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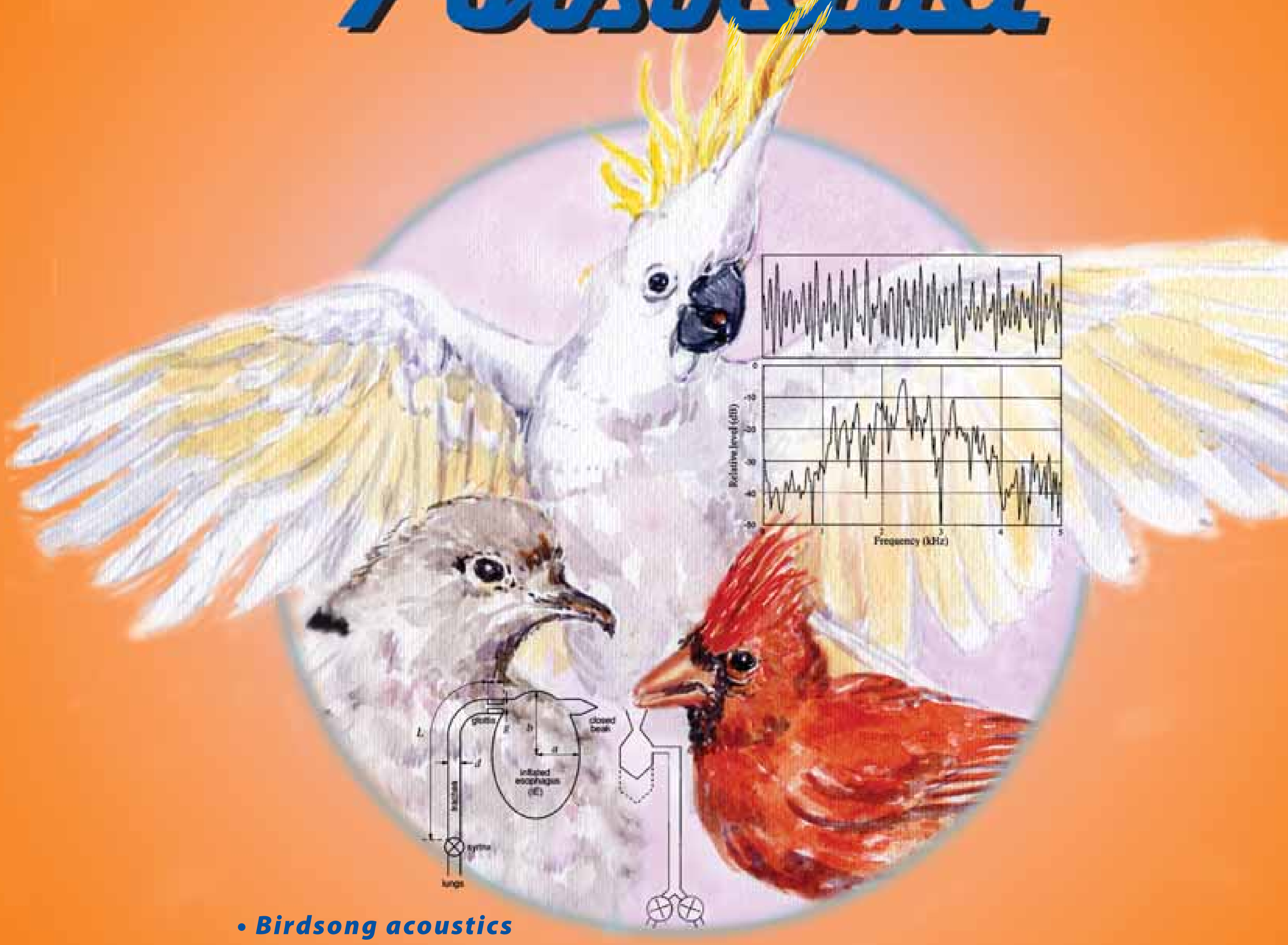

- *Birdsong acoustics*
- *Monitoring aircraft noise*
- *Acoustic performance of foam-filled mufflers*
- *Multiple-leaf microperforated panel sound absorbers*
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Message from the President

As this issue is going to press in time for the 20th ICA and will be seen by many of our international visitors, I am very glad to take this opportunity to welcome all our guests to Sydney for this very exciting Conference. I hope that our international visitors will be able to see first-hand the very high level of acoustics that Australia offers and will also enjoy their stay in Sydney and any other destinations in Australia that they may choose to travel to. A lot of work has gone into this Conference and I want to especially thank the Conference Chair, Marion Burgess, and her dedicated team. As you will no doubt experience, their efforts will have resulted in a very successful conference. And don't forget the associated conferences ISMA in Katoomba, ISRA in Melbourne and ISSA in Auckland.

I would also like to take this opportunity to thank Byron Martin, our outgoing General Secretary and welcome Richard Booker as the new GS. Byron has been a stalwart of the Society, acting in many positions in the SA Division and also acting as Federal treasurer for many years. He recently also did a stint as GS. We wish Byron all the best for the future and hope to see you around at future AAS functions. Richard will have big shoes to fill following David Watkins and Byron and we wish him well in this new appointment.

I recently attended a technical meeting in which road traffic noise and criteria were discussed. One of the things that crossed my mind after the presentation is that we still don't have a unified approach across Australia (and New Zealand) with differing criteria adopted in differing States. It seems a little strange that we can't all agree on what traffic noise criteria should be. In addition, yet once more, the question of engine brake noise and what to do about the resulting annoyance came up. I remember one of my first projects in consulting back in 1979 was a project documenting engine brake noise for what was then the NSW SPCC with a view to determining how best to manage this problem. Have we not progressed?

Another question that cropped up was the issue of accreditation of acoustic consultants. Some Government departments and councils request that members or companies proposing to do work for them be accredited in some way. The AAS does not accredit its members. We do ask members to "self nominate" areas where they have professional expertise but we do not endorse this self proclaimed expertise. Should the Society move to creating some form of assessment of professional experience which will allow its members to be accredited? This is not an easy step and would involve a lot of work. The question is whether members feel such a move is warranted and whether you have any ideas how to implement such a scheme. Please advise your thoughts to me by email. I hope to see many of you at the 20th ICA in Sydney.

Norm Broner

Message from the Editor

Welcome to the August issue of the Acoustics Australia journal and my first as editor. I'd like to thank the previous editorial team – Joe Wolfe, John Smith, Marion Burgess and Emery Schubert, for their excellence and dedication in presenting the journal over the last five years. I hope that I can do justice to this demanding and distinguished role. To briefly introduce myself, I have a keen interest in vibrations and acoustics which stems from my undergraduate and PhD studies in mechanical engineering at the University of Western Australia. A chance discovery of the book *Structure-borne sound* by Cremer, Heckl, Ungar was the pivotal point in my postgraduate studies and in launching an academic career in this field. In 1999, I took up a lecturing position at James Cook University and in 2003 I joined the University of New South Wales as a senior lecturer, primarily to teach vibrations and acoustics courses in the School of Mechanical and Manufacturing Engineering. Acoustics and noise continue to play an increasingly larger role in my life both at home and at work. On a professional level, my research interests have progressed from topics in structural vibration to those directly associated with noise and its control, primarily for signature management of maritime platforms and optimisation of mufflers for improved acoustic performance. In the meantime, at home I enjoy the nocturnal calls of young children and the fortune of living directly under one of Sydney's noisy flight paths!

As I started to write this editorial message I was wondering about the history of the journal – when it started and who the previous editors were. Thankfully I didn't need to wonder for very long as a quick search revealed an article by Howard Pollard and Marion Burgess titled 'History of the Journal of the Australian Acoustical Society', *Acoustics Australia*, Vol. 28, No. 3, pp. 87-88 (2000). The journal was established in 1972 with the title of *The Bulletin of the Australian Acoustical Society*. In 1985 the title of the journal was changed to *Acoustics Australia*. Whilst the journal originated in NSW, it was initially intended that alternative issues would be produced by the NSW and Victoria Divisions, however this posed problems associated with the journal production. In the first few years of the journal there were four issues produced to coincide with the seasons – autumn, winter, spring and summer – each issue for the handsome price of $1.00! Over the years, the chief editors include; Ted (Edward) Weston (from 1972) Richard Heggie and Fergus Fricke (from 1975), Robin Alfredson (1979), Rob Law (1980), Don Gibson (1981), Howard Pollard (from 1982), Neville Fletcher (from 1993) and Joe Wolfe (2005-2010). These chief editors worked with a team of three or four, thus numerous AAS members have contributed to the production of the journal over the last 4 decades. *{As I have only listed the chief editors I apologise if I have missed someone's significant contribution to the journal.}*

A significant step forward for this journal (as described in the News section on page 99) is that all back copies of the Acoustics Australia journal have been scanned and are available for download from the AAS website. For this August issue, I'd like to thank Marion Burgess for providing the news items (on top of organising ICA 2010!) and John Smith for general editing. Thanks are also extended to the business manager Leigh Wallbank, Tracy Gowen (the new journal treasurer) and Louise Fraenkel of Cliff Lewis Printing. Most importantly, I'd like to thank all authors for their contributions as well as the reviewers of the articles. I gladly welcome submissions to the journal. I hope you enjoy reading this issue and look forward to your feedback.

Nicole Kessissoglou

ACOUSTICAL BACKGROUND TO THE MANY VARIETIES OF BIRDSONG

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Birdsong can be broadly classified into two categories: biphonic in which two different notes can be sung simultaneously, and monophonic in which only one note is sung. Monophonic song can be further divided into rich harmonic song, single-frequency song, and chaotic song. While some of these distinctions arise from clear anatomical features of the bird, others are more subtle and pose problems for physical scientists working in collaboration with biologists. This paper provides information on the physics and acoustics underlying these differences and shows how quantitative predictions can be made.

INTRODUCTION

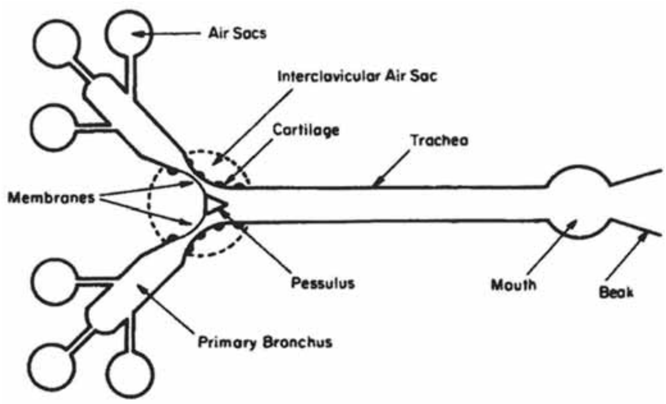
Birdsong is a biological phenomenon of general interest because of its variety and auditory attractiveness, a classic survey having been given in a book by Greenewalt [1] and a collection of articles edited by Kroodsma and Miller [2] and a more recent survey by Marler and Slabbekoorn [3]. In biology 'birdsong' can be produced only by 'songbirds' which are defined to be birds with five syringeal muscles, other birds producing 'calls', or more generically 'vocalisations'. In this paper I will not be greatly concerned with this subtle distinction – I hope it will not annoy the biologists!

Some birds produce songs with individual notes or 'syllables' that are almost pure tone in some species but rich in harmonics in others, with the spectrum shaped into emphasised formant bands like the vowels of human speech. The calls of some birds such as cockatoos, however, have a truly chaotic waveform rather than simply a broad spectrum [4]. Finally, those species known as 'songbirds' are actually able to sing two different notes at the same time, a skill for which there is a simple anatomical explanation as is discussed later.

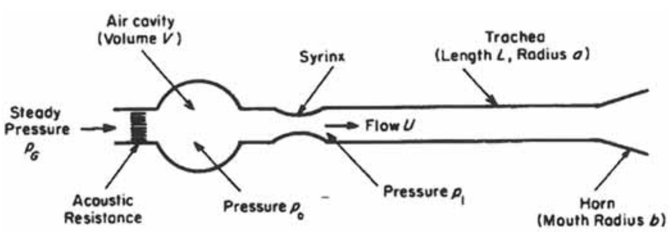
Birdsong is a field of study in which there can be great advantages in collaboration between biological and physical scientists, because both the physiology and the acoustics involved are quite complex. My own involvement in such collaborations has provided the stimulus for a detailed analysis from an acoustic viewpoint [5] which provides a quantitative model for sound production in various anatomical cases. More recent collaborations have examined various special cases, some of which will be discussed here.

BASIC SOUND PRODUCTION PHYSIOLOGY

The basic mechanism for sound production is essentially the same in all air-breathing animals – perhaps omitting those that live under water. Air stored in the lungs at an overpressure of order 1 kPa (10 cm water-gauge) is exhaled through the vocal tract which consists of two tubes, the 'bronchi', leading through a single exit tube, the 'trachea', to the mouth. The vocal tract in mammals contains an adjustable and flexible constriction, the larynx, near the top of the trachea that can be maintained in vibration by the combination of aerodynamic and elastic forces acting upon it. In some birds the anatomy is similar, but in song-birds there is one such valve in each of the bronchial tubes rather than a single one in the trachea, as shown in Fig. 1(a).



(a)



(b)

Figure 1. (a) Vocal anatomy of a songbird, showing the two valves of the syrinx. (b) A simplified single-valve model used for calculation of the behaviour. (From ref. [5]) The symbols show the quantities used in the model analysis.

In the case of humans and other mammals, this valve is known as the larynx and contains two opposed flaps termed 'vocal folds' which are constricted by tension in two 'vocal cords'. The fundamental vibration frequency is determined by the mass and tension of the vocal folds and is about proportional to the mass of the animal to the power –0.4, a slight variation on simple inverse scaling with length [6]. This general rule applies to birds as well as to other animals up to the size of elephants. In the case of birds, the vocal organ is called the 'syrinx', and two different anatomies exist, as mentioned above. This division of bird species explains why songbirds can sing two notes at once, since the two bronchial valves of the syrinx are largely independent. Operation of the songbird vocal system is essentially the same as that of other birds except for two things. Firstly, the distance from the syringeal valve to the open beak is much larger than is the case for a similar bird with the syringeal valve in the trachea. This means that the vocal tract resonances are lower in frequency for a songbird than they are for an ordinary bird of the same general size, a feature that will change the tonal properties of the sound in a way that has not yet been explored in detail. Because birds are generally small anyway, giving resonances of rather high frequency, this may give a more 'mellow' tone to the songbird song. Secondly, there is the possibility of acoustic interaction between the two valves through the air column. Again this has not been explored in detail, and some songbirds avoid the possible problem by singing high-pitched notes from one bronchus and low-pitched notes from the other with little temporal overlap between them.

OPERATION OF THE SYRINGEAL VALVE

A good understanding of the basic acoustics of bird song can be reached by studying the behaviour of a single syringeal valve linked to an upstream air reservoir and a downstream cylinder representing the trachea [5]. The subtleties of the influence of the tongue and beak opening upon the tracheal resonances can also be explored [7] but this is a refinement of the basic model. Examination of such valve models for the human voice has a long history, an excellent summary being given in a book by Sundberg [8]. A 'source/filter' approach in which the vibration of the vocal folds provides an independent source and the resonances of the vocal tract a filter appears to be adequate to explain most aspects and certainly simplifies the analysis. The first resonance of the human vocal tract is at about 500 Hz, with higher resonances near 1500 and 2500 Hz, all these frequencies being widely adjustable by changes in the mouth opening and tongue position. The fundamental voiced frequency is usually well below this so the resonances produce bands of emphasised frequencies known as formants. A particularly interesting case is that of 'coloratura' sopranos who can sing notes with fundamental frequency as high as 2 kHz. To achieve this, the singer changes tongue and lip configurations so that one of the vocal tract resonances matches the frequency of the note being sung, thus reinforcing it [9]. We will see later that something similar happens in the songs of some bird species.

Treating the syringeal valve as a simple isolated mechanism, it is still necessary to take account of the influence of small pressure oscillations in the upstream reservoir, since the pressure there is influenced by the flow through the valve. The valve can then be maintained in oscillation by the influence of up-stream and downstream pressure variations and their effects both through simple pressure exposure to the upstream portion of the valve and also by Bernoulli flow through the valve aperture. The general features of such behaviour are well understood [10] and apply to human vocalisation and to lip motion in the playing of trumpets as well as to birds. A detailed model [5] for the song of birds with rich-harmonic vocalisations, as sketched in Fig. 1(b), predicts results, as shown in Fig. 2, that are in good agreement with observation. For the parameter values selected, the syringeal valve is found to close completely on each cycle at a frequency of about 200 Hz and there are emphasised formant bands near 1 kHz and 2 kHz, which correspond to resonances of the vocal tract [7]. These formants are important to sound quality but are not essential for the production of sound. The radiated sound will retain these formant features, but its waveform will be much more 'continuous' in structure than are the pressure and flow in the figure. Each of the two syringeal valves in the song-bird anatomy of Fig. 1(a) will function in a very similar manner.

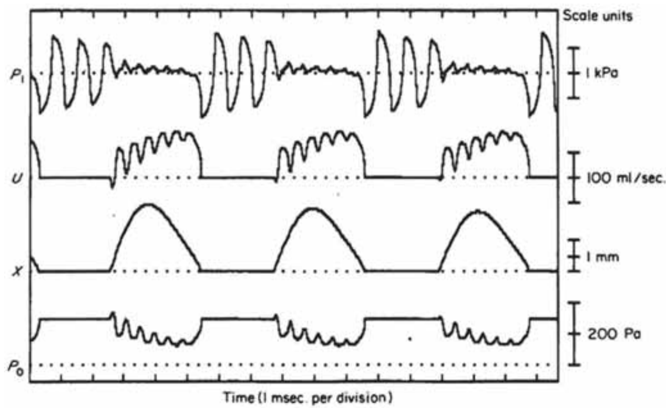


Figure 2. Calculated acoustic behaviour of a syringeal valve as in Fig. 1(b). Variable p_1 is the acoustic pressure in the trachea above the valve, U is the acoustic flow through the valve, x is the valve opening, and p_0 is the pressure in the reservoir below the valve. (From ref. [5])

PURE-TONE SONG

One puzzling feature of the model proposed for the production of normal bird song is the fact that it fails to provide an explanation for the almost pure-tone songs produced by some birds. The only way in which this can be achieved with the model is to use a very low blowing pressure so that the valve fails to close on each cycle, but this results in an extremely low sound level, which is not what is observed in practice.

Two alternative mechanisms for pure-tone song production have been identified. In the first case [11], which applies to birds such as the Northern Cardinal *Cardinalis cardinalis* which are able to produce sweeping whistle-like sounds over a 1–8 kHz frequency range in a single 'syllable' of song, the mechanism relies upon a tuneable resonant cavity near the junction of the trachea and the mouth as shown in Fig. 3(a). The cavity itself is actually in the oesophagus, which leads to the stomach, and is referred to as an 'oropharyngeal esophageal cavity' or OEC. This cavity can be stretched over a wide volume range by attached muscles. The bird then essentially tunes this vocal tract resonance to match the frequency of the note being sung, in much the same way as do human coloratura sopranos and with the same result, though with different anatomical features. A model for this vocalisation using the electric network analog shown in Fig. 3(b) is able to produce good agreement with observations [12].

The second mechanism, which applies to the 'coo' sound of doves, is quite different, since doves, particularly Ring doves, sing with their beaks closed. The 'coo' itself is a rather short syllable, lasting not much more than one second, and typically has a frequency around 600 Hz that remains almost steady. Observations show that the dove expands a sac in its neck when producing the call, and this sac expands somewhat during the call since the bird is exhaling into it. Once again it is possible to devise a theoretical model [13] to accommodate this anatomy and behaviour, though this model is rather different from that for the Cardinal since the beak is closed and air is simply transferred from the lungs to the vocal sac, which is once more located in the oesophagus near its junction with the mouth. The resonance of prime importance in this case is almost that of a Helmholtz resonator comprising the sac and the tubular connection through the glottis between it and the trachea, though this has to be modified to allow for vibration of the thin walls of the sac. The great difference between the dove and the Cardinal is that the dove's beak is closed, so that sound radiation is not through the beak but directly from the vibrating thin walls of the inflated oesophageal sac. Surprisingly, perhaps, the resonant frequency changes very little as the sac expands, because the decrease in the mass per unit area of the walls nearly compensates for the increase in sac volume. The song therefore maintains a nearly constant pitch.

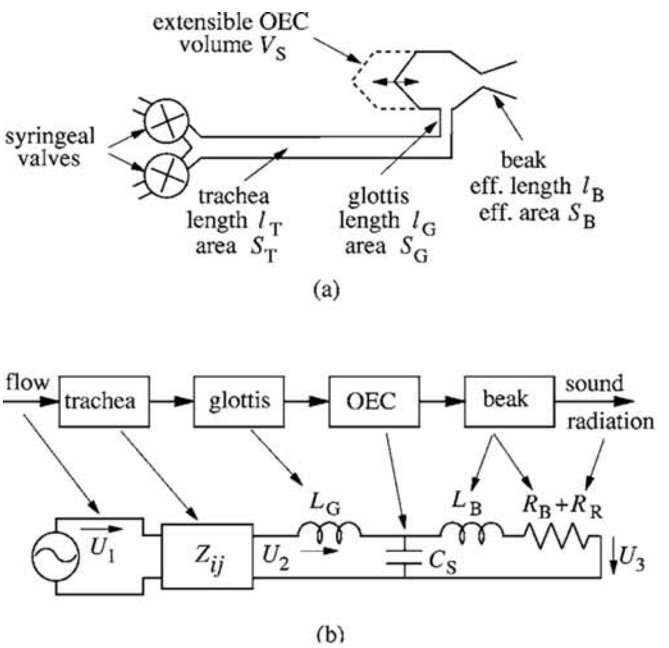


Figure 3. (a) Anatomy of the vocal tract of a pure-tone songbird such as the Cardinal. (b) Simple electrical network model used to analyse the vocal tract behaviour. The bird adjusts the resonance of the OEC and beak aperture to match the frequency of the air flow produced by the syringeal valves. (From ref. [12])

CHAOTIC VOCALISATION

The third type of song of interest can be termed chaotic song, because the waveform is genuinely chaotic rather than being just random noise [4] and typically has a Lyapunov exponent of 0.28 ± 0.06 and a correlation dimension in the range 3.2 to 3.8, values that are comparable with those for standard computed chaotic signals. Australian birds with this type of song are mostly cockatoos, the most prominent being the sulphur-crested cockatoo *Cacatua galerita,* which is also known for its beautiful appearance and its destructive behaviour when a group flies into a tree and pulls off any flowers and new shoots, or even when they attack the rubber gaskets in street lights. The cry of these cockatoos is also very loud – about 80 dB at a distance of 10 metres, which corresponds to a radiated power of about 100 mW. The spectral distribution of the sound is broad, with a maximum near 2.5 kHz and a 10 dB bandwidth from 1 to about 3.5 kHz, so that it sounds very loud and 'harsh' to human ears. An example of the waveform and spectrum is shown in Fig. 4.

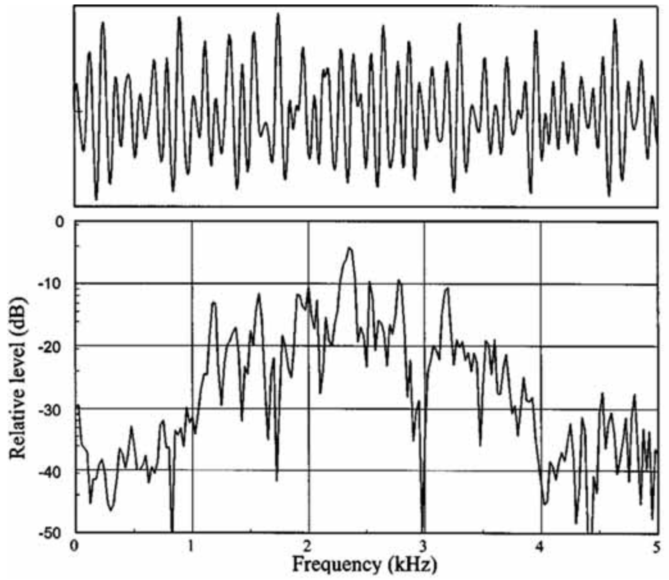


Figure 4. Waveform and spectrum of a short sample of the call of a sulphur-crested cockatoo *Cacatua galerita*. Plots for the gang-gang cockatoo are very similar. (From ref. [4])

Modelling of the production of such a sound is difficult because of lack of information about the detailed anatomy of the vocal valve in these birds. The standard model described above, however, in which the restoring force on the valve membranes under pressure is proportional to its deflection but there is a large increase in both restoring force and damping when the two membranes come into contact, does show some chaotic behaviour when the lung pressure is made very high so that the nonlinearity is emphasised [4]. This suggests that an extended model in which a nonlinear term is added to the membrane restoring force might adequately reproduce the observed chaotic behaviour at lower lung pressures. It is easy to suggest the origin of such a nonlinear term because biological structures are rarely linear in behaviour once the stretching or bending becomes nontrivial. This is because the structures are complex assemblies of cells with quite different elastic properties. To date no anatomical data on this question has been available.

SONG STRUCTURE AND MIMICRY

Setting aside the detailed acoustics of sound production in birds, there is a wealth of information encoded in the songs that has provoked great interest among behavioural biologists. While the information content of vocalisations can be formally defined and measured [14], there is much more interest in the structure of the 'conversation' of birds, conveying information to each other, and in the way in which some species are adept at imitating the calls of birds or animals of other species, or even the sounds of non-biological sources such as chainsaws.

Various species of parrots and cockatoos have been known for a very long time to be able to imitate human speech by tuning the formants and articulation patterns to match human phrases such as "Pretty Polly". This is perhaps not surprising from an acoustical point of view, since the vocal formants, except perhaps the first, can be appropriately tuned by varying tongue position and beak opening [7], and there are generally clear rewards provided by their human hosts for those birds that excel at this imitation.

Of particular interest is the vocal behaviour of several species of Australian birds, particularly the lyrebird *Menura novaehollandiae*, which is adept at imitating a huge variety of birdsongs from other species as well as mechanical and other sounds [15], the pied butcherbird [16] and the magpie [17]. A true understanding of the reasons for this mimicry behaviour involves physiology and psychology as well as acoustics [18] and is outside the scope of this review.

CONCLUSIONS

Birdsong is one of the most interesting and varied forms of vocalisation produced by any animals and almost rivals human speech and song in complexity. Even the acoustics of sound production by birds is complex and varied, as has been summarised here. Collaborative studies between many biological and physical scientists have now achieved a basic understanding of the subject, but there is an immense field of research available on the information content and 'cultural' background involved as well as on the vocal anatomy and physiology of individual bird species.

REFERENCES

I must apologise for the fact that the reference list given below is largely restricted to studies with which I have had a personal involvement. The biological literature on birdsong is very large and I cannot hope to refer to it adequately here. References 1–3 have extensive citation lists. I am grateful to one of the Referees for drawing my attention to several additional recent publications.

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MONITORING AIRCRAFT NOISE LEVELS TO THE SIDE OF FLIGHT PATHS

Marion Burgess and Matthew McCarty
Acoustics and Vibration Unit, University of New South Wales at the Australian Defence Force Academy, Canberra, ACT 2600

Assessment of aircraft noise levels near to the main flight paths has been successfully implemented at many airports and a good indication of the aircraft noise levels at those locations is obtained. Monitoring of aircraft noise levels away from the main flight paths is sometimes required to meet community demands and is more challenging as the aircraft noise may not be clearly defined above the background noise level in the area. This paper reviews the recommendations for monitoring aircraft noise in such *'acoustically unfavourable'* locations with particular reference to the findings from analysis of data from such a placement. The outcomes indicate that more sophisticated analysis is required for such placements to achieve a fair and accurate assessment of the aircraft noise levels.

INTRODUCTION

The noise impact from aircraft operations is of concern to those government agencies which have the responsibility for managing environmental noise and to airport operators which have the goal to have maximum utilisation of the airport. In Australia, the basic guidance for planning relating to aircraft noise impact is set out in the Australian Standard "Aircraft noise intrusion – Building siting and construction" [1]. This standard uses the Australian Noise Exposure Forecast system (ANEF) which leads to contour lines on a map around the airport based on the information on the future operations of the airport. The ANEF contour information is used in planning considerations for future developments and to identify those areas which may be subject to mitigation from expansion or changes in the airport operations. As discussed in the paper on "Expanding Ways to Describe and Assess Aircraft Noise" [2], the ANEF system has limitations in community consultation as it does not provide guidance on the actual noise level for different types of operations.

Monitoring aircraft noise around airports is required to obtain information on actual aircraft noise levels. The use of this aircraft noise level data includes identification of those aircraft which do not comply with applicable noise abatement procedures and which may be penalised for such infringements. The data is also used to monitor the actual noise exposure at the location for comparison with estimations, such as that from the ANEF contours, and for future planning.

The basis of any aircraft noise monitoring system is a noise monitoring terminal, which consists of an all-weather microphone located on top of a mast with an attached data logger. The noise level is continuously measured and then transmitted to a central computer for processing. This approach to monitoring was proposed in the 1970s [3] and developed considerably over subsequent decades. The noise monitor will record the data on all the noise in the area so post processing to extract that data that has originated from an aircraft noise event is required. In a basic system this is achieved by rejecting noise events that do not satisfy the acoustic parameters that have been determined to be applicable to an aircraft noise event. The primary parameter is that the noise level of the event must be above a threshold noise level. Then parameters relating to the time profile of the noise event such as the minimum and maximum rise time and fall time of the event are used to extract those most likely to be aircraft noise events.

An improvement in the identification of the aircraft noise events can be achieved with a noise and flight path monitoring system (NFPMS). In such systems, flight path data from the airport on each aircraft movement is used for correlation with the noise event data that has first satisfied the acoustic parameters. The flight path data is usually in the form of radar tracks and altitudes. A 'correlation area' is defined around the noise monitor and checks are made to see if an aircraft was within that area at the time of the potential aircraft noise event. If there were no aircraft movements within the predefined area, the noise event is rejected as not being due to an aircraft. If there was an aircraft movement within the correlation area any noise event that meets the acoustic parameters is considered to be caused by that aircraft.

Most major airports have some form of aircraft noise monitoring installed and there are various commercial systems available. The ANOMS system from Lochard [4] is one such system and is widely implemented around Australian airports. The system is managed by Airservices Australia, which states that the data is used to: [5]

- *determine the contribution of aircraft to overall noise exposure*
- *detect occurrences of excessive noise levels from aircraft operations*
- *assess the effects of operational and administrative procedures for noise control and compliance with these procedures*
- *assist in planning of airspace usage*
- *validate noise forecasts and forecasting techniques*
- *assist relevant authorities in land use planning for developments on areas in the vicinity of an airport*
- *generate reports and provide responses to questions from Government, industry organisations, community groups and individuals.*

Each quarter, reports on the findings from each terminal are made publically available from the Airservices Australia website.

REQUIREMENTS FOR UNATTENDED MONITORING OF AIRCRAFT NOISE

The International Standards Organisation (ISO) released a standard in 2010 on *"Unattended monitoring of aircraft sound in the vicinity of airports"* [6]. This standard aims to specify the "requirements for reliable measurements of aircraft sound" and includes guidance on installations, performance specifications' quantities to be determined, reporting and procedure for determining uncertainty in reported data.

In most cases the monitoring stations are located under or close to the flight paths where the noise from each aircraft movement is well above the general community noise levels, for example positions 1 and 2 in Figure 1. When the system includes flight path data it is not difficult to comply with the requirements of the standard and correctly identify aircraft noise levels for such locations. However for many airports there are increasing complaints about aircraft noise impact from residents at some distance from the main flight paths, for example positions 3 and 4 in Figure 1. To properly assess these concerns the regulatory authority needs quantitative data and so may seek to locate noise monitoring terminals within these residential areas. It is then more difficult to comply with the requirements of the standard and correctly identify aircraft noise levels. Some of the key requirements of the standard which highlight the increased difficulty are discussed in the following sections.

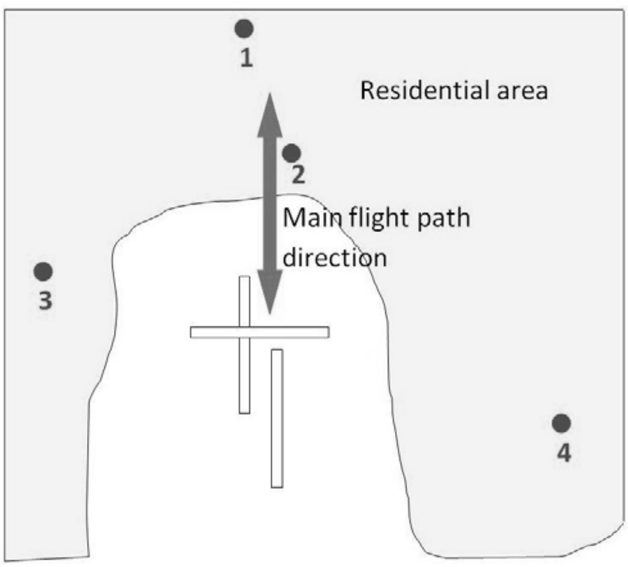

Figure 1. Schematic diagram showing a residential area around an airport with potential locations for noise monitoring terminals. Positions 1 and 2 are close to the flight path and the noise from aircraft would likely be the dominant noise. Positions 3 and 4 are in the residential areas and separating aircraft noise events from general community noise events is more challenging.

Site selection

The standard specifies that a site be selected such that the maximum sound levels from the quietest aircraft to be detected are 15 dB greater than the residual long-time-average sound pressure level. Note that the definition of 'residual sound' is the *'Total sound remaining at a given position in a given situation when the specific sounds under consideration are suppressed'*

The 15 dB guideline allows for the noise to be at least 5 dB above the residual sound before measurement starts and then a clear 10 dB above this value for the determination of the noise event metrics. Figure 2 has been extracted from the standard and shows the critical features in the determination of site suitability. For a slant distance, *s*, assuming spherical spreading, a 10 dB drop will correspond to a flight path distance of 3*s* and an approximate angle, ω, of 70°. For locations close to the flight paths, both s and the portion of the line of sight portion of the flight path within 70° on both sides of the closest point are small. For more distant locations, this portion of the line of sight flight path becomes greater and there is an increased chance that obstacles will be in the way. This will make it more difficult to ensure a 15 dB excess on the residual sound.

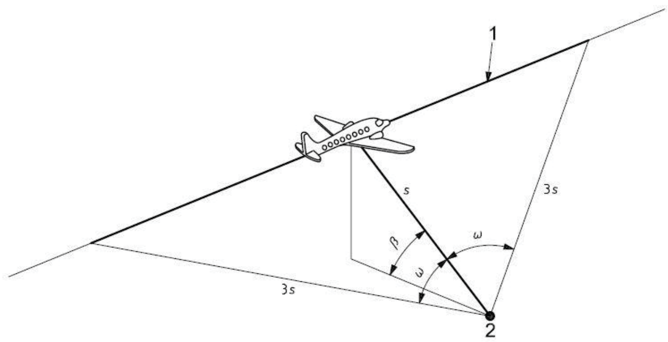

Figure 2. Example for lines of sight to the sound monitor that should be free of obstacles as presented as Figure 3 in ISO 20906 [6]. *s* is the slant distance, *ω* is the line of sight angle, β is the elevation angle relative to the ground plane.

In terms of placement at the location, the microphone needs to be at least 6 metres above the ground and at least 10 metres away from relevant acoustic reflecting surfaces, other than the ground. For microphones mounted on roofs (a common practice with noise monitor terminals in residential areas), it can be difficult to avoid acoustic interference from the roof surface and so a higher uncertainty must be accepted. These factors need to be considered no matter where the monitor is located.

Another recommendation of the standard is that the elevation angle between the ground plane and the sound ray from an aircraft, i.e. angle β in Figure 2, be greater than 30° to reduce ground effects. For distant sideline positions, such as 3 or 4 on Figure 1, it is unlikely that this recommendation will be met.

Sound monitor performance

The sound monitoring part of the system needs to conform to the performance specifications in ISO/IEC 61672 [7] for a class 1 sound level meter. As it is designed to be left unattended in the environment, protection from rain, wind, birds, lightening etc must be provided without affecting the sound level data obtained. These requirements are the same for near and distant locations and are complied with by most aircraft noise monitoring systems.

Measured quantities

A basic requirement is that the sound monitor should be capable of measuring continuous, A-weighted sound pressure levels. The standard provides a listing of the preferred the quantities to be determined to characterise a noise event and these include the sound exposure level, $L_{E,A}$ (to 0.1 dB resolution) and the maximum level, $L_{p,AS,max}$ or $L_{p,A,eq,1s,max}$. A reliable clock is required to enable all sound events to be time-stamped. These requirements are the same for near and distant locations and are complied with by commercial aircraft noise monitoring systems.

Aircraft classification

ISO 20906 [6] states that the main function of a sound monitoring system is to *"reliably and precisely detect and classify aircraft sound events"*. The standard recognises that there are several techniques for correlating measurements from a sound-monitoring system with aircraft movements. The chosen technique should meet the following criteria:

a) *The expanded uncertainty of the measured exposure level for an aircraft sound event shall not exceed 3dB.*
b) *At least 50% of the true aircraft sound events should correctly be classified as aircraft sound events.*
c) *The number of non-aircraft events which are incorrectly classified as aircraft sound events shall be less than 50% of the true number of aircraft sound events.*

At distant locations compliance with these requirements is more challenging as it becomes difficult to separate the aircraft noise events from other noise events in the area using acoustic parameters alone.

The standard does recommend that there be a period of manual identification of aircraft noise levels at the site with at least 20 events of the same type of aircraft being identified with sound level at least 5 dB above background noise levels. Manual identification is time consuming and costly so when a system includes flight path information there is a tendency to rely on the correlation data. Near to the flight path it is probably reasonable to use the flight path data in lieu of manual identification as a check of the initial set up. However at distant locations the risk of incorrectly attributing a noise level from a local event to an aircraft is considerably greater than for a location close to the flight paths and checking using flight path data alone may not be adequate. Correct identification of aircraft noise events and accurate noise level data for those events are the critical factors for obtaining useful information from a NFPMS so it is essential that this checking be done at the initial set up of the system.

The standard recommends that the primary identification of a noise event being due to an aircraft should be based on acoustical data including knowledge of the typical length of an aircraft sound event for the site, relationship between the maximum sound pressure level and the sound exposure level, spectral information, correlation with events at other sites, listening to the sound event recorded and acceptance that wind speed is not excessive. If non-acoustical data, e.g. flight path data, is available, the standard recommends that the sound data may be further checked to ascertain if the event could be identified with a particular aircraft operation.

As the standard recommends a primary determination of aircraft events be based on acoustical information, the setting of the acoustical parameters for an aircraft noise event is essential. An idealised aircraft noise event time profile shown is shown in Figure 3 [extracted from 6]. At more distant locations from the flight path the aircraft noise event will differ from this idealised profile. Thus identification primarily on acoustic parameters becomes more difficult and therefore there is a tendency in such set ups to rely even more heavily on the flight path data. As discussed in the following sections, this approach can still lead to incorrect attributions of aircraft noise levels.

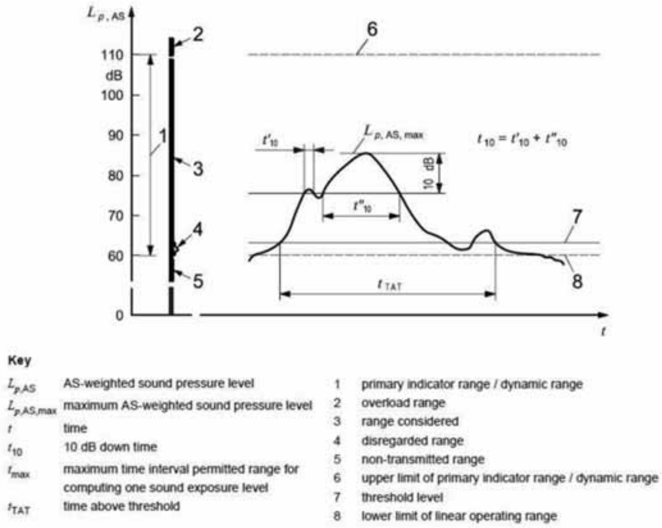

Figure 3. Example of aircraft noise event criteria, Figure 5 in ISO 20906 [6]

CASE STUDY OF NOISE MONITORING AT SOME DISTANCE FROM THE FLIGHT PATHS

Background on installation

The airport under consideration has approximately 910 Regular Public Transport (RPT) aircraft flights per week (47,320 per year) and approximately 70 General Aviation (GA) aircraft flights per week (3,640 per year). There has been an ongoing concern about aircraft noise from RPT aircraft operations and from GA aircraft overflights from the residents approximately 3 km to the side of the flight path to the north of the airport, as indicated in Figure 4.

Such concern about aircraft noise would not be anticipated due to the location and the noise abatement measures in place for the aircraft operations. The airport has a noise abatement zone that requires most RPT aircraft to be higher than 5,000ft above ground level before overflying residential areas. This means that departing and arriving RPT aircraft do not overfly the residential area which is approx 3.8 km from the end of the runway. Outside the ANEF 20 contour around the airport is considered acceptable for residential areas in accordance with Table 2.1 in AS 2021 [1]. The residential area under investigation is approx 2 km outside that contour and also 1.2 km to the side of the 65 dB contour for the noise footprint predictions for a Boeing 737-800 aircraft. Preferred tracks for

GA aircraft attempt to minimise overflights of the residential areas. Those that do overfly the area can be as low as 300m. To investigate the ongoing concerns a portable monitor as part of a NFPMS was installed at the edge of the residential area to obtain data on the aircraft noise levels in the area.

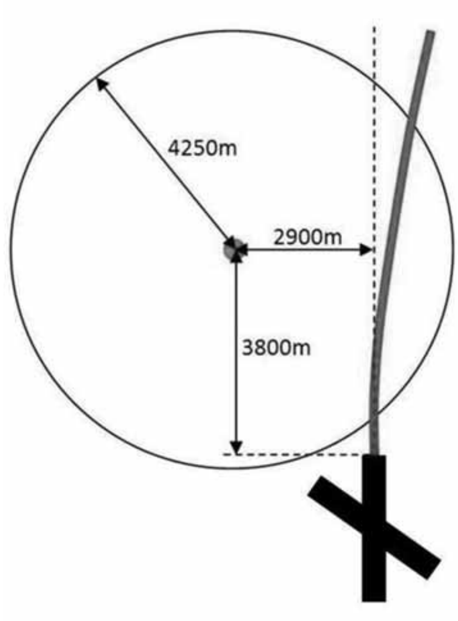

Figure 4. Schematic layout of the monitoring location in the residential area and the typical flight path used by RPT aircraft.

Compliance with the installation requirements

As the site was far to the side of the main flight path where the RPT aircraft were climbing, the elevation angles were well below the 30° minimum required in ISO 20906 [6] and so there was greater risk of ground effects affecting the data. For RPT aircraft operations the slant distance, *s*, was large and so there was an increased chance of obstacles within the angle ω affecting the data. However for the GA aircraft flying near the site the requirements for elevation angle would be met and the slant distance, *s*, was small.

From long term monitoring at the site, the $L_{Aeq,24hr}$ was found to be 49 dB(A) and the $L_{Aeq,night}$ was 42 dB(A). The expected aircraft noise levels for most RPT and GA aircraft operations were not likely to be more than 15dB above the residual noise. .

Thus, in the terms of the ISO Standard, this site is *"acoustically unfavourable"*. However, in view of the ongoing complaints, it is the relevant location for aircraft noise monitoring to try to quantify the aircraft noise levels. It is particularly important under such circumstances for the post processing of the data to not only to optimise the acoustic parameters used to classify an aircraft noise event but also to use as much non acoustic data as possible to minimise incorrect attributions of noise levels to aircraft.

NFPMS installation and Data Reporting

The basic acoustic parameters in the system and settings used at this installation are summarised in Table 1. Those responsible for the installation made the decision to set very broad parameters to avoid missing potential aircraft noise events with the belief that the flight path data correlation would enable rejection of non-aircraft noise events.

Table 1. Acoustic parameters used to identify potential aircraft noise events at the case study installation

Threshold, i.e. the trigger noise level above which the noise data is considered potentially due to an aircraft.	55.0 dB(A)
Minimum rise time before, and minimum fall time after the maximum level for the noise event level	0 dB/sec
Maximum rise time before, and maximum fall time after the maximum level for the noise event level	5 dB/sec
Pre-trigger and post-trigger time which allows for analysis of the data for some time before and after the maximum level has been identified.	5 sec

The need to rely greatly on the flight path data meant that the location of the correlation area was critical. For installations close to the flight path, a circle with radius 1 to 2 km and centred on the noise monitoring will allow for reasonably reliable identification of noise events that satisfy the acoustic criteria and which could be attributed to aircraft operations. For the case study location to the side of the main runway it was necessary to use a single correlation circle with radius 4.2 km to capture the closest approach of the aircraft following prescribed flight paths. This meant that a noise event at the monitor would be tagged as generated by an aircraft if it was operating anywhere within the large correlation area.

Accuracy of Aircraft Noise Data

The main data from the NFPMS is used for the production of reports on aircraft noise levels. The database does include a large amount of additional information on each noise event that is not normally used in the standard reporting process. Detailed analysis by the authors of all the data obtained over a 72 day period allowed for a better understanding of the extent of incorrect attributions of aircraft noise levels.

If there are intense storms to the north of the airport, RPT aircraft are instructed not to follow the usual flight paths as a safety precaution. During the analysis period there were a few RPT aircraft which flew almost directly over the noise monitoring terminal. Figure 5(a) shows the noise level versus time profile for one such event which was correctly identified and classified by the NFPMS and is more than 20 dB above the background noise levels. Figure 5(b) shows the profile for a similar aircraft type following the standard departure track. The maximum noise level during the event is just 10 dB above the background. The noise level profile and the time of the noise event are somewhat comparable and it is reasonable to accept that the noise event in Figure 5(b) is correctly attributed to an aircraft. In contrast Figures 6(a) and (b) show the profiles of two noise events which were also attributed to RPT aircraft that were somewhere in the correlation area. These profiles clearly differ greatly from the profiles in Figures 5(a) and (b); one is extremely short duration and the other has an atypical profile. The standard reporting process would identify these as aircraft noise events but this is unlikely to be a correct attribution.

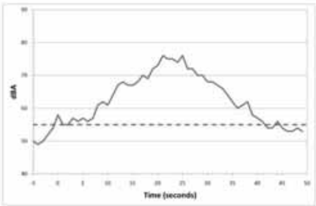

Figure 5(a). Noise profile for a jet aircraft flying directly overhead the NMT,

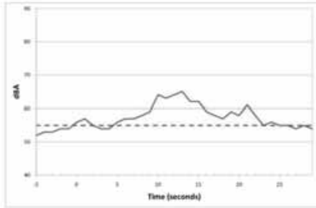

Figure 5(b). Noise profile for a jet aircraft following a standard departure track, putting it approximately 3 km from the monitor

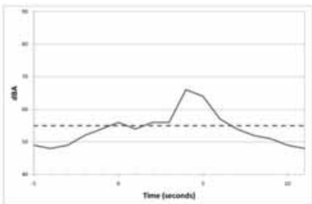

Figure 6(a). Noise profile attributed to a jet aircraft but with very short time duration.

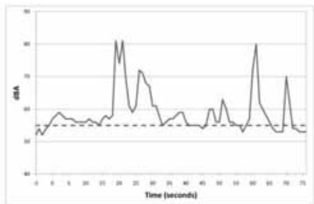

Figure 6(b). Atypical noise profile attributed to a jet aircraft.

GA aircraft could fly near to the noise monitoring terminal and so there was a better chance of accurate attribution of noise level. Figure 7(a) shows the noise profile which is likely to be a correctly attributed event for a helicopter and 7(b) for a small fixed wing aircraft. Figure 8(a) shows a profile attributed to a GA aircraft with an atypical noise profile and which would seem to be from rain noise. The maximum noise level for the event in Figure 8(b) is just above the threshold but the time period is too short for this to be a valid aircraft noise event and is more likely due to a local noise.

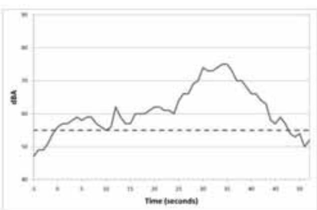

Figure 7(a). Noise profile attributed to a helicopter directly overhead the noise monitoring terminal.

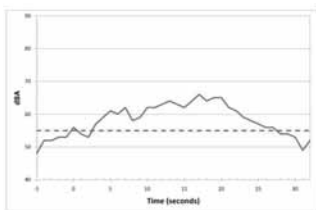

Figure 7(b). Noise profile attributed to a small fixed wing aircraft directly overhead the noise monitoring terminal.

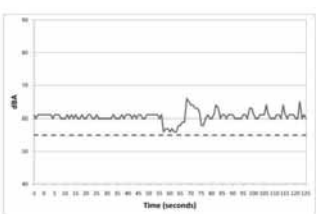

Figure 8(a). Atypical noise profile attributed to a GA aircraft.

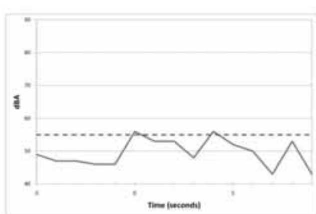

Figure 8(b). Noise profile attributed to a GA aircraft but with a very short time duration and atypical profile.

As the GA aircraft travel at a lower speed they could be within the correlation area for some time and local noise events occurring during that time period could lead to multiple incorrect attributions of aircraft noise levels. From the detailed analysis of 2713 events the data over a 72 day period, multiple noise events occurring within 1 minute were attributed to the one aircraft on 641 occasions. While GA aircraft can make multiple passes over a location these could not occur within such a short time period.

After removing these incorrectly attributed multiple events, 2071 correlated noise events remained and of these only 383 (18.5%) had a maximum noise level at least 10 dB above the threshold of 55 dB(A). As it was impractical to look at every noise event in detail, samples of noise events focussing on those less than 10 dB above the threshold were studied in detail. This analysis used the additional data that is captured but not normally used in the usual reporting process. Included in this analysis was viewing the flight path used by the aircraft to see if it was in the direction of the noise monitor, comparing the time the aircraft passed through specified points for correlation with the higher noise levels in the profiles, comparing the time duration of the noise event with a valid aircraft noise event etc. This detailed analysis confirmed that there was a high rate of incorrect attributions of noise events to aircraft from the NFPMS and this affected the assessment of the aircraft noise levels in the area.

DISCUSSION

Techniques to improve the correct attribution of aircraft noise levels are essential to obtain useful data at *"acoustically unfavourable"* sites. Both Wallis [8] and Adams [9] have emphasised that correct identification of aircraft noise events are critical. The NFPMS captures more data than is used in the normal reporting process. Some of the techniques used in the detailed examination in the case study were time consuming as they were done manually but there is scope for these to be automated. For example, the removal of duplicate noise events attributed to the same aircraft and the removal of events with time duration less than or greater than a prescribed time period. More sophisticated methods need to be employed to remove those profiles that do not comply with the typical noise profile and to provide a better discrimination of aircraft noise events [e.g. 10]. Even with the application of more sophisticated methods for correct identification of aircraft noise levels there may still be incorrect attributions.

The use of a correlation circle for identification of an aircraft in the region has limitations when the monitoring location is to the side of the flight paths and so the area of the circle becomes very large. Alternative shapes for the correlation area could assist to reduce incorrect attributions particularly at busy times for the airport operations when there are a number of aircraft in the vicinity of the airport. Correctly identifying the location of the aircraft at the time of the maximum noise level could assist with correct discrimination of aircraft noise events and such systems are available [e.g. 11].

Audio files for each potential aircraft noise event could be used to identify dubious events remaining in the data base.

While this may be time consuming it would further assist with correctly identifying aircraft noise events for *"acoustically unfavourable"* sites.

CONCLUSIONS

Assessment of aircraft noise levels near to the main flight paths has been successfully implemented at many airports. The development and refinement of aircraft monitoring systems have led to improvements in the analysis and reporting systems so that a good indication of the aircraft noise levels at those locations is obtained. Monitoring of aircraft noise levels away from the main flight paths is more challenging as the aircraft noise may not be clearly defined above the background noise level in the area. However there is a need to obtain data on aircraft noise levels in these *'acoustically unfavourable'* locations.

Following ongoing complaints from residents, a noise monitoring terminal was located in the residential area to the side of the main flight paths for RPT aircraft but subject to overflights by GA aircraft. The detailed analysis of this data has highlighted the difficulties in obtaining accurate attribution of aircraft noise events from the NFPMS. More sophisticated analysis and the use of audio files are required to achieve a fair and accurate assessment of the aircraft noise levels for such *'acoustically unfavourable'* locations.

ACKNOWLEDGEMENT

The authors are grateful to the authority which made the data available for this investigation.

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PREDICTION OF THE ACOUSTIC PERFORMANCE OF SMALL POROELASTIC FOAM FILLED MUFFLERS: A CASE STUDY

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The acoustic performance of small, irregularly shaped mufflers in continuous positive airway pressure (CPAP) devices is often enhanced by the inclusion of dissipative materials. In this study, the acoustic properties of two polyurethane foams were determined using a two-cavity method. Acoustic models of two CPAP device muffler designs incorporating a foam insert have been developed using a commercial finite element analysis software package. Experimental results for the mufflers have been obtained using the two-microphone acoustic pulse method. Results of the transmission loss of the muffler designs obtained from the finite element models are presented and validation of the computational results is discussed.

INTRODUCTION

Continuous positive airway pressure (CPAP) devices generate air flow using a high speed fan and noise from this device is controlled using mufflers situated in the flow path at the fan inlet and the flow generator outlet. While the most significant noise levels are present at frequencies below 4 kHz, the use of dissipative materials is often utilised in order to extend the attenuated frequency range up to 10 kHz.

Foundation work on the theoretical approach to describe sound propagation in porous materials was laid by Zwikker and Kosten [1] who introduced the concept of effective density and bulk modulus. Biot [2, 3] introduced frame elasticity, where the skeleton of the material is not rigid and is capable of transmitting sound waves. A key element of this work was identification of the existence of three types of sound wave for continuous materials: two compression waves and one shear wave. Morse and Ingard [4] developed generic acoustic models for rigid and limp porous materials. Lambert studied low and medium flow resistance foams [5] and this work was extended by Allard *et al*. [6] to high flow resistance foams. Allard and Champoux [7] used the general frequency dependence of the viscous forces in porous materials proposed by Johnson *et al*. [8] to produce expressions incorporating five macroscopic properties of the porous material. Delany and Bazley [9] showed that measured values of characteristic impedance and propagation coefficient for a range of fibrous materials, normalised as a function of frequency divided by flow resistance, could be presented as simple power law functions. Miki [10] found that the Delany-Bazley model produced an unphysical prediction at low frequencies and amended the original equation regression coefficients. Further work was done by Bies and Hansen [11] and Mechel [12] to correct and extend the Delany-Bazley method beyond the bounds recommended by the original authors. Attenborough [13] observed that the normalising parameter used by Delany and Bazley appeared in the theoretical expressions for any pore shape and concluded that empirical relationships of the form proposed by Delany and Bazley should be valid for non-fibrous porous materials. He did however also note that “frame elasticity will be an additional complication” and that the coefficients in the Delany-Bazley model would be unique to each type of porous material. Dunn and Davern [14] followed the same approach used by Delany and Bazley and derived new regression coefficients which applied the power law functions to polyurethane foams. Work by Wu [15] and Ling [16] has resulted in the derivation of further sets of regression coefficients for medium and high flow resistivity foams. Komatsu [17] showed that the coefficients used in the Delany-Bazley model were strongly dependent on the airflow resistivity and introduced a common logarithm term in place of the original non-dimensional normalising parameter.

This study builds on previous work by the authors on acoustic finite element (FE) modelling of reactive muffler designs [18]. The acoustic characteristics of two polyurethane foams were obtained experimentally and the corresponding properties incorporated into FE models of a production CPAP muffler and a prototype integrated chamber design. Results of the transmission loss of the foam-filled mufflers obtained from the FE models are presented. The transmission loss of each of the mufflers was measured using a two-microphone acoustic pulse method which was based on the procedure developed by Seybert and Ross [19]. For the two muffler designs, experimental results for an empty muffler and the muffler containing an insert manufactured from each foam type are compared with results obtained computationally.

MUFFLER DESIGNS

Two muffler designs which were originally presented in the previous paper by the authors [18] were selected for further analysis. The first design shown in Figure 1 is that of a production CPAP device muffler which, while geometrically complex, consists of a single chamber having coaxial inlet and outlet ports located at one end of the chamber. A foam insert, shown by the grey shaded area in Figure 2, occupies the majority of the chamber volume. It is important to note that this insert does not intrude into the direct path between the inlet and outlet ports. The second design shown in Figure 3 consists of two integrated chambers and presents a complex path between the inlet and outlet ports. If air is flowing through the device it would be deflected around a vertical internal baffle before passing through a narrow slot into the final chamber. A foam insert completely fills the volume of the first chamber and sound waves entering from the inlet port must pass through the foam prior to reaching the outlet port.

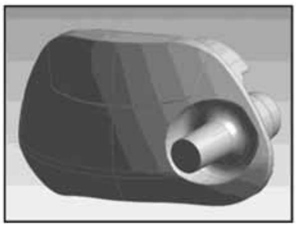

Figure 1a: CPAP muffler air volume

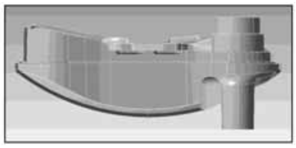

Figure 1b: Muffler cross-section

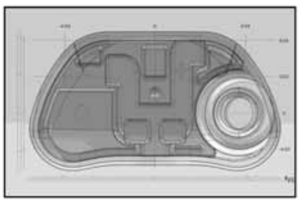

Figure 2a: CPAP muffler foam insert (front)

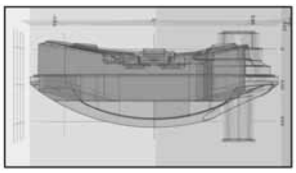

Figure 2b: CPAP muffler foam insert (top)

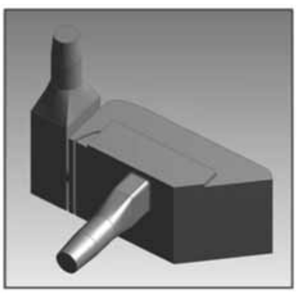

Figure 3a: Integrated muffler air volume

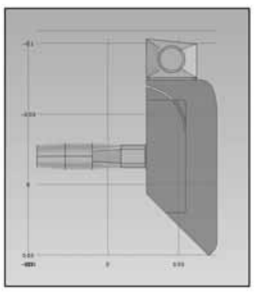

Figure 3b: Muffler foam insert

Two different polyurethane foam materials were selected for comparison. The first foam (light grey) has an apparent density of 34 kg/m^3 and is a material currently being used in CPAP device mufflers. The second foam (dark grey) has an apparent density of 23 kg/m^3 and is more likely to be used in protective packaging. The latter was chosen for inclusion in the assessment as it was anticipated that the acoustic properties would be sufficiently dissimilar to the first to provide an instructive comparison.

FOAM MODELLING METHOD

Characteristic impedance ($Z_{c,f}$) and propagation coefficient (γ_f) of porous materials can be presented as simple power-law functions by [9, 11]:

$$Z_{c,f} = R + jX \ , \quad Z_{c,f} = \rho_a c_a \left[1 + C_1 \left(\frac{\rho_a f}{r_f}\right)^{C_2} - j * C_3 \left(\frac{\rho_a f}{r_f}\right)^{C_4}\right] \tag{1}$$

$$\gamma_f = \propto + j\beta \ , \quad \gamma_f = (\omega / c_a)\left[C_5 \left(\frac{\rho_a f}{r_f}\right)^{C_6} + j\left(1 + C_7 \left(\frac{\rho_a f}{r_f}\right)^{C_8}\right)\right] \tag{2}$$

where ρ_a and c_a are respectively the density and speed of sound in air, f is the frequency and r_f is the airflow resistivity. Delany and Bazley obtained values for the coefficients C_1 to C_8 using a range of fibrous absorbent materials [9]. Several authors have noted that predictions made using Delany and Bazley's original coefficients are not especially accurate when applied to poroelastic materials and have obtained different sets of coefficients [10, 14-16]. In this work, the characteristic impedance, propagation coefficient and airflow resistivity of the two foams materials were measured experimentally. The methodology described by Delany and Bazley was then applied to derive the unknown coefficients C_1 to C_8 for these particular foams. Once the coefficients have been determined and substituted back into Eqs. (1) and (2), the resulting equations are then readily incorporated directly into the finite element model. Further insight into the acoustic performance of the foams may be gained by re-stating Eqs. (1) and (2) in terms of an equivalent fluid having a complex speed of sound (c_f) and complex mean density (ρ_f) by [19]:

$$c_f = j\frac{\omega}{\gamma_f} \ , \quad c_f = \left[\frac{\omega\beta}{(\alpha^2 + \beta^2)}\right] + j\left[\frac{\omega\alpha}{(\alpha^2 + \beta^2)}\right] \tag{3}$$

$$\rho_f = -j\frac{Z_{c,f}\gamma_f}{\omega} \ , \quad \rho_f = \left[\frac{(R\beta + X\alpha)}{\omega}\right] + j\left[\frac{(X\beta - R\alpha)}{\omega}\right] \tag{4}$$

EXPERIMENTAL METHODS

Experimental methods used to obtain the characteristic impedance, propagation constant and flow resistivity of the foams are presented in what follows. Further experiments were then conducted to measure the transmission loss of the mufflers using the two-microphone acoustic pulse method, which has been described previously [18].

Characteristic impedance and propagation constant

The characteristic impedance and propagation constant of porous materials can be measured by applying the transfer function method to a two-cavity approach [20]. A sample of homogeneous porous material was positioned within a Brüel & Kjær Type 4206 impedance tube and against the front face of a moveable plunger. The plunger was then withdrawn away from the sample, producing an air cavity with a known depth L between the rear face of the sample and the plunger (Fig. 4). A random signal was fed to the loudspeaker of the impedance tube and the normal surface acoustic impedance of the sample was measured in accordance with ISO 10534 [21]. The transfer function H_{12} from microphone position 1 to position 2, defined by the complex ratio p_2/p_1, was measured using a two channel Fast Fourier transform. The surface acoustic impedance Z_0 is then obtained by [22]:

$$Z_0 = jZ_{c,a}\left\{\frac{H_{12}\sin[k(Lx + Dx)] - \sin(kLx)}{\cos(kLx) - H_{12}\cos[k(Lx + Dx)]}\right\} \tag{5}$$

where k is the wave number and $Z_{c,a}$ ($=\rho_a c_a$) is the characteristic impedance of air.

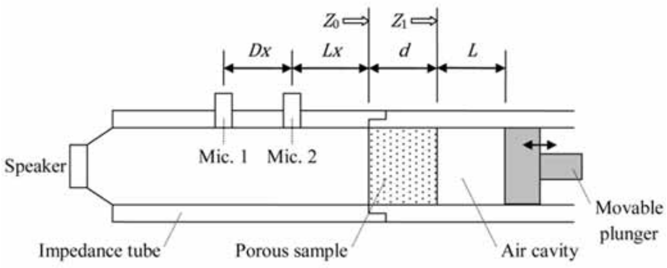

Figure 4: Schematic diagram of the impedance tube configuration

The impedance tube plunger was withdrawn a further distance and the measurement procedure was repeated at depth L' to obtain Z_0'. The theoretical impedance of closed tubes with depths L and L' is given by [20]:

$$Z_1 = -jZ_a\cot(kL)\,, \quad Z_1' = -jZ_a\cot(kL') \tag{6,7}$$

The characteristic impedance and propagation constant of the material can then be calculated by [20]:

$$Z_{c,f} = \pm\sqrt{\frac{Z_0Z_0'(Z_1 - Z_1') - Z_1Z_1'(Z_0 - Z_0')}{(Z_1 - Z_1') - (Z_0 - Z_0')}} \tag{8}$$

$$\gamma_f = \left(\frac{1}{2d}\right)\ln\left[\left(\frac{Z_o + Z_{c,f}}{Z_o - Z_{c,f}}\right)\left(\frac{Z_1 - Z_{c,f}}{Z_1 + Z_{c,f}}\right)\right] \tag{9}$$

where the sign in Eq. (8) is selected so that the real part of $Z_{c,f}$ is positive.

Airflow resistivity

The airflow resistivity of a homogeneous material is given by $r_f = \Delta P / du$, where ΔP is the static pressure drop across the material, d is the unit thickness and u is the linear velocity of air passing through it [23]. Measurements were performed according to the direct airflow method described in ISO 9053 [23]. A unidirectional airflow was passed through cylindrical samples having 25mm thickness and 100mm diameter (see Fig. 5) and the resulting pressure drop between the two free faces of the sample was measured.

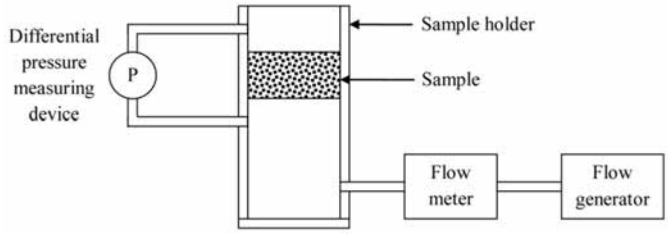

Figure 5: Schematic diagram of the airflow resistivity experimental set-up

FINITE ELEMENT MODELS

Acoustic finite element models of each of the muffler designs were developed using the commercially available finite element analysis package COMSOL (version 4.0). The muffler models were meshed using Lagrange-quadratic elements with controls applied to produce a mesh having at least 6 elements per acoustic wavelength at the upper bound of the frequency range being analysed (limiting case). A harmonic pressure of 1 Pa was specified at the inlet and a radiation condition applied at inlet and outlet. The air was assumed to be non-flowing and inviscid and acoustic damping was not applied at the fluid-structure interface. The foam inserts were modelled using the Delany-Bazley formulation described earlier and having parameters that were obtained experimentally for each of the foam types. Transmission loss is calculated directly in COMSOL using the acoustic power at the inlet and outlet ports of the muffler.

RESULTS AND DISCUSSION

The results are presented in three sub-sections corresponding to the foam airflow resistivity measurements, foam acoustic property measurements (characteristic impedance and propagation constant) and the muffler transmission loss measurements, respectively.

Foam airflow resistivity

Airflow resistivity for each foam type was measured according to the direct airflow method described in ISO 9053. Data was also recorded at linear airflow velocities greater than the 4 mm/s upper limit recommended by the Standard to ascertain the effect of turbulent flow on the apparent airflow resistivity for the foams being studied. The values for airflow resistivity calculated using data within the laminar range are presented in Table 1 and it can be seen that the measured airflow resistivity of the two foam types is significantly different. This finding is consistent with the observed difference in surface pore sizes and spacing.

Table 1: Foam airflow resistivity

Foam	Description	Flow resistivity (Rayls/m)	95% confidence interval
A	Acoustic (light grey)	8,445	182
B	Non-acoustic (dark grey)	2,652	36

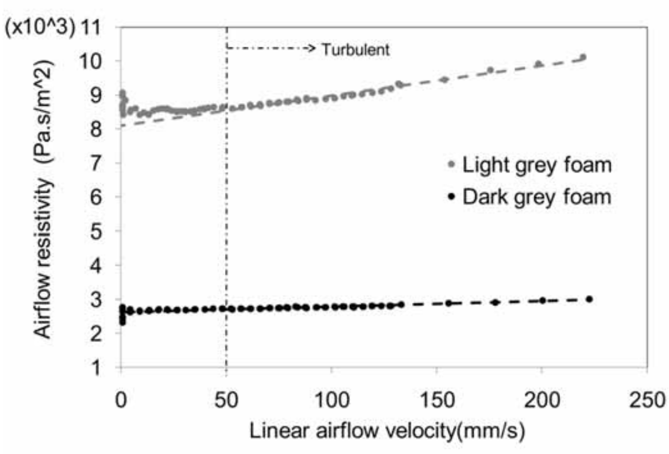

Figure 6: Airflow resistivity of dark and light grey foams

Figure 6 shows that the apparent airflow resistivity for the light grey foam increases as the linear airflow is increased beyond the laminar region, while the apparent airflow resistivity of the dark grey foam remains largely unaffected. This difference in observed behaviour is significant as the Delany-Bazley method uses a single value for flow resistivity to characterise the porous material.

Foam acoustic properties

The normal surface impedance for each foam type was measured and calculated using the test method described in ISO 10534. Measurements were obtained at four cavity depths corresponding to 25mm, 50mm, 75mm, and 100mm, using samples of 25mm thickness. The characteristic impedance and propagation constant were calculated for each of the cavity combinations 25mm/50mm, 50mm/75mm and 75mm/100mm using Eqs. (8) and (9) and the results for the three combinations were averaged. Equations (1) and (2) can be re-stated as:

$$\log_{10}\frac{R}{\rho_a\, c_a} - 1 = C_2\log_{10}\left(\frac{\rho_a f}{r_f}\right) + \log_{10}(C_1) \qquad (10)$$

$$\log_{10}\left(\frac{-X}{\rho_a\, c_a}\right) = C_4\log_{10}\left(\frac{\rho_a f}{r_f}\right) + \log_{10}(C_3) \qquad (11)$$

$$\log_{10}\left(\frac{\alpha c_a}{\omega}\right) = C_6\log_{10}\left(\frac{\rho_a f}{r_f}\right) + \log_{10}(C_5) \qquad (12)$$

$$\log_{10}\left(\frac{\beta c_a}{\omega} - 1\right) = C_8\log_{10}\left(\frac{\rho_a f}{r_f}\right) + \log_{10}(C_7) \qquad (13)$$

As Eqs. (10) to (13) are of the form $y = mx + b$, it is possible to obtain the equation coefficients by fitting linear trend lines through the experimental data. The coefficients that were obtained are presented in Table 2 alongside Delany and Bazley's original coefficients. It can be seen that the coefficients for each of the two foam types are significantly different from each other and also from the original Delany-Bazley coefficients, with the exception of the attenuation constant α which shows reasonable agreement. These differences support previous findings that predictions made using the original Delany-Bazley coefficients are not especially accurate when applied to poroelastic materials [10, 14-16] and that the coefficients would be unique to each type of porous material [13]. However it is worth noting that the propagation constant of both foam types correlate well with the flow resistivity, producing correlation coefficients between 0.96 and 0.99. The characteristic impedance of the light grey foam also correlates well, producing correlation coefficients between 0.88 and 0.92. These observations are consistent with the findings of Wu [15] who reported correlation coefficients between 0.85 and 0.99 for porous plastic open-celled foams. While the correlation coefficients for the characteristic impedance of the dark grey foam are less encouraging (0.58 and 0.72), examination of the characteristic impedance curves shows significant departure from linear behaviour at frequencies greater than 1,600 Hz. This suggests that the observed behaviour might be attributed to sample preparation as this frequency coincides with the transition between measurements obtained in the 100mm diameter impedance tube and those obtained in the 29mm diameter impedance tube.

The Delany-Bazley relationships are only considered to be valid over the range $0.012 \leq (\rho_a f / r_f) \leq 1.2$ [11]. Assuming an air density of 1.18 kg/m^3, the valid frequency range for the dark grey foam is 25 Hz to 2,690 Hz, while for the light grey foam it is 85 Hz to 8,500 Hz.

Equations (3) and (4) were used to obtain the complex speed of sound and complex density of the two foam materials based on the coefficients in Table 2. The results for the speed of sound and density of the light grey foam are shown in Figs. 7 and 8, respectively. The Delany-Bazley model shows excellent agreement with the experimental data. This is not unexpected as the model coefficients were derived using the same set of experimental data and the correlation coefficients were good. The results for the dark grey foam show a comparable agreement between the model and the experimental data, with only slight deviation noted between the model and the data at frequencies below 250 Hz. This deviation is attributed to the lower correlation coefficients associated with the impedance equation.

Table 2: Delany-Bazley equation coefficients

Parameters		Dark grey foam		Light grey foam		Delany & Bazley
		Coefficient	R^2	Coefficient	R^2	Coefficient
$R\ (Z_c)$	C_1	0.2051	0.58	0.2824	0.92	0.0571
	C_2	-0.2249		-0.3659		-0.7540
$X\ (Z_c)$	C_3	0.1175	0.72	0.0980	0.88	0.0870
	C_4	-0.4851		-0.6144		-0.7320
$\alpha\ (\gamma)$	C_5	0.2039	0.98	0.1692	0.99	0.1890
	C_6	-0.5416		-0.5728		-0.5950
$\beta\ (\gamma)$	C_7	0.2688	0.96	0.2561	0.97	0.0978
	C_8	-0.3111		-0.4657		-0.7000

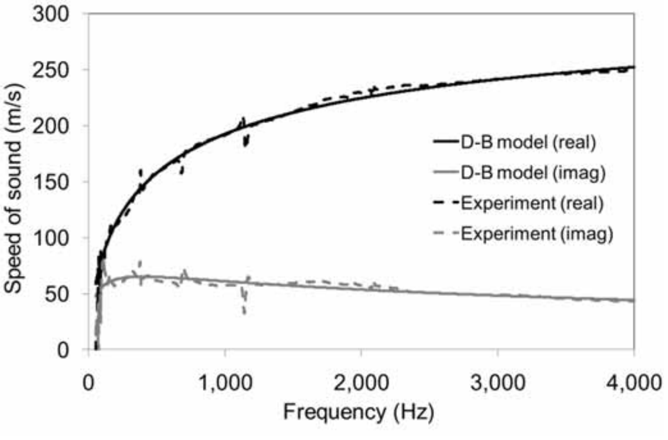

Figure 7: Equivalent fluid speed of sound of light grey foam

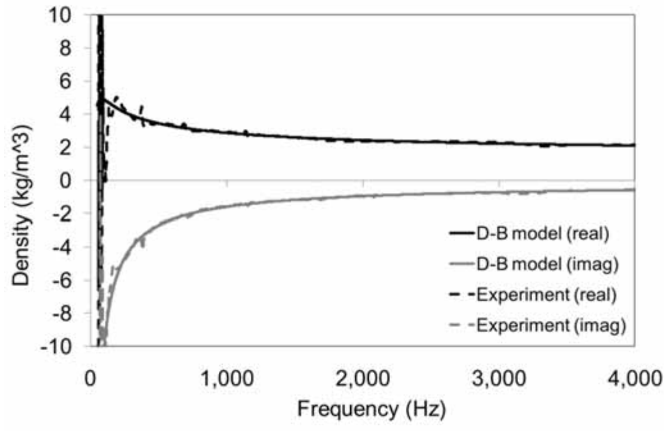

Figure 8: Equivalent fluid density of light grey foam

Muffler transmission loss

Figure 9 contains the transmission loss obtained experimentally for the CPAP device muffler, with and without a foam insert present, and the transmission loss predicted by the COMSOL finite element model. The FE results show good agreement with the experimental results over the frequency range assessed. The inclusion of the foam insert results in slight degradation of performance at the lower frequencies, especially about the peak centred at 800 Hz, but also results in a transmission loss of at least 10 dB over a broadband frequency range above that peak. Figure 10 compares the transmission loss obtained computationally and experimentally for the CPAP device muffler using the light and dark grey foams. The results show that the foam inserts have a very similar impact on the acoustic performance of this muffler design despite a difference in apparent density of approximately 50%.

Figure 11 contains the transmission loss predicted by the finite element model for the integrated chamber muffler both with and without the first chamber filled with foam. The results show that the presence of foam has little effect on the acoustic performance of the muffler below 500 Hz but contributes significantly to increased transmission loss at higher frequencies. In contrast to the observations made in respect to the CPAP device muffler, the results show that the two foam materials make differing contributions to the acoustic performance of this muffler design.

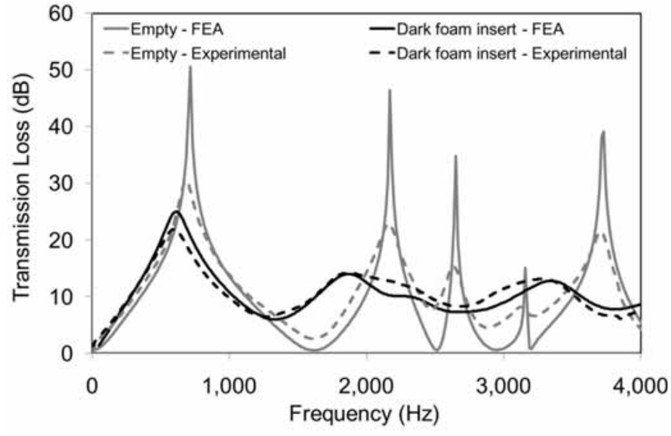

Figure 9: Transmission loss for the CPAP device muffler with and without dark foam insert, comparing computational results (solid lines) and experimental results (dashed lines)

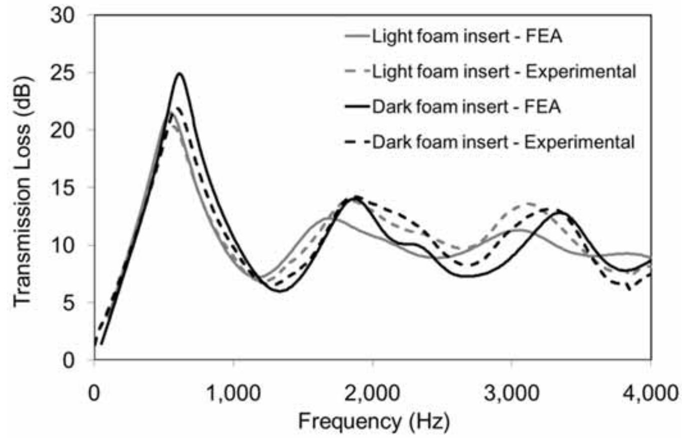

Figure 10: Transmission loss for the CPAP device muffler with foam inserts, comparing computational results (solid lines) and experimental results (dashed lines)

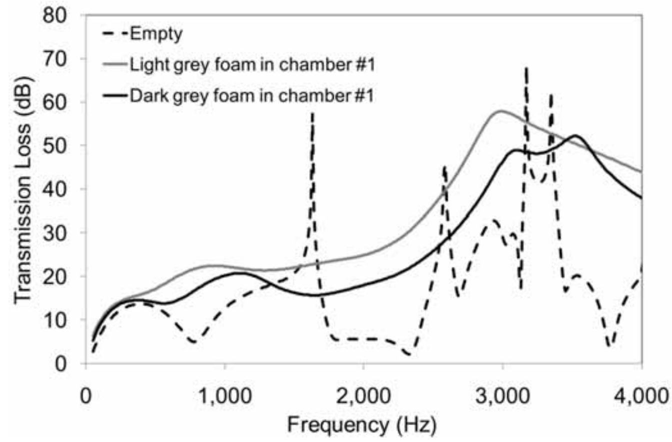

Figure 11: Transmission loss results obtained computationally for the integrated muffler without foam (dashed line) and with the first chamber foam filled (solid lines)

This observation is attributed to the muffler designs and location of the foam inserts. In the case of the integrated chamber design, sound waves travelling between the inlet and outlet ports are required to pass through the foam while in the CPAP device design, they only graze the surface of the foam insert. The greater contribution made by the light grey foam is consistent with the higher apparent density and flow resistivity when compared to the dark grey foam.

CONCLUSIONS

The characteristic impedance and propagation constant of two polyurethane foams have been determined experimentally using a two-cavity impedance tube method. Airflow resistivity of the two foams has been determined experimentally using the direct airflow method described in ISO 9053. Acoustic models of a production CPAP device muffler and an integrated chamber muffler design, both incorporating poroelastic foam inserts, have been developed using a commercial finite element analysis software package. Transmission loss results for the mufflers have been experimentally obtained using the two-microphone acoustic pulse method.

The magnitudes of the airflow resistivity measured for each of the two foam types are significantly different and they also exhibit differing sensitivity to linear airflow variations. The Delany-Bazley equation coefficients calculated for each of the two foam types differ from the original Delany-Bazley coefficients and also from each other. As the original Delany-Bazley model assumes a single value for flow resistivity to characterise the porous material and applies a fixed set of equation coefficients to model all porous materials, use of the original Delany-Bazley model to represent these foams will lead to inaccurate predictions.

Transmission loss results for the two muffler designs with and without the foam inserts were presented. The transmission loss results obtained computationally incorporated the derived Delany-Bazley coefficients. Good agreement between the numerical and experimental results was obtained for both muffler designs across the entire considered frequency range. The foam inserts make little impact on the acoustic performance of either muffler design below 500 Hz and results in slight degradation of performance about the peak centred at 800 Hz in the case of the CPAP device muffler. The inserts make a positive contribution to the transmission loss of both muffler designs at higher frequencies. The light grey foam makes a greater contribution than the dark grey foam which is consistent with its higher apparent density and flow resistivity. The effect of the foam inserts on the muffler acoustic performance is more significant in the integrated chamber design, which is attributed to the sound waves passing through approximately 20cm of foam between the inlet and the outlet ports whereas in the CPAP device design the sound waves only graze the surface of the foam insert.

By characterising foam as an equivalent fluid using straightforward airflow resistivity and impedance tube measurements, it has been shown that accurate predictions of the acoustic performance of foam inserts in small mufflers can be achieved using finite element modelling.

ACKNOWLEDGEMENTS

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MULTIPLE-LEAF SOUND ABSORBERS WITH MICROPERFORATED PANELS: AN OVERVIEW

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Since the pioneering work by Maa, multiple-leaf microperforated panel (MPP) sound absorbers of various configurations with different materials have been studied. Multiple-leaf structures are primarily employed to obtain wideband sound absorption. The authors have proposed double-leaf microperforated panel space absorbers (DLMPP), which consist of two MPPs and an air-cavity in-between, without a back wall. A DLMPP is a wideband sound absorber, which is also effective at low frequencies. However, an MPP is still expensive. If one of the MPPs in such a structure can be substituted with another material, such as a permeable membrane, it can be effective and also economical. The authors, therefore, have been exploring various multiple-leaf structures including both MPPs and permeable membranes. This paper gives an overview of our studies on such multiple-leaf sound absorbing structures with MPPs, including a DLMPP, a triple-leaf MPP space absorber, a space sound absorber consisting of an MPP and a permeable membrane. Also it includes a multiple-leaf structure with MPPs and membranes backed by a rigid wall.

INTRODUCTION

Microperforated panels (MPP) are one of the most promising alternatives among various next-generation sound absorbing materials. MPPs were first intensively studied by Maa [1-4] and intensively studied for room acoustical applications by Fuchs [5-7]. Recent studies also include the applications for low-frequency sound absorbers, duct muffling devices, acoustic window systems, highway noise barriers, etc [8-11].

Attempts in development of new-type MPP absorbers for wider absorption frequency range have been made by using multiple-leaf absorbers [2, 12-15], two MPP absorbers arranged in parallel [16], etc. In Maa's early work, he proposed a double-leaf MPP with a rigid-back wall, which offers wider absorption frequency range due to two resonances [1,2]. The authors proposed a double-leaf MPP space absorber (DLMPP) which consists of two MPPs and an air-cavity in-between without a rigid backing [12,13]. This shows a single peak resonance absorption at mid-high frequencies and moderate non-resonance absorption at low to mid frequencies caused by acoustic flow resistance of the leaves. Thus, DLMPPs can offer much wider sound absorption frequency range. This additional low-frequency absorption due to the leaves' acoustic flow resistances is similar to that of single/multiple-leaf permeable membrane space absorbers [17]. This fact suggests that the low frequency absorption can still be caused even if one of the MPPs in a DLMPP is replaced by a permeable membrane, as an MPP can also be regarded as an acoustical permeable material. Therefore, the authors also proposed a double-leaf space absorber composed of an MPP and a permeable membrane [18]. This structure shows characteristics similar to those of a DLMPP when the sound is incident upon the MPP side, and shows those similar to porous-type absorbers when the sound is incident upon the membrane side – thus, it shows moderately high flat absorption characteristics when placed in a diffuse sound field in which the sound is incident from the both sides [18]. Such variations of a DLMPP, including triple-leaf MPP space absorbers (TLMPP) [15] and space absorbers with a combination of an MPP and a permeable membrane, can be used for various purposes as an effective alternative to classical sound absorbers. Also a combination of an MPP with a permeable membrane backed by a rigid wall has been studied by the authors [19].

In order to enhance the resonance absorption, the authors have proposed the use of a honeycomb in the air-cavity in the MPP sound absorbing structures [20,21]. The authors also examined its effects on the sound absorption performance of the multiple-leaf MPP sound absorbers and confirmed that the honeycomb can effectively improve the multiple-leaf MPP absorbers' sound absorption performance [14].

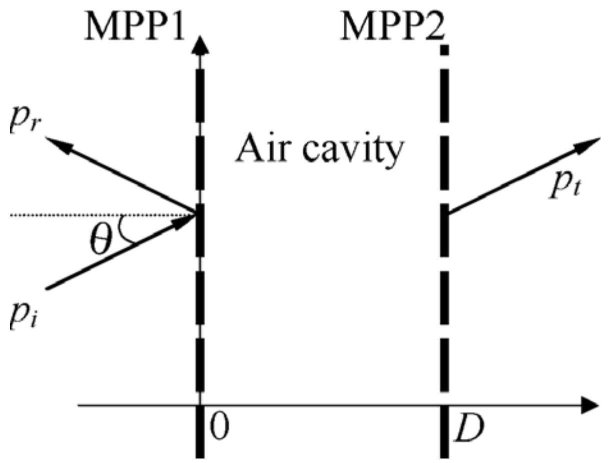
In this paper, the authors' studies on the multiple-leaf MPP sound absorbing structures mentioned above are reviewed. First, the studies on a DLMPP and its variations are reviewed. Secondly, the sound absorbing structures with a combination of an MPP and a permeable membrane, with and without a rigid-back wall, are introduced. Furthermore the studies on the effect of a honeycomb on these absorbers are reviewed.

MULTIPLE-LEAF SPACE SOUND ABSORBERS WITH MPPS

The most basic form of multiple-leaf MPP absorbers is the double-leaf MPP absorber with a rigid-back wall proposed by Maa [1,2]. By using two leaves, two resonance peaks occur which are merged into a broader peak, and it can offer wider absorption frequency range than a single absorber. However, as long as the sound absorption is solely caused by Helmholtz-type resonance, the absorption frequency range is limited in its resonance frequency range. On the other hand, permeable membranes can offer a flat frequency response at low frequencies with moderate absorption coefficients [17]. Considering the fact that MPPs are also permeable materials with acoustic flow resistance, which should behave similarly to permeable membranes, multiple-leaf MPPs without a rigid backing are expected to show a similar behaviour to a double-leaf permeable membrane. Hence, the studies on multiple-leaf MPPs without rigid-backing were triggered.

Double-leaf MPP space absorbers (DLMPP)

Figure 1 shows a sketch and the photograph of an experimental specimen of a DLMPP. Two MPPs are placed in parallel with an air-cavity in-between. Theoretical analyses were performed by using a Helmholtz integral formulation considering the sound-induced vibration of the leaves [13]. A typical result of the sound absorption characteristics of a DLMPP (a theoretical result in comparison with an experimental one) is shown in Fig. 2. In Fig. 2 the theoretical results of the absorption characteristics are shown in the difference between the absorption and transmission coefficients ($\alpha-\tau$), which indicates the ratio of the energy dissipated in the sound absorbing system. This is proven to correspond to the diffuse sound field absorption coefficient measured in a reverberation chamber [22].



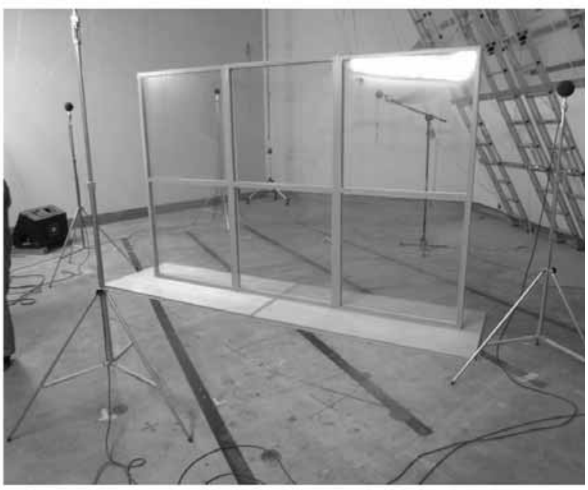


Figure 1: A sketch of a DLMPP (top) and a photograph of its experimental specimen of DLMPP (bottom).

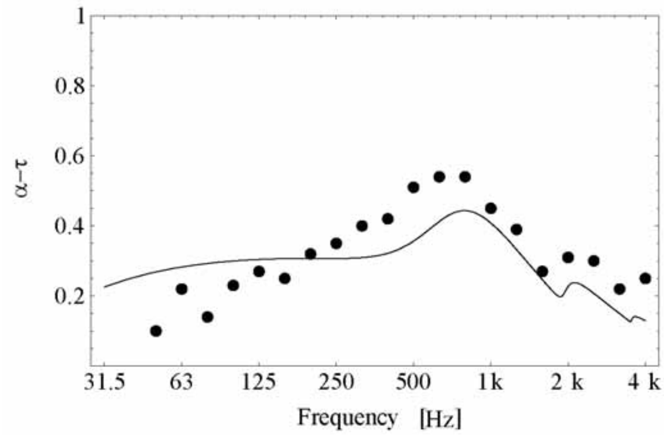


Figure 2: An example of a calculated result of the sound absorption characteristics ($\alpha-\tau$: solid line) of a DLMPP in comparison with experimental results measured in a reverberation chamber (dots). The two leaves have the same parameters: hole diameters 0.5 mm, thicknesses 1.0 mm, perforation ratios 1.23 %, surface densities 1.2 kg m^{-2}, and air cavity depth 100 mm.

As shown in the figure, a peak caused by the resonance, similar to a single MPP absorber, is shown at mid and high frequencies. This infers that a single resonator is produced by the MPP on the illuminated side with the MPP on the back side which plays the role of the back wall. An additional sound absorption at low frequencies is observed, which is not seen in other typical wall-backed MPP absorbers. Thus, a DLMPP can be shown to be an effective wide-band absorber.

Triple-leaf MPP space absorbers (TLMPP)

As Maa proposed [1,2], a wall-backed double-leaf MPP absorber using two MPP leaves makes two resonators, which produces two resonance peaks. When its air-cavity depths are adjusted so that the two peaks occur close enough to be merged into one broader peak, it offers wider sound absorbing frequency range [1,2]. Applying this idea to an MPP space absorber, replacing the rigid-back wall of a Maa's wall-backed double-leaf absorber with the third MPP makes a triple-leaf MPP space absorber (TLMPP), which is expected to produce a broader peak due to two resonances with an additional non-resonance low-frequency absorption from the leaves' acoustic flow resistances.

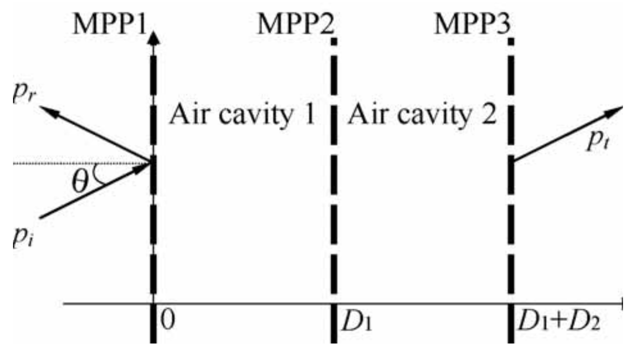


Figure 3: A sketch of a triple-leaf MPP space absorber (TLMPP).

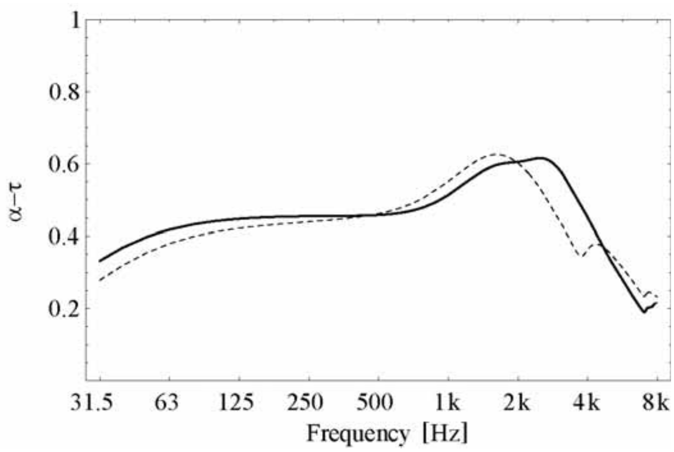


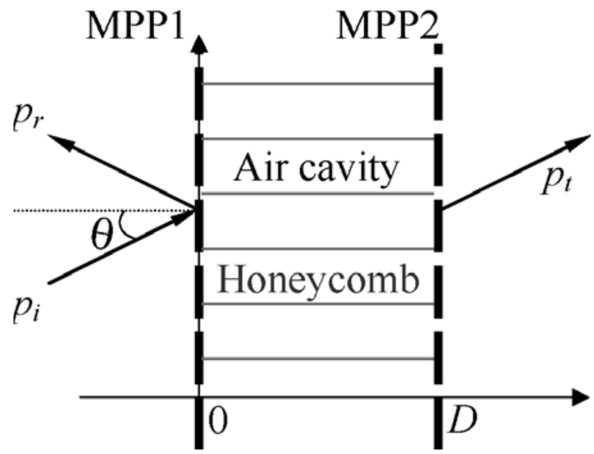
Figure 4: A comparison of the calculated field-incidence averaged sound absorptivity (α−τ) of a TLMPP (solid line), with a DLMPP (dashed). Hole diameters: 0.2 mm; thicknesses: 0.2 mm; perforation ratios: 0.8 %; depths of each cavity of the TLMPP: 25 mm; cavity depth of the DLMPP: 50 mm; surface densities: 1.8 kgm^{-2}.

Figure 3 shows a sketch of a TLMPP. Theoretical analyses have been made using a Helmholtz integral formulation, which is similar to that of DLMPPs, and a closed form solution for the difference of the absorption and transmission coefficients, α−τ is obtained. An example of the theoretical result is shown in Fig. 4. As is seen at mid-high frequencies there is a broader peak in which the two resonance peaks are merged. Therefore, the resonance absorption becomes somewhat broader than a DLMPP. Also in this case, the additional non-resonance sound absorption due to acoustic flow resistances of the MPPs appears as similar to a DLMPP at low frequencies. Thus, a TLMPP can also be effective as a wide-band space sound absorber.

Effect of honeycomb in the air-space

Use of a honeycomb is known to enhance the sound absorption of a porous material [23]. This is known as a "locally reacting absorber". A similar effect is also observed in a single MPP absorber (with a rigid-back wall) [20,21]. A sound wave obliquely incident upon the MPP is forced to travel normally to the incidence surface, which makes the absorption system to show characteristics similar to those in the normal incidence case. This results in a higher and broader resonance peak that is shifted to lower frequencies. Thus, a honeycomb can improve the sound absorption performance of an MPP sound absorber.

The honeycomb is also applied to multiple-leaf MPP space sound absorbers: Figure 5 shows a sketch and a photograph of an experimental specimen of a DLMPP with a honeycomb in the air-cavity. In Fig. 6 an example of the calculated and experimental absorption characteristics of the DLMPP with a honeycomb in the air-cavity, as well as those for the same DLMPP without the honeycomb are shown for comparison. Comparing these two results it is observed that the resonance peak is enhanced and shifted to lower frequencies, although no change is observed in the additional low frequency absorption. As mentioned above the low-frequency non-resonance absorption is caused by the acoustic flow resistance of the leaves, and does not depend on the cavity condition, whereas the resonance peak is largely affected by the honeycomb: a honeycomb makes the sound incident from the back side from the cavity normal to the leaf, and the sound incidence condition becomes close to that in the case of normal incidence (in which the peak is in general larger and appears at lower frequencies). A detailed study reveals that the optimal range of the MPP parameters becomes wider due to the honeycomb. This implies that the optimisation of the MPP parameters is less critical in the honeycomb attached case.



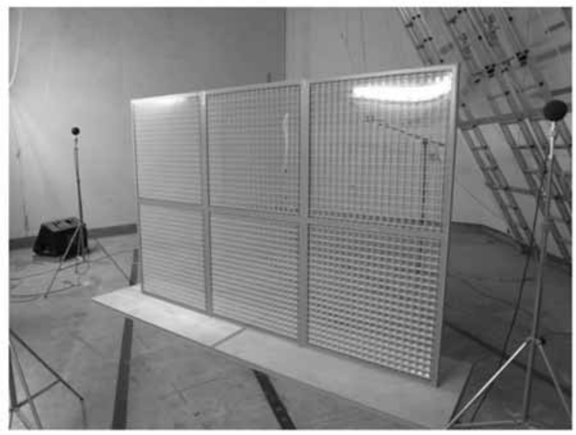


Figure 5: A sketch of a DLMPP with a honeycomb (top) and a photograph of its experimental specimen (bottom).

MULTIPLE-LEAF SOUND ABSORBERS WITH COMBINATION OF MPPS AND PERMEABLE MEMBRANES

One of the demerits of multiple-leaf MPP absorbers is that it uses more MPPs than a single absorber, which costs more than simple absorbers. In order to avoid this problem it can be useful if one of the MPPs in a multiple-leaf structure can be replaced by other less expensive materials. A possible alternative is permeable membranes. As an MPP and a permeable membrane are both acoustically permeable materials with a certain acoustic flow resistance, at least permeable membranes can act as a resistive element to give the additional low-frequency absorption in space absorbers. Also in wall-backed absorbers it can be effective to replace one MPP with a permeable membrane. Here, the possibility of the replacement of an MPP with a permeable membrane is discussed.

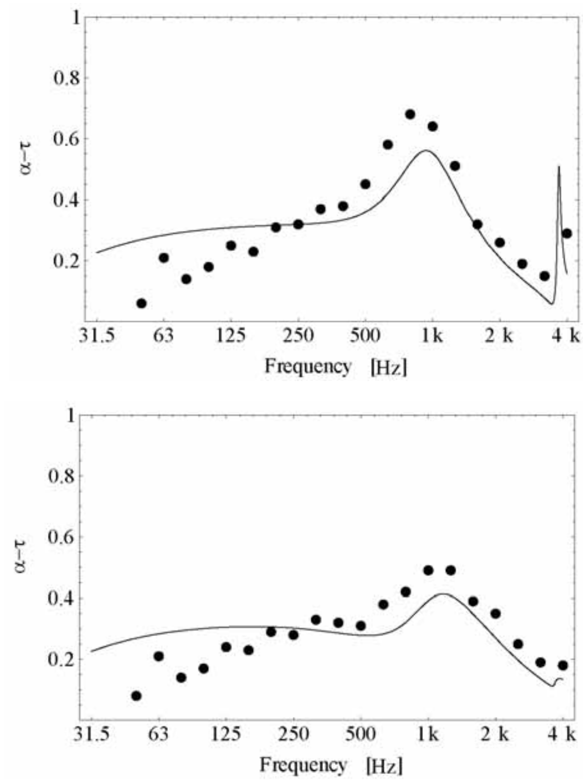


Figure 6: A comparison of the calculated (field-incidence averaged: solid line) and experimental (measured in a reverberation chamber: dots) results for a DLMPP with honeycomb (top) and without honeycomb (bottom). The two leaves have the same parameters: hole diameters 0.5 mm, thicknesses 1.0 mm, perforation ratios 1.23%, surface densities 1.2 kg m^{-2} and air-cavity depths 50 mm.

Sound absorbers with a rigid-back wall

The original form of a double-leaf MPP absorber proposed by Maa [2] consists of two MPPs with air-layer in-between with an air-back cavity with a rigid-back wall. This type uses two MPPs, which costs more than a simple MPP absorber. Hence, we consider the possibility of substituting one of those MPPs with a permeable membrane.

In this case, two alternative structures can be considered (Fig. 7). In Case A the second MPP (inside the air-cavity) is replaced with a permeable membrane, and in Case B the illuminated side MPP is replaced with a permeable membrane. The calculated examples of their sound absorption coefficients are shown in Fig. 8. In these figures the characteristics of the ordinary double-leaf MPP with a back wall are also shown for comparison. In Case A the absorption peak becomes broader and higher which offers more effective absorption in wider frequency range than the ordinary wall-backed double-leaf MPP. In Case B the characteristics are more similar to those of a porous blanket which shows higher absorption at high frequencies. Also it is noted that the contribution of the MPP is not significant because the resonance does not appear clearly. Hence, replacing the second MPP in the cavity with a permeable membrane can be a good alternative which can offer better absorption performance than the ordinary wall-backed double-leaf MPP absorbers.

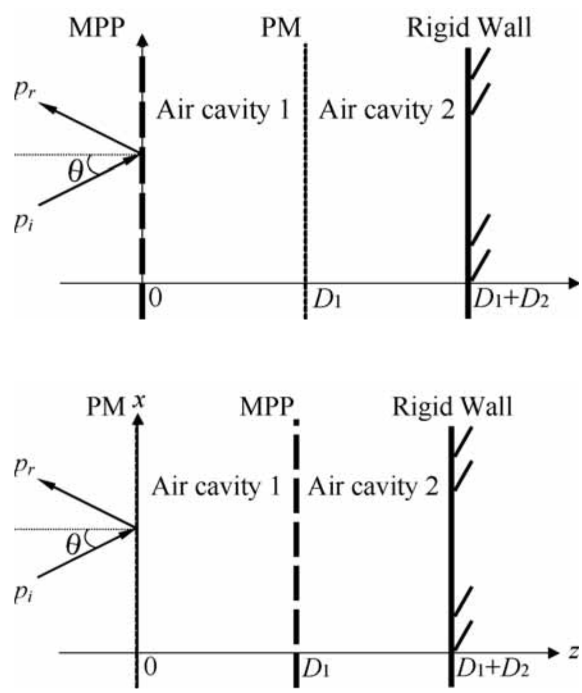


Figure 7: A sketch of wall-backed MPP-membrane combination absorbers: (Case A) MPP on the illuminated side with permeable membrane (PM) in the cavity (top); (Case B) Permeable membrane on the illuminated side with MPP in the cavity (bottom).

Space sound absorbers

The same idea as above can be also applied to multiple-leaf MPP space absorbers such as a DLMPP. Here, the absorption performance of space absorbers with a combination of an MPP and a permeable membrane is examined. One of the MPPs in a DLMPP (Fig. 1) is now replaced with a permeable membrane.

Figure 9 shows a calculated example of the absorption characteristics (α−τ) of multiple-leaf space absorber with a combination of MPP and permeable membrane (PM). Figure 9 compares the characteristics for a sound incidence on the MPP side, those for a sound incidence on the PM side, and the average of (a) and (b) which corresponds to the diffuse sound incidence to the both side (i.e., reverberation absorption coefficient [22]).

Figure 9 shows a typical resonance peak quite similar to a DLMPP in the case of MPP side incidence, whereas a high absorptivity plateau at high frequencies similar to porous materials appear in the case of PM side incidence. Actual absorbing characteristics are considered as are the averaged values, which show moderately high resonance absorption with a low-frequency absorption typical for DLMPP. Thus, this type space absorber can be a good substitution for a DLMPP and can be produced at lower cost.

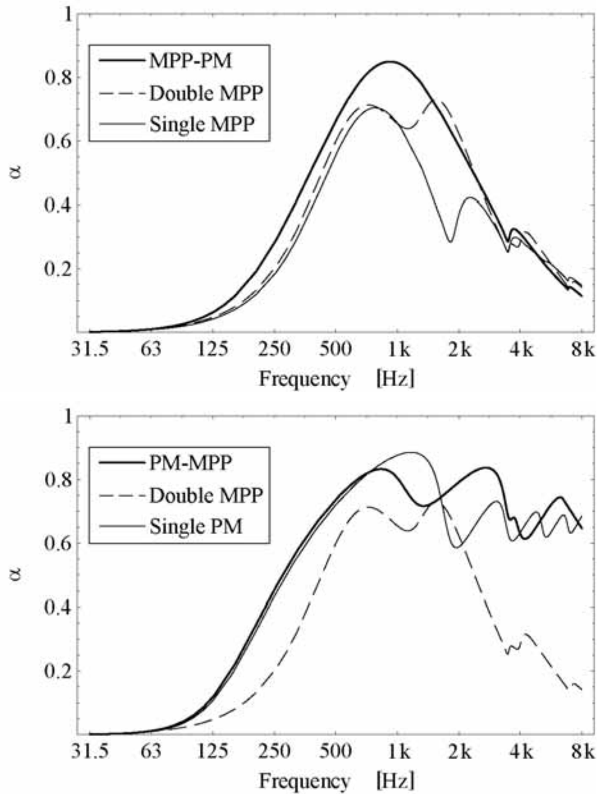


Figure 8: Calculated examples of the field-incidence averaged absorption coefficients of Cases A (top) and B (bottom), both indicated by thick lines, in comparison with the ordinary wall-backed single- MPP (left) and single PM absorber backed by a wall (thin lines) and wall-backed double-leaf MPP absorber (dashed line). MPP: hole diameter: 0.3 mm, thickness: 0.3 mm perforation ratio: 1.0 % surface density: 1.0 kgm^{-2}; PM : flow resistance: 816 Pa sm^{-1}; PM: surface density: 1.0 kgm^{-2} air cavity depths: 50 mm. The tension of the membrane is assumed to be zero.

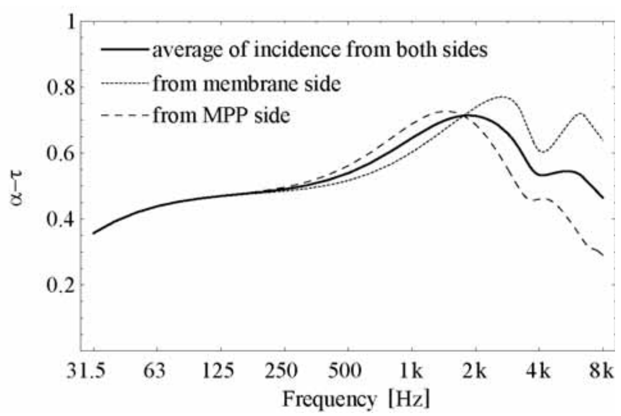


Figure 9: A calculated example of the field-incidence averaged absorption characteristics (α−τ) of a multiple-leaf space absorber with a combination of MPP and permeable membrane (PM). Dashed line: MPP on the illuminated side; Dotted line: PM on the illuminated side; Solid line: Sound incidence from both sides (averaged) which corresponds to reverberation absorption coefficient. MPP: hole diameter: 0.15 mm, thickness: 0.4 mm, perforation ratio: 1.5 %. PM : flow resistance: 816 Pa sm^{-1}; surface density: 3.0 kgm^{-2} air cavity depth: 50 mm. The tension of the membrane is assumed to be zero.

CONCLUDING REMARKS

In this paper, a series of our studies on multiple-leaf sound absorbers using an MPP is reviewed. A multiple-leaf MPP, particularly space absorber type, can be one of the effective alternatives for wideband sound absorbers. Also, a combination of an MPP and a permeable membrane can be a good alternative for multiple-leaf MPP structures: it can be of lower manufacturing cost and still offers reasonably high sound absorption performance.

ACKNOWLEDGEMENTS

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VIBRATIONAL CHARACTERISTICS OF ROLL SWAGE JOINTED PLATES

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The aim of this work is to model the vibrational behaviour of thin plates joined to a stiff orthogonal side plate using the technique of ‘roll swaging’. Swage joints are typically found in plate-type fuel assemblies for nuclear reactors. Since they are potentially liable to flow-induced vibrations, it is crucial to be able to predict their dynamic characteristics. It is shown that the contact between the plates resulting from the swage can be modelled assuming a perfect clamp of all the degrees of freedom except for the rotation around the axis parallel to the swage which is elastically restrained with a torsional spring. A modal analysis was performed on different specimens and the values of the first natural frequencies are used to find the equivalent stiffness of the torsional spring restraint by matching these frequencies with the results obtained from a finite element model (FEM).

INTRODUCTION

Plate-type fuel assemblies are used in several research reactors where a high neutron flux is desired. A plate-type assembly consists of several thin plates containing a uranium mixture, clad with aluminium and mounted in a box-type assembly. They are potentially affected by structural instabilities due to the interaction with the coolant flow [1-3]. Miller [4] used the wide beam theory to investigate the static instability of the plates. Other researchers modelled the plates using the thin plate theory assuming simply-supported boundary conditions [5] or fully-clamped edges [6]. Kim and Davis [7] improved the previous works assuming plates as laterally fixed but elastically restrained in rotation. The aim of this work is to give a theoretical and experimental justification of the model used in Ref. [7]. In a typical fuel assembly the plates are inserted into slots machined into the side walls of the fuel box. The clamping of the plates to the box is generally assured by a swage between adjacent plates. The swage is obtained by forcing a swage cutting wheel into the aluminium ridge between the slots. The shape of the cutting edge results in plastic deformation of the area surrounding the swage, and presses the material onto the plate, creating the clamp as shown in Figure 1.

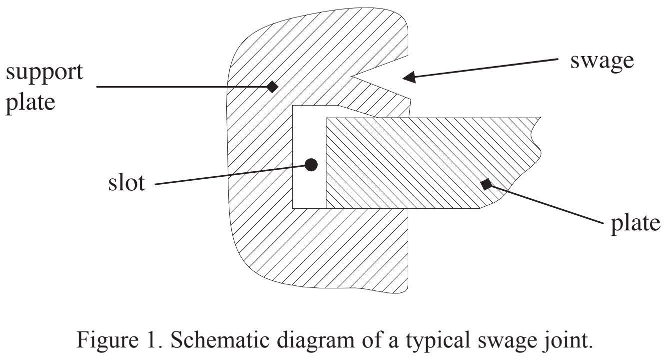

Figure 1. Schematic diagram of a typical swage joint.

The nature of the clamping is crucial to predict the vibrational behaviour of a fuel assembly. A perfect clamp completely constrains all six degrees of freedom at the edges of the fuel plates. In this work it is suggested that the swaging process results in a clamp that fixes all the degrees of freedom but the rotation around the axis parallel to the swage. For small rotations, as assumed by the linear vibration theory, the effect of the swage joint is shown to be a torsional spring whose stiffness is related to the quality of the swage. A good swage leads to a very high stiffness that approaches the ideal case of a perfect clamp, while a poor swage is likely to result in a lower stiffness value tending to the case of a simple support. The model used in this work is built according to the data of some specimens that were made at the Australian Nuclear Science and Technology Organisation (ANSTO) in order to show that the changing of some parameters in the swaging process results in a shift of the natural frequencies. The aim of this work is to present a theoretical explanation of the abovementioned experimental evidence.

The specimens, shown in Figure 2, were tested by clamping the bottom of the support in a vise and performing an impact hammer test to detect the natural frequencies by inspection of the Frequency Response Function (FRF). A laser vibrometer was used to measure the response close to the corner of the plate to maximise the visibility of all the modes.

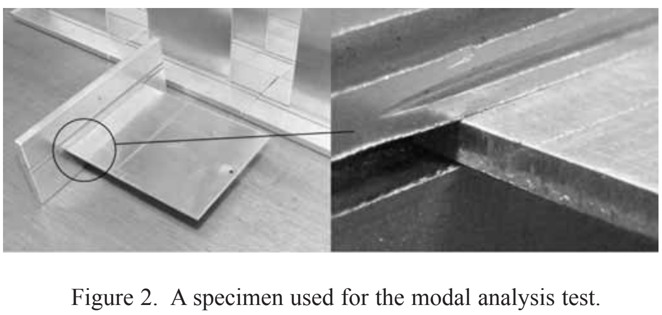

Figure 2. A specimen used for the modal analysis test.

SWAGING PROCESS SIMULATION

To understand the nature of the contact between the plate and the support, a simple two dimensional model is created to simulate the swaging process. It is solved using Nastran Implicit Non-Linear (Solver 600) [8]. The swage wheel is modelled as a rigid wedge and is moved toward the support. The material is Aluminium with density ρ = 2700 kgm^{-3}, Youngs modulus E = 69 MPa, Poisson ratio v = 0.33 and perfect plastic behaviour with Yield stress of 280 MPa. A more precise stress-strain curve should be used in order to improve the results. Figure 3 shows the deformation from the simulation. It can be seen that the deformed material presses the top of the plate to create the swage clamp. The grey scale variation is related to the Von Mises stress, details are not given since is not relevant for the following analysis.

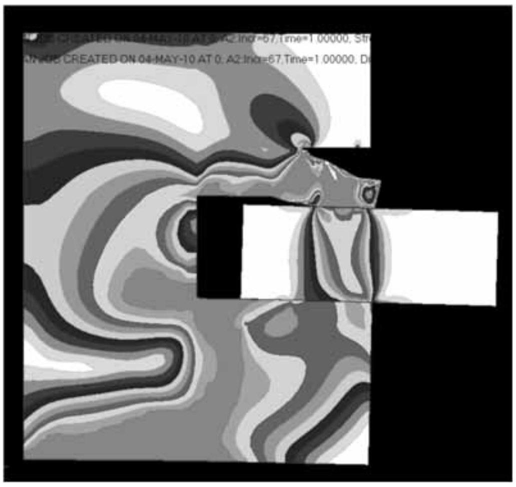

Figure 3. Swaging process simulation with Nastran.

Although the practical reaction force during the blade passing event may have a more complicated time history than a series of simple rectangular impulses and multiple force components, the general features of blade passing frequency component sound pressure due to the component force of equally spaced rotor blades passing a stator blade are qualitatively explained using this simple analysis.

DYNAMIC BEHAVIOUR OF THE SWAGE

Inspecting the results from the swaging simulation, a scheme is reproduced in Figure 4 to study the kinematics of the joint. It is reasonable to assume that after the swage is completed the plate can not translate along the y and z directions and it can not rotate around the y axis. In theory a translation in the x direction and a rotation around z are also possible but it would not affect the dynamics of the plate in bending vibrations. Only the in plane motion will be altered, but this happens at much higher frequency and is therefore considered constrained in this work. A small rotation θ in the x direction is possible around the contact line passing through point C_0 allowing a compression along the contact line L, as shown in Figure 4.

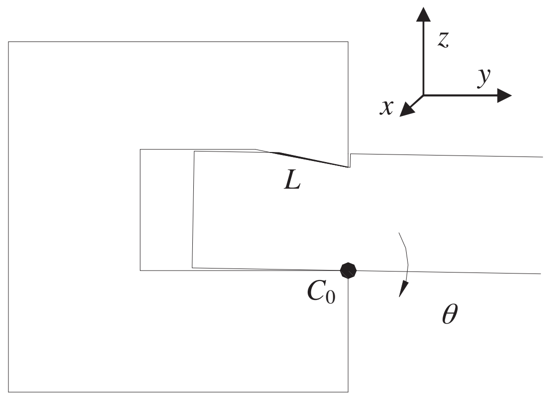

Figure 4. Small rotation of the plate in the swage joint.

Considering a small clockwise rotation θ around the point C_0, the component of the displacement normal to the contact line, the coordinate of which is $\xi_i \in [0,L]$, results in the compression of the material and then in a reaction force per unit rotation given by

$$\mathrm{d}f_i = \varepsilon E(\mathrm{d}A) r_i \cos\delta\ , \qquad \mathrm{d}A = w(\mathrm{d}\xi) \tag{1}$$

where ε is the strain, A and w are the area and width of the contact respectively. Other parameters arise from the geometry of the swage and are given by

$$r_i^2 = \xi_i^2 + r_0^2 - 2\xi_i r_0 \cos\gamma\ ,$$
$$\gamma = \frac{\pi}{2} + \varepsilon\ , \quad \varepsilon = \tan^{-1}(h_s / d_s) \tag{2}$$

$$\delta_i = \cos^{-1}\left(\frac{r_N^2 - r_0^2 + \xi_i^2}{2 r_N \xi_i}\right) \tag{3}$$

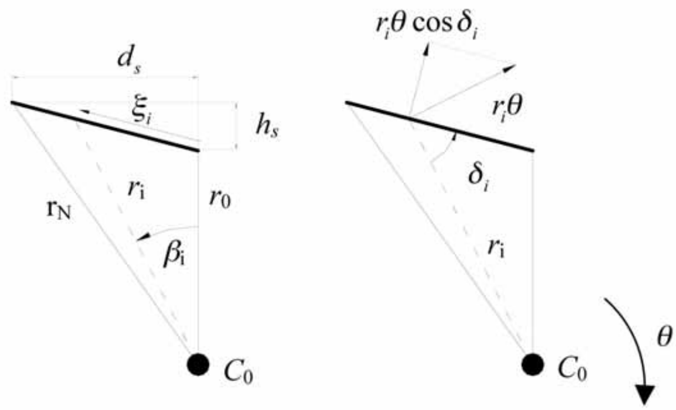

Figure 5. Kinematics of the swage joint.

The resultant force normal to the contact line is given by the integration of the elemental force, such that:

$$F_s = \int_{\xi=0}^{\xi=L} df_i \tag{4}$$

and with a lever arm b with respect to point C_0 given by $b = H_b \sin\varepsilon + 2/3L$. H_b is the height of the plate from point C_0 to point C_{-0} shown in Figure 6. The global effect is then a moment about point C_0 given by $M_s = F_s\theta b$ that can be rewritten as

$$M_s = K_s\theta\,, \qquad K_s = F_s b \tag{5}$$

K_s is the equivalent torsional stiffness of the swage joint. The exact value has to be found by matching experimental and FEM results for the first natural frequency.

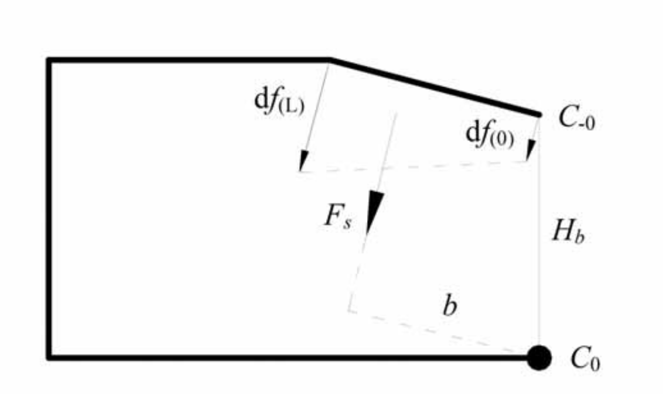

Figure 6. Force resultant from a small rotation of the plate.

A similar model can be used to calculate the reaction moment when the plate is rotating anticlockwise. In this case the plate is likely to rotate around the point C_{-0}. The moment is likely to have a different value leading to a non-linear spring characteristic, with different stiffness for the positive and negative rotations.

FE MODEL OF THE JOINED PLATES

A finite element model is built according to the dimension of the specimens. The plate and the support are meshed using QUADR plate elements. The support is clamped at the base to simulate the clamping of the real model to a vice. The plate is connected to the support using bush elements with adjustable torsional stiffness around the x axis. The first five mode shapes are reported in Figure 7 for a perfect clamp situation. The grey scale variation is related to the out of plane displacement.

In the case where a simple support was used, the first mode degenerated into a rigid body rotation (0 Hz) around the swage axis. The other mode shapes were practically the same except for a slightly bigger rotation at the connection with the support.

Different values for the torsional spring were used to simulate the conditions between a perfect clamp ($K_s \to \infty$) and a simple support ($K_s = 0$). The first five natural frequencies were normalised with respect to the perfect clamp case and are plotted versus the spring stiffness in Figure 8. Convergence to the perfect clamp case is achieved for a value of around K_s = 200 Nm/rad. Observing the slope of the curves it can be seen that the sensitivity first increases, reaching a maximum at 10 Nm/rad for the first natural frequency, and at 100 Nm/rad for the fifth natural frequency. The slope decreases again approaching the ideal clamped case. The lower order modes are more affected by the spring stiffness.

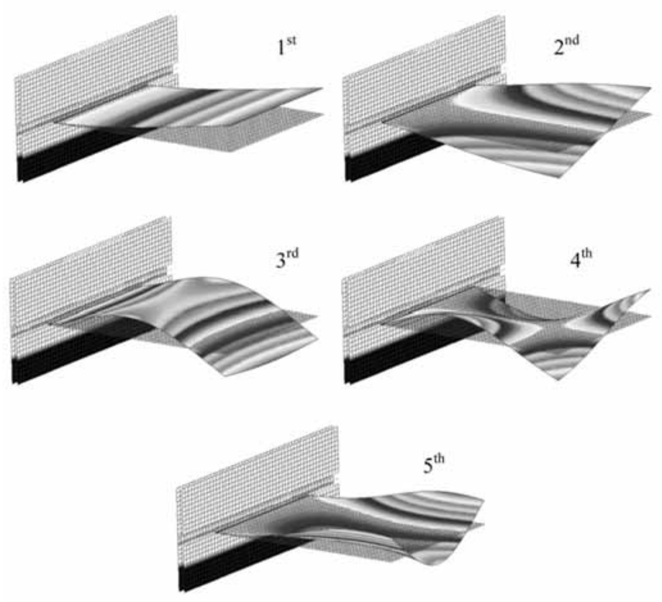

Figure 7. The first five mode shapes for a perfectly clamped plate.

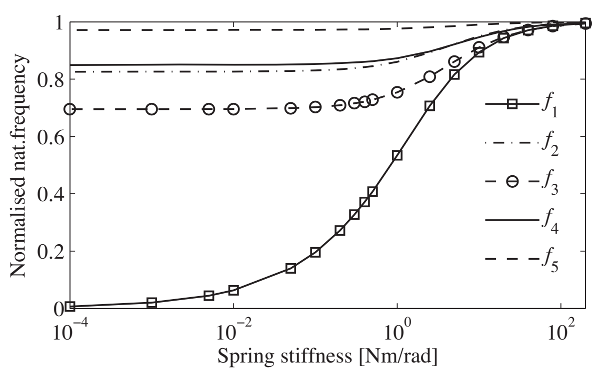

Figure 8. Variation of the natural frequencies with the spring stiffness (FEM results).

Figure 9 shows the same data as in the previous figure but arranged with respect to the natural frequency order. The presented arrangement of the results is useful to compare the frequencies with the experimental results.

Figure 10 shows the first five experimental natural of some specimens obtained by setting the roll swaging wheel at different heights with respect to the plates. It can be seen that the trend of the curves is similar to the FEM results presented in Figure 9.

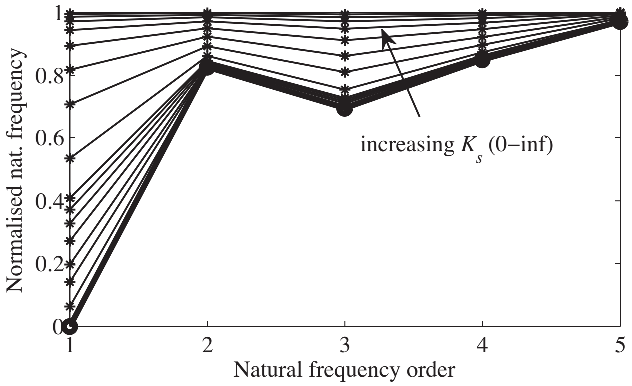

Figure 9. Normalised first five natural frequencies (FEM results).

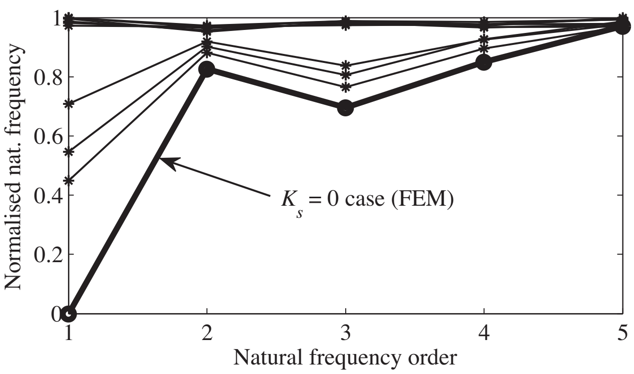

Figure 10. Normalised first five natural frequencies (experimental results).

Figure 11 shows the results updating the value of the torsional spring stiffness in the FE model to match the first natural frequency of two experimental results. The maximum error for the other frequencies is around 3%. A better estimation of the spring stiffness would need an update considering the global effect on more resonances in the frequency range of interest.

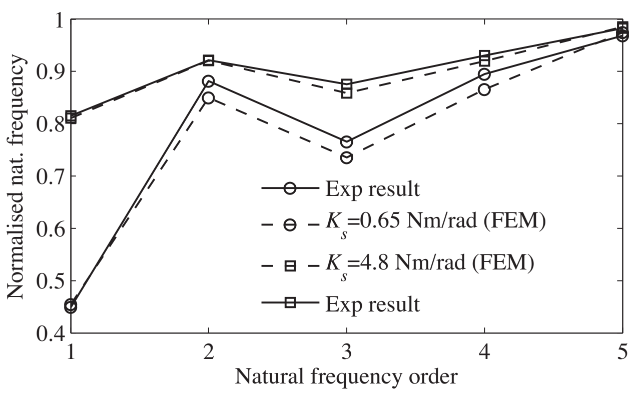

Figure 11. Matching of the natural frequencies by FEM updating.

The Frequency Response Function between the velocity of the plate surface and the impact force is shown in Figure 12 for a specimens with a good swage and another with a poor swage. The measurement was done close to a free corner of the plate using a laser vibrometer. In the case of the good swage, the specimen was hit on the plate itself, in the other case the specimen was hit at the base because otherwise was not possible to get clean results. The shift of the natural frequencies is clearly shown. A non linear behaviour is evident in the poor swaged plate by the leaning forward of the third and fourth resonance peaks. The reason could be the asymmetry of the torsional moment as discussed before or a hardening effect as given by cubic stiffness as in the Duffing oscillator [9].

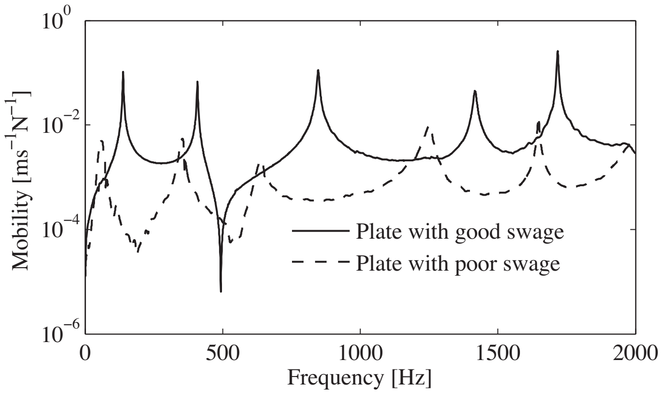

Figure 12. Mobility for two specimens

CONCLUSIONS

The work presented shows a first step to model the dynamic behaviour of swage joints and open a variety of issues that need to be studied in more depth. In particular a more precise model of the swaging process using the non-linear capabilities of FE modelling is required. A more sophisticated FEM updating process able to match the results of more natural frequencies will give an improved value for the equivalent torsional spring. In order to validate the results it is also necessary to set up a consistent method of swaging the specimens with the aim of finding the sensitivity of the natural frequencies to parameters such as height and depth (applied force) of the swage and the cutting profile of the swage wheel. The results can then be used to validate the swaging simulations from FEM. From a reliable FE model it may be possible to figure out the equivalent torsional spring stiffness without requiring an updating of the linear model with the experimental results.

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Technical Note

Note: Technical notes are aimed at promoting discussion. The views expressed are not necessarily those of the editors or the Australian Acoustical Society. Contributions are not formally peer-reviewed.

NOTE ON THE APPLICATIONS OF A SIMPLE ACOUSTIC IMMERSION INDEX

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The applicability of a simple acoustic immersion index for halls is investigated by calculating the values of the index for many well-documented halls and theatres. Correlations between the immersion index S_1 and other auditorium acoustic parameters are investigated, as well as the effectiveness of the index in describing subjective evaluations of halls. The index S_1 appears to broadly correspond to subjective ratings of halls, both from published data and from experiences in Australian halls, but is only weakly correlated with technical parameters describing the immersion of the sound field. However, the index appears to be reasonably correlated with Binaural Quality Index (BQI), and therefore may be useful as a "spaciousness index" as a means of estimating BQI during the early design of a hall.

INTRODUCTION

The immersion index S_1 has been proposed as a simple parameter able to describe the degree of immersion in the sound field experienced by a listener [1]. S_1 is able to be calculated using basic hall dimensions, hall volume, and reverberation time, and therefore can be obtained very early in the design process for a new hall, without requiring the detailed acoustic modelling necessary to calculate other more detailed acoustic parameters. S_1 therefore offers potential benefits in being an early design tool for hall designers, being able to be used with RT and details of the room geometry (e.g. volume per seat) as a "high level" estimate of the room acoustic properties.

S_1 was calculated for five well-known concert halls in [1], with the values of S_1 obtained corresponding with the subjective characteristics of the halls – e.g. Royal Festival Hall, with a low value of S_1, is subjectively dry and less enveloping, while Concertgebouw, with a higher value of S_1, is more reverberant.

The purpose of this study is to calculate the value of the immersion index S_1 for several halls where acoustic data is available, particularly from reference texts such as [2] and [3], and to investigate correlations between S_1 and more complicated room acoustic parameters, and between S_1 and subjective rankings of halls.

IMMERSION INDEX, S_1

S_1 in its original form was derived for rectangular hall geometries. In this study, S_1 will be calculated for actual concert halls, which generally are not purely rectangular in plan. Therefore, the modified form of S_1, (denoted S_{1A} in this paper; Equation (5) in [1]), which is applicable for any shape, will be used:

$$S_{1A} = 10 \log_{10} (25T_{60}/L) \quad \text{dB} \tag{1}$$

where T_{60} is the reverberation time of the hall and L is the average length of the hall.

Although S_1 was developed for evaluating the immersion of organ music, in principle it may be used for the evaluation of halls for other types of music provided that the assumptions used to derive the index are met. The main assumption used to derive the first-order index S_1 is that the entire acoustic power of the source is spread uniformly across the entire cross-section of the hall as it propagates (assuming that losses from surfaces near the source are negligible).

For solo organ music with an organ at one end of a hall, or for unaccompanied choir music, in both cases with no orchestra on the platform, the assumption that reflection losses close to the source are negligible seems reasonable as a first estimate for high-frequency sound, considering that typical hall materials are highly-reflective at mid to high frequencies. This assumption may also be useful for small chamber groups where the source size is small relative to the platform area and therefore the area surrounding the source is largely acoustically-reflective. However, for orchestral music, or for halls with audience seating surrounding the stage, the presence of the orchestra or audience would mean there would be significant high-frequency absorption close to the source, and this assumption would be less valid. Therefore, S_1 is expected to be more applicable for organ music, small ensembles and unaccompanied choir music than for orchestral music.

ACOUSTIC PARAMETERS CONSIDERED

Being an "immersion index", S_1 would be expected to provide some description of how uniform is the sound field in a hall. Where a diffuse reverberant field is dominant, a mostly-uniform sound field would be expected, resulting in a high degree of subjective "immersion" in the sound field, as well as a high value of S_1.

S_1 would be expected to relate to other acoustic parameters which describe the spatial quality of the sound field (Interaural cross-correlation coefficient IACC and lateral energy fraction, LF_{80}) or which describe the balance between early and reverberant sound (clarity, C_{80}). IACC in particular has become increasingly used to describe the degree to which the sound field in a hall is uniform. Two main time periods are used for IACC: $IACC_E$, which is based on energy up to 80 ms after the direct sound, and $IACC_L$, which is based on the energy received between 80 and 1000 ms after the direct sound. Higher values of IACC indicate that the sound field experienced by the two ears is more uniform. $IACC_L$ theoretically may be used as an index for the envelopment/"immersion" experienced by listeners, but it has been found to not vary significantly from hall to hall and is therefore not considered a useful index. $IACC_E$, usually expressed as the Binaural Quality Index (BQI; 1-$IACC_E$), can be used to represent the spaciousness of a hall. Lateral energy fraction, LF_{80}, is a measure of the proportion of energy arriving at a receiver from the sides (lateral directions) within the first 80 ms after the direct sound. Musical clarity, C_{80}, is the ratio (expressed in decibels) of the acoustic energy arriving within the first 80 ms after the direct sound to the acoustic energy arriving after 80 ms.

CALCULATION OF INDEX

S_1 has been calculated for 94 concert halls, recital halls and opera houses, using published room data from reference books ([2], [3]), or from Arup measurements from completed projects. The hall volume, reverberation time (occupied) averaged over 500 Hz and 1 kHz and the hall length have been used to calculate S_1 using the modified form of the index given in Equation (1). The data used to calculate S_1 and the calculated value of S_1 using Equation (1) for each hall is listed in Table 1, with values of other acoustic parameters of interest (BQI, $IACC_L$, LF_{80} and C_{80}) for each hall, where available. Due to the different sources of the hall data, not all parameters are available for every hall. A three letter code is assigned to each hall to assist in labelling data points on graphs.

The calculated values of S_1 for the halls have been compared to other acoustic parameters and to subjective rankings for the halls, sourced from the conductor surveys presented in Beranek [2].

Table 1: Data used to calculate immersion index S_1, and other room acoustic parameters of interest for each hall.

City	Hall	Type	Code	V (m^3)	L (m)	T_{60} (s)	S_{1A} (dB)	BQI	1-$IACC_L$	LF_{80}	C_{80}
Aldeburgh	Snape Maltings	Recital Hall	ASM	7,590	41	1.80	0.4			0.24	
Amsterdam	Concertgebouw	Concert Hall	CBW	18,780	43	2.00	0.7	0.56	0.88	0.18	-3.6
Baltimore	Meyerhoff Hall	Concert Hall	BMH	21,524	49	2.00	0.1	0.52	0.86	0.17	-2.0
Basel	Stadt-Casino	Concert Hall	BSC	10,471	33	1.75	1.2	0.60	0.87		-2.6
Bayreuth	Festspielhaus	Opera Theatre	FES	10,308	32	1.55	0.8				
Belfast	Waterfront Hall	Concert Hall	BEL	30,800	51	1.88	-0.4			0.195	0.0
Berlin	Deutsches Oper	Opera Theatre	BDO	10,800	33	1.35	0.1				
Berlin	Kammermusiksaal	Recital Hall	KAM	12,500	45	1.70	-0.2				-1.8
Berlin	Konzerthaus	Concert Hall	KHS	15,500	49	2.05	0.2	0.67	0.85		-3.1
Berlin	Philharmonie	Concert Hall	BPH	21,000	66	1.90	-1.4	0.45	0.86		-0.6
Bonn	Beethovenhalle	Concert Hall	BBH	15,716	35	1.70	0.9				
Boston	Symphony Hall	Concert Hall	BSH	18,750	49	1.80	-0.4	0.60	0.82	0.235	-2.6
Bristol	Colston Hall	Concert Hall	BCH	13,450	48	1.70	-0.5			0.185	0.2
Buenos Aires	Teatro Colon	Opera Theatre	BTC	20,570	34	1.63	0.7	0.62	0.80		0.8
Buffalo	Kleinhans Music Hall	Concert Hall	KMH	18,240	52	1.35	-1.9	0.30	0.65	0.1	2.8
Buxton	Buxton Opera House	Opera Theatre	BOH	3,100	18	0.90	1.0			0.23	
Cambridge	Faculty of Music	Recital Hall	CFM	4,100	29	1.50	1.1			0.25	
Canberra	Llewellyn Hall	Concert Hall	LLH	10,500	43	2.00	0.7			0.25	-0.6
Cardiff	St Davids Hall	Concert Hall	STD	22,000	48	1.90	0.0			0.17	-0.7
Cardiff	Wales Millennium Centre	Opera Theatre	WMC	11,500	34	1.30	-0.1				
Chicago	Orchestra Hall	Concert Hall	COH	17,410	40	1.60	0.0				
Christchurch	Christchurch Town Hall	Concert Hall	CTH	20,500	43	1.80	0.2			0.14	1.6
Cleveland	Severance Hall	Concert Hall	SEV	15,690	33	1.48	0.5	0.54	0.79	0.14	0.0
Copenhagen	Opera House	Opera Theatre	COP	10,700	30	1.40	0.7				2.1
Costa Mesa	Segerstrom Hall	Opera Theatre	SEG	27,800	49	2.20	0.5	0.58	0.86	0.225	-0.7
Croydon	Fairfield Hall	Concert Hall	CFH	15,400	48	1.70	-0.5			0.15	
Derby	Assembly Rooms	Concert Hall	DAR	15,401	43	1.10	-1.9			0.2	
Denver	Boettcher Hall	Concert Hall	DBH	37,444	46	2.40	1.2	0.25		0.11	0.6
Dresden	Semperoper	Opera Theatre	DSO	12,500	26	1.68	2.1	0.71			
Edinburgh	Usher Hall	Concert Hall	USH	16,000	45	1.70	-0.2			0.3	-1.3
Edmonton/Calgary	Alberta Jubilee Auditoria (before renovations)	Opera Theatre	EJA	21,492	58	1.40	-2.2			0.135	3.7
Fort Worth	Bass Performance Hall	Opera Theatre	BFW	27,300	48	1.95	0.1	0.46	0.71		-2.0
Glasgow	Royal Concert Hall	Concert Hall	GCH	22,700	43	1.75	0.1			0.215	0.9

Glyndebourne	Festival Opera	Opera Theatre	GFO	8,287	29	1.25	0.4			0.155	4.5
Jerusalem	Congress Hall	Concert Hall	JCH	24,700	60	1.75	-1.4	0.53	0.84		-0.4
Leipzig	Altes Gewandhaus (1781)	Recital Hall	LAG	2,130	23	1.30	1.5				
Leipzig	Gewandhaus (1981)	Concert Hall	LGH	21,560	54	2.00	-0.3				
Leipzig	Neus Gewandhaus (1884)	Concert Hall	LNG	10,620	39	1.60	0.1				
Lenox	Koussetitzkty Music Shed	Concert Hall	KMS	42,480	76	1.89	-2.1	0.32	0.76	0.11	-3.8
Liverpool	Philharmonic Hall	Concert Hall	LPH	13,560	50	1.50	-1.2			0.17	1.0
London	Barbican Hall (Before 2001)	Concert Hall	BAR	17,750	44	1.65	-0.3			0.12	-1.2
London	Barbican Hall (Renovated 2001)	Concert Hall	BHR	17,000	44	1.40	-1.0				0.3
London	Coliseum	Opera Theatre	COL	13,600	33	1.40	0.3			0.18	
London	Kings Place	Recital Hall	LKP	3,540	25	1.70	2.3				
London	Queen Elizabeth Hall	Recital Hall	QEH	9,600	44	2.05	0.7			0.18	
London	Royal Albert Hall	Concert Hall	RAH	86,650	67	2.50	-0.3			0.14	0.5
London	Royal Festival Hall	Concert Hall	RFH	21,950	51	1.45	-1.5				1.0
London	Royal Festival Hall (assisted resonance)	Concert Hall	RFH_R	21,950	51	1.80	-0.5			0.195	0.8
London	Royal Opera House	Opera Theatre	ROH	12,250	28	1.10	0.0			0.19	4.8
London	Wigmore Hall	Recital Hall	WIG	2,900	24	1.50	1.9			0.25	
Madrid	Auditorio Nacional de Música	Concert Hall	ANM	20,000	54	1.74	-1.0			0.31	-0.6
Manchester	Free Trade Hall	Concert Hall	FTH	15,430	48	1.50	-1.0			0.24	1.1
Manchester	Bridgewater Hall	Concert Hall	BWH	25,000	49	2.00	0.1			0.25	-1.5
Melbourne	Hamer Hall	Concert Hall	HAM	26,900	53	2.20	0.2				
Melbourne	Melbourne Recital Centre	Recital Hall	MRC	9,000	37	1.90	1.1				-2.5
Minneapolis	Minnesota Orchestra Hall	Concert Hall	MOH	18,975	49	1.85	-0.2				
Milan	La Scala	Opera Theatre	LSC	11,252	30	1.25	0.1	0.49	0.74		2.9
Munich	Gasteig Philharmonie	Concert Hall	GAS	29,737	48	2.10	0.4			0.11	-0.4
Munich	Herkulessaal	Concert Hall	HKS	13,592	42	2.00	0.8				
New York	Avery Fisher Hall	Concert Hall	AVF	18,691	52	1.80	-0.6			0.12	-2.2
New York	Carnegie Hall	Concert Hall	CAR	24,270	52	1.70	-0.9				
New York	Metropolitan Opera House	Opera Theatre	MET	24,724	40	1.55	-0.1	0.60	0.83		1.5
Northampton	Derngate	Concert Hall	DER	13,500	45	1.80	0.0			0.17	
Nottingham	Royal Concert Hall	Concert Hall	RCN	17,510	50	1.90	-0.2			0.21	
Oslo	Opera House	Opera Theatre	OOH	11,789	31	1.70	1.4				
Paris	Opera Garnier	Opera Theatre	PAR	10,000	28	1.10	0.0	0.47	0.79		4.4
Poole	Wessex Hall	Concert Hall	PWH	12,430	41	1.70	0.1			0.2	
San Francisco	Davies Hall	Concert Hall	SFD	24,070	54	1.85	-0.7	0.41	0.84		-1.5
San Francisco	War Memorial Opera	Opera Theatre	WMO	20,900	37	1.50	0.1				
Salt Lake City	Abranavel Hall	Concert Hall	SLC	19,500	38	1.75	0.6	0.56	0.84		-2.0
Salzburg	Festspielhaus	Opera Theatre	SFH	15,500	30	1.50	1.0			0.14	-0.7
Sapporo	Kitaka Concert Hall	Concert Hall	SKH	28,800	50	1.80	-0.5	0.44	0.83	0.12	0.7
Stuttgart	Liederhalle	Recital Hall	SLH	16,000	42	1.65	-0.1			0.13	2.0
Sydney	City Recital Hall	Recital Hall	APL	10,850	35	1.75	1.0				-1.0
Sydney	SOH Concert Hall	Concert Hall	SOH	24,600	67	2.20	-0.8				
Sydney	SOH Opera Theatre	Opera Theatre	SOP	8,200	33	1.10	-0.8				
Taipei	Taipei Cultural Centre Concert Hall	Concert Hall	TCC	16,700	45	2.00	0.4	0.83		0.245	-4.0
Tel Aviv	Mann Auditorium	Concert Hall	MNN	21,238	47	1.50	-0.9	0.37	0.82		-0.9
Tokyo	Asahi Hall	Recital Hall	AHT	5,800	31	1.73	1.4	0.66	0.85		-0.6
Tokyo	Bunka Kaikan	Concert Hall	TBK	17,300	47	1.50	-1.0	0.56	0.85	0.19	-0.7
Tokyo	Dai-Ichi Seimei Hall	Recital Hall	DAS	6,800	31	1.56	0.9	0.86			-0.5
Tokyo	Metropolitan Art Space	Concert Hall	MAS	25,000	48	2.15	0.5	0.58	0.86		-1.2
Tokyo	New National Theatre	Opera Theatre	NNT	14,500	31	1.50	0.8	0.64	0.84		1.7

Tokyo	Opera City Concert Hall	Concert Hall	TOC	15,300	47	1.96	0.2	0.70	0.88		-2.8
Tokyo	Suntory Hall	Concert Hall	TSH	21,000	55	2.00	-0.4	0.51	0.84	0.165	-0.9
Valencia	Palau de la Música	Concert Hall	PMV	15,400	40	2.05	1.0			0.35	-4.0
Vienna	Grosser Musikveriensaal	Concert Hall	MKV	15,000	53	2.00	-0.3	0.63	0.86	0.18	-4.3
Vienna	Konzerthaus	Concert Hall	VKH	16,600	37	1.88	1.0	0.66			-1.2
Vienna	Staatsoper	Opera Theatre	VSO	10,665	27	1.30	0.9	0.61	0.80		-0.7
Washington	Kennedy Center Concert Hall (pre-renovation)	Concert Hall	KCW	22,300	37	1.85	1.0	0.59	0.86	0.22	-0.4
Washington	Kennedy Center Opera House	Opera Theatre	KCO	13,027	32	1.50	0.7				
Watford	Watford Town Hall	Concert Hall	WTH	11,600	50	1.45	-1.4			0.15	
Wellington	Michael Fowler Centre	Concert Hall	MFC	22,700	48	2.00	0.2				
Worcester	Mechanics Hall	Concert Hall	WMH	10,760	41	1.55	-0.2	0.54	0.78	0.2	-1.5
Zurich	Grosser Tonhalle	Concert Hall	ZGT	11,398	38	2.05	1.3	0.63	0.88		-4.0

SUBJECTIVE RANKINGS

Beranek [2] presents two subjective rankings of halls, one for concert halls and one for opera theatres, based on interviews and questionnaires of conductors and music critics. These subjective rankings have been used to examine the calculated S_1 values for these halls to investigate whether there is a relationship between S_1 and the acoustic quality of a hall.

Subjective Rankings – Concert Halls

For concert halls, Beranek ranks 58 concert halls in order of perceived quality. Acoustic data was available for 36 of these halls, and S_1 has been calculated for these halls. The subjective rankings divide the halls into three groups:

- 20 "upper group" halls (here denoted Group A), which are considered to be of highest quality
- 19 "middle group" halls (here denoted Group B), which are judged to lie below the Group A hall in quality. Note that no ranking order was given for the Group B halls as they were not considered to be clearly separated in acoustic quality; the ranking numbers used in this paper were based on the alphabetical listing of the halls;
- 19 "lower group" halls (here denoted Group C), which were ranked as being below both Group A and Group B in quality.

A summary of the 36 halls included in Beranek's ranking, including calculated S1 values is provided in Table 2.

Figure 1 presents an overview of the relationship between the S_1 index for each hall and the subjective hall ranking from Beranek. It can be seen from Figure 1 that generally speaking, the higher ranked halls have higher calculated values of S_1. However, there is considerable spread in the dataset, as can be seen when a linear regression curve is applied to the data, as shown in Figure 2. The correlation between S_1 and subjective ranking is relatively weak, with a coefficient of determination (R^2 value) for a linear regression of 0.25, and a standard deviation of 0.7 dB.

Table 2: Subjective Concert Hall Rankings (from Beranek [2])

City	Hall	Code	Subjective Ranking	S_{1A} (dB)
Vienna	Grosser Musikveriensaal	MKV	1	-0.3
Boston	Symphony Hall	BSH	2	-0.4
Berlin	Konzerthaus	KHS	4	0.2
Amsterdam	Concertgebouw	CBW	5	0.7
Tokyo	Opera City Concert Hall	TOC	6	0.2
Zurich	Grosser Tonhalle	ZGT	7	1.3
New York	Carnegie Hall	CAR	8	-0.9
Basel	Stadt-Casino	BSC	9	1.2
Cardiff	St Davids Hall	STD	10	0.0
Bristol	Colston Hall	BCH	12	-0.5
Costa Mesa	Segerstrom Hall	SEG	14	0.5
Salt Lake City	Abranavel Hall	SLC	15	0.6
Berlin	Philharmonie	BPH	16	-1.4
Tokyo	Suntory Hall	TSH	17	-0.4
Tokyo	Bunka Kaikan	TBK	18	-1.0
Baltimore	Meyerhoff Hall	BMH	20	0.1
Christchurch	Christchurch Town Hall	CTH	24	0.2
Cleveland	Severance Hall	SEV	25	0.5
Jerusalem	Congress Hall	JCH	27	-1.4
Leipzig	Gewandhaus	LGH	29	-0.3
Munich	Gasteig Philharmonie	GAS	31	0.4
Tokyo	Metropolitan Art Space	MAS	34	0.5
Stuttgart	Liederhalle	SLH	41	-0.1
New York	Avery Fisher Hall	AVF	42	-0.6
Edinburgh	Usher Hall	USH	44	-0.2
Glasgow	Royal Concert Hall	GCH	45	0.1
London	Royal Festival Hall	RFH	46	-1.5
Liverpool	Philharmonic Hall	LPH	47	-1.2
Manchester	Free Trade Hall	FTH	48	-1.0
Edmonton/Calgary	Alberta Jubilee Auditoria (before renovations)	EJA	50	-2.2
Sydney	SOH Concert Hall	SOH	53	-0.8
San Francisco	Davies Hall	SFD	54	-0.7
Tel Aviv	Mann Auditorium	MNN	55	-0.9
London	Barbican Hall	BAR	56	-0.3
Buffalo	Kleinhans Music Hall	KMH	57	-1.9
London	Royal Albert Hall	RAH	58	-0.3

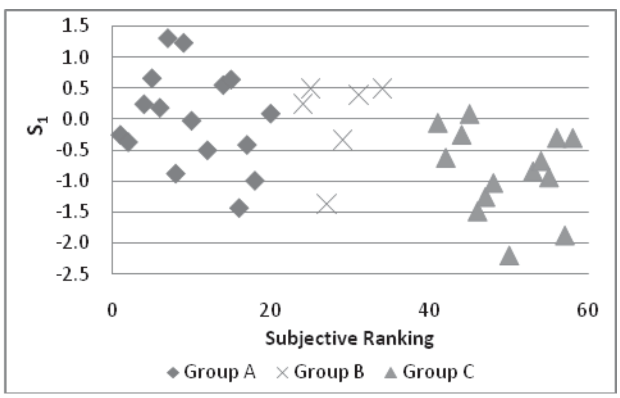

Figure 1: Comparison of calculated S_1 values against subjective concert hall ranking.

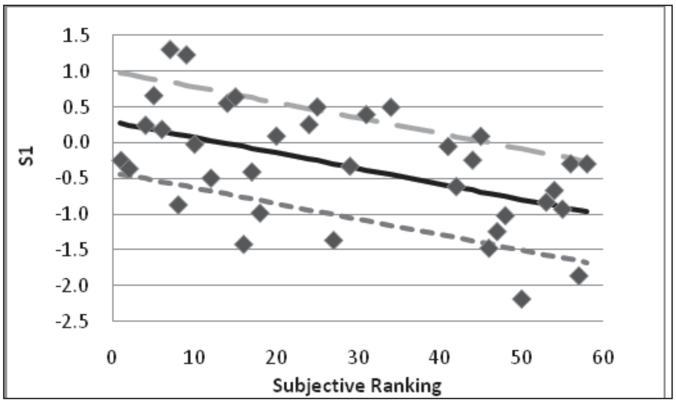

Figure 2: Relationship between S_1 index and subjective hall ranking, showing linear regression relationship (solid black line) and standard deviation of the dataset (dashed grey lines).

The average value of S_1 for each of the three groups has been calculated:

- Group A: 0.1 dB (standard deviation 0.8 dB)
- Group B: 0.0 dB (standard deviation 0.7 dB)
- Group C: -0.8 dB (standard deviation 0.7 dB)

The average values of S_1 for each group match the subjective rankings for these halls, however there is considerable overlap between the three groups. Therefore, S_1 does not appear to distinctly separate concert halls in the different subjective groups, although the overall trend is for higher S_1 for higher-rated halls.

Subjective Rankings – Opera Theatres

Beranek also presents a subjective ranking of 21 opera theatres, based on surveys of conductors. Acoustic data for 11 theatres was available, and was used to calculate S_1. A summary of the rankings and calculated S_1 values for the 11 opera theatres is provided in Table 3.

Table 3: Subjective Opera House Rankings (from Beranek [2])

City	Opera House	Code	Subjective Ranking	S_{1A} (dB)
Buenos Aires	Teatro Colón	BTC	1	0.7
Dresden	Semperoper	DSO	2	2.1
Milan	La Scala	LSC	3	0.1
Tokyo	New National Theatre	NNT	4	0.8
Paris	Opéra Garnier	PAR	7	0.0
Vienna	Staatsoper	VSO	9	0.9
New York	Metropolitan Opera House	MET	10	-0.1
Salzburg	Festspielhaus	SFH	11	1.0
San Francisco	War Memorial Opera House	WMO	13	0.1
London	Royal Opera House	ROH	14	0.0
Berlin	Deutsches Opera	BDO	17	0.1

A comparison of the predicted S_1 values and the subjective ranking for the 11 opera theatres is presented in Figure 3. As for concert halls, generally speaking the higher-ranked opera theatres have higher values of S_1. However, there again is considerable spread in the data, with a R^2 value for a linear regression of 0.26, and a standard deviation of 0.6 dB.

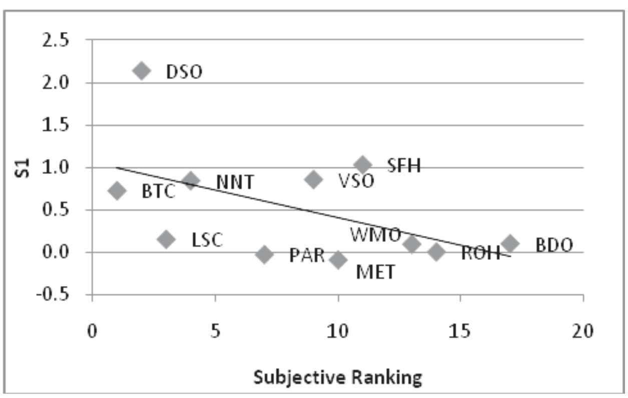

Figure 3: Comparison of S_1 index and subjective ranking for opera theatres, showing linear regression relationship (black line)

AUSTRALIAN HALLS

It is considered informative to focus on Australian halls, which may be more familiar, in order to investigate the subjective aspects of the S_1 index. The predicted values for the six Australian halls included in this study range between -0.8 dB (Sydney Opera House Concert Hall and Opera Theatre), to 1.1 dB (Melbourne Recital Centre). Subjectively, the high values of ~1 dB for the Sydney City Recital Hall and Melbourne Recital Centre corresponds to the spacious and enveloping sound in these halls, which is perhaps assisted by these halls being essentially "shoebox" shape.

For the almost fan-shaped geometry of Llewellyn Hall, and the Sydney Opera House Opera Theatre, the complicated surround shape of the Sydney Opera House Concert Hall, and the wide outer walls and high ceiling of Melbourne Hamer Hall, the resultant sound field is less enveloping and spacious, corresponding to the lower values of S_1 for these spaces. The calculated values of S_1 for Australian halls appear to match the subjective experiences of the sound field in these halls.

BINAURAL QUALITY INDEX, BQI ($1\text{-}IACC_E$)

Although S_1 is intended as an index for describing the listener envelopment due to the late reverberant sound, it is informative to consider its usefulness in describing the spaciousness of the early sound field. Accordingly, the calculated S_1 values for 36 halls has been compared to the BQI values for halls (where available), calculated using the unoccupied $IACC_E$ data, as shown in Figure 4. Interestingly, BQI and S_1 appear to be reasonably well correlated (R^2 0.61), with increased S_1 generally corresponding to an increased value of BQI. The standard deviation in the dataset is 0.09 BQI. The correlation is stronger than seen between S_1 and the subjective ranking of the hall. Three halls were excluded from the dataset used to generate the regression relationship: Tokyo Dai-Ichi Seimei Hall (DAS) and Taipei Cultural Centre (TCC), which both have significantly higher values of BQI than other halls, and Denver Boettcher Hall (DBH), which has a significantly lower value for BQI (perhaps due to its "surround" plan form). Beranek notes that BQI greater than 0.5 is associated with satisfactory halls, with the highest rated halls having BQI over 0.6. This roughly corresponds to S_1 values greater than ~0 dB.

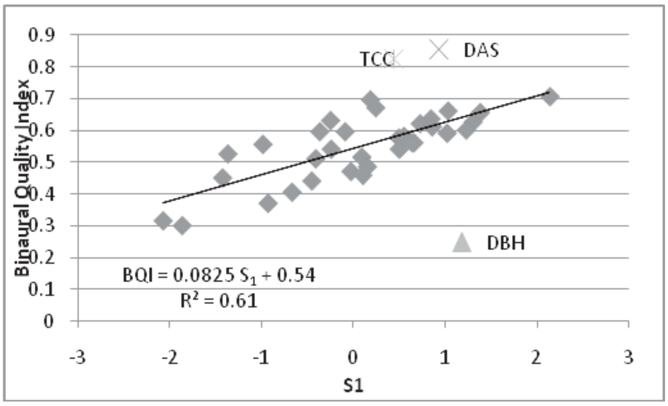

Figure 4: Comparison of S_1 index and Binaural Quality Index, showing linear regression relationship (solid black line). Data points not included in the regression are shown separately and labelled with hall code.

LATERAL ENERGY FRACTION

Although lateral energy fraction (LF_{80}) was found by Beranek to be less useful in accounting for the subjective ranking of halls, given the correlation seen between S_1 and BQI, it is of interest to see whether there is a similar relationship with LF_{80}.

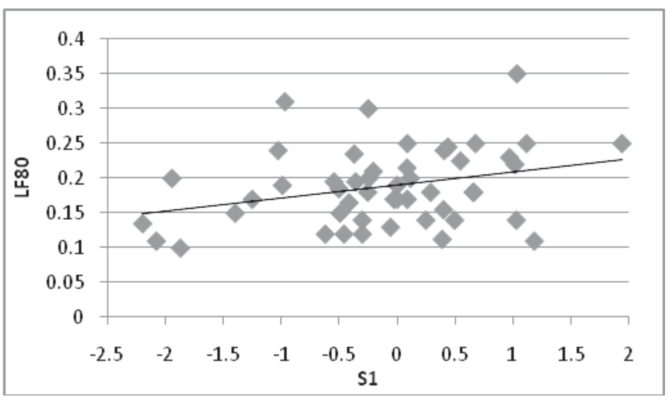

Figure 5: Comparison of S_1 index and Lateral Energy Fraction LF_{80}, showing linear regression relationship

There is only a very weak correlation (R^2 0.09) between S_1 and LF_{80}, with significant spread in the dataset (standard deviation of 0.05 LF_{80}). Other than a very broad trend for increasing LF_{80} with increasing S_1, there seems to be no real relationship between the parameters. This is perhaps not surprising, since LF_{80} is not a particularly useful parameter for resolving differences between halls [2], and since S_1 is only calculated with the hall RT and length and therefore does not consider the more detailed room shape, which may have a significant effect on LF_{80}.

IMMERSION, 1-$IACC_L$

One of the only numerical parameters suggested to describe the degree of listener envelopment or immersion in the sound field is the IACC (late, from 80 ms to 1000 ms), $IACC_L$, (usually expressed as the parameter 1-$IACC_L$), although this was found to be approximately constant for most halls and was therefore not considered to be a particularly useful parameter. However, in the absence of other technical parameters to describe immersion, the S_1 and 1-$IACC_L$ values for 31 halls where $IACC_L$ data was available have been compared.

As seen in Figure 6, the 1-$IACC_L$ values for most halls lie within the range 0.8-0.9, independent of the change in S_1. This is not surprising, since Beranek also found that $IACC_L$ does not vary significantly between highly-rated halls and lower-rated halls. The lowest values of (1-$IACC_L$) occur for Kleinhans Music Hall, Buffalo (KMH), Koussetitzky Music Shed, Lenox (KMS), which are both large fan-shaped halls, and for Bass Performance Hall, Fort Worth (BFW) and La Scala, Milan (LSC), which are both horseshoe-shaped opera theatres.

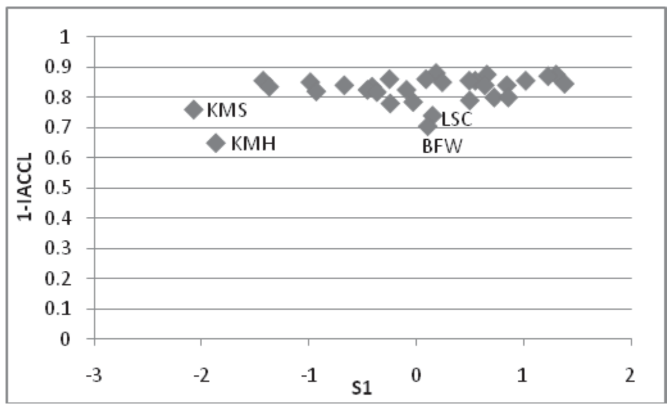

Figure 6: Comparison of S_1 and (1-$IACC_L$). Halls discussed in the text are labelled with identification codes on the graph.

Both Kleinhans Hall and Koussetitzky Music Shed have significant areas of smooth boundary surfaces, which may contribute to the reverberant field being less diffuse in these halls, and hence the lower $IACC_L$ values. However, the surface properties of the room are not taken into account (directly) in calculating S_1, and therefore it would not be expected to account for the reduced (1-$IACC_L$) values for these halls.

A less-diffuse reverberant field is a characteristic of traditional horseshoe-form opera houses, since balcony overhangs and the flytower opening limit the angles from which the reverberant field may be "seen" by seats. This likely explains why the (1-$IACC_L$) values for traditional opera theatres are lower (e.g. La Scala and Bass Hall; Paris Opera Garnier and Teatro Colon, Buenos Aires also have 1-$IACC_L$ values of 0.80 or lower). Again, the factors contributing to the lower "immersion" in traditional opera theatres are more complicated than the assumptions used to derive S_1, and therefore it would not be expected to account for these effects.

It is clear that for most halls, (1-$IACC_L$) is not a particularly useful descriptor of the degree of listener envelopment. Therefore, comparison with (1-$IACC_L$) may not provide a meaningful evaluation of the effectiveness of S_1 as a descriptor for listener envelopment/"immersion".

MUSICAL CLARITY, C_{80}

In the derivation of S_1, it was described as the "inverse of various types of 'clarity index'" [1], and theoretically S_1 indeed is calculated from the ratio of reverberant to "prompt" sound (not necessarily sound arriving within 80 ms as in the clarity index C_{80}). S_1 would therefore be expected to be inversely related to C_{80}, since S_1 is the ratio of reverberant sound to "prompt" sound, whereas C_{80} is the ratio of early sound to late sound.

Figure 7 presents an overview of the relationship between the predicted S_1 values and the measured C_{80} values for the 63 halls where C_{80} data was available. There is significant variation in the data for C_{80}, with only a very slight inverse relationship between S_1 and C_{80} visible, and several outlying data points. The correlation between C_{80} and S_1 is very weak (R^2 0.04), with a standard deviation of 2 dB. This indicates that S_1 and C_{80} appear to be essentially independent. A reason for S_1 and C_{80} not being more closely correlated (as expected) may be that the "early" sound has different definitions in the two parameters. S_1 includes only first-order reflections in its definition of "prompt" sound, while C_{80} includes all energy received up to 80 ms, regardless of the order of reflection. Additionally, the variation may be accounted for by the influence of the room shaping on C_{80} – the dimensions and orientation of the room boundaries can have a significant impact on how much energy is received within 80 ms, whereas in S_1 essentially only reflections from the stage zone are included in the "prompt" sound.

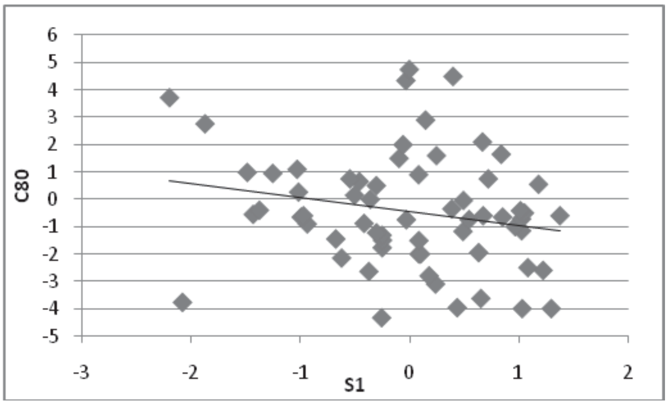

Figure 7: Comparison of S_1 and C_{80}, showing linear regression relationship.

DISCUSSION

Because only subjective evaluations are available for evaluating S_1, and because the subjective evaluations did not specifically focus on the degree of listener envelopment, but on overall hall quality, it is difficult to comment on its effectiveness as an "immersion" index.

From the subjective evaluations, S_1 values in the range ~0 1 dB appear to be associated with the highest rated halls. These are generally halls where the reverberant field is perceived as being rich and enveloping (e.g. Musikverein), although other aspects of the room acoustic may also contribute to the high subjective ratings of these halls.

Although the (1-$IACC_L$) data is not particularly useful, for the few halls where (1-$IACC_L$) is lower, S_1 does not reflect this change (e.g. traditional opera theatres such as La Scala where S_1~0 dB have similar values of (1-$IACC_L$) to Koussetitzky Music Shed with S_1 ~ -2 dB). This suggests that S_1 may not be very effective at capturing the degree to which the reverberant field is diffusive and "immersive". Additional subjective evaluations focussing on listener envelopment or comparison with another parameter that better reflects the listener envelopment than (1-$IACC_L$) would assist in gaining more understanding into the applicability of S_1.

The assumptions inherent in calculating S_1 have the consequence that S_1 is expected to be less effective for "surround"-type halls, or halls with unusual geometry or material finishes than for traditional halls, particularly "shoebox"-type halls or other rectangular-plan halls. Due to these simplifying assumptions, which do not take into account more detailed aspects of the room shape, surface finishes etc that are considered in other, more detailed parameters, there is no strong correlation between S_1 and more detailed acoustic parameters, even C_{80}, which theoretically is close to being the "inverse" parameter of S_1.

The strongest correlation between S_1 and other parameters is between S_1 and Binaural Quality Index (BQI). This suggests that S_1 may be useful as a "spaciousness" index during early design, and as a means of gaining a first estimation of the Binaural Quality Index for a hall before undertaking detailed acoustic modelling. This suggests that S_1 may be able to be used with reverberation time (RT), and room geometry ratios (such as volume per seat, V/N) as an initial design parameter for use in evaluating concepts for a hall design, and is an unexpected result in that S_1 is not intended as a "spaciousness" index!

S_1 can provide a useful supplement to existing design tools in that it would allow the spaciousness of the hall (as expressed as Binaural Quality Index) to be estimated via a simple calculation, before detailed acoustic modelling is conducted. Comparison with highly-rated halls suggests that a S_1 value of ~0-1 dB would be desirable. Further subjective studies of listener envelopment would allow the usefulness of S_1 as a parameter describing "immersion" in the reverberant field to be determined further. Initial findings by comparing with the (admittedly less useful) parameter (1-IACCL) suggests that S_1 may not be particularly useful in instances where the listener envelopment is low.

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Technical Note

Note: Technical notes are aimed at promoting discussion. The views expressed are not necessarily those of the editors or the Australian Acoustical Society. Contributions are not formally peer-reviewed.

MODELLING SOUNDSCAPES

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(This technical note is a summary of a presentation to the NSW branch of the Australian Acoustical Society earlier this year)

Most effort in architectural acoustics goes into spaces where acoustics is critical – concert halls, lecture theatres, places of worship, etc. This means that the vast majority of spaces, where most people spend most of their time – cafés, call centres, foyers, dance studios, courtyards etc. - have little or no serious thought given to how the space will sound.

Obviously the detailed acoustic modelling and assessment techniques used for a concert hall are inappropriate for a café. Acoustic design for such spaces needs to be a very quick and simple process. Preferably it should be done by an acoustic professional, but in cases where the project cannot afford that, approximate design by a non-professional (e.g. an architect) with some training may be better than nothing. What we need for these "also-ran" spaces is:

- a simple set of parameters that will encapsulate the important acoustic properties of ANY space; and
- a simple way to model the space, view the values of those parameters and listen to an approximate simulation of the sound.

As a set of parameters to describe the acoustic properties of a general space, I propose the following:

- background SPL (note "background" means "constant", not "quiet");
- sounds that should be inaudible (i.e. L_{Amax} < background -10dB), and sounds that should not be intrusive (i.e. L_{Amax} < background + 5 dB);
- reverberation time; and
- a measure of speech intelligibility (STI?).

These parameters provide a systematic way to summarise the acoustic design intent for the space, and can be used to guide the design process. The last is controversial – or at least there is dispute about the best units to use to describe speech intelligibility. Resolving this is regarded as an important priority. For example, for a café the design parameters might be:

- Background SPL = 45 dBA because we want a relatively quiet cafe. This can be partially from external traffic noise and partially from air-conditioning.
- Sound from the kitchen, and from the bus stop outside, should not be intrusive (therefore LAmax < 50 dBA). Sound from PA in the adjacent retail space should be inaudible (therefore LAmax < 35 dBA).
- Reverberation time 0.5 secs because this café is intended to be "calm", not "buzzy".
- Patrons' speech to be intelligible within 3m.

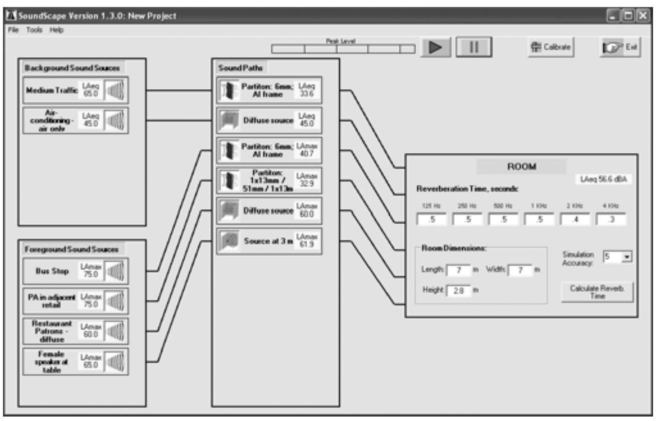
SoundScience has produced a program called SoundScape that is intended to fulfil the requirements for modelling of general acoustic spaces. The program allows you to:

- select sound samples from various background and foreground sounds (you can add more), and define their absolute level;
- place them either in the room at a distance, in the room as a diffuse source, or outside the room (in which case you select the construction for the partition);
- adjust the reverberation time; and
- listen to the result.

For the café example, you can select samples representing all the relevant sources, and enter them into a basic model (room size and shape defined only by length, width and height). You can play with, for example, the external glazing, to achieve the right internal level from a bus. You can design the reverberation time from material properties and areas (Sabine approximation). And finally you can push "Play" and listen to the room. A screen shot from such a model is below.

Other features include:

- good generic reverberation simulation, including direct



sound, using stereo 25-delay feedback delay network with frequency-dependent reverberation time;
- ability to add new sound sources (as 16-bit WAV files), partition types and surface finishes; and
- ability to calibrate, so the simulation is reproduced at the correct SPL.

How can you get a copy of the SoundScape program? Just go to http://www.soundscience.com.au/products/soundscape.htm

Technical Note

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IMPLICATIONS OF UPDATING THE VIBRATION ASSESSMENT METHODOLOGY OF BS6472 FROM THE 1992 TO THE REVISED 2008 VERSION

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INTRODUCTION

In New South Wales, human comfort from tactile vibration is usually assessed against The New South Wales Department of Environment, Climate Change and Water (DECCW) guideline "Assessing Vibration: a technical guideline", dated February 2006. The methodology contained in this document is based upon the guidelines contained in British Standard 6472:1992, "Evaluation of human exposure to vibration in buildings (1-80 Hz)". This British Standard was superseded in 2008 with BS 6472-1:2008 "Guide to evaluation of human exposure to vibration in buildings – Part 1: Vibration sources other than blasting" and the 1992 version of the Standard was withdrawn. The new Standard contains some significant differences to the older Standard including a change of the vertical frequency weighting function. Vibration assessed according to the older Standard will therefore differ from assessments made in accordance with the new Standard.

This technical note highlights how assessing vibration in accordance with the 2008 Standard (rather than the 1992 Standard) will result in a 1.5 to 2-fold increase in the Vibration Dose Values (VDVs) of common vibration sources assessed in building vibration such as plantrooms and other indoor vibration sources (gyms, escalators, etc.), road and rail traffic and construction activities. In particular for the latter, these changes will directly impact on safe working distances which may in some cases result in reduced working hours/increased respite periods.

Although a new version of BS 6472 has been published, the DECCW still requires vibration to be assessed in accordance with the 1992 version of the Standard at this point in time.

SUMMARY OF BS 6472-1:2008 ASSESSMENT PROCEDURE

In this section a brief overview of BS 6472-1:2008 is given with a focus on highlighting the differences between this version and its predecessor, BS 6472:1992.

The Vibration Dose Value

BS 6472-1:2008 assesses the probability of adverse comment from vibration by means of VDVs. Unlike its predecessor, BS 6472:1992, the new Standard allows for assessing continuous, intermittent and impulsive vibration events with a unified procedure. This represents a considerable simplification from the 1992 Standard which used different procedures for continuous, intermittent and impulsive vibration events and blasting. In particular the use of weighted summed acceleration and weighted root mean square (rms) acceleration added complexity at no or little benefit as did providing criteria in both the acceleration and velocity domain.

The VDV is given by the fourth root of the time integral of the fourth power of the acceleration level after it has been frequency weighted (effects of frequency weighting are discussed in the following section). This is expressed mathematically as:

$$VDV = \left(\int_0^T a_w^4(t)\,dt \right)^{0.25} \tag{1}$$

The VDV is much more strongly influenced by vibration magnitude than by duration. A doubling (or halving) in the vibration magnitude results in a sixteen fold decrease (or increase) in the exposure duration for a VDV with the same magnitude.

The VDV is a cumulative measure and increases as the exposure duration increases. It is not an averaging procedure. An X% increase in VDV can be directly related to an X% increase in vibration discomfort (Griffin 1986).

In the case that vibration conditions are constant or repeated regularly, only one representative sample VDV needs to be measured to determine the overall VDV of the assessment period. Similarly, VDVs of different events can easily be added. Corresponding formulas are provided in BS 6472, Griffin (1986) and the ANC guideline (2001).

Principal Difference: Weighting Functions

The frequency weighting used for vertical vibration has changed from W_g, used in BS 6472:1992, to W_b in BS 6472-1:2008. The frequency weighting functions are defined in BS 6841:1987 and are plotted in Figure 1. The thin and thick lines show the asymptotic approximation and the actual modulus of the transfer function, respectively. The 12 dB per octave roll-offs of the band-limits (below 0.5 & 1 Hz and above 80 Hz), which filters one-third of an octave outside the nominal frequency limits, are not plotted. Formulas for the modulus are provided in BS 6841:1987 and Griffin (1986). Griffin (1986) also provides formulas for the asymptotic approximation.

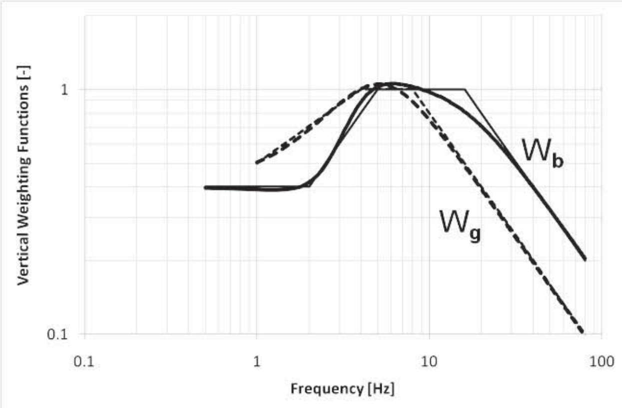

Figure 1 - BS 6841:1987 weighting function in vertical direction - W_g (dashed) and W_b (solid).

The use of the W_b-weighting function improves the consistency between BS 6472 and BS 6841 (W_g-weighting is the preferred weighting for the assessment of hand control and vision in BS 6841:1987). Furthermore, while the W_g weighting was related to activity interference, the W_b weighting was related to comfort which intuitively would suggest the W_b weighting would be more stringent.

The difference in these two weighting functions is frequency dependent as can be seen in Figure 1. In between 4 Hz and 10 Hz the differences arising from the change in weighting functions are small. The weighted acceleration will increase by a factor of up to two at high frequencies, i.e. $W_b > W_g$, while at low frequencies the weighted acceleration is reduced by a factor of up to 1.4, i.e. $W_b < W_g$. The 2008 version of the Standard also assesses vibration down to 0.5 Hz whereas the 1992 version of the Standard has a higher frequency limit of 1 Hz.

The frequency shift in weighting functions means that the revised 2008 Standard is more sensitive to vibration levels above 10 Hz whilst the 1992 standard is more sensitive to vibration below 4 Hz.

Estimated Vibration Dose Values

Actual VDVs may be estimated by eVDVs for continuous vibration that is not time-varying in magnitude and has a crest factor between three and six. The new Standard has distanced itself from the eVDV procedure by discouraging the use of eVDVs for vibration with time-varying characteristics or shocks.

eVDVs can be calculated from the following equation:

$$eVDV = 1.4 \times a_w \times t^{0.25} \qquad (2)$$

where a_w is the frequency weighted rms acceleration in m/s^2 and t is the period over which a_w has been evaluated in seconds.

Historical data

BS 6472-1:2008 provides information on how to appropriately use historical data in situations where it is desirable to examine results derived in the past in light of the revisions introduced with the new Standard.

Considerable care must be taken when only historical spectra (W_g-weighted or unweighted, one-third octave or narrowband) are available. In particular, if the data has been 1-80 Hz band-limited (i.e. information between 0.5 Hz and 1 Hz has been filtered out), this data can only be analysed if there are no low-frequency contributions in the signal.

Recommended Levels

The probability of adverse comment from occupants exposed to a particular level of vibration is given in Table 1. The daytime results in Table 1 do not represent a change to the old version, however for the night-time period a range is now presented as compared to discrete values.

Table 1 – Vibration dose value ranges which might result in various probabilities of adverse comment within residential buildings

Place and time	Low probability of adverse comment (ms-$^{1.75}$)	Adverse comment possible (ms-$^{1.75}$)	Adverse comment probable (ms-$^{1.75}$)
Residential buildings, 16hr day	0.2 to 0.4	0.4 to 0.8	0.8 to 1.6
Residential buildings, 8hr night	0.1 to 0.2	0.2 to 0.4	0.4 to 0.8

Note: For offices and workshops, multiplying factors of 2 and 4 respectively should be applied to the above vibration dose value ranges for a 16 hr day.

The new Standard acknowledges that there is widely differing susceptibility to vibration in the community and accordingly, ranges rather than discrete values are provided.

Coordinate System

In addition to the changes outlined above, the 2008 version of the Standard no longer uses a co ordinate system that is referenced to the human body (i.e. foot-to-head) but uses a Standard geocentric earth based coordinate system.

IMPLICATIONS – WORKED EXAMPLES

Table 2 presents results for some typical construction activities in addition to train pass-by vibration spectra on different track forms. It is important to keep in mind that these changes are indicative and will vary depending on the particular plant used and to some extent on the local geotechnical conditions.

In some cases the historical data was available as one-third octave data band-limited between 1 Hz and 80 Hz. The historical eVDVs were multiplied by $\sqrt{20/23}$ to ensure consistency in the comparison of eVDVs[1]. For all considered cases, there was no low frequency energy contribution and the crest factors were acceptable.

The increases were calculated by dividing a W_b-weighted eVDV by a W_g-weighted eVDV. Almost identical results would

[1]If the W_g-weighted rms acceleration is based on 20 one-third octave bands (i.e. 1 Hz to 80 Hz) then post-multiplication by $\sqrt{20/23}$ is deemed appropriate since W_b-weighted rms acceleration is based on 23 one-third octave bands.

have been obtained had VDVs been compared. Accordingly, the term VDVs in the subsequent discussion and sections relates to both VDVs and eVDVs.

For the considered cases, the shift from the W_g to the W_b frequency weighting implies a 1.6- to 2-fold increase in VDV magnitudes.

In the special case of a tonal vibration source, the change in VDVs can be approximated simply by scaling the historical VDV according to the W_b/W_g ratio at the dominant frequency. For the case that the vibration is of broadband character, a calculation is required to determine the impact of a change from the old to the new Standard.

Table 2 – Expected typical VDV increases associated with a move from W_g to W_b frequency weighting.

Vibration	Typical Increase
Trains at grade, ballasted track	1.75 to 1.9
Trains in tunnels, direct fixation (very stiff pads)	1.9 to 2.0
Vibratory piling	2.0
Hammer piling	1.8
Jackhammer	1.6
Vibratory Roller	1.9
Rockbreaking	1.9
Tunnel Boring	1.9

The data presented in Table 2 is based on ground measurements. It is reasonable to expect that the typical increase in VDV inside buildings on suspended long-span floors will be somewhat less than predicted in Table 2. This is because a floor's fundamental frequency will put more weight to the no-change regime between 4 Hz and 10 Hz and may attenuate vibration at higher frequencies. Similarly, the use of highly resilient rail pads may reduce the 'impact' of W_b-weightings compared to W_g-weightings.

Some typical vibration spectra are presented in Figure 2, Figure 3 and Figure 4. The weighted overall rms acceleration is indicated by the symbols on the right hand side.

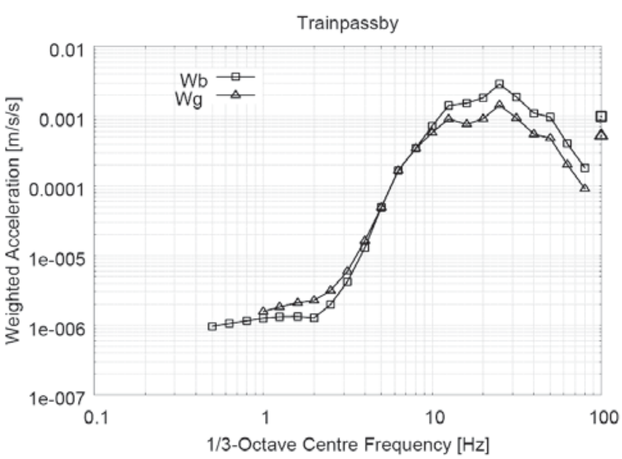

Figure 2 – Train pass-by on ballasted track.

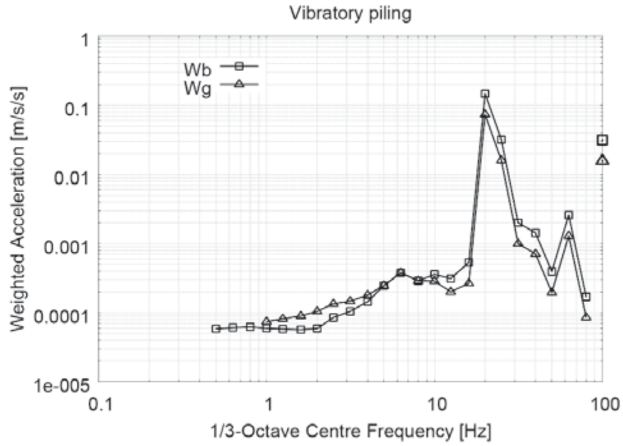

Figure 3 – Vibratory piling (ICE 416L vibratory hammer).

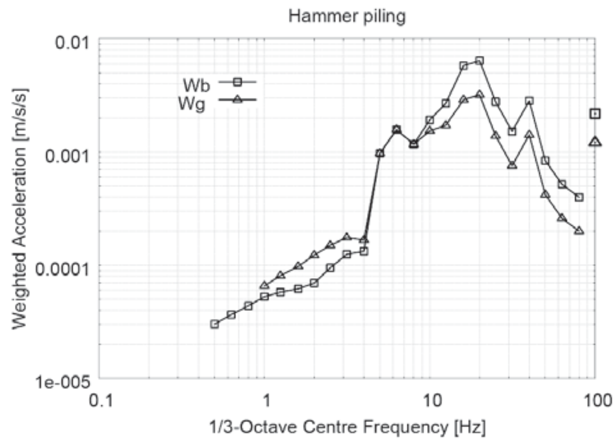

Figure 4 – Hammer piling (BSP 357 hydraulic hammer).

DISCUSSION

Due to the change in frequency weightings the impact of vibration assessed in accordance with the 1992 Standard will be different to assessments in accordance with the 2008 Standard. For most building vibrations the VDVs will increase. In many cases the VDVs may increase by up to a factor of 2.

As a consequence, required offset distances for construction works will increase with the 2008 version of the Standard. We expect that generally accepted minimum offset distances, such as those presented in the Transport Infrastructure Development Corporation's (TIDC) publication entitled "Construction Noise Strategy (Rail Projects)" (CNS) would increase by a factor of approximately 1.2-1.6. Minimum offset distances for human response given in the TIDC's CNS are shown in brackets in the right hand column of Table 3 underneath the safe working distances for human response based on BS6472:2008. These distances are based on continuous vibration, are indicative and will vary depending on the particular item of plant and local geotechnical conditions.

Table 3 TIDC Recommended Safe Working Distances for Human Response.

Plant/Item	Rating/Description	Safe Working Distances 2008 / (1992)
Vibratory Roller	< 50 kN (Typically 1-2 Tonnes)	20m to 25m (15m to 20m)
	< 100 kN (Typically 2-4 Tonnes)	25m (20m)
	< 200 kN (Typically 4-6 Tonnes)	50m (40m)
	< 300 kN (Typically 7-13 Tonnes)	130m - 150m (100m)
	> 300 kN (Typically 13-18 Tonnes)	130m - 150m (100m)
	> 300 kN (> 18 Tonnes)	130m - 150m (100m)
Small Hydraulic Hammer	(300 kg – 5 to 12t excavator)	10m (7m)
Medium Hydraulic Hammer	(900 kg – 12 to 18t excavator)	30m (23m)
Large Hydraulic Hammer	(1600 kg – 18 to 34t excavator)	90m (73m)
Vibratory Pile Driver	Sheet piles	30m (20m)

Similarly, the exposure duration (or the number of vibration events) will be decreased. For instance, one W_b-weighted train pass-by (2008 Standard) induces the same vibration dose as 16 of the same pass-bys using the W_g-weighting (1992 Standard)[2].

CONCLUSION

In comparison to BS 6472:1992, the 2008 version of the Standard represents a simplification of the assessment methodology since a unified VDV procedure is used to assess the impact of vibration in relation to human comfort.

The shift away from the W_g to the W_b vertical frequency weighting function is based on the latest knowledge and experience and is believed to better correlate with human comfort response to vibration, rather than activity disturbance. The change in weightings results in appreciably higher VDVs than those predicted with the old Standard for common sources of vibration.

If the DECCW was to incorporate the 2008 revisions into their Assessing Vibration guideline, this would result in the prediction of higher VDV levels for most building vibration events assessed for human comfort. Greater safe working distances than those currently recommended in TIDC's Construction Noise Strategy would also follow.

REFERENCES

[1] BS 6472:1992, "Evaluation of human exposure to vibration in buildings (1-80 Hz)", 1992.

[2] BS 6472-1:2008 "Guide to evaluation of human exposure to vibration in buildings – Part 1: Vibration sources other than blasting", 2008.

[3] BS 6841:1987 "Guide to measurement and evaluation of human exposure to whole-body mechanical vibration and repeated shock", 1987.

[4] M. J. Griffin "Evaluation of Vibration with Respect to human Response", SAE paper no. 860047, 1986.

[5] Association of Noise Consultants "Measurement & Assessment of Groundborne Noise & Vibration", ANC guidelines, 2001.

[6] Department of Environment and Climate Change "Assessing Vibration: A Technical Guideline", 2006.

[7] Transport Infrastructure Development Corporation "Construction Noise Strategy (Rail Projects)", 2007.

[2]Based on a 2-fold increase in VDV.

LETTER TO THE EDITOR

What is Offensive Noise? A Case Study in NSW

Renzo Tonin, Renzo Tonin & Associates (NSW) Pty Ltd,
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I refer to my paper "What is Offensive Noise? A Case Study in NSW" published in Acoustics Australia Vol. 38 No. 1, April 2010, and wish to clarify a statement made on page 32 that *"Private schools (unlike public schools) are subject to the POEO Act"*. This should have read *"Private schools (unlike public schools) are subject to Noise Abatement Orders"*. Public schools are subject to other provisions of the POEO Act, notably a Noise Control Notice issued by the regulatory authority (DECCW in the case of public schools) and a Noise Abatement Direction (given by an authorised officer appointed by DECCW).

NEWS

New AAS General Secretary

Richard Booker is the new General Secretary of the Australian Acoustical Society, having taken over the position from Byron Martin. Richard has a background primarily in IT Project Management, having worked for BHP Aerospace & Electronics, NSW Education Dept, Optus Communications, and was the IT Manager for the University of Wollongong in Dubai for 5 years. Recently he has been consulting on Body Corporate redevelopment projects. He looks forward to working closely with the society and improving organisational systems.

Acoustics Australia Back Copies on Web

All the back copies of the Acoustics Australia journal have been scanned and are available for download from the AAS website. Access is via the link to the Journal on the AAS website www.acoustics.asn.au/joomla and then follow the link to Available Back Issues.

Access is free for all up to 2 years before the current issues, that is, for 2010 the access is available for all issues up to and including 2008. Members can log in and then go to the same link and see all the issues to the current time.

As the issues are now all available on line, the archivist has a complete spare set of the journal. If anyone is interested in obtaining this hard copy set, for the cost of the postage, please contact Charles Don at charlesd@virginbroadband.com.au

Australian Acoustical Society Conference Proceedings on Web

Fully searchable copies of early AAS conference proceedings have started to become available on our web site over the past months, thanks to the persistence of Charles Don. The earliest is from 1968, when an "International Acoustics Conference" was held in September at the Wentworth Hotel, Sydney, hosted by the NSW Division. Twelve papers were presented, involving topics familiar even today, such as Traffic and Office Noise, Community Annoyance due to low frequencies, Hearing conservation and Jet Airplane noise. Among the presenters were some well-known early members of the Society, including Louis Challis, Anita Lawrence and Peter Knowland.

In 1971 there was a "Noise Zoning" conference in Victoria, a "Noise Legislation & Regulation" in NSW during 1972, then "Noise, Shock & Vibration" was the topic for the Vic. Conference in 1974. An interesting feature of these and other early meetings was that they focussed on a relatively narrow area of acoustics compared to today's conferences, with many of the speakers being drawn from outside the immediate acoustics area. For example in the 1972 conference, a judge of the Supreme Court of N.S.W., the Hon. Mr. Justice R. Else-Mitchell gave the opening session paper on the topic "Noise Control and the Law", with following speakers including a barrister, a Federal Medical Officer and a Worker's Compensation Commission judge

More recently, "The Economics of Noise Control", presented in SA in February, 1983, considered topics such as the cost of traffic noise abatement, a local government noise control scheme, acoustic insulation in passenger cars and the situation in an Oil Refinery. While views may have altered over the intervening years, many of the points being made are still very relevant and provide an interesting insight into the progress, or lack if it, that have occurred over the intervening years. Indeed, these past conference proceedings provide a valuable source of measurements, techniques and developments that have become the foundations of current acoustic practice and are well worth a glance at by all modern acousticians.

It anticipated that, gradually, a complete set of AAS conference proceedings will be available on line in the form of searchable pdf files. The AAS archives started to keep the proceedings in the mid-1970s and some of the very early proceedings have disappeared from sight. A list of known conferences and symposiums is on the web and this indicates that there was a Symposium on Noise in Victoria and the 2nd Building Research Congress in N.S.W. back in 1964 and several other gatherings. If anyone has copies or knows of the existence of copies of these old proceedings in a dusty drawer or bookshelf, Charles Don would be delighted to learn about them and arrange to get a copy made. So if you should find something please check with the earlier issues already on the web, and if it is not there, please make your find known to Charles Don at charlesd@virginbroadband.com.au.

Hear Us: Inquiry into Hearing Health in Australia

The Senate Community Affairs References Committee tabled its report titled *Hear Us: Inquiry into Hearing Health in Australia*, on May 13 2010, after an eight month inquiry that included written submissions from interested parties and public hearings.

The report focuses on the prevalence, causes and cost of hearing loss in Australia and the issues faced by those who are hearing impaired. These issues include access and services, educational opportunities and lack of support in the criminal justice system. The evidence shows there is a crisis in Indigenous ear and hearing health in Australia with Indigenous people suffering ear disease and hearing loss up to ten times the rate of non-Indigenous Australians. The report also looks at the adequacy of research, education and awareness programs.

The committee makes a number of recommendations which, if implemented, will address many of the issues raised during the inquiry. Those that may be of interest to Acoustics Australia readers are:

In relation to workplace noise:

No 16 *The committee recommends that Australian Governments continue to prioritise and fund research into occupational noise exposure. The focus of research should be informed by the results of the 'Getting heard: effective prevention of hazardous occupational noise' project, currently being undertaken by Safe Work Australia, and include investigation into the effectiveness of current legislation in limiting occupational noise exposure. Research should continue to develop understanding about the design of workplace equipment, hearing protection, and the long-term effects of acoustic shock and acoustic trauma.*

No 18 *The committee recommends that the Department of Health and Ageing work closely with Safe Work Australia to investigate the relationships between ototoxic substances and hearing impairment, and the possible implications for workplace safety practices.*

No 20 *The committee recommends that the Department of Health and Ageing provides funding for Australian Hearing to develop, in close consultation with major hearing health stakeholders, a national hearing health awareness and prevention education campaign. This campaign should have three dimensions. It should:*

(a) target those at highest risk of acquired hearing loss (including employers and employees in high-risk industries, farmers and rural workers, and young people) to improve their knowledge about hearing health and change risky behaviours;

(b) raise the level of awareness about hearing health issues among the broader Australian population to help de-stigmatise hearing loss; and (c) promote access to support services for people who are hearing impaired.

In relation to recreational noise:

No 1 *The committee recommends that the Department of Health and Ageing work with the appropriate agencies and authorities to devise recreational noise safety regulations for entertainment venues. Specifically, where music is expected to be louder than a recommended safe level, that the venues be required to:*

(a) post prominent notices warning patrons that the noise level at that venue may be loud enough to cause hearing damage; and

(b) make ear plugs freely available to all patrons.

No 15 *The committee recommends that the Australian Government fund the National Acoustic Laboratory to undertake longitudinal research into the long-term impacts of recreational noise, particularly exposure to personal music players.*

Also of some relevance are:

No 2 *The committee recommends that the Department of Education, Employment and Workplace Relations engage with state and territory jurisdictions, and with employment and hearing loss peak bodies, to develop a 10 year strategy to better support, engage and retain hearing impaired Australians in the workforce. The strategy should be made publicly available, and detail annual performance targets and the level of resources committed to achieving them.*

No 13 *The committee recommends that the public counters in all government service shopfronts be accessible to people with a hearing impairment through the provision of hearing loop technology. The committee recommends that the Office of Hearing Services coordinate a project which sets targets toward that end for all government agencies, at all levels of government, and that these be publicly reported upon.*

The report, submissions and hearing transcripts can be accessed at: http://www.aph.gov.au/SENATE/COMMITTEE/CLAC_CTTE/hearing_health/index.htm

Pam Gunn

SLR Heggies acquires Hoefler Consulting Group

SLR Management Ltd announces the acquisition of Hoefler Consulting Group (HCG), a leading Alaska-based provider of environmental consulting services to clients in the energy, mining and natural resource management sectors. Headquartered in Anchorage, HCG has 55 employees and offices in Fairbanks, Portland and Seattle. The company is a provider of air quality permits and compliance, and land quality services. SLR acquired Australian consultancy Heggies Pty Ltd in February this year and has 140 employees working from 12 offices in Australia, New Zealand and Singapore. The Hoefler acquisition continues the integration and expansion of SLR's technical expertise across North America, Europe and Australasia. SLR provide planning and environmental services in the energy, mining, oil and gas industries as well as renewable power including wind, solar, energy-from-waste, hydropower and biomass.

New Products

Low Cost Portable Peak Vibration Meter

Dytran's new portable vibration meter D4190 is a tool designed for field testing of accelerometers, other peak vibration measurements and is available from Kingdom Pty Ltd. The D4190 can be used in conjunction with a hand-held shaker for sensor validation. The D4190 converts the vibration signal from a 10mV/g or 100mV/g IEPE accelerometer to the Peak value of the vibration signals generated by the sensors. The RMS value of the peak can be easily derived. The D4190 has a built in power supply that provides 2mA of current at +18VDC, able to power either the 10mV/g or 100mV/g IEPE accelerometer and two rotary switches which allow the user to select either of the accelerometer inputs or to select a full scale range of 2g, 20g and 50g. There is a large digital front panel display, a bias test switch to verify proper accelerometer operation and an output jack for monitoring the output signal of the accelerometer with an oscilloscope or analyser. It weighs just 320 grams with dimensions of 135 x 84 x 57 mm. Two replaceable 9V batteries provide up to 10 hours of continuous operation. For more information contact Kingdom Pty Ltd on (02) 9975 3272 or visit www.kingdom.com.au

Low Cost Integrating Sound Level Meter

The BSWA308 complies with the IEC61672 international standards and features an industrially designed housing with ergonomic sculpturing. The 1/2" measurement microphone which is supported by an ICCP (IEPE) preamplifier is equipped with a TNC connector and can therefore be operated attached or detached from the main processing unit via a microphone extension cable. The BSWA308 has a dynamic range of 102 dB and can always measure noise from 29 dBA to 131 dBA in a single range. It can measure three parameters simultaneously with the A, C, and Z frequency weightings and with F, S, and I time weightings. In addition, the equivalent continuous sound pressure level and maximum and minimum values are calculated. The integration time for integral sound quantities can be set. The BSWA308 is designed as a sound level instrument for general purposes noise measurements where Class 1 accuracy is required or desirable. For more information contact Kingdom Pty Ltd on (02) 9975 3272 or visit www.kingdom.com.au

New CCLD Tacho Probe MM-0360

Brüel & Kjær announce a new Tacho Probe, the MM-0360 that can work directly with LAN-XI modules. It is especially designed for contact-free speed measurements on rotating or reciprocating machine parts, producing a voltage pulse for each rotation of a shaft or cycle of a machine part. In addition, MM-0360 is powered by CCLD so it can be used with any DeltaTron or ICP® constant current line drive (CCLD) supply that provides 3mA to power the tacho probe. CCLD power means no separate power supply or special cabling. The MM-0360 uses a continuous wave laser to provide the precise rotational speed and phase information required for order tracking, phase or balancing applications. Brüel & Kjær claim speed measurements down to 0 RPM for wind turbine, ship propulsion and paper machine applications with an appropriate power supply. For more information visit www.bksv.com.au

Cost Effective, High Channel Density Measurement Modules

Brüel & Kjærs offers a new 12 channel data analyser for multi-channel noise and vibration testing; LAN-XI Module Type 3053-B-120. It is the only data analyser currently available which offers this number of channels in such a small package; each module measures just 132.6 x 27.5 x 250 mm (5.22 x 1.08 x 9.84 inches). The frequency range of the LAN-XI Module from DC to 25.6 kHz makes it suitable for a wide range of acoustic and vibration measurements, including structural measurements on large structures such as wind turbines. The high frequency bandwidth also makes it suitable for sound power measurements such as those made to International Standards ISO 3741 to ISO 3743 (comparison method) and ISO 3744 to ISO 3746. As part of Brüel & Kjær's LAN-XI measurement hardware family, Type 3053-B-120 is almost infinitely expandable with other LAN-XI input, input/output modules and mounting frames. It can also be a key building block for high-density, high-channel count systems, such as those for array acoustics and noise source identification. When used in an array system, the module's detachable connector panel allows it to be fitted into a LAN-XI frame, with a multi-pin connector, to reduce the number of cables. The same modules can also be removed from the frame and fitted with the standard 12 SMB connector panel to create individual, 12-channel systems to maximize the flexibility of the system. This allows the same measurement hardware to fulfil multiple testing roles: a high-channel system, a small portable field recorder or front-end. As part of the LAN-XI family, the Type 3053-B-120 can be upgraded to a stand-alone recorder to make high-quality data recording supremely simple in situations where a normal PC could not survive, such in-flight, in-vehicle testing or other harsh or restricted environments. For more information visit www.bksv.com.au

EchoHush Makes Spaces of Great Beauty and Sound Quality

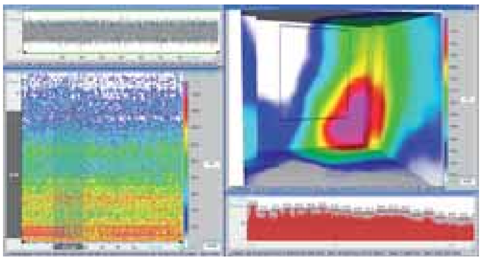
EchoHush, just released by Pyrotek Noise Control, is a new acoustic panel product that enables the architect or designer to enhance both a space's sound quality and its aesthetics. EchoHush invites designers to apply their own creativity through choice of colour and selection of design for Pyrotek to craft into acoustic panels that will fulfil the artistic vision for the spaces being created. In the new trademarked brand's two forms - EchoHush Profile and EchoHush Metro - this customisable aspect makes EchoHush a stand-out option for restaurants, public areas, hallways, showrooms, corridors, home theatres, sports halls, kindergartens, lecture theatres, recording studios, office receptions, open plan offices, bars, church halls, science school rooms, cinemas, work

AC easy

acoustic camera
simply perfect – perfectly simple

acoustic camera

acoustic image 3D
noise emmission via mechanical, external stimulation of the building



AC easy and the high-end system AC pro do not differ software-wise as both use NoiseImage4. Differences only exist in hardware equipment.

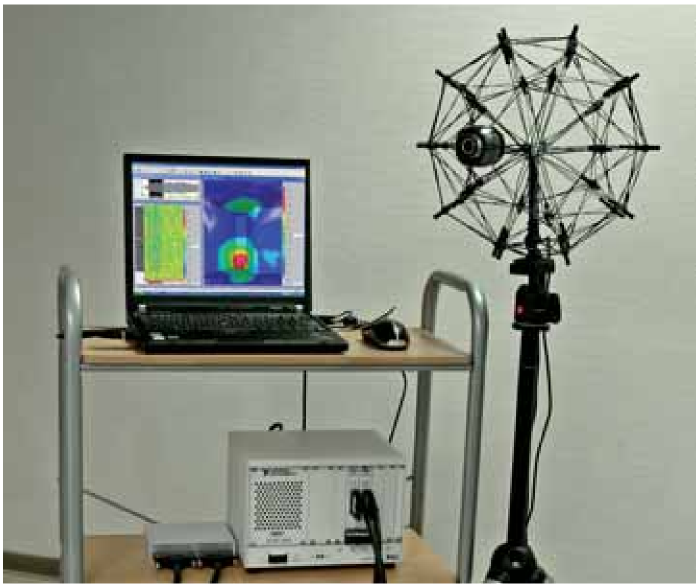
Depending on the desired area of application the user can choose from three different microphone arrays. The two offered AC easy basic configurations can be combined with a Sphere32-35 easy, Ring32-75 easy or a Ring33-35 easy Array.

AC easy key features:

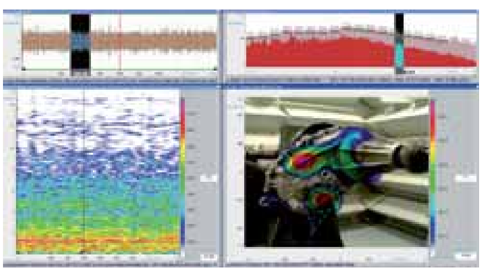
- Live streaming (the recorded data is transferred from the data recorder to the PC/ notebook in real-time, a life-analysis is immediately visible)
- High usability, easy and intelligible handling
- Shows sound sources quickly and reliably
- Small, mobile and comacpt

Specifications – system with notebook

- Software NoiseImage4 for PCs, starting at Windows XP / 7
- Standard notebook
- Microphone arrays; Sphere32-35 easy, Ring32-75 easy or Ring32-35 easy
- National Instruments NI PXI-1033 Chassis with two microphone measurement cards (NI PXI 6250; 48kHz data recording, 16bit resolution)



acoustic image 2D
sound source of a transmission unit at 1.700 rotations/min



Specifications – system with desktop PC

- Software NoiseImage4 for PCs, starting at Windows XP / 7
- Microphone arrays; Sphere32-35 easy, Ring32-75 easy or Ring32-35 easy
- Standard PC with two National Instruments micorphone measurement cards (NI PCI 6250; 48kHz data recording, 16bit resolution)

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areas, call centres, music rooms, nightclubs, practice rooms, public spaces and everywhere controlled sound response is expected from a habitable space.

EchoHush Profile panels are cut from combustion-modified acoustic foam before being treated then painted to the specifier's colour choice. EchoHush Metro comprises metal panels router-milled to offer six design options produced to the specifier's choice of colour and finish, with a framed cavity absorber. The results are invariably stunning, expressing style through design and colour. Moreover, the use of EchoHush in sports halls and public areas often proves the more durable and robust option. This is not to forget the primary function of EchoHush: to provide, along with the aesthetics, solutions to acoustic problems within a space. Already Pyrotek Noise Control can point to projects in which EchoHush in both its main forms has enabled architects, interior designers and acoustic engineers to provide innovative designs enhancing a space's aesthetic ambience as well as tuning the space's acoustics to suit its function. Pyrotek Noise Control offers free advice on EchoHush via its new hotline on 1300WAVEBAR (1300 928 322).

MEETING REPORTS

NSW Division

The AAS NSW Division welcomes two new committee members, Jeff Parnell (NSW Department of Planning) and Matthew Harrison (AECOM).

The AAS NSW technical meetings were presented by Nigel Holmes on the topic of 'The effects of the atmosphere on noise propagation and how to deal with it in environmental assessments', and Mark Latal of the NSW Department of Environment, Climate Change and Water (DECCW) on the DECCW draft road noise policy.

NSW Division Student Grants

Eleven students have been awarded the AAS NSW Young Scientist Award to attend the ICA 2010 congress in Sydney, 23-27 August 2010. The successful recipients of the award are Sebastian Oberst (ADFA, UNSW), Paul Croaker (UNSW), Herwig Peters (UNSW), Francisco Ferreiro (UNSW), Rodney Phillips (UTS), David Sallak (UTS), Doheon Lee (USyd), Peter Gangemi (UNSW), Michael Coats (UNSW), Jian Chen (UNSW), and Zhifang Zhang (ADFA, UNSW).

VIC Division

The third AAS Victoria Division technical meeting for 2010 was held on 22 June at the SKM Theatrette, Armadale. James McIntosh of VicRoads described the VicRoads traffic noise reduction policy and spoke about associated questions concerning coordination with property developers, traffic noise issues with a Melbourne population of 5 million, and noise issues with multistory buildings near arterial roads. A summary of his presentation is given in what follows.

The VicRoads Traffic Noise Policy applies to traffic on expressways (as toll-ways or freeways) and arterial roads adjacent to residences, hospitals, schools and places of worship. The traffic noise limits apply for 18 hours (06:00-00:00hrs) near residences and hospitals, and for 12 hours (06:00-18:00hrs) near schools and places of worship. There are no noise limits for traffic on such roads passing through commercial and industrial areas. The specified noise limits are given in terms of L_{10} measured over either the 18 or 12 hour period, depending on whether the area contains residences and hospitals, or schools, etc. If L_{10} in the noise sensitive area exceeds the specified limit, remedial action such as noise barriers are required. In 1983, the specified L_{10} noise limit was set at 68 dB(A) over 18 or 12 hours as applicable. In 1989, the L_{10} limit was reduced to 63 dB(A) for new and upgraded roads, but remained at 68 dB(A) for existing roads. These remained as draft regulations until formalised in 1997. Because this policy is not completely explicit, municipalities and developers must refer plans for buildings near expressways to VicRoads.

VicRoads generally requires noise barriers located preferably at a property boundary or in the road reserve. Supplementary architectural treatments may be acceptable when AS 2107:2000 (Recommended design sound levels and reverberation times for building interiors) provides an appropriate criterion. With multi-storey buildings, the VicRoads requirements apply only to the lowest habitable floor. Barriers, mostly of 28 mm timber plywood, are currently designed for a life of 25 years, eventually to be extended to 50 years. Property owners are required to pay for a barrier's first ten years maintenance, after which VicRoads takes over these costs. Finally, there are numerous traffic and vehicle noise challenges requiring attention into the future. These include truck drivers using truck engine brakes outside peak periods, especially at night. There is no current Australian Design Rules (ADR) limiting this noise. The closest is that faulty engine exhausts fail the ADR 83 test. Other challenges include increased urban densities and high rise living, more people living near busy roads, the need for quieter vehicles generally, and the need for low-noise road pavements in urban areas. However, the pressing additional need for more extensive public transport in urban areas (for environmental, urban and transport-efficiency reasons) was not mentioned.

STANDARDS AUSTRALIA

Wind Generator Noise

AS 4959-2010 Acoustics-Measurement, prediction and assessment of noise from wind generators has been released in early 2010. It provides a methodology for assessing the impact of noise from wind turbine generators (WTGs) at all receivers in the vicinity of a wind farm. It was developed in collaboration with the National Environment Protection Council to support new guidelines for wind farm development which seek to address community concerns on issues including wind turbine noise and the impact on landscapes and threatened species.

Wind farms are predicted to contribute considerably to achieving the Federal Government's target of 20 per cent renewable energy generation by 2020. The new Standard provides wind farm developers and relevant regulatory authorities with a suitable framework to develop a method for the measurement, prediction and assessment of noise from wind farms. While it sets out assessment methods and a framework for noise level limits based on background noise limits and wind speed, the Standard does not explicitly prescribe noise limits. Areas covered include:

- noise emission prediction procedures;
- background noise level monitoring;
- post-construction noise monitoring; and
- documentation requirements.

It is expected that the Standard will be referenced in the National Wind Farm Development Guidelines and form the basis for any wind farm noise impact assessment.

Standards Australia Magazine Vol 15, 2010

Construction noise and vibration

In May 2010, a revised version of AS 2436 "Guide to noise and vibration control on construction, demolition and maintenance sites" was released. This Standard is concerned with noise and vibration from construction, demolition and maintenance sites as it affects persons working on these sites and also those living and working in the neighbourhood. This Standard provides guidance in noise and vibration control on such sites and includes guidance in investigation and identification of sources, measurements of sound and vibration, and guidance in assessment with a view to planning measures for noise control and monitoring of their effectiveness. This Standard is applicable to a wide range of different activities associated with construction, demolition and maintenance works. It is intended to assist local and/or regulatory authorities, developers, builders, architects, engineers, planners, designers and contractors in planning and/or assessing measures for control of noise and vibration on and from construction, demolition and maintenance sites.

Procedure for Work on New or Revised Standards After an extensive consultation period, Standards Australia has finalised the Standards Australia Guide to the Project Prioritisation Process and Criteria to assist stakeholders in understanding the Project Prioritisation and Selection Process and applying to have their project considered as a Standards Australia resourced Standards Development project.

Stakeholders interested in submitting a project proposal as part of the Prioritisation and Selection Process are encouraged to familiarise themselves with the process by reading the Prioritisation Guide. A revised Pathways Guide is also available, providing guidance on the Standards Development pathways offered by Standards Australia.

Standards Aust Bulletin #8

FASTS

The 2010/11 Federal Budget handed down on 11 May 2010 announced an unexpected policy change to the Commonwealth Grant arrangements enjoyed by FASTS over the last 4 years. FASTS will receive a one off payment of $216,000 from the Federal Department of Innovation, Industry, Science and Research before 30 June 2010, after which there will be no further grant funding under existing Commonwealth schemes and programs provided to FASTS. As a consequence FASTS Board has proposed a sustainable funding model. In the immediate term, it was agreed that FASTS' activities would be limited to the following initiatives until such time as human and financial resources allowed otherwise:

- Undertake the *Communicating the Future: Climate Change Summit* (June 2010)
- Develop an 2010 election policy for presentation to Government (immediately)
- Roll out Science meets Parliament 2011 (first half of 2011)
- Undertake a workshop to develop a strategic plan (September 2010)
- Pursue phase 2 of the EMC research that is already underway (August - Sept)
- Undertake revenue generating activities (ongoing)

It was acknowledged that a new growth strategy should be accompanied by a review of the FASTS' constitution to ensure the most appropriate membership and governance structures are in place to support strategy. These operational and structural issues will be addressed in the context of the strategic planning workshop.

In other respects, the 2010-11 budget contained no major announcements for science and innovation. The substantial increase in Australia's research funding seen last year was honoured in this year's Budget. The total Commonwealth commitment to research and innovation is expected to increase by 4 per cent in 2010-11. Funding for the ARC is neutral. The Budget is silent on:

- On-going funding for the International Science Linkages Program which stands to jeopardise important international agreements.
- Investment in maths, science and engineering education and training.
- The recommendations in Inspiring Australia – a national strategy for engagement with the sciences are yet to be implemented

FUTURE CONFERENCES AND WORKSHOPS

ICBEN 2011

The 10th International Congress on Noise as a Public Health Problem will be held between 24-28 July 2011 in London, UK, organized by the UK Institute of Acoustics on behalf of the International Commission on the Biological Effects of Noise (ICBEN). This congress aims to present the current state of the art in research on the biological effects of noise on health and is suitable for research scientists, policy makers and industry concerned with the effects of noise. Papers and posters will be welcome on topics including noise induced hearing loss, noise and communication, non-auditory physiological effects of noise on health, influence of noise on performance and behaviour, effects of noise on sleep, community responses to noise, noise and animals, interactions with other agents and contextual factors and noise policy and economics.

More information from

http://www.icben2011.org/

Inter-Noise 2011

The 40th International Congress and Exposition on Noise Control Engineering (Inter-Noise 2011) will be held in Osaka, Japan from 4-7 September 2011. The Congress is sponsored by the International Institute of Noise Control Engineering (I-INCE) and co-organised by the Institute of Noise Control Engineering Japan (INCE/J) and the Acoustical Society of Japan (ASJ). The organisers and the Organising Committee of the Congress extend a warm welcome to all prospective participants world-wide and invite all to join them in Osaka to discuss the latest advancements in noise and vibration control engineering and technology, focusing on the congress theme of "Sound Environment as a Global Issue".

More information from

http://www.internoise2011.com

Wind Turbine Noise 2011, Rome, Italy

The fourth international conference on wind turbine noise and its effects on people will be held in Rome, Italy from 12-14 April 2011. The conference is organised by INCE/Europe and the previous conference in 2009 involved more than 160 delegates from 25 countries representing manufacturers, developers, researchers in noise and vibration, environmentalists, pressure groups and consultants. There is an introductory course on noise to be held in the afternoon prior to the conference, which has proved to be very popular in previous years. Offers of papers for this conference are invited and prospective authors should send a 200 word abstract by 1 November 2010 to organiser@windturbinenoise2011.org. A template for abstracts can be found on the conference website and those wanting to attend may also register to receiver further information as the organisation of the conference progresses. The CDs of the Proceedings of WTN 2009, WTN 2007 and WTN 2005 are available from the INCE Europe secretariat, contact Cathy@cmmsoffice.demon.co.uk. More information from www.windturbinenoise2011.org

Renzo Tonin to Host CadnaA Workshops in Sydney/Brisbane/Melbourne

Renzo Tonin & Associates is sponsoring a free one-half day workshop on CadnaA noise prediction modeling in Sydney, Brisbane and Melbourne on the following dates:

Sydney	Monday 30th August 2010
Brisbane	Tuesday 31st August 2010
Melbourne	Wednesday 1st September 2010
Time:	10am to 2 pm (lunch included)

Topics covered include an introduction to industrial and traffic noise prediction, a basic modeling tutorial for new users demonstrating modeling techniques and calculation methods such as segmentation, projection, reflection and diffraction. More advanced topics include dynamic 3D visualization, Google Earth interface, dynamic noise map, GIS-Integration, project management and data handling including object trees, variants and groups. The workshops will be presented by Ingo Rabe, Training Manager from DataKustik Germany. All existing CadnaA users are invited as well as users of other modeling packages who may be interested in finding out how adding CadnaA can make their work profitable. Graduate engineers are especially invited to make up their CPD points and to meet others in the profession.

For bookings please contact Lesley Barratt by email on lbarratt@renzotonin.com.au or telephone (02) 8218 0500. Bookings must be made before Friday 20th August 2010.

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Norman Disney & Young is Australia's largest privately owned building services consultancy. We also have a major presence in New Zealand and the UK. NDY Sound, the firm's specialist group of experienced acoustics engineers, is committed to sharing knowledge and developing new technologies. This enhances the service we offer our clients and aids professional development.

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Applicants should have degrees in mechanical engineering, which are acceptable for CPEng status with the Institution of Engineers Australia and/or post graduate qualifications in Acoustics. Applicants with other qualifications relevant to the field of acoustics are also encouraged to apply.

Your experience in architectural, building, audio visual/production, environmental, industrial and infrastructure projects are of particular interest.

Applications

Applications will be treated in the strictest confidence. Submit a covering letter and resume detailing your qualifications and experience to:

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Telephone: +61 2 9928 6800
Email: d.obrien@ndy.com
Web: www.ndy.com/careers

Norman Disney & Young

For more information about NDY Sound please visit to our website:

www.ndy.com/acoustics

ICA 2010

This is the fifteenth in a series of items in the lead up to ICA in Sydney 23-27 August 2010.

It is likely that by the time you are reading this item the ICA 2010 congress will either be in progress or a recent memory. In the last weeks before the congress there is the satisfaction as the program has developed with a large number of high standard papers. But this is coupled with the concern to attend to all the final details so that delegates find it a worthwhile experience and that this event reflects well on the Australian Acoustical Society

The vision to bid for the ICA 2010 came from Ken Mikl back in 2004, however Ken was unfortunately unable to continue as Chair for the congress. After the congress we will report on the statistics and give thanks to all those who have assisted. At this time I would like to particularly note the ongoing commitment from the core organizing committee who have worked on this task steadily over the years since 2004: David Anderson as Secretary and Chris Schulten as Treasurer. Their input has been essential in the planning and organisation of this major event for acoustics in Australia.

The website **www.ica2010sydney.org** has been the source for information in the lead up to the congress and in due course will have the reports and summaries of the event. On behalf of everyone involved with the organisation of the congress we hope that all the delegates find ICA 2010 a stimulating and worthwhile experience.

Marion Burgess, Chair ICA 2010

Diary

2010

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

26 – 31 August, Sydney and Katoomba
ISMA 2010 International Symposium on Musical Acoustics
http://isma2010.phys.unsw.edu.au/

29 – 31 August, Melbourne
ISRA 2010 International Symposium on Room Acoustics
http://www.isra2010.org/

29 – 31 August, Auckland, New Zealand
ISSA 2010 International Symposium on Sustainability in Acoustics
http://issa.acoustics.ac.nz

23 – 27 August, Seattle, USA
11th International Conference on Music Perception and Cognition
http://depts.washington.edu/icmpc11

26 – 30 September, Makuhari, Japan
Interspeech 2010
http://www.interspeech2010.org

11 – 14 October, San Diego, USA
2010 IEEE International Ultrasonics Symposium (IUS)
http://ewh.ieee.org/conf/ius_2010

18 – 19 November, Cardiff, UK
Reproduced Sound 2010
http://www.reproducedsound.co.uk

2011

12 – 14 April, Rome, Italy
Wind Turbine Noise 2011
http://www.windturbinenoise2011.org

22 – 25 May, Prague, Czech Republic
International Conference on Acoustics, Speech, and Signal Processing (IEEE ICASSP 2011).
http://www.icassp2011.com

23 – 27 May, Seattle, USA
161st Meeting of the Acoustical Society of America
http://asa.aip.org/meetings.html

27 June – 1 July, Aalborg, Denmark
Forum Acusticum 2011
http://www.fa2011.org

10 – 14 July, Rio de Janeiro, Brazil
18th International Congress on Sound and Vibration (ICSV18)
http://www.iiav.org/index.php?va=congresses

24 – 28 July, Tokyo
19th International Symposium on Nonlinear Acoustics (ISNA)
http://www.isna19.com

24 – 28 July, London, UK
10th International Congress on Noise as a Public Health Problem (ICBEN)
http://www.icben2011.org

27 – 31 August, Florence, Italy
Interspeech 2011
http://www.interspeech2011.org

4 – 7 September, Osaka, Japan
Inter-Noise 2011 - Sound Environment as a Global Issue
http://www.internoise2011.com

5 - 8 September, Gdansk, Poland
International Congress on Ultrasonics (2011 ICU)
http://icu2011.ug.edu.pl/index.html

31 October – 4 November, San Diego, USA
162nd Meeting of the Acoustical Society of America
http://asa.aip.org/meetings.html

2012

20 – 25 March, Kyoto, Japan
IEEE International Conference on Acoustics, Speech, and Signal Processing (ICASSP 2012)
http://www.icassp2012.com

8 – 12 July, Vilnius, Lithuania
19th International Congress on Sound and Vibration (ICSV19)
http://www.iiav.org/index.php?va=congresses

12 – 15 August, New York, USA
Inter-Noise 2012
http://www.internoise2012.com

9 – 13 September, Portland, USA
Interspeech 2012
http://www.interspeech2012.org

2013

26 – 31 March, Vancouver, Canada
IEEE International Conference on Acoustics, Speech, and Signal Processing (ICASSP)
http://www.icassp2013.com

2 – 7 June, Montréal, Canada
21st International Congress on Acoustics (ICA 2013)
http://www.ica2013montreal.org

Meeting dates can change so please ensure you check the conference website: http://www.icacommission.org/calendar.html

SUSTAINING MEMBERS

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

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www.3m.com

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www.acousticresearch.com.au

ACRAN
www.acran.com.au

ACU-VIB ELECTRONICS
www.acu-vib.com.au

ADAMSSON ENGINEERING
www.adamsson.com.au

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www.aaac.org.au

BORAL PLASTERBOARD
www.boral.com.au

BRUEL & KJAER AUSTRALIA
www.bksv.com.au

CSR BRADFORD INSULATION
www.csr.com.au/bradford

EMBELTON
www.embelton.com.au

ENERFLEX ENVIRONMENTAL
www.enerflexglobal.com

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www.colpro.com.au

NSW DEPT OF ENVIRONMENT & CLIMATE CHANGE
www.environment.nsw.gov.au

PEACE ENGINEERING
www.peaceengineering.com

PYROTEK NOISE CONTROL
www.pyroteknc.com

SINCLAIR KNIGHT MERZ
www.skm.com.au

SOUND CONTROL
www.soundcontrol.com.au

SOUND SCIENCE
www.soundscience.com.au

VIPAC ENGINEERS AND SCIENTISTS
www.vipac.com.au

OBITUARY

Graham Thirkell

8 April 1937 to 14 April 2010

Graham Thirkell began his professional life as an electronics engineer however he is remembered equally for his contributions to electronics and studio acoustics. Graham won the Victorian top apprentice of the year in 1956 (electrical fitting and armature winding). At 18, he was chief engineer at the electronics company Byer, supervising a staff of about 12. He went on to head research and development teams at Byer and later the loudspeaker company Rola which bought out Byer in 1957. During his time at Byer Graham was involved in the development of transportable reel to reel tape recorders. The technology was new and there was incentive from many quarters to have the machines ready in time for the 1956 Melbourne Olympics. Graham left Rola in 1960 to work at Telefil Sound and Film Recording Studios in St Kilda (Melbourne). In the studio environment Graham's passions for high end audio and physics found expression in acoustics. While continuing to work in the field of audio electronics, Graham began to educate himself about acoustics. Graham left Telefil to return to head research and development at Rola Plessey when Rola was bought by Plessey in the mid 1960s. At Rola Plessey, Graham was involved in the development of 4 and 8 track tape recording machines. During this period he also began building mixing consoles and audio processors at home under the Sontron banner. Sontron started in the lounge room of the Thirkell family home and expanded to the passage and ultimately the kitchen before Graham gave up his day job. He and his wife and business partner Katherine moved the business to a shopfront in Huntingdale in the late 1960s. The business focus was on responding to the niche electronic needs of film, television and audio studios. Graham was at the international forefront in the adaptation of integrated circuits and micro-controller technology to recording industry requirements. As well as servicing many studios, he was involved in the setting up of Armstrong Studios (later AAV). During this period Graham built up a reputation for being inventive, practical and good in a crisis. Maintenance of state of the art equipment in an industry where talented people are working under pressure required all these skills.

For many, Graham's most memorable invention of this era was the Editron audio and video time code synchronisation system. Every major film mixing theatre in Australia used the Editron. It found an international market and is reportedly still relied on in some studios after more than 20 years of operation.

Graham worked closely with recording engineers and he listened to what they had to say. During the late 70s there were ground breaking developments in recording technology; such as 24 track recording and the 24 track mixing desk, and standards in the industry rose suddenly. Initially the studio building was left behind. Different recording spaces and monitoring rooms resulted in different sounds, and led to disappointment that the potential of the new technology was not being realised. Graham responded by turning his attentions to room acoustics. He encouraged better bass absorption and was an early proponent of the 'live end/dead end' control room. Listening to engineers also led Graham to investigate alternative methods for measuring and quantifying the acoustics of a space. Frustrated that the available measuring methods could not account for the nuances in listening rooms articulated by the recording engineers he worked with, Graham imported possibly the first TEF Analyser into Australia. (The TEF is an instrument designed to perform time, energy and frequency (TEF) measurements using time delay spectrometry). Among other things, the TEF enabled him to investigate phasing issues and bass build up within a room and to identify early and delayed reflections.

From room acoustic trouble shooting Graham's involvement in studio acoustics moved to sound isolation and complete studio design. He has had creative input into most major television and recording studios established on the east coast of Australia between 1970 and 2000 and independently designed some 100+ facilities. Additionally Graham worked with Melbourne Architect Peter Brown on facilities such as the ABC Southbank, and Soundfirm Melbourne and Sydney. Throughout his career Graham continued to undertake electro-acoustic projects, working with cinemas, museums and exhibition spaces as well as studios. His company Acoustisearch completed the acoustic and electro-acoustic design of large format and special cinemas, including the large format cinema 'The Edge' at Katoomba, and the cinemas at ACMI, which conform to THX certification requirements. In 2007 the Australian Screen Sound Guild awarded Graham its most prestigious award, The Syd Butterworth Lifetime Achievement Award.

With his background in electro-acoustics and his knowledge of acoustics Graham was uniquely positioned to address acoustic problems that arose in studio and similar high end listening spaces. He is remembered by many professionals in these fields for his willingness to take on any challenge, his inventiveness and for his successes. Graham is survived by Katherine Thirkell, his children Christine, Linda and Gerrard and grand children Sarah, Donovan, Bronte and Aliesha.

Dianne Williams
(Dianne worked with Graham Thirkell from 1999 to 2003)

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