a strong professional interest in science education including science and mathematics teachers associations, Deans of Science, the Academies, professional societies and public sector research agencies. The meeting will open with a presentation by DEST giving an overview of science education programs and preliminary discussion of the audit of science skills shortages. FASTS will use the outcomes from this meeting in their discussion with the Government on science education policy.

One finding from survey is the importance of showing at the high school level that there are interesting, challenging and stable career prospects. This is surely relevant for the AAS and should be the prompt for the membership to take any opportunities to show the opportunities that are available for those entering the acoustics profession.

Noise on the menu.

The recent CitySounds survey conducted in Melbourne found that over one half of the respondents felt that cafes had become noisier in the last 3 years.

More than half the diners under 35 had experienced difficulty holding a conversation and 78% over 35 had experienced difficulty.

See http://www.melbourne.vic.gov.au/info.cfm?top=46&pg=3009

AUSTRALIAN ACOUSTICAL SOCIETY TRAVELLING GRANT 2006 for Participation in ICA 2007

The Australian Acoustical Society Travelling Grant has been established to encourage members of the Society to attend the International Congress on Acoustics in Madrid, August 2007. Two travelling grants will be awarded, each of $1,000. The grant is open to any member of the Society of Grade Fellow, Member, Graduate, Associate or Student.

What is required for entry?

The submission should not be greater than two A4 pages and should include:-

- A brief background of the applicant.
- An outline of how attendance at ICA 2007 will enhance their career and/or studies in acoustics.
- An outline of how their participation at the ICA will enhance the profile of Australian acoustics.
- Applicants proposing to present a paper at the congress should include an abstract

The selection committee may seek additional information from the applicant as part of its selection process. Each applicant must provide a report (500 words) on return from ICA 2007 on the experience. These may be printed in Acoustics Australia.

Submission of entry.

Entires should be forwarded by 30 September 2006 in electronic form, to:-

GeneralSecretary@acoustics.asn.au

Standards Australia

Standards Australia has a very extensive list of standards and the AAS has representation on most committees with responsibilities for production of standards primarily on acoustics and vibration. However there are a great number of Australian Standards that have only a small section on acoustics and vibration and its not always assured that these sections of the standards will be referred to those with expertise in acoustics and vibration prior to publication. The problems this can lead to were raised in a previous issue of the journal (Acoustics Aust Vol 33, No 3 p113) when Ian Hillock identified errors in the vibration related appendix to a new explosives Standard and Marion Burgess identified sections of old marine related standards that were inconsistent with recent noise management standards.

The AAS Council has recently written to Standards Australia seeking that AAS be advised during the preparation and public comment period of all standards that may have any content on acoustics or vibration. The Council will then call upon the advice of appropriate representatives from the AAS to review the proposed standards and provide comment to Standards Australia. With this support from the membership it is hoped that in due course all the acoustics and vibration sections of Australian Standards will be accurate and consistent.

New Products

QUEST

SoundPro Series

The SoundPro® SE/DL Series includes Advanced Hand Held Sound Level Meters and Real-Time Frequency Analyzers. When used with QuestSuite® Professional II applications software, you can simply and reliably perform advanced noise analyses. All meters include two virtual sound level meters to simultaneously measure multiple regulatory requirements. Meters equipped with 1/1 and/or 1/3 octave real-time frequency analysis allow you to choose either real time analyzer or broadband measurements for one of the virtual sound level meters. Use high speed USB or removable SD memory to transfer data from your sound level meter to your computer for further analysis. With a total of twelve available models in the the SoundPro® SE/DL Series, one of these noise exposure meters is certain to meet your specific requirements.

Information: Air-Met Scientific Pty Ltd, peter.shiels@airmet.com.au, www.airmet.com.au

LMS

Mobile front-ends and Test.Xpress analyser

At the annual LMS user conference in Munich, Germany, LMS introduced a series of new products. The LMS SCADAS family has been extended with a new LMS SCADAS Mobile series of mobile and ultra-portable data acquisition front-ends. LMS Test.Lab is tightly integrated with the new SCADAS Mobile front-end and gains a new solution for mobile noise and vibration testing. LMS also strengthens its offering in the areas of qualification, and troubleshooting testing with LMS Test.Xpress, a powerful sound and vibration analyser and data recording solution, based on the same LMS SCADAS Mobile front-end.

Information: info@davidson.com.auNEW

New Members

Sustaining Member
Sinclair Knight Merz (Vic)

Member
Preston Adams (Qld)
Rebecca Donovan (Qld)
Mark Latal (NSW)
Matthew Goodfellow (Qld)
Savithr Shimada (NSW)

Graduate
Simon de Lisle (vic)
Aaron Lepp (Qld)

Student
Matthew Vance (Vic)

Book Reviews

Sound Images of the Ocean in Research and Monitoring,

Peter C. Wille.

Springer-Verlag, 2005, 472 pages (hard cover), 452 illustrations, 391 in colour with CD-ROM. Hardcover. ISBN: 3-540-24122-1 Price approx A$187 from DA Information Service, www.dadirect.com.au,

Only in recent decades has the detailed structure of portions of the oceanic deeps been realised through swath sonar acoustic instrumentation, but the imagery obtained has been confined largely to specialists. This book aims in particular to introduce the spectacular geological formations and sedimentary bedforms in the deep oceans revealed by the acoustic methods to a much wider audience. The bathymetric features imaged range from the global scale (with the help of other techniques such as gravity anomaly detection), spreading ridges and associated trenches, continental shelves and canyons, to finer scales such as mud volcanos, reef structures, and seagrass beds.

In pursuit of its main aim, the book presents an impressive collection of colour-coded imagery of the seabed obtained by multibeam swath bathymetric sonar. The power of acoustics to provide 3-D seabed imagery via Digital Elevation Models and fly-throughs is demonstrated with files on CD-ROM which can be viewed and manipulated as 3-D images on computer screens using the Fledermaus software (*Hint: the vertical exaggeration can be changed by clicking on the widget*). 2-D imagery obtained by another type of swath sonar, sidescan sonar, is used to complement the multibeam data. Mention is made of the increasing role of Autonomous Undersea Vehicles (AUVs) in carrying these acoustic instruments to the deep seabed, enabling higher resolution data to be obtained than from surface vessels.

However, the book is not confined to the seabed, nor to geology. Acoustic remote sensing techniques can also be used to probe the sediment column, a process known as sub-bottom profiling. The ocean volume can be acoustically probed to gain information on ocean currents, fish stocks, and other phenomena such as bubbles and gas transfer near the ocean surface. Acoustic devices such as Acoustic Doppler Current Meters (ADCPs), and sub-bottom profilers measure a time series of acoustic response from a single direction, rather than across a swath, and usually point downwards into the ocean or into the seabed. ADCPs estimate ocean current vectors through the water column by measuring the Doppler shift off particles in the water. Parametric sonars use nonlinear acoustic effects to generate low frequencies which can penetrate the seabed, to reflect at the interfaces between sediments of different acoustic impedances, revealing seabed structure. Attaching these instruments to moving vessels allows cross-sections of water column or seabed parameters to be visualised. Sounds emitted by marine creatures can be imaged as sonograms. Man-made objects such as wrecks, pipelines, and archaeological structures may be imaged by acoustic methods. Acoustic tomography can be used to reconstruct the three-dimensional temperature field, or changes in heat content over particular paths. Subsurface drifters can be acoustically tracked. Acoustics provides a window to the sea at many levels.

Who would benefit by reading this book? The foreword says the book is written for the non-expert in marine acoustics who may be a scientist from a neighbouring faculty e.g. researchers and lecturers in oceanography, marine geology, environmental sciences, geophysics and related fields. Is it likely to be useful for that audience? My assessment is on the whole quite positive, while noting some gaps in coverage.

The book barely mentions the use of acoustic backscatter from single beam or swath systems to assist in characterisation of sediment type or benthic habitat, but this is an ongoing topic of intense interest in many fields e.g. environmental protection, geological classification of surficial sediments, and so on. The book is certainly not a text book on marine acoustics, nor on swath sonar. It does not aim to be such, but details on how the principal sources of the images (swath sonars) work, is glossed over. If you do not know how they work before you read the book, then you will probably not know afterwards. Nor will you know much about their capabilities, weaknesses, data processing requirements, nor even that different types of multibeam sonars exist which use different principles of operation. The book considers there to be four lines of evolution of ocean echosounding: the multibeam echosounder, sidescan sonar, sub-bottom profiler, and the acoustic Doppler current profiler. I can think of at least two more marine acoustic technologies amenable to visualisation techniques worth a mention: monitoring of suspended sediment concentrations through the water column by MHz frequency backscatter systems, and Acoustic Daylight, the process of using ambient noise to image objects. Local examples of these two applications for Acoustics Australia readers may be found in Hamilton (1998) and Readhead (2001). Acoustic cameras have also been produced, one type of which is essentially a miniature multibeam sounder. Surprisingly, this receives no mention.

The book could also do with some more editing. For example, it is stated more than once that *Posidonia* seagrass is found only in the Mediterranean, but this should be *Posidonia oceanica*. The book eschews equations, and the qualitative descriptions of acoustic processes offered in introductory sections are consequently sometimes rather opaque. The introductory sections strive to be more than bare science, and are more interesting than straight text books, but they are rather long, and there seems to be a breathless rush to get through them to the main issues while throwing in as much as possible. Some reconstruction of these sections would benefit the reader. I find the lack of simple equations difficult to understand, given that professionals are the major intended audience. If explanations of acoustical processes and principles of the various sonars described are required, then companion texts are needed.

Publicity for this book says it is the first of its kind, offering a comprehensive overview of acoustic imaging applications in the various fields of marine research, including utilization, surveillance, and protection, and that it is written to be accessible to professionals in diverse related fields. I would say that it largely succeeds in these aims. It is in the category of books which make for compulsive reading. It is well worth adding to a research library for the overviews it provides, largely by way of case studies, particularly of the geological processes. For many, the illustrations and descriptions of the various underwater processes will be quite a revelation, however, this visually compelling book is far more than a collection of pretty pictures. The book should be a useful addition for those interested in applications of acoustics to marine science.

References

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Readhead, M. (2001). Acoustic daylight – using ambient noise to see underwater. Acoustics Australia 29(2), 63-68.

L.J. Hamilton

Les Hamilton is a Senior Researcher for the Defence Science & Technology Organisation (DSTO). He works in various aspects of physical oceanography, including acoustic seabed classification, the process of using echoes stimulated by single beam, sidescan, and multibeam sonars, to characterise seabed sediments.

Managing Noise and Vibration at Work: A practical guide to assessment, measurement and control

Tim South

Elsevier, 2004, 268 pages (soft cover). ISBN: 0 7506 6342 1
Price approx A$77 from Elsevier, www.elsevier.com.au,

Tim South has been teaching the courses on workplace noise and vibration assessment as part of the UK Institute of Acoustics programs. The book is a comprehensive text and reference for those courses. It has a well planned structure and a practical approach to presenting the material. In the preface, the author states that he spent some time considering how much mathematics to include in the book. In my opinion he has achieved the right balance.

Part 1 on Noise commences with noise, human response and measurement then proceeds with methods for assessment of exposure. It is at this point a problem arises for the Australian reader for all the references are to the UK or EU standards, legislation and directives. Similarly, Parts 2 and 3 on vibration commence with the concepts and proceed to assessment methods. As there is effectively no legislation in Australia regarding human vibration, the UK and EU criteria are an appropriate default reference. Part 4 deals with methods for reducing noise and vibration risks.

The strengths of this book are that it includes both consideration of noise and vibration and that it gives practical examples including photographs of measurement, assessment and control in the workplace. With the reservation that the Australian methods of assessment are not discussed, I recommend this book to anyone involved with managing workplace noise and vibration.

Enhancing Occupational Safety and Health

Geoff Taylor, Kellie Easter and Roy Hegney

Elsevier, 2004, 600 pages (soft cover). ISBN: 0 7506 6197 6
Price approx A$99 from Elsevier, www.elsevier.com.au,

This book aims to address all aspects of occupational health and safety (OHS). As such it provides little more than an overview of occupational noise and vibration with only 10 pages on workplace noise and 2 pages on vibration. The authors state that the book has been written as a practical guide for students and those working in OHS. The three authors are based in Perth and the Australian link is promoted. However the authors state that the book has been written to appeal to "*anyone dealing with health and safety in any country which has a substantial English-speaking connection.*" They rely very much on the UK documentation and unfortunately do not include any reference to relevant Australian standards or codes. Further, there is no reference to the EU directive for occupational noise which one would expect to see in a new book. The coverage of vibration is very superficial and again there is no mention of the EU directive for workplace vibration. It is obvious that a guide to all aspects of OHS cannot cover every aspect in detail but it should direct the reader to sources of additional information.

This book does provide a good overview of management of the various aspects of OHS including the background on legislation etc. As such it would be interesting background reading for acoustic consultants who are working extensively in the area of occupational noise assessment.

Marion Burgess

Marion Burgess is a Research Officer for the University of NSW at the Australian Defence Force Academy in Canberra. She has experience with occupational noise surveys and with presenting courses on the topic.

ICA 2010

ICA2007 Madrid,
2-7 September, 2007

This is the fourth in a series of regular items in the lead up to ICA in Sydney in 2010.

The previous item in this series (Dec 2005 issue) summarised the purpose and governance of the International Commission on Acoustics and the nature of the primary activity, namely the International Conference on Acoustics. This item provides information on the Congress to be held in 2007 in Madrid - the precursor to ICA2010 in Sydney.

The International Commission on Acoustics (ICA) is held every three years and the Commission takes great care with the selection of the venue for the congress and the capability of the organizing committee to meet the high standard of technical quality that the ICA has become known for. As an acoustics congress the ICA is similar to a number of acoustics conferences being held simultaneously.

The ICA 2007 to be held in Madrid, Spain from 2-7 September 2007 has the theme "Acoustics for the 21st Century". It will follow the tradition of an ICA congress with a full program over 5 days. The program will be confirmed over the coming year and it will comprise at least one plenary lecture each day. The plenary lecturers will be selected for their ability to provide an interesting and stimulating review of the latest technical advances in their particular area of acoustics. Thus attending the plenary lectures is one way to broaden your knowledge of acoustics. Distinguished lectures scattered throughout the day may be in parallel providing a choice. The remainder of the program comprises parallel sessions in the various topic areas either in the form of structured sessions and contributed papers. In addition there will be poster sessions, demonstration sessions and a technical exhibition.

Following the ICA there will be satellite symposia which will provide a focus for two specific topic areas in acoustics. ISMA 2007 will be the International Symposium on Musical Acoustics and will be held from 9 to 12 September in Barcelona. ISRA will be the International Symposium on Room Acoustics and will be held from 9 to 12 September in Sevilla. Registrations for these satellite conferences can be made in conjunction with the ICA or independently.

September 2007 may seem a long time away but the planning for attendance at this important event on the Acoustics calendar needs to start soon. The date for abstract submission date is 30 January 2007 and the full papers will be due in May.

The Australian Acoustical Society is encouraging a high representation at this congress to show case the high standard of acoustics in our region and to encourage a high international participation at the ICA 2010 in Sydney. To further encourage attendance, the AAS has established two traveling grants for attending ICA 2007. Each grant is worth $1,000 and the application details can be found from www.acoustics.asn.au. The main assessment will be based on how attendance at ICA 2007 will enhance the applicants career and/or studies in acoustics and how participation at the ICA will enhance the profile of acoustics in our region. The deadline for application for these grants is 30 October 2006.

Participation in ICA 2007 provides a great opportunity to justify a trip to Europe in late summer and to extend to stay to enjoy a holiday in the region. Spain is an exciting, historical, fascinating country and a wonderful holiday destination and September is just after the main European summer season so travel becomes a little easier. The organizers of ICA2010 in Sydney hope to see many Australians and New Zealanders at ICA 2007 in Madrid.

Marion Burgess

Diary

2006

18-20 September, Leuven
ISMA2006, Noise and Vibration Engineering Conference
http://www.isma-isaac.be

18 - 20 September, Adelaide
ACTIVE 2006
http://www.active2006.com

18 - 20 September, Bristol.
12th Int Conference on Low Frequency Noise & Vib and Control.
http://www.lowfrequency2006.org

18 - 21 September, Pittsburgh
INTERSPEECH 2006 - ICSLP.
www.interspeech2006.org

27-29 September, Sydney
8th National Injury Prevention Conference
secretariat@aipn.com.au

3 - 6 October, Vancouver
IEEE International Ultrasonics Symposium.
http://www.ieee-ultrasonics2006.org

19-20 October, Cincinnati
Noise induced hearing loss in children at work and play
http://www.hearingconservation.org/conf_childrenconf.html

20 – 22 November Christchurch
1st Joint Australian/New Zealand Acoustical Societies Conference
"Noise of Progress"
http://www.conference.co.nz/index.cfm/acoustics06

28 November - 02 December, Honolulu
Acoustical Soc of America & Acoustical Soc of Japan Joint Meeting.
http://asa.aip.org

3-6 December, Honolulu
Inter-Noise 2006
Engineering a Quieter World
www.internoise2006.org

3-8 December, Brisbane
AIP Conference
www.aipc2006.com

2007

10 - 12 April, Loughboro
4th Int Conf on Bio-Acoustics.
http://www.ioa.org.uk

16 - 18 April, Kanagawa
29th Intl Symp on Acoustical Imaging.
http://publicweb.shonan-it.ac.jp/ai29/AI29.html

16 - 20 April, Honolulu
IEEE Intl Conf on Acoustics, Speech, and Signal
Processing (IEEE ICASSP 2007).
www.icassp2007.org

16 - 20 May, Honolulu,
IEEE Int Conf on Acoustics, Speech & Signal Processing (IEEE ICASSP 2007)
http://www.icassp2007.org

03 - 07 June, Bologna
11th Int Conf on Hand-Arm Vibration.
www.associazioneitalianadiacustica.it/HAV2007/index.htm

18 - 21 June, Aberdeen
Oceans07 Conf.
www.oceans07ieeeaberdeen.org

25 - 29 June, Heraklion
2nd Int Conf Underwater Acoustic Measurements: Technologies and Results.
www.uam2007.gr

9-12 July, Cairns
ICSV14 incorporating AAS Annual Conference
www.icsv14.com

26-29 August, Istanbul.
Inter-noise 2007.
http://www.internoise2007.org.tr

27-31 August, Antwerp
INTERSPEECH 2007.
conf@isca-speech.org

2-7 September, Madrid
ICA2007
www.ica2007madrid.org

9-12 September, Barcelona.
Symposium on Musical Acoustics (ISMA2007)
www.ica2007madrid.org

9 - 12 September, Sevilla
Symposium on Room Acoustics
www.ica2007madrid.org

17-19 September, Lyon
Fan noise 2007
www.fannoise2007.org

28 - 31 October, New York
IEEE Int Ultrasonics Symposium.
www.ieee-uffc.org/ulmain.asp?page=symposia

2008

30 June - 4 July, Paris,
Acoustis'08 Paris
http://www.acoustics08-Paris.org

28 July - 1 August, Mashantucket
ICBEN 9 Int Cong Noise as a Public Health Problem.
www.icben.org

22 - 26 September, Brisbane
INTERSPEECH 2008 - 10th Intl Conf on Spoken Language Processing (ICSLP).
www.interspeech2008.org

2010

23-27 August, Sydney
ICA2010
http://www.ica2010sydney.org/

Meeting dates can change so please ensure you check the www pages. Meeting Calendars are available on http://www.icacommission.org

Late News

500 Australian standards to be cut.

Standards Australia is to withdraw 500 standards of which 29 are specifically acoustics and vibration and others may well include relevant sections on acoustics. The listing of those to be culled is http://www.standards.org.au/cat.asp?catid=135&ContentId=79

For some of these withdrawal may well be appropriate as they are outdated or have been substantially replaced by other standards. However Members may be surprised at the withdrawal of those which are referenced in specifications and for which there will be no equivalent AS - for example the entire AS 1217 series on measuring sound power!

If you are concerned about the deletion of any of these please advise Standards Australia directly, copying your message to GeneralSecretary@acoustics.asn.au so that the AAS can keep track of the comments and provide additional follow up as necessary.

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* Payment of annual subscription
* Proceedings of annual conferences

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