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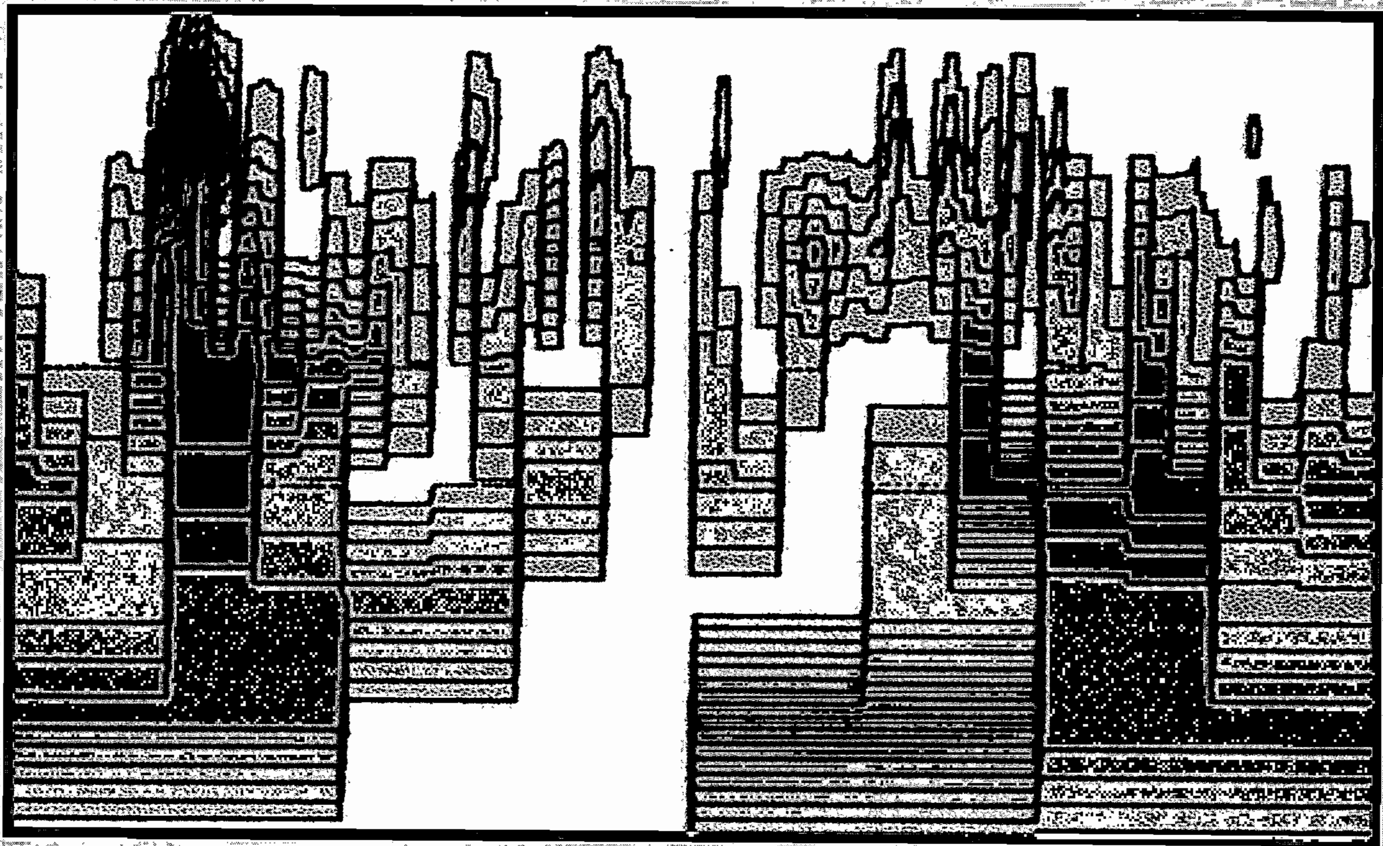

- Heart Sounds
- Impact Noise
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Australian Acoustical Society

Vol. 25 No. 1 April 1997

Acoustics Australia

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http://www.adfa.oz.au/~mxb/aa

Acoustics Australia is published by the Australian Acoustical Society
(A.C.N. 000 712 658)

Printed by
Cronulla Printing Co Pty Ltd,
16 Cronulla Plaza,
CRONULLA 2230
Tel (02) 9523 5954,
Fax (02) 9523 9637
email: cprint@printnet.com.au

ISSN 0814-6039

Vol 25 No 1 April 1997

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From the President

Lobbying is a fact of life. We do it when seeking support to join a golf club or when we feel strongly about some political matter. Unfortunately, one solo voice is unlikely to gain much attention in the halls of power, so it is necessary to join forces with like minded groups in order to make an impact. For this reason the Society became a member of FASTS, the Federation of Australian Scientific and Technological Societies. After a slow start, this group has become an active body, developing a policy document, giving out press releases and holding discussions with relevant Government personnel. A recent FASTS activity, a one day forum in Canberra on the state of mathematics and science education, gained considerable notice and media coverage. Current FASTS activities include formulating a response to the West Review of Higher Education and also to the Stocker Review of publicly-funded Science and Technology in Australia. The latter review is seeking to identify national priorities and locate any gaps and overlaps in funding.

Consultants and manufacturers of acoustic related products "use" the output from the education programs and/or the results from technological developments. Do you know of shortcomings or examples where it is necessary to maintain or improve standards in education? Are you aware of misdirected or inadequately funded research and development, particularly in areas relevant to acoustics ? While we may "know" things, often we sit back and let events happen about us. This could be dangerous. Implications of Government decisions made today can carry through the next decade. By yourself, your voice may be insignificant, however, when joined with others from our Society and combined with many more from other groups, perhaps attention can be raised in Canberra. The Society can only act effectively if we have adequate input from its members. So let us know your thoughts by writing to the Editors of Acoustics Australia who also act as AAS contacts with FASTS.

This year Council intends to produce an updated version of the Memorandum and Articles of Association of the Society. We wish to make them gender neutral, eliminate repeating such expressions as "unless repugnant to the context or provisions of these articles" and replace much of the text with simpler English. Also, parts of the articles unnecessarily stress the NSW or Victorian Divisions: which is irrelevant now we have active Divisions in other states. It is not intended to alter the general thrust of the articles, rather we wish to bring them in line with current practice. The proposed changes will be prepared by a subcommittee and Members will be given the opportunity to vote on the changes later in the year. The subcommittee would appreciate any comments and these can be forwarded to the General Secretary (address on information page of this journal)

Charles Don, President

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Wavelets and Heart Sounds

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ABSTRACT: Simple quantitative procedures for analysing heart sounds have been investigated. They are based on the discrete wavelet transform, DWT, particularly those developed within the past decade or so. The early indications are that these transforms are particularly useful in being able to identify clearly the presence of murmur and of some low frequency phenomena associated with abnormal heart sounds. While the origin of the first is well understood, the source of the latter requires further investigation. It occurs at frequencies well below the audio-frequency range and would not be detected with normal auscultatory procedures. The work is continuing.

1. INTRODUCTION

While the ear is adept at perceiving patterns in non-stationary acoustic signals such as those produced by the heart, it lacks the ability to respond to frequencies in the infrasonic region and to quantify signals in the audio-frequency range. Short-time Fourier transforms (STFT) have been shown to be useful for presenting visual and quantitative images of heart sounds [1-3], and can easily identify certain abnormalities. They are limited however in their ability to resolve low frequency components although they are excellent in highlighting higher frequency phenomena such as murmur.

The development of the discrete wavelet transform (DWT) over the past couple of decades [4], has introduced a method for analysing sounds from a different perspective. In addition, they have a constant percentage type of character (unlike the constant bandwidth approach of the STFT), which opens up the possibility of examining low frequency phenomena in more detail than is possible with the STFT. There are tradeoffs of course at the high frequency end and time will tell if, on balance, they provide more insight than do STFTs for identifying a wide range of abnormal heart sounds. The early stages of the study reported here suggest that DWTs do indeed have potential and there is no difficulty in identifying abnormal sounds. The diagnostic importance of the significant amount of infra-sound associated with abnormal hearts has yet to be established.

2. HEART SOUNDS

The functioning of the heart and the origin of heart sounds is only briefly discussed here and is available in much more detail elsewhere [5].

Heart sounds are of two essential types. The first is of a transient nature and is of short duration. It is associated with the commencement and cessation of blood flow as the result of opening and closing of various valves – the familiar "water hammer" effect. The second type can be more continuous and is associated with turbulent flow which gives rise to murmurs.

In a normal cardiac cycle the transient heart sounds occur twice – "lubb-dupp". The first sound is associated with vibrations in the atrio-ventricular chambers and the contained blood. They are caused by the abrupt rise in ventricular pressure resulting from ventricular contraction. That is immediately followed by a release in pressure at the onset of the ejection of blood through the semi-lunar valves. The vibrations of the ventricles and blood are transmitted through the surrounding tissues and reach the chest wall where they can be detected.

The second heart sound occurs when the semi-lunar valves close, usually fairly suddenly, causing a rapid cessation in blood flow. The resulting vibrations of the blood and vessels similarly transmit through to the chest wall. This sound is usually of a slightly higher frequency than the first and is, of course, of shorter duration.

Murmurs are caused, as noted earlier, by turbulent flow. This could be the result of imperfect closure of valves, restrictions to the flow paths, pathologic communications in the cardio-vascular system or ruptured intracardiac structures. They are usually of significance although "innocent" murmurs sometimes occur in younger children.

Clearly the transmission path from the heart to the chest wall will affect the character of the sound detected at the chest. Some researchers advocate invasive procedures to minimise this effect [6], but this study has not followed that approach at this stage æ a simple non-invasive procedure is preferred.

3. DISCRETE WAVELET TRANSFORMS

The discrete wavelet transform is just one of many ways available for decomposing a signal into some more basic components. There are several different families of wavelets, and the ones considered here are relatively recent and are due to Daubechies [7,8]. Other families are still under investigation.

Each family of wavelets consists of its own basic shape characterised by the "mother" wavelet. This is required to have very specific properties as discussed [9]. There are in addition, a whole series of related wavelets, all of the same

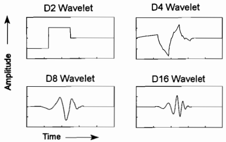

Figure 1. Some typical Daubechies wavelets.

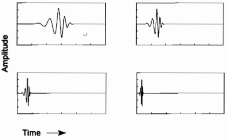

Figure 2. Some members of the D16 wavelet family.

basic shape as the mother wavelet, but translated in time, and contracted by various amounts depending on their respective levels.

An appreciation of some of the different Daubechies wavelets can be obtained by reference to Figure 1. There, four of the wavelets are shown, namely D2, D4, D8 and D16.

They are indicative of this group. It happens also that the D2 wavelet is also referred to as the Haar wavelet due to its earlier development, [10]. Some of the members of the D16 wavelet are shown in Figure 2. They are all of the same general shape, as noted above, but are of shorter and shorter duration. They thus belong to different "levels". Level 0 contains one wavelet which occupies the entire period of the signal under analysis. Level 1 contains two wavelets, level 2 contains four wavelets, level 3 contains eight wavelets, and so on. For a signal containing 512 data points, the highest level, level 8, will contain 256 wavelets, all equally spaced and partly overlapping, as they are for the lower levels.

Each individual wavelet at each level is adjusted in amplitude so that, when they are all summed, the resulting signal will closely represent the original signal.

It is worth noting in passing that since the various wavelets are "compact" in the time domain, they will not be compact in the frequency domain. Not only does this vary with level, it also varies from family to family. Some idea of this latter variation can be obtained by reference to Figure 3. There, the spectral representations of four different wavelets at level 5 are given. Note that the most compact, the D2, has the most extended spectra, while the least compact, the D16, has the most compact spectra, as expected.

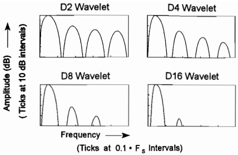

Figure 3. Spectra of some Daubechies wavelets.

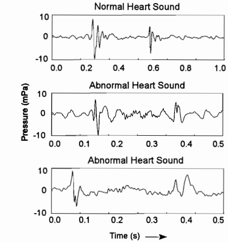

Figure 4. Some heart sounds.

It is not immediately obvious which of the above wavelets is the most helpful when analysing heart sounds and when attempting to identify abnormalities. Clearly for very abrupt changes in level, the D2 series would be likely to be more useful but this series would struggle when representing more gradual changes. The D8 and the D16 families may have some advantage since their general shape and character are more like those of the normal heart sound as discussed later. However, any of the wavelet families can be used legitimately to decompose the heart sounds and each will present its own particular interpretation.

4. APPLICATION OF DISCRETE WAVELET TRANSFORMS TO HEART SOUNDS.

The three heart sounds shown in Figure 4 are representative of a normal heart, and two levels of abnormality. The first was produced by a young adult (20 years) with no known abnormalities. The second came from a child (2 years) who was known to have a ventricular septal defect giving rise to a murmur between the first and second heart sounds. The third belonged to an adult (32 years) who was suffering from hypertension.

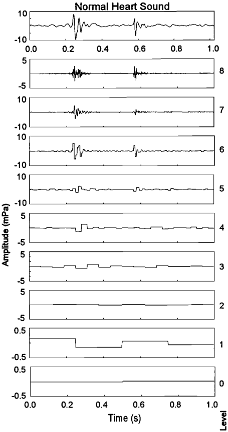

Figure 5. D2 series of a normal heart sound.

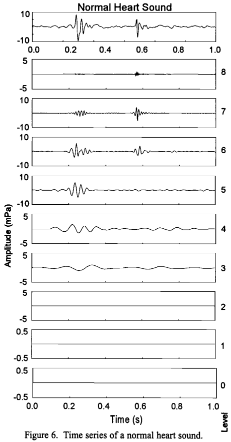

Figure 6. Time series of a normal heart sound.

The signals were collected by a B & K type 4146 microphone and associated power supply each capable of recording very low frequencies. The signals were originally sampled at 1000 Hz but subsequently resampled so that each cardiac cycle consisted of 512 data points.

Signal analysis was carried out using programs from [9] in conjunction with the Matlab and DADiSP suite of programs. Final graphical representation made use of the Wordperfect package Presentations.

4.1 Normal Heart Sounds

The decomposition of the normal heart sound using D2 wavelets is shown in Figure 5. Note that all components exist at all levels although the greatest amplitudes occur at level 6. This decomposition can be considered as a filtering process with the amplitudes at each level depending on the type of filter (D2 in this case) as well as the character of the original signal.

The decomposition using the D16 wavelet is given for comparison in Figure 6. Levels 5 and 6 now have about equal amplitudes and there is less activity at level 8. The amplitudes at the other levels are relatively comparable. Figures 5 and 6 thus represent different and legitimate ways of decomposing the signal.

Space does not permit presentations of all results for all of the wavelet series but the D4 and D8 wavelets did not appear to offer any particular insight which was not already reasonably apparent from these two figures.

The information shown in these figures can be presented more compactly using the method developed in [11]. Thus discrete wavelet "maps" or "scalograms" offer essentially the same information except that the various amplitudes at the various levels are plotted as contours on a decibel scale. The abscissa again is time while the ordinate is wavelet level æ the higher the level, the finer the scale.

Scalograms for the D2, D8 and D16 wavelets are shown in Figures 7(a), 7(b) and 7(c) using contours at 3dB intervals. Each figure clearly shows the first and second heart sounds with the first sound extending over more levels than the second as expected. Each also shows periods of relative calm. There is some low level activity in the time period from 0.6 to 0.8 seconds at levels 3 to 6. The significance of these has not been established.

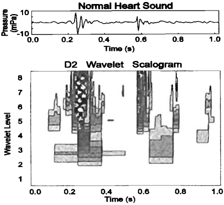

Figure 7(a)

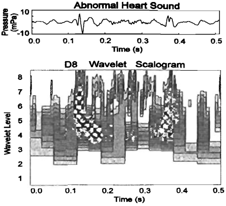

Figure 7(d)

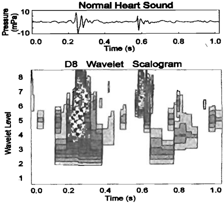

Figure 7(b)

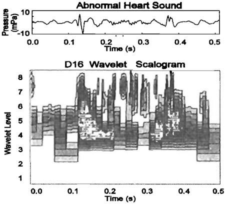

Figure 7(e)

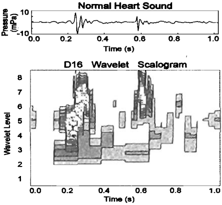

Figure 7(c)

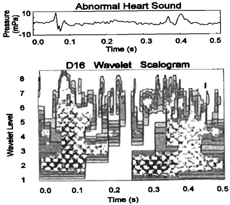

Figure 7(f)

Figure 7. Discrete wavelet transforms of heart sounds.

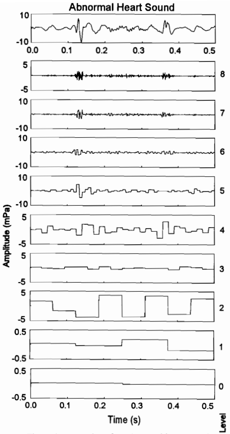

Figure 8. D2 series of an abnormal heart sound.

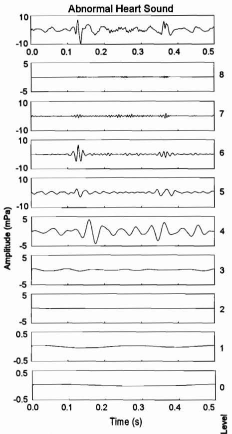

Figure 9. D16 series of an abnormal heart sound.

All of the scalograms give a clear quantitative impression of a normal heart sound and are offered as a basis for comparison with the abnormal heart sounds.

4.2 Abnormal Heart Sounds

The time series decomposition of the first of the abnormal heart sounds of Figure 4 using the D2 wavelet is given in Figure 8. This signal now contains more continuous activity at levels 7 and 8 which can be related to the murmur produced by blood flow through the ventricular septal defect. There is also considerable activity at the lower wavelet levels, eg levels 2 to 4, although the origin of those signals is not known. These components are below the audio-frequency range.

The D16 representation, Figure 9, confirms the changed pattern at level 4 although the level 2 component is comparable to that of the normal heart. The continuous activity at levels 7 and 8 are again apparent and can be distinguished from those of the normal heart sound given in Figure 6.

Scalograms based on the D8 and D16 wavelets are given in Figures 7(d) and 7(e). There are some clear differences in the abnormal heart when compared with those of the normal heart sound, Figures 7(b) and 7(c).

The components due to the murmur are evident at about 0.25 seconds at levels 6 to 8. The continuous activity at level 4 noted

previously is again apparent. There is no doubt that the appearance of these scalograms is considerably different to those for the normal heart. There is no difficulty in identifying abnormal heart sounds.

The final example of a wavelet time decomposition of an abnormal heart sound, that of the third signal given in Figure 4, is shown in Figure 10 using the D16 wavelet. There are distinctive features.

Continuous activity occurs at very low wavelet levels, namely levels 0,1 and 2. These are all well below the audio-frequency range. The sharpness of the second heart sound is missing æ there are now no conspicuous peaks at levels 6 and 7, for example, as there are in say Figure 6. Most activity occurs at level 4. The first heart sound has very little activity at level 6 in contrast to that of the normal heart.

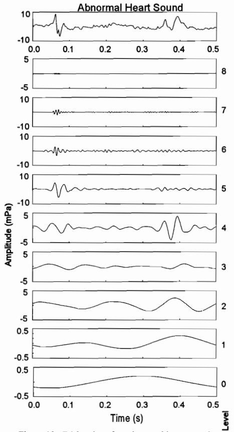

Figure 10. D16 series of an abnormal heart sound.

The corresponding scalogram is given for this heart sound in Figure 7(f), and may be compared with the normal heart sound of Figure 7(c). The most striking differences are highlighted, as expected, at the lower wavelet levels. Thus levels 1,2,3 and 4 are fairly continuous. The first and second sounds are not nearly as distinct and there also appears some evidence of murmur at levels 6 and 7 during a fair proportion of the cardiac cycle. The overall impression is one of disorder in stark contrast to the uncluttered simple pattern of the normal heart sound pattern.

5. CONCLUSIONS

Discrete wavelet transforms are useful for identifying normal and abnormal heart sounds. The use of scalograms in particular presents clear quantitative impressions of the variation in heart sound with time, and also from heart to heart.

Each of the four wavelets investigated was easily able to detect convincingly abnormalities in heart sounds and it is still not clear which will finally prove to be the most useful. Further experience with a wider range of heart sounds is needed.

The discrete wavelet transform is particularly attractive in being able to detect the presence of activity at low wavelet levels, i.e. at very low frequencies. The origin of these components in the abnormal heart sounds is not known. These components would not be detected with conventional auscultatory procedures. At the same time heart murmur is also easily detected with components appearing conspicuously at the higher wavelet levels. Their position in time is clearly distinguishable (unlike the low level components).

There is good reason to explore and to exploit discrete wavelet transforms for analysing heart sounds. They appear to be ideally suited for the purpose.

ACKNOWLEDGMENTS

The work described here is part of a collaborative program with the Monash Medical Centre and the Centre for Biomedical Engineering at Monash University. Thanks are due to Dr. Dan Penny and Associate Professor Ian Brown at these respective organisations.

REFERENCES

1. M.S. Obadial, "Phonocardiogram signal analysis: Techniques and performance comparison". *J. Med. Eng. and Tech.* **17**(6), 221-227 (1993).
2. R.J. Alfredson and K.M. Smith, "Time dependent spectral analysis procedures for analysing heart sounds" *Proc. Fifth Western Pacific Regional Acoustics Conference*, Korea, 1994, pp. 247-252.
3. J.C. Wood and D.T. Barry, "Time frequency analysis for the first heart sound", *IEEE Eng. in Med. and Biol.*, March/April, 1995, pp. 144-151.
4. I. Daubechies, "Where do wavelets come from? æ A personal point of view", *Proc. IEEE*, **84**(4), 510-513. (1996)
5. A.J. Vander, J.H. Sherman and D.S. Luciano, *Human Physiology*, McGraw-Hill, New York, 1990, pp. 362-380.
6. J.C. Garcia, S.A. Layton and B.J. Rubal, "Comparison of the frequency response characteristics of catheter mounted piezoelectric and micromanometric phonotransducers", *Cathet. Cardiovasc. Diagn.* **17**, 48-55 (1989)
7. I. Daubechies, "The wavelet transform, time-frequency localisation and signal analysis", *IEEE Trans. Inf. Theory* **36**, 961-1005 (1990)
8. I. Daubechies, S. Mallat and A.S. Willsky, "Wavelet transforms and multiresolution signal analysis", *IEEE Trans. Inf. Theory* **38**, 529-531 (1992).
9. D.E. Newlands, *An Introduction to Random Vibrations, Spectral and Wavelet Analysis* Longman, Essex 1993, 3rd ed., pp. 295-370.
10. A. Haar, "Zur Theorie der orthogonalen Funktionensysteme", *Math. Ann.* **69**, 331-371 (1910).
11. D.E. Newlands, "Some properties of discrete wavelet maps", *Prob. Eng. Mech.* **9**, 59-69 (1994).

Application of Active Noise Control to Noise Barriers

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This paper is based on the 1996 PRESIDENT'S PRIZE. This prize, established in 1990 by the Australian Acoustical Society, is awarded to the best technical paper presented at the Australian Acoustical Society Conference.

ABSTRACT: Previous work has shown that active noise control technology may improve the low frequency performance of noise barriers. In this paper, such a possibility is confirmed. A multi-channel active control system has been used to create quiet zones on the top of a barrier in order to reduce the diffraction along the top, and to increase the insertion loss of the barrier. Both the simulation and experimental results obtained showed that a barrier assisted with an active noise control device achieves extra noise attenuation when the control system is optimally arranged. The results also demonstrate that active noise control is particularly effective at low frequencies in increasing the insertion loss of a noise barrier. This feature of active noise control overcomes the weakness of noise barriers at low frequencies.

1. INTRODUCTION

Barriers are classical devices in the field of noise control. When a noise barrier is interposed between the noise sources and the receivers, the direct sound wave will be blocked. Only the diffracted sound wave will contribute to the noise level in the area behind the barrier. Behind the barrier, the contribution of the diffracted field is relatively weak compared with that from the original direct field. Therefore, an area of quiet, or a 'dark' area, can be created.

The effectiveness of a barrier in blocking the noise depends upon many factors such as the characteristics of the noise source, the shape and dimensions of the barrier, and environmental conditions. It has been found that while the barrier is very effective in attenuating high frequency noise, it becomes ineffective at low frequencies where the wavelength of the noise is comparable with the height and length of the barrier. Increasing the height of the barrier can improve the low-frequency performance of the barrier, but it is usually not practical. As a result, the improvement in the performance of a barrier, especially in the low-frequency range, has been a research topic in the field of acoustics for more than 20 years.

Although the concept of using noise to cancel noise is not new [1], the recent developments of the control technique have made the implementation of active noise control practically possible. Since active noise control (ANC) is very effective in attenuating low frequency noise [2], it is reasonable to believe that the low-frequency performance of a barrier can be improved by this technique.

Ise [3] applied an adaptive control system into a half-scale model of a passive barrier. In Ise's system, a speaker was used as a monopole control source and positioned on the top of the barrier. The error microphone was set in the desired space behind the barrier. He was able to create a quieter area around the error microphone at very low frequencies (125 Hz or lower). Omoto [4] used a multiple channel adaptive controller in his control system. In a different approach from Ise's arrangement, Omoto put all the error microphones on the top of the barrier. As the sound pressure at the top behaves like virtual sources of the diffracted field, the mechanism of this arrangement was to cancel the sources of the diffractive noise around the top of the barrier. For his specific configuration, Omoto concludes that when the interval of the error microphones on the diffraction edge is less than half of the wavelength, the active noise barrier works effectively.

The present authors [5,6] have thoroughly investigated active noise control in open space. It is found that a large volume (in terms of the wavelength) of noise attenuation can be obtained when the control system is optimally arranged. In this paper, we apply our findings about active noise control in open spaces to a noise barrier, and illustrate the effectiveness of ANC in improving the low-frequency performance of noise barriers.

2. INSERTION LOSS OF NOISE BARRIER

Many theories may be used to predict the sound insertion loss of noise barriers. The basic ones are the Huygen's principle and the Kirchhoff's diffraction formulation [7, 8]. For the reflective noise barrier shown in Fig. 1, and using a point noise source with pressure field of

$$P_0 = \frac{A}{kr}e^{ikr}, \tag{1}$$

the diffracted field can be approximately expressed as [9]

$$P_d = -\sqrt{\frac{2}{\pi k R_1}}A\,e^{-i\pi/4}\{\mathrm{sgn}(\pi+\alpha-\phi)\frac{e^{ikR}}{\sqrt{k(R_1+R)}}F\left[\sqrt{k(R_1-R)}\right] + \mathrm{sgn}(\pi-\alpha-\phi)\frac{e^{ikR'}}{\sqrt{k(R_1+R')}}F\left[\sqrt{k(R_1-R')}\right]\} \tag{2}$$

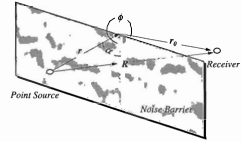

Figure 1. Schematic of a noise barrier.

for $kR_1 >> 1$, where k is the wave number of the sound, R and R' are respectively the distances from the receiver directly to the source and to the source mirror image in the barrier. $R_1 = r + r_0$ is the shortest distance from the source to the receiver over the barrier top, $A = -iZ_0 q$ where q is the source strength, $Z_0 = \omega^2 \rho_0 / 4\pi c_0$, and

$$F(\mu) = \int_{\mu}^{\infty} e^{j\chi^2} d\chi \qquad (3)$$

is the Fresnel integral. The symbol sgn is the sign function, and α and ϕ are the angles defined in Fig. 1.

The sound insertion loss caused by the barrier then can be given by

$$\Delta L = 20 \log_{10} (|P_d / P_0|), \qquad (4)$$

where P_0 is the sound pressure at the receiver's position when the barrier is absent, as expressed by Eq. (1). A widely used engineering approximation for the sound insertion loss of the barrier is Maekawa's asymptotic expression [10]

$$\Delta L = -10 \log_{10} (3 + 20N), \qquad (5)$$

where N is Fresnel's zone number of the barrier, expressed as

$$N = \frac{2}{\lambda} \delta, \qquad (6)$$

where $\delta = r + r_0 - R$ is the path difference and λ the wave-length of the diffractive sound.

The insertion losses of the barrier described by Eqs. (4) and (5) are shown as a function of frequency in Fig. 2 for a typical noise barrier of height of 1 m. Located on different sides of the barrier, the noise source and receiver are both 0.5 m high, and both at a distance of 2 m away from the barrier.

It is shown in Fig. 2 that the sound attenuation of the barrier at a point in the quiet area is only 5 dB at the low frequencies, while the insertion loss at the high frequencies is more than 10 dB. These indicate that the effort of improving the performance of the noise barrier should be focused on the low frequency range.

3. ACTIVE NOISE CONTROL IN OPEN SPACE

Active noise control in an open space can be implemented by either global control or local control. It has been found that global control can only be achieved when the control sources and the primary sources are closely located. In many practical applications, this condition may not be satisfied, in which case, the local control strategy seems to be the only choice.

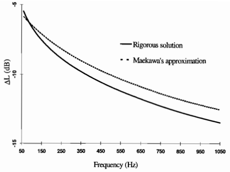

Figure 2. Sound insertion loss of a specific barrier as a function of frequencies.

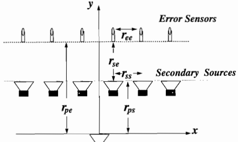

Figure 3. A typical arrangement of a multi-channel active noise control system in open space.

It has been shown [5] that the total sound power output of the local control system usually increases after control, which means that while a 'quieter' area can be created in some places, there must be other areas with an increase in sound pressure. The objectives of local control of the sound pressure field are (1) to create large quiet zones at required positions (where error sensors are located), and (2) to minimise the increase of the sound pressure at other locations (or to minimise the increase in the total power flow from all sources). The quiet zone is defined as the area where the primary sound pressure level is attenuated by more than 10 dB.

Figure 3 is a typical control system configuration, where the equally spaced N secondary sources and N error sensors are placed in two parallel lines. A monopole primary source is located on the central axis of the arrays of secondary sources and error sensors. The distance between the primary source and the secondary source array in the y direction is r_{ps}, and that between the secondary source array to the error sensor array is r_{se}. The secondary sources and the error sensors are separated respectively by r_{ss} and r_{ee}, with $r_{ss} = r_{ee}$ in this arrangement. The sum of the squared sound pressures at the microphone positions is selected as the cost function. Our research on this control system has found that when both the distances from the noise source to the control sources, and from the control sources to the error microphones are given, there exists an optimal range of intervals among the adjacent

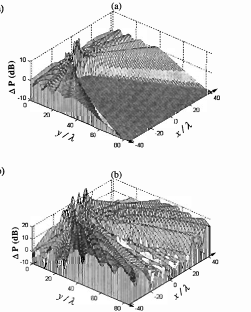

Figure 4. Sound pressure attenuation of a control system with 21 secondary sources and 21 error sensors when r_{ps}=5λ, r_{se}=5λ, and (a) r_{ss}=0.715λ, and (b) r_{ss}=0.88λ.

control sources and error microphones. Within this range, the increase of the total sound power output is minimised, and the largest area of quiet zone can be obtained, which resembles a wedge with its edge along the error microphones. The upper and lower limits of this range are expressed [6] as

$$r_{ss-\max} = \begin{cases} \dfrac{\lambda}{2}\sqrt{1+\dfrac{4r_{se}}{N\lambda}}, & N = 2,\ 4,\ 6,\cdots, \\ \dfrac{\lambda}{2}\sqrt{1+\dfrac{N+1}{N-1}\dfrac{4r_{se}}{N\lambda}}, & N = 3,\ 5,\ 7,\cdots, \end{cases} \tag{7}$$

and

$$r_{ss-\min} = \begin{cases} \dfrac{5\lambda}{2}\exp\left\{-\left[\dfrac{3(\lambda+0.04r_{ps})}{2r_{se}-\lambda}+\dfrac{20\lambda}{15\lambda+r_{ps}}\right]\right\}, & N = 4,\ 6,\ 8,\cdots, \\ \dfrac{3\lambda(N+1)}{N}\exp\left\{-\left[\dfrac{\lambda+2r_{ps}}{2(2r_{se}-\lambda)}+\dfrac{12\lambda}{5\lambda+r_{ps}}\right]\right\}, & N = 3,\ 5,\ 7\cdots. \end{cases} \tag{8}$$

It has also been found that for the configuration with the intervals outside the above range of limits, the system is not able to create a large area of quiet zone, and a large sound power output is often observed.

Figure 4 gives examples of quiet zones actively created in free space (no barriers) by a multiple control system with 21 control sources and 21 error microphones. The system is arranged with the primary source at the position (0, 0, 0), the 21 secondary sources at (5λ, $(i-11)r_{ss}$, 0) (i=1,2…,21), and the 21 error sensors at (10λ, $(i-11)r_{ss}$, 0) (i=1,2…,21), where r_{ps}=5λ and r_{se}=5λ. The upper limit and the lower limit for optimum performance of the system can be calculated by Eqs.

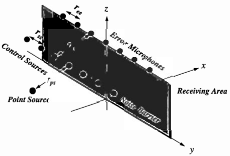

Figure 5. Active noise control system on a noise barrier.

(7) and (8) to correspond to r_{ss}-min=0.5λ and $r_{ss\text{-}max}$=0.715λ. Fig. 4(a) is the configuration within the optimal range (r_{ss}= $r_{ss\text{-}max}$) and Fig. 4(b) is the configuration outside the optimal range (r_{ss}=0.88λ> $r_{ss\text{-}max}$).

It can be seen that the quiet zone created by the optimal configuration is quite large, with no significant increase of sound pressure level outside the quiet zone (except for the area close to x=0 and y=0, where the secondary sources are located). For the configuration just outside the limits, there is hardly any evident quiet zone, and most of the areas suffer from a large increase of sound pressure level, as shown in Fig. 4(b). It can also be seen that the area of quiet zone created by the optimally arranged control system is scaled in terms of the wavelength of the noise, which means, in a practical application, the lower the frequency of the noise, the larger the area of quiet zone.

4. ACTIVE NOISE BARRIER

As the sound around the top of the barrier contributes a diffracted sound field in the quiet area behind the barrier, it is reasonable to believe that if the sound over the top can be cancelled, the diffracted sound in the quiet area behind the barrier should also be attenuated. This approach should have the equivalent effect to an increase of height of the noise barrier. Consequently the insertion loss of the barrier can be increased.

The multi-channel system used for open space noise control is applied here to a noise barrier. The control system consists of N control sources and the same number of error microphones, as shown in Fig. 5. The control sources and error microphones are equally spaced in two parallel lines. The array of error microphones is located just on the top of the barrier. The control source array is located between the primary source and error microphone array, and in the same plane containing both primary source and error microphone array.

When the ANC system is on, the sound pressure at the error microphones is cancelled, and a quiet zone along the top of the barrier is then created. This also increases the noise attenuation in the receiving area behind the barrier. The total diffracted sound pressure becomes

$$P = P_{pd} + \sum_{i=1}^{N} P_{sd}^{(i)}. \tag{9}$$

P_{pd} is the diffraction caused by the primary noise source only, which also represents the diffractive sound while the active control system is off, and $P_{sd}^{(i)}$ the diffraction caused by the i control source. Both P_{pd} and $P_{sd}^{(i)}$ are expressed in the form of Eq. (2). The extra sound insertion loss created by the active multiple control system can then be described as

$$\Delta L = 20\,\log_{10}(|P/P_{pd}|). \tag{10}$$

When applying the multi-channel active noise control system to the noise barrier, the configuration of the control system, such as the intervals of the adjacent control sources and the adjacent of the error microphones, is extremely important. It has been found that the optimal configurations of the control system in open space also apply to the active noise barrier shown in Fig. 5. The specific active noise barrier used in this analysis is 1 m high and located along the y axis. The location of the primary source is (-1.376, 0, 0.5) and the control sources are located at (-0.688, $(i-(N+1)/2)r_{ss}$, 0.75). The error microphones are located at (0, $(i-(N+1)/2)r_{ss}$, 1). For the control system with 3 control sources and 3 error microphones and an operating frequency of 500 Hz, the optimal range of r_{ss} is [0.22λ, 0.98λ] according to Eqs. (7)-(8). The extra sound attenuations of two configurations (one with r_{ss} within the limits as r_{ss}=0.75λ, another with r_{ss} outside the limits as r_{ss}=1.75λ) are given in Fig. 6. For this case, the ground on both sides of the barrier is assumed to be non-reflective.

It is clear that an active noise control system can effectively improve the insertion loss of the barrier if the system is optimally arranged. When the intervals of the control sources are within the optimal range, the extra sound attenuation of the barrier due to the control system can be more than 10 dB in the area behind the barrier, as shown in Fig. 6(a). If the intervals of the control sources are outside the optimal range, the large extra sound attenuation may not be achieved, and the control system may degrade the insertion loss of the barrier, as shown in Fig. 6(b). These examples demonstrate that the configuration of the control system is most important for the active noise barrier.

Although the above conclusions are made from the observation of a simple case of multi-channel active control system (3 control sources and 3 error microphones are used in this simulation), it can be shown that they are also applicable to the cases of more control sources and error microphones.

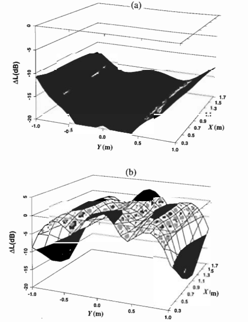

Figure 6. Extra sound attenuation due to the active noise control system when (a) r_{ss}=0.75λ and (b) r_{ss}=1.5λ.

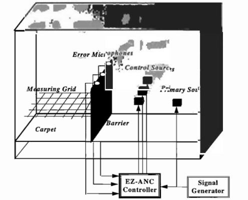

Figure 7. Experiment setup in an anechoic chamber.

5. EXPERIMENTS

Experiments were carried out in an anechoic chamber with the size of 4.2m×4.2m×4.2m. The barrier consists of 2 pieces of plywood plates sandwiching a foam. The barrier of size 0.05m thick, 1.0m high and 4.2m long, was put on the suspended metal grid floor of the anechoic chamber. To prevent the sound propagating underneath the barrier, and to prevent reflection from the floors on both sides, the metal grid floor was covered with thick carpet. The primary noise source was about 1.4 m away from the barrier, 0.5 m above the floor and on the central line of the chamber. The control system consists of 3 half-enclosed speakers as control sources, 3 microphones as error sensors, and a multi-channel EZ-ANC as the controller. The arrangement of the control system is shown in Fig. 7. The sound signal used in the experiment was a pure tone of 500 Hz. The pure tone signal was fed into the primary source directly, and was also provided to the controller as a reference signal. Three control channels of the controller were used to cancel the total sound pressure at the position of 3 error microphones.

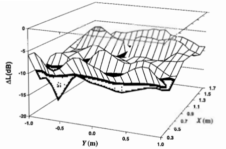

Figure 8. Insertion loss of the barrier from experimental measurement.

The experimental insertion loss of the noise barrier without active control in a measuring plane (0.5 m above the floor) is shown in Fig. 8. It can be seen that due to the reflections from the floor carpet and the walls of the anechoic chamber, the insertion loss of the barrier is not as large and smooth as predicted by the theory. It will be shown later that these reflections also decrease the effectiveness of the active noise barrier.

The coordinates of the control sources and error microphones are the same as in the computer simulation discussed previously [(-0.688, $(i-(N+1)/2)r_{ss}$, 0.75) and (0, $(i-(N+1)/2)r_{ss}$, 1) respectively, where i=1,2,3, and N=3]. Two different intervals of the control sources were used to test the effectiveness of the active noise barrier. One was within the optimal range at $r_{ss}=0.75\lambda$, and the other was outside the optimal range at $r_{ss}=1.75\lambda$. The extra sound attenuation achieved by these two configurations of active noise control system is chosen in Fig. 9.

Figure 9 shows a significant difference in the extra sound attenuation of the active noise barrier for the different configurations of the control system. When the system is optimally arranged, the extra sound attenuation has been achieved at every position in the measuring plane, as shown in Fig. 9(a). When the system is arranged outside the optimal range, the active noise control system may even decrease the insertion loss of the passive barrier in some locations, as shown in Fig. 9(b).

Comparing Fig. 9 with Fig. 6, it can be easily seen that the extra sound attenuation of the active noise barrier in the experiments is not as significant as that of the theoretical analysis. This is due to the reflection from the carpet, as well as from the walls of the anechoic chamber. Practical situations such as a highway barrier and an industrial barrier often have the reflections from the grounds and nearby reflective objects. It is expected that the characteristics of the quiet zone and the optimal arrangement of control sources will relate to the ground impedance and properties of the nearby reflective objects. Further investigation in those aspects is under way.

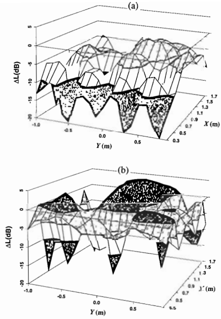

Figure 9. Extra attenuations due to the active noise control system when (a) $r_{ss}=0.75\lambda$ and (b) $r_{ss}=1.5\lambda$.

6. CONCLUSIONS

The effectiveness of applying active noise control system to improve the sound insertion loss of the barrier has been demonstrated in this paper. To create a large area of quiet zone around the diffraction edge of the barrier is a good approach to increase the sound insertion loss of the barrier. An optimal multi-channel active control system developed for active noise control in free space can be successfully applied to the barrier. Similar to the cases of ANC in open space, the configuration of the active noise control system used with noise barriers is extremely important in terms of optimising performance. The extra sound attenuation due to the active noise control system can be significant only when the system is optimally arranged, otherwise, the active control system may be ineffective or even reduce the insertion loss of the barrier.

The size of the improved quiet area is proportional to the number of control channels used in the active control system. As the size of the improved quiet area is scaled in terms of the wavelength of the sound, it can be concluded that the active noise barrier is more useful in the low frequency range where the passive noise barrier is not as effective.

Although the results of the analysis presented in this paper are explained in terms of single frequency sound waves, the optimal arrangement discovered can be used for the design of a control system for practical source with multiple frequency components (such as those from power transformers). Taking the steady state primary sources as an example, the optimal separation distance between control sources is determined by the shortest wavelength of the noise component to be controlled and it is also important to arrange control system so that wavelength of the dominant frequency component is within the optimal range. For those frequency components with wavelength outside the optimal range, band-pass filtering can be used to avoid the effect of control on these components and consequent unnecessary increase of sound level.

In this research, a good coherence between the primary source signal (reference signal) and the error signal is assumed. In all the practical applications of ANC, this is the basic requirement for achieving any significant noise reduction. In the case when the coherence time of wave trains is not very long, spatial causality will set a limit to the arrangement of the control sources and to the processing time of the controller [11].

REFERENCES

1. P. Lueg, *Process of silencing sound oscillations,* U.S. Patent 2043416, 1936.
2. P.A. Nelson and S.J. Elliott, *Active Control of Sound,* Academic Press, San Diego, 1992.
3. S. Ise, H. Yano and H. Tachibana, "Basic study on active noise barrier", *J. Acoust. Soc. Jpn.* **12**, 299-306 (1991).
4. A. Omoto and K. Fujiwara, "A study of an actively controlled noise barrier", *J. Acoust. Soc. Am.* **94**, 2173-2180 (1993).
5. J. Guo and J. Pan, "Analysis of active noise control in a free field", *Proc. Active* **95**, Newport Beach, California, pp. 649-660, 1995.
6. J. Guo, J. Pan and C. Bao, "Actively created quiet zones by multiple control sources in free space", *J. Acoust. Soc. Am.* **101** 1492-1501, 1997.
7. A. Sommerfeld, *Optics,* Academic Press, New York, 1954.
8. M. Born and E. Wolf, *Principles of Optics,* Pergamon Press, London, 1959.
9. J. J. Bowman, T. B. A. Senior and P. L. E. Uslenghi, *Electromagnetic and Acoustic Scattering by Simple Shapes,* North-Holland, Amsterdam, 1969.
10. Z. Maekawa, "Noise reduction by screens", *Applied Acoustics.* **1**, 157-173 (1968).
11. D. G. Pederson and T. A. Laak, "Method for optimizing error filtering based on coherence", *Proc. Active 95*, Newport Beach, California, pp. 649-660, 1995.

Impact Sound Insulation of Building Partitions: Its Measurement and Inherent Difficulties*

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ABSTRACT: Part F5 of the Building Code of Australia 1990 contains performance requirements in relation to the resistance of building partitions to the transmission of impact sounds. The Code specifies the method required under laboratory conditions. The construction is deemed-to-satisfy if it is no less resistant to impact sound than one of three specified walls. Measurements have been carried out on a number of commercial partitions. Great difficulties are experienced so far as details/requirements are concerned, and even more in the interpretation of results. In this paper these difficulties are pointed out, with suggestions for modification of the Code and possible adoption of a means of the expression of results by a single number. Some alarming errors in measured values due to unexpected flanking paths have been detected. Accordingly it has been decided to eliminate the reporting of actual measured values of impact insulation.

1. INTRODUCTION

One of the basic objectives stated by the Building Code of Australia [1] is to "...ensure that acceptable standards of amenity are maintained for the benefit of the community". On the aspects of noise transmission and insulation, Part F5 sets out specifications for the sound insulation offered by partitions, and outlines the method of measurement of impact insulation.

The original purpose of this paper was to present the results and comments given in two papers of the Australian Acoustical Society in Brisbane in November 1996. Paper 1 was confined to measurement of impact insulation with the Code in mind while Paper 2 discussed shortcomings of the Code both in measurement and interpretation. In view of the many Code shortcomings, and some alarming discoveries, this paper will not present measurement results for fear of creating wrong impressions for measuring laboratories desirous of commercially testing partitions in conformity with the Code.

This paper examines measurement procedures for commercial partitions and shows why current procedures are quite unworkable. The interpretation of results poses a far greater problem especially due to the doubtful validity of measured quantities. The question of a possible single-value rating for impact insulation is discussed and questions are posed on the likely benefit to the occupants of an adjoining room.

* The substance of material for this paper was embodied in two papers presented at the Australian Acoustical Society 1996 conference "Making ends meet. Innovation and legislation", Brisbane Nov 13-15, 1996: "Building Code of Australia 1990 - impact sound insulation of building partitions". M.Debevc, "Measurement of impact insulation," and K.R. Cook, "Shortcomings of the code in measurement and interpretation."

2. MEASUREMENT OF IMPACT

2.1 Measuring facilities

In order to measure the airborne sound transmission loss and impact insulation of partitions, Code specification F5.5 requires a measuring facility which complies with AS 1191 [2]. The facilities for the study described in this paper comprise two pentagonal-plan reverberation rooms each of volume 110-120 m^3 and includes an aperture for a sample of 10.69 m^2. This suite satisfies all requirements of AS 1191 which include the testing to ensure that the sound transmitted by any path other than through the partition shall be negligible compared with that through the partition. Valid measurements are possible in one-third octave bands of centre frequency 100 to 5000 Hz. Even though precision is compromised, the measurements may be extended down to centre frequency 50 Hz.

2.2 Application to impact insulation measurements

In following the Code, a horizontal steel platform is placed with one long edge in contact with the source-room side of the test wall. To generate the impacts a tapping machine complying with ISO140/VI-1978 [3] is mounted on the platform. The impact sound pressure levels resulting in the receiving room are measured, in particular at four microphone positions each for a period of 128 seconds. These are then normalised using a reference equivalent absorption area, A_0 of 10 m^2

$$L_{nf} = L_{pf} - 10 \log_{10}(A_0/A_f) \quad (1)$$

where L_{nf} is the normalised sound pressure level in dB at centre frequency f, A_f is the receiving room equivalent absorption area in square metres at centre frequency f, given by

$$A_f = 0.16V/T_{60,f} \quad (2)$$

where V is the receiving room volume in cubic metres and $T_{60,f}$ is the reverberation time in seconds at centre frequency f.

Because of anomalous behaviour detected and discussed later in this paper the actual results of measurement are not presented. However some actual results of measurement of impact insulation by Debevc [4] are available.

2.3 Test of equivalence of results

Table F5.5 of the Code sets out construction details of three walls which have been deemed to possess impact insulating properties which are suitable for dividing walls. The results of measurement of a sample partition are to be compared with those of one of the Table F5.5 walls. The sample partition is deemed to satisfy the Code only if its impact insulation is at least as high as that of one of the standard walls.

2.4 Measurement-procedure shortcomings of Code

At first glance it may appear a particular measuring laboratory need only take comparative measurements of a sample partition and a Table F5.5 partition, though the means of comparison is not stipulated. It is nevertheless important that measurement results are to some extent comparable between measuring laboratories. This will only be feasible if the method of measurement is laid down with some rigidity, not the currently-worded version of the Code. Some of the measurement-method features which may greatly influence the results are as follows:

(a) Location of the impacting of the sample

- (i) In the vertical direction, the distance between the bottom of sample and the source-room floor for the steel platform will vary between measuring laboratories
- (ii) In the horizontal direction, the number of locations for the impacting plate has not been stated so the minimum number must be specified. For the results in this paper the number was chosen as four.

What is of greater significance is the influence on the resulting impact levels due to studs in a cavity wall construction. An average effect will not be obtained unless the impacting plate locations chosen comprise some over the studs and some between the studs.

(b) Nature of the impact – the impact on the sample caused by the tapping machine is in fact not a direct one (in contrast with that used for testing floors). It is an indirect one via the 510 × 10 mm horizontal steel edge of the platform. Should this edge not be a strictly continuous one then the imparted impact energy will not always be constant. In real partitions it may be very difficult for the installer to achieve a perfectly plane surface over each and every region of edge contact.

(c) Precision in equivalent absorption area of the receiving room – A_f in eqn.(1) is required in the normalising of measured sound pressure levels. To enable some valid comparison of results between measuring laboratories the precision in terms say of the 95% confidence interval should be prescribed. (In measurements for this paper, between 100 and 5000 Hz, four microphone positions are used. This number is increased when investigations are carried out down to 50 Hz.)

(d) Frequency range – the Code fails to state the appropriate frequency range. Such a choice will be influenced by the intended destination of results, particularly if there is a desire/intention to derive a single-value for a partition.

2.5 Possible alternative measurement method

The original method of using a tapping machine to investigate floors to simulate footstep noise is not relevant in the study of the effect of common single impacts on walls. It is even less relevant if impacts on a wall are not direct ones but via a plate as required by the Code. For real walls it would be more appropriate to produce single impacts and then to measure the direct energy transmitted to the receiving room. This anomaly is pointed out by Craik [5], Schultz [6], Tanaka et al [7]; this latter work recommends use of a rubber ball and is applicable to a Japanese standard [8]. In practice, generally the disturbance to the amenity of an occupant of a room is due to single impacts caused for instance by closing doors and use of sinks attached to an adjoining wall. Two alternatives that have been investigated are:

Impulse by a falling rod. A polycarbonate rod 30mm diameter length 375mm and mass 0.335kg is attached to a horizontal reference plate by two sisal strings each of length 280mm. The rod is released and completes one-quarter revolution before impacting the sample wall (horizontally). On rebounding the rod is prevented from further action.

As with work with the tapping machine, four locations for the impact by the rod are chosen at the same height above the base of the sample as the tapping method plate. The measurement of the effect of impact in the adjoining room is by measurement of peak sound pressure levels. In this case however the microphone is located in the receiving room direct field at a distance of 535mm from the sample surface and behind the point of impact. Note that for this method the sound pressure levels in each one-third octave band of centre frequency between 50 and 5000 Hz are not normalised.

The advantage of this method is that it becomes possible to standardise it completely. The impacting rod details, the method of impacting and the technique of measurement of the effect in the adjoining room may all be specified.

A similar-type method employed by Craik [5] uses a simple plastic-headed hammer to produce single impacts on walls In work reported by Tanaka et al [7] a special rubber ball is the impact source and is used to drop onto a floor. It would seem reasonable that such a ball could be used as a source in preference to the polycarbonate rod earlier described because its design has been optimised.

Impacts due to running water. A garden tap is fixed to the sample wall on the source room side and water is supplied via a 19mm garden hose. The flow rate of approximately 12 litres/minute is regulated and measured. Details are provided by Debevc[4]. This method resembles conditions in practice where plumbing noise impacts an adjoining partition of a building. For this method only one tap location is used. The resulting receiving room sound pressure levels are measured at three locations in the reverberant field and normalised using eqn. (1). The one-third octave band levels are measured in centre frequency range of 100 to 5000 Hz.

2.6 Effects of flanking transmission

In measuring the sound transmission loss, clause 2.7(b) of AS 1191 [2] requires the edge conditions of a test sample to resemble those of an actual installation. Since it is beneficial to incorporate this test with the work on impact insulation the sample remains in the same condition. However when impacts are made on the wall it is probable that some impact energy incident on the wall will in part be transferred to the steel aperture and thence to the wall of the source room. This will in turn exert some influence on the sound pressure levels present in the receiving room. This part-bypass may be prevented by use at the sample perimeter of some isolating material but this then will contravene the requirements in sound transmission loss determination. Some detective work was carried out to see whether this possible flanking exists but is not a detailed study:

(a) *Using tapping machine:* Following normal measures the steel impact plate was moved slightly from contact with the sample and measures repeated. These show that significant energy is entering the receiving room which has not originated solely from the partition; this is more noticeable above 800 Hz.

(b) *Using impact rod:* After normal measures, the polycarbonate rod was moved so that the impact was on a wall of the source room some 3 metres from the sample wall. Measures of non-normalised levels in the receiving room close to the sample partition show that above 160 Hz there were no significant differences from when the rod made contact with the sample wall. This is quite a staggering discovery because it appears to negate any attempt to measure impact insulation of a wall.

Should further investigation reveal that flanking transmission is restricted to the sample perimeter then it effectively means any determination of impact insulation without rigid fixture to the aperture may not later be used to determine airborne sound transmission loss. This coupling of a test sample to surrounding structures is pointed out by Craik[9] who takes special care to see that such coupling errors are minimised.

3. INTERPRETATION OF THE CODE

3.1 Sample performance requirements

In F5.5 of the Code a wall is required to possess a stated single-number rating for airborne sound transmission. However for impact insulation it is required to be no less resistant to the transmission of impact sound than a wall listed in Table F5.5. The shortcomings of the Code are three-fold:

(a) The frequency range of investigation is not stated. Even though airborne transmission deals with a range 100 to 5000 Hz it does not necessarily follow that the appropriate range for impact testing be the same. It is necessary to devise a frequency range after studying the subjective effect on occupants. Current thoughts by ISO suggest an extension of the range down to 50 Hz.

(b) No information is provided of the likely normalised sound pressure levels of each of the Table F5.5 walls when tested by the method of No.3 of Specification F5.5. Nevertheless, this presents no real difficulties, so long as further shortcomings are addressed, since the laboratory is required to compare the results with those for a sample with a Table wall.

(c) Short of any clarification, currently a comparison of two walls means a sample fails the Code if its normalised sound pressure level exceeds that of a Table wall at any centre frequency. This is clearly a ridiculous state of affairs in resolving the practical performance of a given partition.

3.2 Possible single-number value of impact insulation

The above Code shortcomings could be overcome if the impact insulating properties of partitions could be expressed by a single-number value. The essential requirement of such a rating is that it addresses the subjective response of an occupant in the receiving room. This is of course contingent on each of the Table F5.5 walls providing satisfactory impact insulation. Initially, one should review some possible known methods of rating impact insulation.

Impact insulation class, IIC. The means of deriving this rating is set out in ASTM E989 [10]. It is based on use of the tapping machine but in direct contact as distinct from the Code indirect contact method via the steel plate. It is specifically applicable to the comparison of floor assemblies and is not designed to simulate any one type of impact. This is clearly then not suitable or applicable to impact on walls.

Normalised impact sound pressure level, L'_n or standardised impact sound pressure level, L'_{nT}. These values are derived as set out in ISO/DIS 717-2.2 [11] and measured using a tapping machine. However this standard is intended for field and not laboratory measurements and is intended just for floor assemblies. This is therefore an unsatisfactory rating for the Code.

Bodlund's index, I_s The above ratings compare normalised levels with a shifted reference curve whose values vary with frequency from 100 to 3150 Hz. For walls in practice this is considered to be not the most appropriate one because of the different exciting mechanisms impacting the walls. In addition such impacts are commonly single events rather than the repetitive one imposed by a tapping machine. In an attempt to resolve these differences Bodlund [12] studied possible variations in the shape of the reference curve also the frequency range. His work is based on work both on floors and on party walls.

Bodlund derived a modified reference curve to yield a new single-value rating I_s which yields better correlation with his judged subjective ratings of intrusive impact sounds. The important differences for this index are the appropriate frequency range for measurements namely 50 to 1000 Hz, as well as the shape of the curve. Following necessary investigations it might then be beneficial to adopt a single-value rating for impact insulation. It would be necessary also to measure the impact behaviour of the three Table F5.5 walls along with any subjective effects of the intrusive impact noise. This of course still leaves the questions of whether the tapping machine is the most suitable impactor and of the appropriate measures to be made in the receiving room.

When an alternative reference curve rating method has been derived it may require alternative shape and rules depending on the means used to impact a wall. Such means will depend on a building code identifying the most significant type of intrusive impact sounds likely to cause a loss of amenity to occupants of a room.

4. ALTERNATIVE TYPES OF IMPACT SOUND

When the most suitable single-value rating for impact insulation has been established, it now requires a method or methods by means of which a wall may be impacted. It is also necessary to decide on the most appropriate measures in the receiving room. It is felt essential that the method for measuring impact insulation be laboratory-based if it is the intention of the Code to have a sample partition deemed-to-satisfy the performance of a Table F5.5 type. This comparison of partitions will not be possible if field measurements are permitted.

As discussed earlier, possibly the most serious type of intrusive impact noise is due to single events. Accordingly it is more appropriate to use a single-impact source. Such single impacts feature in a Japanese industrial standard [13] even though it is a field method designed for floors. From this standard a working group has been set up, a round robin conducted, and an interim report produced [14]. The basis of the source is a rubber ball to produce a single impact by dropping and the consequent ball design allows standardisation.

In order to utilise the Japanese method as a means of producing a single impact on walls, it would be possible for a suspended ball to describe an arc before striking a partition as discussed in section 2.5 of this paper. By this means the actual impact energy may be specified in terms of the arc travelled and height fallen of the ball.

5. SUBJECTIVE EFFECT IN RECEIVING ROOM

When one first reads the Code, the impression is gained that in terms of impact insulation there is a benchmark of three magical walls which will provide adequate protection for the occupants in an adjoining room from the large range of possible impact noises. All sample partitions are then divided into "pass" or deemed-to-satisfy and "fail" when compared with this benchmark. Unless guidance is provided concerning the method of comparison and the actual measured quantities the whole aim of giving acceptable standards of amenity is doomed to failure. Short of Code guidelines the acceptance or rejection of a partition could well be based largely on the personal opinion of a measuring laboratory together with a few measurements, meaningful or meaningless.

5.1 Which quantity to measure in receiving room?

By including a tapping machine in testing methods it is probable the Code is assuming the use of normalised sound pressure levels in the adjoining room. There are various possible sound levels such as peak level dB, dB(A), dB(C), long-term average dB, dB(A), dB(C), sound exposure levels etc. As pointed out by Akay [15] the human auditory system has difficulty in accurately judging impact noises by just comparing the loudness with steady state levels. This then seems to condemn the Code practice of just measuring normalised sound pressure levels.

Each and all of the above types have appeared in work by various authors over many years. For instance in the investigation of a ratings system for impact noise transmission Bowles has suggested [16] the use of the C-weighting level for large-amplitude responses having dominant low-frequency components. Schultz [6] discusses the various types of level measures and that is likely to achieve a better correlation with the subjective assessment if the impact used is a real-life one. This then appears to support replacement of the tapping machine by some mechanism as discussed in section 2.5 of this paper. Kumagai [17] has investigated various types of single-impact noises and states that influencing parameters are the peak level, decay and rise times of the impact energy. Even though support is given to the measure of peak levels due to impact, the influence of the time-response mechanism appears to add complication to the measurement process.

5.2 Impact noise generation mechanisms

It is necessary to better understand the impact noise generation mechanisms if the most appropriate measurement system is to be chosen. Considerable work has been carried out over a number of years into these mechanisms. However this work has largely been confined to the subjective effect due to sonic booms, aircraft noise, drop-forge operation and not to single impacts on interior building partitions with adjoining rooms. There has been general agreement on the importance for impact noise nature of the energy level the impact rise-time and decay-time. Studies however have concluded that the greatest influencing factor has been the significant variations between the people in matching the loudnesses of two impact sounds.

For the response to impulse noise, Schomer [18] discusses some objective measures though work appears restricted to external sources. A particular quantity discussed is the sound exposure level, SEL where

$$\mathrm{SEL} = 10 \log_{10} [1/p_{ref}^2 t_0^2 \int p^2 (t)\, dt] \qquad (3)$$

where p_{ref} is reference pressure of 20 μPa, t_0 = 1s and $p(t)$ is a time-varying sound pressure, A- or C-weighted. This term has been chosen by the Environmental Protection Agency of USA as a better indicator of annoyance estimates. Schomer concludes that the most appropriate measure is the C-weighted SEL for impulse noises external to the building. In fact its measurement is possible by incorporation in a sound level meter. However it has been pointed out by others that an ordinary sound-level meter does not incorporate an adequate time constant to address the impact rise time. Schultz [6] has suggested instead a measurement of peak impact levels using a time constant of 35 milliseconds. Considerable work has been carried out by Carter [19] on the auditory response to impact noise and its controlling factors, in addition to considerable other studies reported by him. It would be beneficial to develop and further such studies in an attempt to decide on the most appropriate quantity for measurement in the receiving room.

6. CONCLUSIONS

This paper illustrates the enormous problems facing the Australian Building Codes Board in the production of a practical and useful code for the measurement of impact insulation of building partitions. It is considered that the present status of the Code relegates it to the guide classification and not to a Code. Perhaps difficulties have emerged because, to a large extent, the question of impact insulation of walls in a quantitative sense has generally not emerged as a problem in other parts of the world, so that there has not been pressure for International Standards that can be applied.

The Code in its present form is not likely to point those measuring impact insulation in the same direction so naturally they will probably produce significantly different results. The use of the tapping machine has been shown to be inappropriate and it is recommended it be replaced by a more suitable type of single impact. To achieve some degree of uniformity nationally in the results, it is felt advisable to require some stated precision in the measured response in the receiving room from which a confident Code can be derived.

A most serious problem present is that a measurement facility for airborne sound transmission is probably not one suitable for impact insulation determination with the same sample mounting. Any duplication of sample mounting to dismiss flanking would of course complicate the laboratory determination in an economic sense. It is essential that the transfer of impact energy to the receiving room structure via the partition is minimised. Such flanking has been shown to be significant when a partition has been installed primarily to measure airborne sound transmission.

In the event that the difficulties of measurement have been addressed by the inclusion of more stringent operating procedures with achievement goals there remains the problem of interpretation of results. The introduction of a single-value rating of impact insulation would not only be in harmony with the single-value STC for airborne sound transmission but would greatly simplify the question of meeting or not meeting the Code. It has been shown that some ratings are inappropriate for impacts on walls and their relevance can be influenced by the nature of the impact. Before selection of this single-value rating it is necessary to investigate the subjective effect of impact sounds on room occupants. In conjunction with this work the three walls of Code Table F5.5 should be checked to ensure that they satisfy criteria for impact-insulating partitions. It should be a task of such a study to recommend the rating that truly reflects the effect of that type of impact considered to be most intrusive upon room occupants.

REFERENCES

1. *Building Code of Australia* 1990.
2. AS 1191 Acoustics - *Method for laboratory measurement of airborne sound transmission loss of building partitions.*
3. ISO/DIS 140-6:1996, *Measure of sound insulation in buildings and of building elements - Part 6. Laboratory measurements of impact sound insulation of floors.*
4. M Debevc, "Measurement of impact insulation," *Aust. Acoust. Soc. Conf.*, Brisbane Nov 13-15, 1996.
5. R.J.M. Craik, "The measurement of structure-borne sound transmission using impulsive sources," *App. Acoust.*, **15**, 355-361 (1982).
6. T.J.Schultz, "Alternative method for evaluating impact noise," *J. Acoust. Soc. Am.*, **60**, 645-655 (1976).
7. H.Tanaka, H.Tachibana, M. Koyasu, *Interim report of the round robin on heavy and soft floor impact sources performed in Japan*, ISO Working Group.
8. Japanese Industrial Standard JIS A 1418-1978, *Method for field measurement of floor impact sound level.*
9. R.J.M. Craik and P.J. Berry, "The internal damping of building materials," *Appl. Acoust.* **35**, 139-148 (1992).
10. ASTM E989-89, *Determination of impact insulation class (IIC)*
11. ISO/DIS 717-2.2, *Rating of sound insulation in buildings and of building elements - Part 2: impact sound insulation.*
12. K. Bodlund, "Alternative reference curves for evaluation of the impact sound insulation between dwellings," *J. Sound Vibr.* **102**, 381-402 (1985).
13. Japanese Industrial Standard JIS A 1418-1978, *Method for field measurement of floor impact sound level*, 1978
14. H. Tanaka, H. Tachibana and M. Koyasu, *Interim report of the round robin test on heavy and soft floor impact sources performed in Japan.*
15. A. Akay, "A review of impact noise," *J. Acoust. Soc. Am.* **64**, 977-987 (1978).
16. S. Bowles, *Impact noise transmission through wall systems*,Thesis, RMIT, (1983).
17. M. Kumagai, M. Ebata, and T. Sone, "Effect of some physical parameters of impact sound on its loudness (a study on the loudness of impact sound) I," *J. Acoust. Soc. Japan (E)* **2**, 15-26 (1981).
18. P.D. Schomer, "Evaluation of C-weighted Ldn for assessment of impulse noise," *J. Acoust. Soc. Am.* **62**, 386-399 (1977).
19. N.L. Carter, "Effect of rise time and repetition rate on the loudness of acoustic transients," *J. Sound Vibr.* **21**, 227-239 (1972).

Design of a Test Facility for Vibration Isolator Characterisation

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ABSTRACT: Vibration isolators are an important element in the reduction of structure-borne noise transmission. The dynamic properties need to be determined at the pre-load and over the frequency range experienced in normal operation. The four-pole parameters description of the isolator dynamic properties is independent of the testing arrangement. A test facility has been designed to measure the four-pole parameters of vibration isolators with pre-loads up to 30 kN and over a frequency range from 10 Hz to 2000 Hz. This paper describes the dynamic design of the test facility using modal analysis and harmonic response analysis.

1. INTRODUCTION

The vibratory power transmitted from machinery mounted on isolators depends upon the dynamic properties of the foundation, the machinery mounting point and the vibration isolation mounts. A knowledge of the frequency dependent dynamic properties of vibration isolators is necessary in order to be able to predict their isolation performance. In particular for naval surface ships and submarines determination of the vibration characteristics is vital for acoustic signature management.

Most isolators contain elastomeric material, the stiffness and damping properties of which are both frequency and pre-load dependent. In addition it is possible for standing waves to be set up within the isolator at certain frequencies, which greatly reduce its effectiveness at those frequencies. It is therefore necessary to determine the isolator properties under the pre-load conditions and frequency range experienced in normal operation.

The transfer impedance or mobility of an isolator has traditionally been used to describe its dynamic properties and provide a measure of its isolation performance. This measure does not necessarily provide a full description and may also be dependent upon the test conditions. An alternative description is provided by the four-pole parameters, which relate the force and velocity above the isolator to the force and velocity below. This parameter set is particularly adapted to analysing composite systems.

The significance of the four-pole parameter description is that it provides a measure of the isolator dynamic properties that is not dependent upon the measuring set-up, and it can be used to give an estimate of the isolator's effectiveness.

A vibration isolator test facility was developed at the Aeronautical and Maritime Research Laboratory to measure the frequency dependent vibration transmission characteristics of flexible isolation mounts used for machinery on-board ships and submarines. The test facility has been designed to measure the four-pole parameters at pre-loads of up to 30 kN and over the frequency range from 10 Hz to 2000 Hz. In undertaking the experimental measurements the force and velocity above and below the isolator need to be determined, and it is vital that the dynamics of the testing machine structure do not affect the results.

This paper describes the modelling of the modal behaviour of the elements of the test rig and the use of harmonic response analysis modelling to ensure there is no effect on the experimental measurements from the structural modes of the test rig. The results show that while the structural framework and other elements of the test rig have natural modes within the frequency range of interest, careful design can ensure that these do not affect the results.

2. FOUR-POLE PARAMETERS

The four-pole parameters description of the dynamic properties of a vibration isolator relates the forces F_1 and velocity V_1 at the isolator's input to the force F_2 and velocity V_2 at the isolator's output, [1, 2, 3] is

$$\begin{bmatrix} F_1 \\ V_1 \end{bmatrix} = \begin{bmatrix} \alpha_{11} & \alpha_{12} \\ \alpha_{21} & \alpha_{22} \end{bmatrix} \begin{bmatrix} F_2 \\ V_2 \end{bmatrix} \quad (1)$$

where α_{11}, α_{12}, α_{21}, and α_{22} are the four-pole parameters, and are complex, time-invariant functions of the angular frequency ω.

Application of Maxwell's law of reciprocal deflections to the isolator leads to the relationship:

$$\alpha_{11}\alpha_{22} - \alpha_{12}\alpha_{21} = 1 \quad (2)$$

A symmetric isolator is one that behaves the same if the input and output ports are interchanged. For this case an additional relation is applicable:

$$\alpha_{11}=\alpha_{22} \quad (3)$$

From equations (2) and (3) it is evident that, for a symmetric isolator, only two independent four-pole parameters need to be measured in order to completely characterise it. At lower frequencies an isolator may be assumed to be a massless spring of dynamic stiffness k. This assumption yields $\alpha_{11}=\alpha_{22}=1$, $\alpha_{12}=0$ and $\alpha_{21}=j\omega/k$, where $j=\sqrt{-1}$ and ω is the circular frequency.

From equation (1) two particular cases can be derived. Case 1 is for the output to be free, $F_2=0$, which yields $F_1=\alpha_{12}V_2$ and $V_1=\alpha_{22}V_2$. Case 2 is for the output blocked, $V_2=0$, which yields $F_1=\alpha_{11}F_2$ and $V_1=\alpha_{21}F_2$. While the first case is experimentally convenient it does not allow the determination of the isolator properties under pre-load, and therefore the properties measured in this way will not be representative of those for the installed isolator.

Verheij [4] developed a method for determining the blocked transfer function $F_2/j\omega V_1$. Dickens and Norwood [5] developed a system that followed Verheij's basic method but determined the four-pole parameters. Improvements to the original measurement technique correcting for the small but finite velocity of the blocking mass, and measuring the force directly were suggested by Dickens and Norwood, [6, 7]. It was decided to implement these measurement improvements in an upgraded test rig. Additional improvements would also include increasing the upper frequency limit to 2 kHz and the dynamic force capacity to 5.3 kN.

3. TEST RIG

The developed test rig is shown schematically in Figure 1. The test isolator under test is mounted between two large masses; static pre-load is applied by air-bags top and bottom and the dynamic load by an electro-dynamic shaker. The rig has two supporting frames, the upper frame that supports the shaker and the lower frame that provides the reaction forces for the upper pre-loading air-bags. The lower pre-loading air-bags sit on a base plate mounted on top of a seismic mass.

Two frames are used to reduce coupling between the pre-loading structure and the shaker. The shaker is decoupled from its supporting frame by four isolation hangers and drives the excitation mass through a single centrally located connecting rod. The seismic mass is a block of reinforced concrete of dimensions 3m × 3m × 1m, supported on four air-bags connected to air reservoirs and having a mounted natural frequency of approximately 1.2 Hz. The use of the seismic mass decouples the blocking mass from the laboratory floor, reducing the input of extraneous forces and transmissions from the two supporting frames.

Air-bags are used to provide the pre-load as the static force can be easily adjusted, while at the same time giving a degree of isolation between the masses and the supporting structure. The dynamic force between the excitation mass and the isolator is measured directly by an assembly of force transducers. The motions of the excitation and blocking masses are measured using accelerometers.

The rig was required to be able to test isolators with

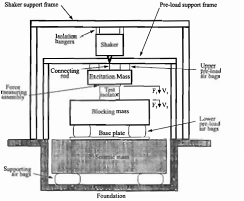

Figure 1. Schematic of Vibration Isolation Test Rig.

dynamic stifffnesses in the range from 1×10^5 to 2×10^7 N/m, with pre-loads adjustable over the range from 1.5 to 30 kN and over a frequency range from 10 to 2000 Hz. In designing the test rig it was important that the dynamic response of the test rig did not affect the measurement of the isolator's four-pole parameters. This implies that where possible the components of the test rig should not have structural modes within or near the frequency band of interest, and where this is not possible the structural response of the rig should not affect the results.

4. DYNAMIC ANALYSIS OF THE TEST RIG

The analysis is divided into three parts. Firstly modal analyses were conducted to determine the natural frequencies and mode shapes of the test rig components and sub-components. Where possible, components were designed not to have structural modes within or near the test frequency band; these included the excitation mass, the blocking mass, the table extension for the shaker, the force measuring assembly and the base plate for the seismic mass.

For components that could not be designed to satisfy the modal frequency criterion, a harmonic response analysis was performed. The effect of the modal behaviour of the various components on the measurements was determined. Items in this group included the supporting frames and the seismic mass.

Finally a harmonic response analysis for the entire assembly was performed and compared with the response from an idealised spring/mass system. This was done to ascertain if there were any effects from the modal behaviour of the individual test rig components on the assembly as a whole.

The ANSYS finite element analysis program was used in the analyses.

4.1 Modal Analyses

In this series of analyses the natural frequencies and mode shapes of the excitation mass, blocking mass, shaker table extension and the air bag base plate were determined to ensure they did not occur within the frequency range of interest. The components were modelled using eight noded brick elements, and where necessary, the parts of the test rig in contact with the component under study were included in the model to ensure appropriate boundary conditions.

(a) *Blocking mass:* The model included the supporting air-bags and the isolator. Two extreme case were considered, the stiffest isolator under the maximum pre-load and the softest isolator under the minimum pre-load. The air-bags and the isolator were modelled as springs spread over the contact areas. The optimum design selected for the blocking mass was a steel cylinder of diameter 480 mm and height 398 mm, which gave a first modal frequency of 4.00 kHz for the (1,0) mode. To apply the two mass method described by Dickens and Norwood [2] for testing non-symmetric isolators, two different blocking masses are required. A second blocking mass made from an aluminium alloy was analysed to ensure its modal properties were adequate. It had the same height but a smaller diameter compared to the first mass, and slightly higher predicted modal frequencies.

(b) *Excitation mass:* This model included the pre-loading air-bags and the isolator. As for the blocking mass the air-bags and isolator were modelled as spring elements, and the two extreme cases described above were analysed. Similarly to the blocking mass, the optimum shape selected was a cylinder; in this case it had a diameter of 360 mm and a height of 355 mm, with the first natural frequency of 4.58 kHz. The first mode was torsional, and the second mode was the (1,0) mode at 5.66 kHz. To accommodate low pre-loading forces down to 1.5 kN, a second excitation mass having a lower mass was required. The second mass had the same dimensions as the first excitation mass but was made from an aluminium alloy, and gave marginally higher predicted modal frequencies.

(c) *Table extension:* In order to limit the length of the rod connecting the shaker's table to the excitation mass, it was necessary to provide an extension to the shaker table. This would be bolted to the shaker table in six locations and was required to have a minimum height of 175 mm. The extension was made of aluminium to minimise its mass. The optimum design selected was a base cylinder of diameter 125 mm leading into a cone and tapering to a 20 mm diameter cylinder at the top. This had a first mode at 4.26 kHz.

(d) *Base plate:* A base plate attached to the seismic mass was to be used to locate the six lower pre-loading air-bags under the blocking mass. The diameter of the plate was set at 480 mm diameter, which was the minimum required to support the air-bags. The plate was initially modelled as being attached to the seismic mass by 15 bolts on three concentric circles. The optimum solution for this arrangement was a steel plate with thickness of 45 mm giving a first mode at 3.57 kHz. One of the problems to be resolved with this model was that the mode shape would be rectified. The plate could deflect away from the mass only. Unfortunately this one-way motion was not able to be modelled, so the mode shapes were calculated as for normal bi-directional deflection. It was reasoned that the uni-directional motion constraint would increase the modal frequency so the results calculated would be conservative.

A second model using adhesive to attach the plate to the seismic mass was made. This consisted of a plate 40 mm thick and three locating bolts, with a continuous layer of epoxy adhesive between the plate and the mass. The adhesive layer was modelled as a continuous compliant layer and the first structural frequency was 11.86 kHz. This design was selected in the construction of the test rig.

4.2 Harmonic Response Analyses

The harmonic responses of the two support frames and the seismic mass were modelled to ensure that the modes which occur within the desired test frequency range did not affect the results. The two frames were modelled using three dimensional beam elements, and the seismic mass was modelled using eight noded brick elements. Initially the modal analysis was performed and then the harmonic response analysis was performed using modal superposition, with clustering at the modal frequencies to give better resolution of the expected peak responses. The forcing function modelled the input from the shaker, and each analysis was carried out using several different damping conditions to assess the importance of damping to the peak responses.

(a) *Shaker support frame:* The model of the frame included the columns, cross beams and braces for the frame itself and the isolation hangers on which the shaker was supported. The shaker was modelled as a lumped mass and the input force was applied to it. The response analysis was designed to determine the level of force transmitted to the floor by the frame, and hence the amount of feedback to the air-bags supporting the blocking mass through the seismic mass. The reaction forces at the base of the columns were calculated, and multiplied by the measured transfer function between the mounting points for the columns on the floor and the top of the seismic block to give displacement levels on the top of the seismic block.

The displacements predicted by this method were at least 120 dB less than those predicted on the seismic block via the direct path through the isolator and the blocking mass over the frequency range of interest. Therefore it was concluded that feedback via the shaker supporting frame was not a problem.

(b) *Pre-load support frame:* The pre-load support frame consisted of a pair of portal frames cross connected by an "H" structure which carried the pre-loading air-bags. The model included the upper and lower pre-loading air-bags, the excitation mass, isolator, blocking mass, seismic mass and supporting air-bags. The air-bags and isolator were modelled as spring elements and the masses were modelled as rigid mass elements. The system was excited by applying an harmonic force to the excitation mass. The analysis was intended to predict the errors caused by the feedback of the reaction forces at the base of the supporting frame columns via the laboratory floor and the seismic mass' base.

The calculated reaction forces were combined with measured transfer functions to predict the displacements on the top of the seismic mass. Over the frequency range of interest it was found that the predicted displacement peaks due to the feedback path were at least 90 dB below the amplitude of those due to the forward path via the blocking mass. This shows that the pre-load support frame is more important then the shaker support frame as a potential error path. However the levels predicted are so far below the displacement levels via the forward path that errors due to feedback through the frame will not be significant.

(c) *Seismic mass:* A modal analysis of the seismic mass showed that it had a considerable number of modes below 2 kHz, so that the important consideration would be that these modes did not affect the results. It is also possible for the seismic mass to be excited by the forward path forces through the pre-loading air-bags under the blocking mass, and these motions could affect the results. Therefore a model of the seismic mass, excitation mass, base plate, blocking mass, adhesive layer, locating bolts, air-bags and isolator was made. The seismic mass, base plate and blocking mass were modelled using eight-noded brick elements and the excitation mass was modelled as a lumped mass, while the air-bags and the isolator were idealised as spring elements.

This was compared to an idealised system where the seismic mass modelled as a rigid mass element and its supporting air-bags combined together. In addition, the bottom of the adhesive layer in contact with the seismic mass was constrained to move rigidly in sympathy with the seismic mass. The other elements in the model remained the same as above.

The displacements on the blocking mass and the contact forces between the blocking mass and the isolator for the actual system were compared for the two models. For the two models the displacements and forces differed by less than 0.03% and 0.01% respectively, indicating that the modal behaviour of the seismic mass has an insignificant effect on both the predicted displacements and forces.

5. MODELLING AND ANALYSIS FOR COMPLETE SYSTEM

The system was idealised as a series of springs and rigid masses to determine the system frequencies and establish if these modes would affect the measurements, Figure 2. The model didn't include the supporting frames, so the stiffness elements attached to the frames were assumed to be fixed at these points.

The isolation hangers are made up of multi-layer rubber isolation elements and steel springs and are represented by the springs k_1 and k_2 and the mass m_1. The shaker is made up of the trunnion, body, driving magnetic coil and extension table and is represented by and the masses m_2, m_3, m_4 and m_5 and by the springs k_3, k_4 and k_5. The force F_{in} produced by the shaker is generated between the body and the driving magnetic coil.

The shaker drives the excitation mass through a connecting rod, represented by the spring k_6, while the upper pre-loading air-bags are modelled by k_7. The excitation mass including the top mass of the force measuring assembly, the mass of the end supports for the pre-loading air-bags and one half of the air bag mass is represented by m_6. The stiffness of the force measuring assembly is modelled by the spring k_8.

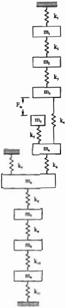

Figure 2. Spring-mass respresentation of Test Rig.

The isolator under test is considered to comprise upper and lower plates separated by an elastomeric element represented by the stiffness k_9. The mass m_7 includes the upper plate, the bottom plate of the force measuring assembly plus half the mass of the elastomer. The blocking mass, isolator lower plate, half of the elastomer mass, the masses of the end supports for the air-bags and half of the masses of the supporting air-bags are represented by mass m_8. The stiffness of the air-bags is modelled by k_{10}. The mass m_9 represents the seismic mass, base plate, mass of the end supports for the air-bags and half the mass of the supporting air-bags. The spring k_{11} represents the stiffness of the air-bags that support the seismic mass.

Two limiting cases need to be considered. The softest case, comprising the softest expected isolator under minimum pre-load with the lighter excitation mass; and the stiffest case, comprising the stiffest expected isolator under maximum pre-load and with the heavier excitation mass. It is not possible to know a priori the masses of all the isolators to be tested, representative minimum and maximum masses were selected from isolators already tested.

The system as modelled has nine degrees of freedom and hence nine modal frequencies. The modal frequencies were solved for using MATLAB software. Masses and stiffnesses for the two limiting cases investigated are given in Tables 1 and 2 and the modal frequencies are given in Table 3.

Modes 4, 5, 6, 7 and 8 fall near or within the desired frequency range of measurement from 10 Hz to 2 kHz. The fourth mode has the table, coil, excitation mass and the top plate of the isolator in-phase with each other, but out-of-phase with the isolator's bottom plate and blocking mass. The stiffness of the isolator has a major influence on the modal frequency f_4. The hanger exhibits the dominant motion in the fifth mode, and is out-of-phase with the trunnion. The frequency f_5 is determined by the stiffnesses of the rubber and steel spring elements, and thus is the same for both cases. In the sixth mode, the table and coil are in-phase with each other and out-of-phase with the excitation mass. The frequency f_6 is predominantly determined by the stiffness k_6 of the connecting rod. In the seventh mode, the shaker's trunnion is out-of-phase with its body and the frequency f_7 critically depends on the stiffness of the toroidal elastomeric isolators, and remains the same for both cases. The table and coil are out-of-phase with each other in the eighth mode, and the frequency f_8 is critically dependant upon the stiffness k_5, and so is equal for both cases.

Table 1
Masses

Mass (kg)	m_1	m_2	m_3	m_4	m_5	m_6	m_7	m_8	m_9
Softest case	16.6	41.1	607	4.34	6.38	109	1.65	572	22200
Stiffest case	16.6	41.1	607	4.34	6.38	292	10.3	581	22200

Table 2
Stiffnesses

Stiffness (N/m)	k_1	k_2	k_3	k_4	k_5	k_6	k_7	k_8	k_9	k_{10}	k_{11}
Softest case	1.57×10^7	1.63×10^5	2.65×10^9	1.37×10^5	5.49×10^8	6.77×10^7	2.62×10^4	2.00×10^9	1.00×10^5	2.31×10^5	2.87×10^6
Stiffest case	1.57×10^7	1.63×10^5	2.65×10^9	1.37×10^5	5.49×10^8	6.77×10^7	4.37×10^5	3.00×10^9	2.00×10^7	7.80×10^5	2.87×10^6

Table 3
Natural frequencies

Frequency (Hz)	f_1	f_2	f_3	f_4	f_5	f_6	f_7	f_8	f_9
Softest case	1.80	2.72	3.74	7.75	156	410	1320	2370	5580
Stiffest case	1.88	3.31	6.30	50.2	156	399	1320	2370	2770

From the work by Dickens and Norwood [3] and since the direct force at the input of the sample isolator is being measured, modes 4, 5, 6, 7 and 8 will not adversely affect the test measurements. The sixth and eighth modes dominate the behaviour of the excitation mass, which shows maximum acceleration levels at the frequencies f_6 and f_8. When testing at or near these frequencies, care must be exercised to prevent excessive acceleration and force amplitudes of the table and coil. The upper frequency limit of the measurements is determined by the modal behaviour of the assembly of force transducers used to measure the input force, given by the ninth mode. In this mode the excitation mass and the top mass of the assembly are in-phase with each other but out of phase with the bottom mass of the assembly and the top plate of the isolator. The force transducer assembly measures the direct forces satisfactorily until it exhibits modal behaviour, and so the upper limit is determined by f_9, which predominantly depends upon the axial stiffnesses of the force transducers and the masses of the isolator's top end plate and elastomer. Therefore the upper frequency limits of the measurements for the softest and stiffest cases are 5.58 and 2.77 kHz, respectively.

Dickens and Norwood [3] derived an expression for the lower frequency limit of the measurements using direct forces. This limit is governed by the square root of the sum of the stiffnesses of the isolator and the lower pre-loading air-bags divided by the sum of the masses of the blocking mass and the isolator's lower plate. For the softest and stiffest cases they equate to 3.83 and 30.1 Hz, respectively.

Allowing factors of approximately two for the lower, and one half for the upper frequency limits gives practical testing frequency ranges for the softest and stiffest cases of 10 Hz to 2.7 kHz, and 60 Hz to 1.3 kHz, respectively. Because the modal behaviour of the individual components of the system has been designed for a maximum frequency of 2.00 kHz, the resulting system frequency ranges for the softest and stiffest cases are 10 Hz to 2.0 kHz, and 60 Hz to 1.3 kHz, respectively.

6. CONCLUDING REMARKS

A test facility has been designed to measure the four-pole parameters of vibration isolators with pre-loads of up to 30 kN. The frequency range over which the isolators can be tested is not limited by the structure and construction of the test rig. The lower frequency is governed by the stiffness of the isolator and the lower pre-load air-bags, and the masses of the blocking mass and the isolator's lower plate, and not the structure of the test rig. The only components of the test facility with modal behaviour in the frequency range of interest are the two frames and the seismic mass. The flanking path transmission from the isolator input to the output through the frames, the ground and the seismic mass is at least 90 dB lower than the direct path excitation level. Therefore the modal behaviour of the frames will not influence the test results. The effect of the modal behaviour of the seismic mass on the measured isolator's velocities and forces is predicted to be less than 0.03% and 0.01% respectively, and so can be considered negligible.

The test rig will be used to measure the four-pole parameters of isolators used to control the structure-borne noise transmission in ships and submarines. Using the four-pole parameters and the dynamic properties of the structure above and below the isolator, the effectiveness of the isolator can be determined. An isolator's performance will change over time and with exposure to environmental factors such as oil and heat. The test rig will be used to measure the four-pole parameters for new and aged isolators to determine the degradation and the allowable life between refit. The performance of purpose designed isolators can be determined with the test rig providing a more comprehensive measure of the isolators performance than quality control tests now used.

REFERENCES

1. C.T. Molloy, "Use of four-pole parameters in vibration calculations," *J. Acoust. Soc. Am.* **29**, 842-853 (1957).
2. J.C. Snowdon, "Vibration isolation: Use and characterisation," *J. Acoust. Soc. Am.* **66**, 1245-1274 (1979).
3. J.D. Dickens, C.J. Norwood and R.G. Juniper, "Four pole parameter characterisation of isolator acoustic transmission performance," *Proc. Aust. Acoust. Soc. Conf.*, Glenelg, SA, 1993. pp. 110-117.
4. J.W. Verheij, *Multi-path sound transfer from resiliently mounted shipboard machinery,* Ph.D. thesis, Institute of Applied Physics TNO-TH, Delft, (1982).
5. J.D. Dickens and C.J. Norwood, Vibration isolator test facility, *Defence Science and Technology Organisation, Aeronautical and Maritime Research Laboratory,* report DSTO-TR-0357, (1996).
6. J.D. Dickens and C.J. Norwood, Measurement of four-pole parameters using blocking masses, *Proc. Aust. Acoust. Soc. Conf.*, Canberra, ACT, 1994. pp. 57-65.
7. J.D. Dickens and C.J. Norwood, Measurement of four-pole parameters using direct forces, *Proc. Aust. Acoust. Soc. Conf.*, Fremantle, WA, 1995. pp. 15-21.

Acoustic Design at RMIT University

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Abstract. The Australasian Soundscape Project in the Faculty of Environmental Design and Construction at RMIT University has introduced a new minor study in acoustic design for architecture and design students. The study seeks to emphasise the contribution of sound to the experience of space through a series of practical and theoretical design and analysis subjects.

The Faculty of Environmental Design and Construction (FEDC) at RMIT University has introduced a minor study in acoustic design. The study consists of three theory subjects and a design studio, and may be taken by students studying architecture, landscape architecture, interior design, and industrial design.

The three theory subjects are Constructing Sound, The Sound of Space and Acoustic Environments. Constructing Sound is an introduction to physical, physiological and psychoacoustic processes involved in the experience of sound, and is the only one of the theory subjects taught so far. The Sound of Space will explore sound in relation to architectural spaces and develop new methodologies for designing spaces for sound. Acoustic Environments will outline current cultural theory, legal responsibilities and planning implications relating to sound.

Design studios in sound have been run within the FEDC since 1993. These studios have focused on a range of areas such as class-room acoustics, the design and construction of acoustic installations for cultural events and soundscape analysis. Many of these studios have involved communities outside RMIT. For example, design proposals were developed for the renovation of a pre-school for hearing impaired children in Nathalia in northern Victoria, and a similar studio is currently underway for school rooms in the Aboriginal community at Yirrkala in Arnhem Land. Acoustic installations such as "Perfect Form", built around the De Kooning sculpture in the forecourt of the Victorian Arts Centre, were part of the Next Wave Youth Arts Festival.

The acoustic minor was designed by Robyn Lines, Neil McLachlan and Jonathan Mills. This team of people, with the addition of Peter Clark and Herb Jercher, is developing a program of activities at RMIT under the title of the Australasian Soundscape Project. This project was initiated by Jonathan Mills in 1993 when he joined the FEDC as the recipient of the Lady Beale Fellowship in the Acoustic Arts.

The Australasian Soundscape Project (ASP) seeks to raise the profile of acoustics as an essential part of the experience of space and integral to the practice of design. Its approach involves the generation of new acoustic understanding and practice through the synthesis of the generally discrete bodies of knowledge of architects, designers, acousticians, audiologists, composers, sound artists, cultural theorists and others. Specific outcomes include teaching programs, seminars, documentary and research publications, public exhibitions, installations and performances.

Members of the project have been involved in research on how architects and designers integrate acoustic knowledge into their practice, new educational strategies for acoustics within the design professions, musical acoustics and perception in non-western percussion ensemble music and the design and application to performance and installation of novel musical instrumentation.

To date the teaching program is the central concern of the ASP. It seeks to emphasise the contribution of sound to the experience of architectural spaces. Traditional approaches to acoustic education for
design professionals have resulted in a narrow conception of acoustics as noise control, or as an expensive, optional extra to the main concerns of architecture.

The neglect of sound in architecture may be traced to the dominance of graphical processes in design, which lead to a predominantly visual conceptualisation of architectural spaces. Quantitative acoustic evaluation of design concepts usually becomes relegated to (at best) limited remedies on pre-determined architectural forms.

The introductory subject to the acoustic minor, Constructing Sound, seeks to cultivate an ability in students to imagine the sound experience which would be generated by a given physical system. The vibrational behaviour, timbre, and sound resonating and radiating properties of reference sound sources such as air columns, metal plates, tubes and rods, electronic oscillators and loudspeakers, the human voice and musical instruments are described in a variety of ways. Simple physical and mathematical models such as springs and simple harmonic motion, visual and graphic representations including acoustic spectra and waveforms, musical conventions and textual descriptions of sound are all used. Basic bio- and psychoacoustic principles are also introduced.

Students are encouraged to develop an understanding of sound through the physical manipulation of these reference sources, and a capacity to describe and predict acoustic behaviour (in a general sense) by developing an ability to link knowledge gained in separate experiences of the various contributing phenomena to sound sensations.

The Sound of Space is currently being prepared for the first semester in 1997. It will include the extensive use of case studies where students will make detailed qualitative and quantitative analyses on a range of architectural spaces. These spaces will be modelled using a room acoustic modelling computer program linked to 3-D auto-CAD, and redesigned to achieve specific acoustic design outcomes. A desk top auralisation program will be used to assist students to make qualitative acoustic evaluations of their designs. Simple physical models such as ripple tanks and some basic mathematical concepts and systems of approximation will be used to introduce the principles of room acoustics.

The acoustic minor and other programs of the Australasian Soundscape Project are in the early stages of development and professional input is welcome. Interested people may contact the ASP:
Telephone (03) 9660 1926
Fax on (03) 9660 1820
e-mail mclachlan@fedcmac.edc.rmit.edu.au

AAS Membership Grading

On the Grading of Membership Applications by the Council Standing Committee on Membership (CSCM)

Ken Cook
Chairman of CSCM

This Article has been prepared to assist Grading Committees, new applicants, and existing members transferring their grade of membership, to complete the Society's Application Form.

Admission to the Society is open to people working in all fields of acoustics. There are seven grades of membership in the Society; Fellow, Member, Honorary member, Associate, Subscriber, Student and Sustaining member. More information about these grades of membership can be obtained from the Membership Information leaflet available from the General Secretary or the State Division Secretaries.

The completed Application Form and Entrance Fee should be submitted to the General Secretary, PO Box 4, TALLY HO VIC 3149. The entrance fee is currently $74.00 except in the case of applicants for Student grade when the fee is $20.00.

It should be noted that the Society's Articles of Association are presently being amended. This may result in minor changes to the Articles covering membership and the procedure for assessing applications. However, these changes are unlikely to take affect before 1998.

Enquiries regarding membership and sustaining membership should be directed to the General Secretary or the appropriate State Division Secretary (see back of journal for names and addresses).

More than one article in *Acoustics Australia* has provided guidelines for admission and grading of members. The "Gates" for grading have been an interpretation of Articles 16 and 18 of the Memorandum of Association. CSCM grades an application for membership, taking into account the details of the application form submitted and the recommendation by the referring division.

This article restricts discussion to just two grades of membership, namely Member and Associate, since these are where most of current difficulties arise for CSCM in seeking to support the referring division/applicant.

Remember, if CSCM is not able to establish from the application form the undisputed level of acoustics in understanding, expertise and experience, no grading decision will be made. This means disputed matters need to be pointed out to the referring division, likely causing long delays in eventual grading decisions.

The intention of this article is to provide further guidelines for intending applicants for admission to the Australian Acoustical Society, and for divisions who will forward their recommendations to CSCM. Divisions should be mindful that applicants in general will be seeking guidance in the provision of an application form which correctly follows the intention of the Society Articles.

There are perhaps the following important components in an the application for membership, all of which are brought out in the Application form:

1. Education

There is a distinction on the form between

(a) Formal qualifications, which are

(i) "recognised educational qualifications" in Gates MA and MB for Member grade,

(ii) not specifically mandatory in Gates MC and MD for Member grade.

The Application form requires applicants (whether applicant be for grade Member or Associate) to provide certified copies of qualifications. CSCM is prepared to accept a copy of the qualification(s) so long as it has been signed by a responsible member of the referring-division committee to certify that the original or certified copy has been sighted; and;

(b) Other studies at tertiary level

The application form provides space for applicants to state source title, course duration, stage reached, Institution or Organisation, and relevant dates. Such information could be relevant to applicants for grade Member or Associate.

2. Present position

As shown on the Application form, it is essential to state the proportion of time spent in acoustics. In some other statement on the form it is important to indicate the level of employment time in acoustics:

(a) For each of the gates for grade Member it is mandatory for the applicant to show engagement in the science or practice of acoustics at a professional level. Professional level refers in this context to the degree of difficulty, complexity, understanding, experience of the work in acoustics by the applicant.

(b) For each of the gates for grade Associate it is mandatory for the applicant to show engagement in the science or practice of acoustics at technician level. Technician level refers in this context tothe degree of competence of following established procedures for measurement, testing, calculating or similar.

3. Professional and technical experience

The statements to be included in the application should make it perfectly clear in the extent to which an applicant has adequately developed an understanding and ability to work in, and follow developments in their field of acoustics. It is from such information that CSCM will be in a position to judge both the level of activity/experience in acoustics and the equivalent period in each level of activity.

The requirement of the application is two-fold:

(a) The form needs to "Provide in chronological order a list of positions held and associated duties. Include details of major projects or activities". Note the additional need to state the proportion of time spent in acoustics and the Verification of the statements made. Applicants and division grading committees should recognise that administrative and management work are not recognised as work in acoustics; thus an applicant may typically have to work for 5 years with 40% involvement in acoustics at a professional level to achieve a total of 2 years experience in acoustics.

(b) A most important section for applicants in the grade of Member or Associate is sufficient evidence provided that the experience gained is at the appropriate level stated in the membership gate. It is in this area that CSCM has experienced the greatest difficulty in obtaining sufficient justification. Consequently, the words of the Application form are here shown in full:

"Applicants for the grade of Member or Associate must include details of involvement in acoustical aspects, e.g. in design, analysis, drawing, specification, reports, papers". What is needed is the extent of involvement and degree of difficulty of the acoustical content, and to make the picture clear, examples of work, photocopies of calculations, reports and similar.

Of particular relevance is the application for grade **Member, gate MD** where there is a lack of recognised educational qualifications. In addition, an applicant may currently be employed in a position which is deemed "non-professional", so circumstantially does not seem worthy of consideration of grade Member. On the other hand the applicant may well be involved (in toto, part time) in activities of a professional nature. In such a case the applicant must attempt to convince the Society that the grade Member is appropriate, via the medium of the Application form.

(c) Applicants for the grade of **Associate** are reminded of the need to provide complete details of involvement in *technical* aspects of acoustics. There are the two gates AA and AB for grade Associate, distinguished as follows:

(i) Gate AA the applicant must show evidence of satisfactory completion of an appropriate certificate course or other appropriate post- secondary qualification (refer to (a) above). Applicants must also show that they have been engaged in the science or practice of acoustics at technician level, and have been so for not less than 2 years

(ii) Gate AB - in the absence of approved technical qualifications it is essential for an applicant to demonstrate completely that they have been actively engaged in the science or practice of acoustics at technician level and have been so for not less than 5 years. Entry via this gate is not likely to be entertained unless the applicant's work history has been shown on the Application form in great detail.

For provision of the above required information, CSCM is mindful of two aspects:

A. The information is likely to be confidential/economically sensitive. Such information will be classified by CSCM as highly confidential. Following grading the applicant's form/attachments are closed and sent to Archives via Federal Registrar

B. Information may be part of a project for a client, so would be confidential/strictly not transferable apart from client. CSCM is quite prepared to accept field notes/project/reports etc. where client's name or project location does not appear. Remember that CSCM is

4. Verification

A separate column appears on the Application form under Professional and technical experience, for Verification. A verifying person must give careful consideration as the extent to which they are verifying any statement made by an applicant. Unless this is clearly set out, if necessary by an accompanying statement, there is every likelihood the weight given to the application by CSCM will be zero.

(i) It should be obvious that an applicant may rely heavily on the support of a verifier, especially in cases where the verifier is the direct supervisor and possibly in a position to support project work. This is particularly important in cases where a judgement is made as to whether the applicant's work has been professional, technically adequate or just laboratory assistant grade. Persons verifying claims in a membership application form would assist the applicant if they were to supply a letter advising how they are able to verify the claims, such as through supervision of the applicant and personal knowledge; this may avoid the appearance of verifiers relying on oral information from the applicant himself or herself.

(ii) Verification of proportion of time spent on acoustics, ie. active involvement in science or practice of acoustics at a professional level. Accordingly, employment time spent as managing director or director of a firm dealing in acoustics will contribute nothing in the eyes of CSCM in respect to satisfying time in acoustics referred to in the gates for Member admission.

(iii) Verification of professional and technical experience. Verification should be of the activities in acoustics/proportion of time spent therein, in preference to verification of employment period.

Associate

There are two gates for entry to Associate, so much the same attention to detail as for Member should be provided on the *Application form.*

Grading of applications by CSCM

As pointed out above, the assessment of an application is based on the information provided on the application form and the attachments forming part of this form.

(a) For a correctly-completed application, grading in general is straightforward since there is just the one set of rules.

(b) When the application form/attachments are incomplete, CSCM then makes judgements of the time spent in acoustics under the following columns:

Duration in an employment, months
Percent time in acoustics, as claimed
Claimed time in acoustics, years
Percentage time in acoustics assessed by CSCM
Time in acoustics assessed by CSCM, years

This last column is totalled, then compared with the requirement for a specific gate to judge whether the experience is sufficient. Note that this decision is made only on the condition that CSCM is convinced that the time in acoustics has been an involvement at a professional level (in the case of Member), or in the science or practice of acoustics at a technician level (in the case of Associate).

Books...

Theory of Vibration, An Introduction

A A Shabana

Springer Verlag, 1996, pp 347, Hard Cover ISBN 0 387 945245 Australian Distributor: DA Information Services, PO Box 163, Mitcham Vic 3132. Tel 03 9873 4411 Fax 03 9873 5679. Price A$89.75.

The book's stated aim is to present the theory of vibration at an undergraduate level, and is the first of two volumes by the same author. The second volume expands on the first volume's last chapter and is entitled Theory of Vibration, Discreet and Continuous Systems. The subject book is its second edition and is one of fifteen books that currently form the Mechanical Engineering Series published by Springer-Verlag with Frederick F. Ling as the series editor. The book is divided into seven chapters and has 61 worked examples, 212 figures and problems at the end of each chapter.

The basic vibrational quantities are defined in Chapter 1 and the differential equations for linear and rotational motion derived. A procedure for linearising rotational differential equations is explained in Chapter 1. Methods for solving homogeneous (free vibration) and non-homogeneous (forced vibration) differential equations are described in Chapter 2, and the requirements for stability are developed.

Chapters 3, 4 and 5 consider the free and forced vibrations of single degree of freedom systems for undamped and damped cases. Oscillatory motion, harmonic and non-harmonic forcing functions are treated. Chapter 6 investigates multi-degrees of freedom systems. The equations of motion are presented as matrices, and vibration absorbers explained. Chapter 7 deals with continuous systems, and discusses free and forced longitudinal, torsional and transverse vibrations.

The book develops the subject matter logically and clearly, and the text is supported by well presented figures. The assumptions made are unambiguously stated and the worked examples comprehensively assist the reader in understanding the topics covered. Answers have been provided to selected problems. The book provides an excellent introduction to the theory of mechanical engineering vibration and should prove of value to undergraduate students.

John Dickens

John Dickens is a Senior Research Engineer at the Aeronautical and Maritime Research Laboratory, Defence Science and Technology Organisation, Department of Defence. He is currently undertaking research into the control of noise and vibration in ships and submarines.

Hearing Conservation

Interactive Media Communications, 1996, CD ROM, Australian Distributor, Select Learning, 31 Croydon St Cronulla, NSW 2230 Tel 02 9544 2322, Fax 02 9544 2575. Price $2,500 purchase or rental $950 per month.

This CD ROM is one of a wide range of interactive multimedia training units produced by Interactive Media Communications in the US. Each unit is "stand alone" and designed to give the employees specific skills as well as general information that will help them understand "why" as well as "how".

The CD requires MPEG for the video quality graphics. We managed to get it running on a PC with MPEG simulation rather than a hardware MPEG card but did encounter difficulties. The sound dropped out in a random manner and then would come back again. I understand that these problems would not occur if a hardware MPEG card was used. However software emulation is becoming more widely available, particularly on the smaller transportable computers which would be of greater use when taking the training to the various workplaces.

The only documentation with the Hearing Conservation CD is a short installation note. This is not a disadvantage as the CD guides the user adequately and there is plenty of opportunity to go back over material. There is also a "map" in the main menu which helps the user to know the current location within the unit. After entering employee identification, a pre-course quiz must be completed before commencing the main part of the course. The scores for this quiz allows the user to skip certain parts of the course if the scores for the questions on those topics are high enough. The user can leave and then return to where they left by entering their identification.

The main topics expected for such training, such as types of noises, basics of hearing, engineering noise controls, personal hearing protection, etc are all covered. Use is made of talking heads, graphics, videos of work environments etc and questions are asked at relevant points. It has been well prepared and well presented so that the user is encouraged to continue to the end. It is also well structured so that the user is required to work through all the parts unless allowed to skip because of the results in the pre test quiz. There is a final quiz for which there is immediate feedback on the correctness of each answer.

The major disadvantage is that all reference to legislation and criteria is applicable to the United States. The OSHA guidelines, which use 5 dB doubling, are frequently referred to as the criteria and this is very misleading for Australia, which uses the 3 dB doubling. The brochure for the CD states that the training can be "tailored to suit your operation" and it would be essential for use in Australia that all reference to legislation and criteria be changed. The distributors have advised that a small addition to explain the Australian legislation would cost the client less than $30. The costs for more elaborate changes such as site specific video clips and some site specific test questions could be incorporated for about $500.

Once the changes were made to "Australianise" the legislation sections, this CD would be of assistance in training and providing a refresher course for OH&S officers and supervisors. It would be up to management to decide the appropriateness for the workers on the factory floor. In some respects it goes into too much detail for their requirements and would be unsuitable for those with limited English skills. On this matter the distributor has advised that the CD reviewed was the "Rolls Royce" version and other more simplified versions are available.

Marion Burgess

Marion Burgess is a Research Officer in the Acoustics and Vibration Unit at the Australian Defence Force Academy and has been involved with many courses for hearing conservation. She was pleased to get a high score in the precourse quiz! The assistance of Errol Brown, computer support for the School, is gratefully acknowledged for getting the CD ROM installed and running.

Acoustics in Buildings

Bernard Grehant

Thomas Telford, 1996, pp 305, Hard cover, ISBN 0 7277 2511 4, Australian Distributor, DA Information Services, PO Box 163, Mitcham Vic 3132, Tel 03 9873 4411, Fax 03 9873 5679. Price A$74.25.

This book presents strategies for addressing the specific problems associated with light motorised equipment, such as blinds, shutters etc, in buildings and is aimed at industrial designers and architects. The author has combined his skills developed as lecturer with his experience in industrial research to produce a book that is easy to follow with many practical examples. The book concentrates on the product range from the French based company SOMFY International but the general principles have wider application.

The first seven chapters comprise Part 1 and deal with the basic principles of sound and vibration. There are lots of diagrams to assist with the explanations. In the introduction, the author suggests these chapters can be skipped by readers acquainted with acoustics. The sections on criteria for noise regulations concentrate on the European legislation and standards for equipment noise and it is interesting to compare these with the Australian situation.

Chapters 8 to 11 form Part 2 of the book, entitled Improvement. Here the emphasis is clearly on the motorised units that can be used inside or on the facade of buildings. The motor, the moving unit and the installation are all considered in turn and examples of field and laboratory measurements provided. The figures and tables in this section are a little difficult to follow as the reference data changes from one figure to another. However the concepts of the factors which are important to minimise the noise output are summarised well. There is at least one case of typesetting errors in an equation (p153 the superscript is applied to the subscript for the variable rather than the variable itself).

Part 3 on Recommendations comprises three chapters on product design, installation and consequences for architectural design. These are useful summaries of the detailed findings from Part 2. The appendices cover some more theoretical concepts associated with the units for sound, transmission etc.

Overall this is a readable book and certainly provides a lot of very useful, practical information for anyone faced with a noise problem from light motorised equipment in buildings. I do consider that the title is misleading as the book does not (or only briefly) address many of the important issues related to the broad topic of acoustics in buildings. A bibliography would be of assistance to the reader who seeks more information on particular topics. The translation has been done well although there are some slightly strange phrases which can still be understood. This book is certainly recommended for those involved with the development, installation or noise control of light motorised equipment for buildings.

Marion Burgess

Marion Burgess is a Research Officer in the Acoustics and Vibration Unit at the Australian Defence Force Academy and has been involved with acoustics and buildings for some years.

Research Papers In Violin Acoustics 1975-1993

Edited by Carleen Hutchins and Virginia Benade Acoustical Society of America, Woodbury NY (1996) 2 Volumes, 1340 pp. ISBN 1-56396-609-3. Price US$210 (2 vols).

Of all the instruments of music, the violin has probably attracted the greatest amount of research interest. Much of the reason for this is the place that the violin occupies in Western musical culture, but another reason is the apparent simplicity but hidden subtlety of its construction and acoustic behaviour. The great Italian violin makers of the early 18th century – above all Antonio Stradivari (1644-1737) but also the Amatis, the Guaneris and others – have provided a mystique for the instrument and a standard of acoustic performance that modern makers strive to match. Since many of these instruments are still in good condition and regularly played, they have also provided the raw material for much study. Thanks to the diligent efforts of many researchers around the world, most of them aficionados rather than employees of musical instrument companies, the best modern violins can now be made to a standard that matches, or even surpasses, that of the old Italian masters.

Much of this worldwide research effort is coordinated informally through the work of the appropriately named Catgut Acoustical Society, founded in 1963 by Frederick Saunders of Harvard University and Carleen Hutchins who worked with him. The society now has over 750 members in 35 countries, publishes a journal, and coordinates national and international meetings in musical acoustics. In 1975-76, Carleen Hutchins edited a two-volume set, in the series "Benchmark Papers in Acoustics," that represented the best of all that had been published on violin acoustics from the time of Felix Savart (1840), through the work of Nobel Laureate C.V. Raman on bowed strings around 1920, up to about 1973.

Now Carleen Hutchins, with the assistance of Virginia Benade, has edited a further two massive volumes that bring the record up to 1993. Publication this time is by the American Acoustical Society, and the production (on heavy gloss paper, giving a total weight for the two volumes of about 4.5 kg) is of a very high standard. Altogether the two volumes contain 121 papers, making up 1340 large two-column pages, with contributions from all of the major researchers in the field. Also included are brief biographical notes on all of the authors and a complete author index to the Newsletter and Journal of the Catgut Acoustical Society, 1964-1994.

The collection is organised in a structural way, beginning with some generalities about acoustic radiation and then progressing through sections on the behaviour of the bowed string, the bridge, the soundpost, and so on to the completed instrument. Naturally, emphasis is given to the shaping and resonances of the top and bottom plates of the violin and to their coupling in the completed instrument by mechanical and air links, for this behavior is at the very heart of the distinction between good and poor instruments. Subsequent sections then deal with wood, with varnish, with psychoacoustics research, and with some miscellaneous matters. I could have wished for something other than simple alphabetical arrangement by author in the sections, since this gives no logical progression of subject matter, but otherwise the structure is good.

The papers are drawn largely from the Journal of the Acoustical Society of America, from Acustica, and from the Journal of the Catgut Acoustical Society, but some other publications are represented. All but two of the papers are in English. A survey showed that all the papers I would regard as most important have been included, and there were a few surprises that I had missed when they were first published. The papers have been selected for clarity and interest of presentation, as well as for the importantce of their subject matter, and many are a delight to read.

These two volumes are absolutely essential for anyone interested in research on the acoustics of bowed-string instruments, for they make all the major recent publications in the field available in compact and convenient form. The instrument maker will similarly find much of value, though the technical level might make the going rather heavy. The more general reader will gain insight into the current state of our understanding of what is important in violin acoustics. Such a reader should, however, begin by reading the excellent general papers in the volume – Meyer on "The Sound of the Orchestra", Benade on "Musical Acoustics", McIntyre and Woodhouse on "The Acoustics of Stringed Musical Instruments" and Hutchins on "The Acoustics of Violin Plates" – before plunging into the more technically detailed papers.

Neville Fletcher

Neville Fletcher is a physicist with CSIRO, ANU and ADFA. Although not a violin player himself, he has done extensive research on the physics of musical instruments of all kinds.

Vocal Fold Physiology: Controlling Complexity and Chaos

P.L. Davis and N.H. Fletcher, editors

Singular Publishing Group, San Diego and London, 1996. 433 pp, soft covers, ISBN 1-56593-714-7. Fax +1619 563 9008. Price US$57.50

This book is the ninth volume in an international research series organised by the Voice Foundation. It records the papers presented at the 9th Vocal Fold Physiology Symposium, held in Sydney during May 1995.

The book is divided into the following sections

Physical aspects	(5 chapters)
Physiological aspects	(10 chapters)
Performance aspects	(7 chapters)
Clinical aspects	(3 chapters)
Discussion	(2 chapters)

Each well written chapter is approximately self-contained, with a good selection of references. The authors come mainly from USA, Sweden, Japan, The Netherlands, and Australia (with one each from Germany and France). Since good research in this area is going on in the United Kingdom and Italy, for example, the references are therefore somewhat biased, and I could find no non-English-language references. Nevertheless, the references are very comprehensive. The index, maybe not so, which is quite understandable.

The topics covered are almost as numerous as the chapters, ranging through mechanical modelling of the vocal folds, possible mechanisms of vocal instabilities, respiratory control of the larynx, laryngeal muscle control, replicability and accuracy of pitch patterns in professional singers, and clinical relevance of control, chaos, and complexity, to name but a few.

Anyone wishing to study the physics and physiology of vocal cords should consult this book. It contains a great wealth of information, both researched and speculative (for future research). The level is postgraduate and above. Researchers or would-be researchers are particularly advised to have a personal copy.

Gordon Troup

Gordon Troup is a physicist, singer and voice researcher, based at Monash University and Melba Conservatorium.

OBITUARY
HR Weston

From 1923 to 1983 the Division of Occupational Health, of the then NSW Department of health, importantly assessed and improved working conditions in, for example, factories, agriculture and at the Snowy Mountains.

Horry Weston, a graduate of the Adelaide School of Mines, joined the Division in 1960 and investigated the adverse effects of dust and welding. However, within a few years he became particularly interested in occupational and environmental noise.

Because of staff increases, in 1965, four professional branches were formed in the Division including one concerned with the different technical and scientific aspects of "Industrial Hygiene". In 1969 the Division moved to new and expanded premises at Lidcombe. Horry had a well equipped noise laboratory, a staff of four and had an acoustic chamber built.

Horry professionally travelled overseas to study instruments and technical developments. Also he was a member of a N.S.W. official group charged with producing draft anti-noise legislation, was a member of the Australian National Health and Medical Council's ad hoc Committee about occupational noise and, likewise, was appointed to an Australian Standards Associations Committee.

Extensively Horry's advice was sought by Managements, Associations, Trade Unions, employees and by medical practitioners about persons with hearing impairments. My colleague had the ability to easily develop a meaningful relationship with such persons and to persuade them to "do the right thing". In this respect his ability to assess important aspects of working conditions and explain difficult technical matters were also major assets.

At the division he was very popular, highly respected by doctors, scientists, engineers and by support staff.

His qualities, energy and doggedness extended beyond work places into the community. Horry declined being Gosford Council's Australia Day senior citizen of the year; however this was awarded posthumously to the gratitude of his wife. Also he was an active delegate on the "Keep Australia Beautiful Council".

I have many happy memories of Horry. I wish there were more people like him.

Alan Bell

New Members...

The following new members, or upgrades, are welcomed to the Society.

SA

Member	Dr P Teague, Mr B Martin

NSW

Subscriber	Mr P Gerrard

WA

Member	Mr B Ismail, Dr RA Wilde, Ms S Derry
Associate	Mr E. Fry
Subscriber	Mr M Shead, Mr E Fry,

TAS

Member	Mr C Butler

VIC

Member	Mr P Sanchez

MALAYSIA

Member	Mr M Dowsett

HONG KONG

Member	Mr W Fung

AUSTRALIAN ACOUSTICAL SOCIETY

CHANGES TO MEMORANDUM AND ARTICLES OF ASSOCIATION

Work has commenced to re-write the Memorandum and Articles of Association for the Society (see p114, vol 24).

Comments, advice and recommendations are welcome.

Please forward them directly to:
General Secretary
Australian Acoustical Society
PO Box 4
Tally Ho Vic 3149
Tel/Fax: +61 3 9887 9400
watkins@melbpc.org.au

News...

International Conference and 1997 AAS Conference

Over 300 international registrations of interest have already been received for the combined 5th International Congress on Sound and Vibration and the 1997 Australian Acoustical Society Conference to be held at the University of Adelaide, South Australia, December 15-18, 1997. The technical program covers all aspects of sound and vibration and includes 8 keynote lectures as well as 7 specialist 2-hour tutorials from eminent experts in acoustics and vibration.

The social program (in addition to the banquet) includes a complimentary reception on the evening (December 14) preceding the congress, a complimentary BBQ and entertainment at Cleland Wildlife Park and an evening tour to the famous whispering wall or local Bell Centre.

There is an extensive accompanying persons tour program during the congress as well as a choice of several pre and post congress tours of the highlights of South Australia.

The exhibition accompanying the congress will include all types of acoustical and vibration products and services. It is promising to be quite extensive as more than two thirds of the available exhibition booths have already been booked by international as well as Australian companies wishing to exhibit their products and services.

For those wishing to present papers, the abstract deadline has been extended to April 30, 1997, but the due date for the completed manuscript remains at August 15, 1997.

Further information: Fifth International Congress on Sound and Vibration, Department of Mechanical Engineering, University of Adelaide, Adelaide, 5005 Australia Fax: +61-8-8303-4367 Web site: http://www.icsv5.on.net

INTER-NOISE 97 and ACTIVE 97

Budapest, Hungary, August 1997

The 1997 International Congress on Noise Control Engineering is the 26th in a series of international congresses on noise control engineering. INTER-NOISE 97 is sponsored by the International Institute of Noise Control Engi-neering and is being organized by the Scientific Society for Optics, Acoustics, Motion Pictures and Theater Technology in cooperation with the Acoustical Commis-sion of the Hungarian Academy of Sciences.

The theme of **INTER-NOISE 97**, August 25-27, is **Help Quiet the World for a Higher Quality Life.** Three distinguished lectures will address the theme of the congress, and more than 460 abstracts from authors in 48 country have been accepted for presentation in 8 parallel technical sections and poster sections. In addition to the regular sessions of invited and contributed papers, 11 special sessions on a variety of topics on noise control engineering have also been arranged. Parallel to the lectures an extensive exhibition will also be organized.

The fifth of a series of symposia, devoted to the topic of active noise and vibration control, ACTIVE 97, will be held August 21-23. ACTIVE 97 has been designated a European Acoustics Association (EAA) Symposium as well as a symposium of the International Institute of Noise Control Engineering (I-INCE). More than 130 abstracts from authors in 32 countries have been accepted and six plenary lectures will be given.

The regular participation fee is 400 USD for INTER-NOISE and 380 USD for Active 97, including proceedings, non-optional social events and lunches. Students can obtain a 50% reduction, and a reduction of 100 USD is given to those participating in both events.

Although the number of papers to be presented point to rather intensive working days, all efforts are being made to ensure that delegates have opportunities to enjoy the social program that is being arranged which will include a number of tours and excursions in Budapest and throughout the country, cultural and technical visits, congress banquet at a lucrative site and symposium dinner in a traditional Hungarian village - even visit to a cooking school to learn how to cook a real, delicious Hungarian Goulash.

For more details: Congress Secretariat:, Optical, Acoustical and Filmtechnical Scientific Society (OPAKFI), H-1027 Budapest, Fi u. 68, Hungary, Tel/Fax: Int + 36 1 202 0452 or + 36 1 201 8843, e-mail: ac97.opa@mtesz.hu, http://www.mmt.bme.hu/events/active97/ or http://www.mmt.bme.hu/events/inter-noise97/

WESTPRAC VI 97

WESTPRAC is to be held for the second time in HONG KONG. Over the last decade of rapid urban development, Hong Kong has changes remarkably not only in her urban townscape but also in the integration of many acoustic developments for the well being of her people. It's time for every one who has contributed to the acoustic achievements for a conference to refresh, renew and rally in Hong Kong again. WESTPRAC VI will be held from 19 - 21 November 1997 at Nikko Hotel, Tsimshatsui East, Kowloon. Plenary lectures will be presented by Dr. George Wilson of Wilson, Ihrig & Assoc., Inc (USA), Dr George Wong of National Research Council (Canada) and Prof. Tachibana of the University of Tokyo (Japan). Contributed papers will be considered in all areas of acoustics and noise control engineering.. Full registration fee is US$500, the conference banquet will be extra. Abstracts were due 1 March but late ones may be considered. Full conference papers shall be submitted before 1 July, 1997.

Further Information: Dr. SK Tang, WESTPRAC VI 97 Dept. of Building Services Engineering, The Hong Kong Polytechnic University, Hung Hum, HONG KONG, Fax (852) 27746146, email:besktang@polyu.edu.hk, http://www.polyu.edu.hk/~westprac

FASTS

The Federation of Science and Technology Societies (FASTS), of which the AAS is a member, has released the following summary of its recent achievements.

- Helping Member Societies to raise matters with Government, such as the forum to highlight problems in mathematics and sciences education.
- Speaking out for science and technology (S&T). FASTS argued the case against higher HECS fees for science courses before the Senate Committee on Employment Education and Training; and raised matters such as the lack of a career structure for younger scientists and the funding of industrial R&D.
- Helping S&T groups speak to government by bringing together administrators of the Science Academies, the CRCs

Association, ANZAAS, the AV-CC and Australian Science Communicators to discuss joint responses to the issues of the day. FASTS has organised joint media conferences at Parliament House to give a voice to all science groups.

- Maintaining a strong presence in the media, to ensure that policy-makers and the public are aware of the importance of S&T.
- Continuing contact with the Federal Government and meeting frequently with the Ministers and members of the Opposition. The FASTS' Policy Document has significantly shaped the S&T debate.
- Involving other powerful groups in raising awareness of S&T matters, such as the National Farmers' Federation the Australian Chamber of Manufactures etc.
- Helping scientists provide comprehensive solutions to Government.

The release of the "Ten Top Issues" early in the year received considerable media coverage. The issues covered subjects ranging from restructuring the universities to guarantee access to high-quality science education and research, boosting the scientific exploration of Australia's Ocean Territory in order to exploit marine and seabed resources in a sustainable manner; and addressing the looming shortage of properly trained mathematics and science teachers. Top of the list is a call for the Government to develop a national vision for Australia, with a clear place for science and technology.

VALUING EDUCATION: the Case for Mathematics and Science - The following is an edited version of the report on the forum originally prepared by Jan Thomas.

If Australia is to be competitive in the future technological world, more attention must be paid to mathematics, science and technology education. The Forum stressed the cultural and economic importance of a high level of scientific and mathematical literacy for all. Australia has a proud history of basic and applied research in the sciences. The speakers at the Forum addressed many concerns about the present and the future. There are many challenges ahead if Australia is to remain competitive. The scientific base in other countries in the Asia Pacific region is expanding rapidly while in Australia it is contracting.

The Forum demonstrated that there are many people in education, the universities, business, industry and government organisations who want to work together to improve mathematics and science education for young people. There are other groups, for example parents associations, which were not formally represented and would also want to contribute. The Forum noted the need for some leadership and initiative from government. Participants are anxious to work with government and this requires relevant ministers to engage in on-going dialogue with all the stakeholders. Otherwise government initiatives can be counter-productive. A unity of view and vision was seen as being needed.

The meeting saw as matters of urgency:

- **The supply of teachers** - it is projected that Australia is facing a massive shortfall in the supply of teachers.
- **Professional development and professional status of teachers:**
- **Career advice** - much better information is required about the breadth of careers in areas underpinned by mathematics, science and technology.
- **HECS changes** - differential HECS payment must not affect areas that are of strategic importance.
- **University funding cuts** - the specific affect on mathematics and science related areas must be carefully monitored.
- **Networking** - advantages from continuing communication.

Any issues that could be raised for action by FASTS should be forwarded to Editors of this journal.

WHO Noise Guidelines

The World Health Organization (WHO) will issue new noise guidelines in 1997. Its previous guidelines were established in 1980 and the revision is the result of a growing body of research conducted in many different countries on the harmful effects of noises on general health, not only hearing. Under the new recommended maxima, the average night-time noise level for undisturbed sleep will be set at 30 dB(A), down from 35 dB(A). The guidelines will fix a maximum peak night-time noise level for the first time, at 45 dB(A).

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STANDARDS

Patient Alarm Systems

A nurse call system is an essential item in all health care facilities as it enables patients in need to communicate with medical staff. There are a wide variety of such systems available, from simple bell-push buttons to highly sophisticated electronic devices. Standards Australia is to develop a Standard to harmonise all the bells, buzzers, bleeps and lights, and which will raise the overall quality of the product, and has formed Subcommittee TE/16/1. The Standard will detail the minimum generic requirements for patient alarm systems and have an Appendix outlining options to the basic system which will allow a health care facility to choose a system to sit its required level of sophistication. The one definite restriction of the Standard is that it will pertain to hard-wired systems; another Standard for radio-controlled systems may be developed if necessary.

One of the aims will be industry consistency of signals (both audible and visual), which should help with staff mobility; if a nurse hears or sees a certain type of alarm in Perth, it should indicate the same need as the same sound or light in Brisbane.

The draft Standard is currently being prepared for public comment and it is anticipated that it will be available shortly.

New Standard

AS/NZS 3949 Intruder alarm systems - Road vehicles. Performance requirements. Specifies the requirements and test methods for vehicle security alarm systems intended as original equipment for post-delivery installation for vehicles intended for the carriage of passengers. Appendices are provided for the requirements of sound level test for sounding devices and detection range test for ultrasonic movement detectors. Supersedes AS 3749,1-1990.

New Committee

AV/5/6 Noise Management - Ports, Shipping will have a prime function to develop a New Zealand Standard on port noise management. Projects manager liaison: Grant Cooper (02) 97464821.

Draft for Comment

DR 97013 CP *Ultrasonics-Physiotherapy systems-Performance requirements and methods of measurement in the frequency range 0.5 MHz to 5 MHz.* Proposes the adoption of the IEC 1689:1989 as a Joint Australian/New Zealand Standard. Gives requirements for ultrasonic physiotherapy equipment using single plane transducers, including performance and safety specifications for the generated field.

Further information: Standards Australia, PO Box 1055, Strathfield NSW 2135, Tel +61-02-9746400 Fax +61-02-97463333

ISO Drafts:

ISO 717-1:1996, Acoustics - *Rating of sound insulation in buildings and of building elements - Part 1: Airborne sound insulation* as replacement for AS1276-1979.

ISO 10534-1, Acoustics - *Determination of sound absorption coefficient in impedance tubes - Part 1: Method using standing wave ratio* as replacement for AS 1935-1976.

New ANSI Standards

ANSI S2.13-1996/Part 1 *Mechanical Vibration Of Non-Reciprocating Machines - Measurements Of Rotating Shafts And Evaluation - Part 1: General Guidelines.* Provides the test procedure for the measurement and evaluation of the mechanical vibration of non-reciprocating machines and guidelines for adapting evaluation criteria for different types of machines.

ANSI S12.9-1996/Part 4 *Quantities And Procedures For Description And Measurement Of Environmental Sound - Part 4: Noise Assessment And Prediction Of Long-Term Community Response.* Specifies methods to assess environmental sounds and to predict the annoyance response of communities to long-term noise.

ANSI S12.17-1996 *Impulse Sound Propagation For Environmental Noise Assessment.* Describes engineering methods to calculate the propagation of high energy impulsive sounds through the atmosphere for purposes of assessment of environmental noise.

ANSI/IEEE Std. 260.4-1996 *American National Standard Letter Symbols and Abbreviations For Quantities Used In Acoustics.* Covers letter symbols for physical quantities used in acoustics. Abbreviations for a number of related measures that are in common use are also given.

Available from: Acoustical Society of America, Standards and Publications, PO Box 1020, Sewickley, PA 15143-9998, Fax +412 7410609 asastds@aip.org.

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Sleep Arousal

A NSW Division technical meeting in 1996 comprised a panel of two speakers discussing sleep arousal - what really wakes you up. It was noted that all persons who attended were sufficiently aroused that no one was noted dosing off during the excellent presentation.

Dr Norm Carter of NAL discussed the various traditional methods of measuring sleep and raised a number of questions which still require answers. In summary the measurement of sleep can be conducted either during sleep or the next day. If it is measured during sleep there are three common methods of measurement; polygraphy (EEC eyes chin and skull); actinometry (hand movement); and voluntary response (press a button upon walking). Alternatively, if measured the next day after sleep or sleep deprivation there are also three common methods of measurement; polygraphy; performance tests; and subjective reports. Dr Carter suggested that the sleep state change could be a more preferred method of measurement. He stated that there is strong evidence confirming that a reduction in sleep affects performance and health in the following ways: reduction in Stage 4 sleep is related to sleepiness; reduction in Stage 2 sleep injures some type of memory; loss of stage 4 sleep may impact on the immune system (NAL/Uni Newcastle study); and fragmentation of sleep reduces daytime alertness.

Dr Rob Bullen of ERM Mitchell Cotter proposed a method for assessing sleep disturbance due to intermittent environmental noise. It was based on: the number of individual noise events heard; the maximum levels of events; and the 'emergence' of events above the ambient noise. He defined intermittent noise as individual events such as planes, cars and trucks. Dr Bullen stated that the derivation of the method, and hence the 'sleep disturbance index' (SDI), was based on a number of field trials. He presented number of examples of how the method could be used. The weighting factor, a fundamental feature, was derived from the results of eleven major international studies on the probability of awaking, based on the number of events and maximum internal noise level. Dr Bullen plotted these results and put a best fit second-order polynomial curve through this data to obtain the weighting curve. He noted that this method is more useful than the existing alternative and could form a more solid basis for sleep disturbance assessment.

The two speakers and **Mr Roger Treagus** from the EPA invited questions from the audience. The questions were probing and stimulating and highlighted the fact that, like airport noise, sleep disturbance is a very complex technical issue which is difficult to define as a model that covers all the possible variables. Whatever criteria are finally used to define sleep disturbance will need to be a compromise of a more complex set of variables.

David Eager.

Railway Vibration

The speaker at the December meeting of the Victorian Division was **Dr Hugh Hunt,** University of Cambridge Engineering Department who spoke of his research into reducing the amount of railway vibration, particularly from trains in metropolitan underground tunnels, transmitted to adjacent buildings.

He began by describing the various devices currently used to reduce the noise and vibration from train movement in underground tunnels. The acoustic insulation and resilient suspension of the passenger coaches, while it added to the comfort of those inside, did virtually nothing to reduce the amounts of noise and vibration transmitted into the tunnel and to the tunnel structure and other buildings beyond the tunnel.

Reduction of this noise and vibration required the railway track to be resiliently supported, which reduced both the wheel-on-rail noise in the tunnel and the amount of vibration transmitted from the track to the tunnel structure and beyond. Currently, resilient track supports included rail pads (usually of rubber) between rail and sleeper, together with pads between sleeper and tunnel interior. In some track designs, the rail pads consisted of a pad between rail and baseplate, with a second pad between baseplate and sleeper. The use of anti-reverberant acoustics treatment of the tunnel walls also contributed to reducing noise inside the tunnel.

However, additional anti-vibration treatment was required in order to reduce the amount of vibration transmitted from the tunnel structure through the ground into adjacent buildings, particularly through their foundations. An early means of attenuation was provided by seating a building's foundation columns on rubber springs with a natural frequency as low as practicable to give maximum attenuation to vibration with a spectrum containing frequencies of 50 Hz and less. At this earlier stage of development, it had been assumed both that the tunnel structure was rigid, and that the spring's performance could be predicted from their theoretical behaviour.

Since then, neither assumption has been found to be accurate; for the tunnel structure is not rigid, and buildings (including their support columns) can amplify the vibration. Further, the vibration measured at any particular point in a building has often been found to arrive there by significantly more than one path through the building. Dr Hunt's recent researches have therefore been concerned with developing the mathematics to describe this more complex vibration transmission behaviour. His current computer 'model' relies on repeating element theory, and runs well on a PC. Following several questions to Dr Hunt, those present went to dinner at the Club.

Louis Fouvy.

ACEL & IHS Merge

IHS Australia and ACEL Information have merged to become the largest supplier of technical information and international standards in the Southern Hemisphere. While the telephone, fax and postal addresses remain the same, access will be to a much broader and comprehensive range of products and services. These include: Standards and Commercial publications, OH&S Information, Military Documents and Standards, Electronics/Engineering data and much more.

Further Information: Tel 02 9876 533, 02 9906 5566 or 02 98266099.

Money For Nothing?

Small businesses are again being targeted by operators claiming to represent government agencies or quality accreditation organisations. The latest crop of operators, currently under investigation by the Australian Competition and Consumer Commission (ACCC), are extracting fees from companies under the pretence of including companies in small business registers which they claim are issued to government departments who use the registers to select suppliers and business partners. Some operators and schemes go even further, inferring in letters sent to companies that businesses need to pay for the service being offered in order to meet government regulatory requirements. These latest allegations have been met with strong words of warning from the ACCC to small businesses. The ACCC strongly advises companies to be vigilant and alert to misleading request for funds.

Data Copyright

The President of the Academy of Science, Sir Gustav Nossal, asked the Federal Government to oppose the introduction of new, restrictive copyright rules on databases. A treaty on database extraction rights has been proposed by the World Intellectual Property Organisation. 'Data collected in many different places, often at public expense, is the life-blood of much scientific research' said Sir Gustav. The progress of science and the advancement of many other kinds of human knowledge and endeavour is based upon the principles of free exchange of data and information'. At the moment, huge quantities of information from databases are available for use by researchers at nominal cost. Sir Gustav pointed out that the suggested changes to copyright rules will endanger access by researcher. Australia will be particularly badly affected, as it uses more data than it generates.

It is not only the Academy which is concerned about this proposal. The International Council of Scientific Unions has also called for a delay while it studies the effects of the treaty and scientific academies in the USA have called on their government to oppose the treaty.

On the Web

Catgut Acoustical Society can be found at http://www.marymt.edu/~cas

The Musical Acoustics Reference Library at CCRMA/Stanford University is gradually taking shape.
http://www.ccrma.stanford.edu

CRNet, the communication arm of the Ian Clunies Ross Memorial Foundation, is now live on the Internet. The new website showcases the full spectrum of Australian innovation, science and technology. At the heart of the website is the Innovation GATEWAY which is an online database of science and technology industries, individuals, projects and organisations.
Address: http:///www.crnet.com.au

Science Issues - Nova WWW site, provides up-to-date and accurate information on science, technology, environment and health issues in the news, and is found at http://www.science.org.au/nova/

Nova is aimed primarily at secondary school teachers. It is also valuable to students doing assignments, parents who want to support their children's interest in science, and scientists and engineers whose work is highlighted. There are six topics online at present and the Australian Academy of Science is seeking additional funding from government and corporate sources to add new topics as they make headlines.

New Journal

The International Institute of Acoustics and Vibration (IIAV) has published the first issue of the Institute's important new scientific journal, the International Journal of Acoustics and Vibration (IJAV). It is anticipated that this Journal will exert a major influence within key areas of application and a for all scientists and researchers worldwide, who have something to say and tell their fellow-researchers.

The first issue has 5 articles and 18 pages of abstracts for papers from various other Journals and Conferences. In addition there is information about the Institute and its activities plus book reviews, calendar and other items. Information about the Institute and the Journal can be obtained from:
IIAV Secretariat, PO Box 13, Auburn, AL 36831, USA, Fax 1 334 844 3306
mcrocker@eng.auburn.edu

Ultrasound Terminology

The American Institute of Ultrasound in Medicine (AIUM) announces the publication of the second edition of *Recommended Ultrasound Terminology*. The new edition, a major revision of the first edition published in 1990, includes terms that have been modified or added to reflect the changing times. Available immediately, the 1997 edition's goal is to make available in one source those terms and definitions making up the vocabulary for the growing field of medical ultrasound.

The document, which was prepared by the Terminology Subcommittee of the Technical Standards Committee, was written so that many terms, although technical in nature, are expressed in ways that are comprehensible to individuals without a strong background in physical science. This edition is available for \$20 for AIUM members and \$40 for nonmembers plus shipping and handling through the AIUM Publications.

Further Information: AIUM Publications Department, 14750 Sweitzer Lane, Suite 100, Laurel, MD 20707-5906. Fax: Int+301 4984450.

NIT to LMS

In 1996 the company name was changed from Numerical Integration Technologies to LMS Numerical Technologies. Dr Urbain Vandeurzen, Chairman, president and CEO of LMS International said "The new name reflects the size of the company and the strategic importance of prediction technologies within the LMS Group. LMS can provide tightly integrated testing, modelling and prediction solutions for acoustics, structural dynamics and durability engineering". LMS products include SYSNOISE, RAYNOISE, VIOLINS, MOSART and OPTIMUS. The Australian Agents are COMPUMOD, *Tel 02 92832577 Fax 02 92832585*
email Andrew.Currie@compumod.com.au

New Products...

RTA Technology

ENM Windows

Since ENM DOS was first released in 1986, it has found tremendous success world-wide. ENM Windows presents an improved graphical interface and makes the program simpler to learn using the on-line help system. ENM Windows incorporates its own CAD drawing module and imports to and exports from AUTOCAD.

ENM Windows simulates outdoor sound propagation and predicts noise levels from known sources for close and distant locations. The model takes into account noise sources which are enclosed or unclosed, accounts for distance from the source to the receiver, for the noise source type, size and directivity, geometric spreading, air absorption, wind effects, temperature gradient effects, ground absorption effects and shielding by vegetation, buildings and natural topographical features.

ENM DOS and ENM Windows have been validated independently by a multitude of researchers throughout the world.

Further Information: RTA Technology, Tel 02 92327251, Fax 02 92327260, email rtatech@ozemail.com.au

LMS

LMS OPTIMUS

LMS OPTIMUS is a Computer-Aided-Engineering software for multi-disciplinary optimal design synthesis which enables engineers to develop products that meet several design criteria using commercial simulation packages, without the need to write any interface software. It drives and manages the exchange of data between existing simulation tools, and modifies the product design based on the simulation outputs until an optimal design is found. The number and type of constraints, design variables, simulation software packages and optimisation objectives is unrestrained, specified only by the user.

LMS OPTIMUS is a management framework residing above the simulation codes - it intelligently launches and drives each CAE simulation tool as required and evolves the product design based on the simulation outputs until an optimal design is found. The optimisation engines can be interfaced to any simulation tool. The optimisation engines can be interfaced to any simulation tool currently on the market. Therefore, LMS OPTIMUS re-employs and adds value to the considerable time and money invested in existing simulation technologies and defined procedures.

Further Information: COMPUMOD Pty Ltd, 9th Floor, 309 Pitt St, Sydney 2000, Tel 02 92832577 Fax 02 92832585 email Andrew.Currie@compumod.com.au

KINGDOM

FFT Analyser

The PCMCIA FFT Analyser ACE has been re-packaged to provide 4 MBytes extended memory in the DP104. There is now only one version of ACE and in future, all ACE modules will include the full memory expansion. The standard ACE model provides 1600 lines of spectral signal resolution with the addition of a simple software option ACE will now access the full spectral signal resolution up to 25,600 lines. ACE will operate in various computers notebooks or desktops which have Type III pcmcia sockets.

Further Information: Kingdom Pty Ltd, PO Box 75 Frenchs Forest NSW 2086, Tel: 02 99753272

Letters...

Ototoxic Substances

Members will be aware of the revision, currently in progress, of AS1269, "Hearing Conservation". This has now reached an advanced stage and postal ballot responses to the Draft will shortly be considered.

A current consideration, by the Physical Agents Threshold Limit Values (TLV) committee of the American Conference of Governmental Industrial Hygienists (ACGIH), of the possible role of ototoxic substances in the induction of hearing loss form work activities has led to the inclusion in the introduction section of the draft standard of a clause referring to this possibility, which has not hitherto been addressed in previous versions of AS1269. Nevertheless, it will be apparent that recognition of ototoxic effects, as well as those of noise, may be relevant to the development of hearing conservation programs. Moreover, the question of ototoxicity has, also, not previously been addressed in the setting of exposure standards for materials at the workplace, whether in the Worksafe Standards for workplace atmospheric contaminants or in the widely-used TLV's for chemical substances of ACGIH.

It is, therefore, desirable that any evidence of such effect known to AAS members from Australian experience, be available for study by all three bodies concerned. The international literature suggests that materials involved fall into the following categories - **Ototoxic:** Trichloroethylene, Lead, Toluene, Mercury, Butanol (n), Manganese and Arsenic. **Ototoxic with Noise Exposure:** Carbon disulphide, Carbon tetrachloride and Styrene - and at a recent meeting of AAS it was suggested that the matter be raised in this correspondence so as to reach the largest number of members, and others, who might be able to contribute information.

This might perhaps best be initiated by letters to the Editor, from which the standards-setting bodies could communicate direct with the writers for more detailed evidence relevant to inclusion in Standards.

G.V. Coles MAAS, ACGIH Physical Agents TLV Committee, Occupational Hygiene Unit, Deakin Uni, Geelong, VIC 3217, Fax 03 5227 1040

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Seeking Work

I have decided to write to "Acoustics Australia" in my search for career opportunities in Australia. Currently a research assistant at the Institute of Sound and Vibration Research (ISVR) in Southampton under the supervision of Professor Fahy, my contract is finishing in July 1997 after which I will be available for work. Therefore, I am at present looking for a challenging position either in consultancy or in research. The research I am currently undertaking is concerned with the medium and high frequency vibro-acoustic behaviour of curved window glass of road vehicles. It involves studies of various aspects of vibro-acoustics including SEA. I also obtained an M.Sc. in Sound and Vibration at the ISVR in 1995 (I am teaching a lab on reciprocity to the M.Sc. students). My M.Sc. project enabled me to acquire some experience in BEM, FEM and in intensimetry.

Although I am French, I have lived in England for four years where I have learnt how to communicate fluently in both oral and written English. As a dynamic and well organised person, I believe I would integrate easily into an Australian company and become efficient quickly through immediate application of my technical, language and adaptability skills.

If anyone can help me in my search for job opportunities, please contact me at:

Cédric VEILEX, ISVR-FDAG, University of Southampton, Highfield, Southampton SO171BJ UK Tel :+44 (01703) 592291
Fax: +44 (01703) 593190
E-mail: vc@isvr.soton.ac.uk
Web page:http://www.soton.ac.uk/~cv/

Diary...

CONFERENCES and SEMINARS

* Indicates an Australian Activity

1997

May 1, SYDNEY
Planning and the Environment
Details: Ms Devika Singh, Ph: 02 92327253
Fax: 02 92327260, rtatech@ozemail.com.au,
http://www.ozemail.com.au/~rtatech/building/

May 12-16, GDANSK
13TH FASE Symp on Hydroacoustics
Details: Inst Exp Physics, Ul. Wita Stwosza 80-952 Gdansk, Poland. Fax +48 58 413175,
fizas@halina.univ.gda.pl

May 20-22, TRAVERSE CITY
SAE Noise and Vibration Conf.
Details: SAE/MJA, 3001 W. Big Beaver Rd, Suite 320, Troy, MI 48084, USA. Fax +1 810 649 0425

June 1-3, MIAMI BEACH
2nd bilingual Americas' Conf. on Ultrasound.
Details: AIUM, 14750 Sweitzer Lane,
Suite 100, Laurel Maryland 20707-5906, USA,
pubs_govt@aium.org.

June 3-5, GOTHENBURG
Low Frequency Noise and Vibration,
Details: Multi-Science Publishing Co. Ltd.,
107 High St. Brentwood, Essex CM14 4RX,
Fax: +44 1277 223453

June 4-6, ITALY
Int. Conf. on Computational Acoustics and Its Environment Applications.
Details: L. Morton, Wessex Inst. of Tech.,
Ashurst Lodge, Ashurst, Southampton,
SO4O 7AA UK, Fax: +44 1703292853,
lynn@wessex.witcmi.ac.uk

June 15-17, STATE COLLEGE
Noise-Con 97
Details: INCE, PO Box 320 Arlington Branch,
Poughkeepie, NY 12603, USA.
Fax +1 914 463 0201

June 18-21, PRAGUE
3rd European Conf. on Audiology
Details: Paediatric Otolaryngologic Clinic,
Faculty Hospital Motol, V Uvalu 84, 15018
Prague 5, Czech Republic. Fax +42 2 2443 2620

June 24-27, PRAGUE
1st Eurp Conf on Signal Analysis & Prediction
Details: ESCAP Secretariat, Institute of Chemical Technology, Technicka 5 166 28 Praha 6,
Czech Republic, escap@vscht.cz

June 30-July 4, LA SPEZIA
High Frequency Acoustics in Shallow Water
Details: Anna Bizzarri, SACLANT Undersea Research Centre, Viale San Bartolomeo 400,
19138 La Spezia (SP) Italy.
Fax: +39 187 540 331, pace@saclantc.nato.int

July 14-17, SOUTHAMPTON
6th int. Conf on Recent Adv in Struct Dynamics
Details: N. Ferguson, ISVR, Uni. of Southampton, Southampton S17 IBJ, UK. Fax: +44 1703 593033, mxs@isvr.soton.ac.uk

July 21-25, CHILWORTH MANOR
4th int. Conf. on Natural Physical processes Related to Sea Surface Sound.
Details: Maureen Strickland, ISVR Conf. Secretary, uni. of Southampton, Southampton S)17 1BJ, UK, Fax: +44 1703 592294,
mzs@isvr.soton.ac.uk

August 14-18, USA
3rd Int. Conf. on Theoretical & Comput Acoustics
Details: Yu-chiung Teng, Columbia Uni. School of Mines, Aldridge Lav. of Applied Geophysics, New York, NY 10027, USA, Fax: +1 2028546504

August 19-22, EDINBURGH
Int. Symp. on Musical Acoustics.
Details: Dept. Physics and Astronomy, University of Edinburgh, James Clerk Maxell Building,
Mayfield Rd, Edinburgh EH9 3JZ. Scotland.
Fax +44 131 650 5902, isma.97@ed.ac.uk

August 21-23, BUDAPEST
Int. Sym on Active Control
Details: ACTIVE Secretariat, OPAKFI, Fo u. 68,
1027 Budapest, Hungary. Fax +36 1 202 0452

August 25-27, BUDAPEST
Internoise 97
Details: OPAKFI, H-1027, Budapest FO U68
Hugary, Tel/Fax: +36 1202 0452

August 29-31, GERMANY
Pan-European Voice Conference.
Details: School of Logopedics, Uni of Regensburg, 93042 Regensburg, Germany,
Fax: +49 94195319[illegible]6

September 1-4, JAPAN
IMAC-XV Japan, 'Bridge Over Virtual & Real Design'
Details: IMAC-XV, Dept. of Precision Mechanics,
Chuo University, 1-13-27 Kasuga, Bunkyo-ku,
Tokyo, 112 Japan. Fax 81 3 3817 1820
jmac@okubo.mech.chuo-u.ac.jp

September 10-12 , NEW ZEALAND
Biennial Conference - NZ Acoustical Society
Details: NZ Acoustical Society, PO Box 1181,
Auckland, NZ Fax +64 9 623 3248

September 10-12, STUTTGART
Biomechanics of Hearing
Details: EUROMECH Colloquium 368, W. Schiehlen, Institute B of Mechanics,
Uni of Stuttgart, 70550 Stuttgart, Germany.
wos@mechb.uni-stuttgart.de

September 12-13, USA
Symposium on Sonoluminescence.
Detail: M. brenner, MIT, Dept of Mathermatics,
Cambridge, MA 02139, USA,
Fax +1 617 2534358, brenner@math.mit.edu

September 15-17, ENGLAND
Whole-Body Vibration Injuries.
Details: Mrs Smith, Juman Factors Research Unit, Institute of Sound and Vibration Research,
University of Southampton, Southampton SO17 1BJ England. Ph: +44 1703592277 Fax: (+44) 17035 92927. hfru97@isvr.soton.ac.uk

September 17-19, ENGLAND
32nd Meeting of the UK Group on Human Response to Vibration.
Details: Mrs Smith, Juman Factors Research Unit, Institute of Sound and Vibration Research,
University of Southampton, Southampton SO17 1BJ England. Ph: +44 1703592277
Fax: (+44) 17035 92927. hfru97@isvr.soton.ac.uk

September 18-19, MEXICO
4th Mexican Congress on Acoustics
Details: Mexican Instit. of Acoustics, PO Box 75805, 07300, Mexico City, Mexico,
Fax: +52 55234742, sberista@vmredipn.ipn.mx

September 18-20, GREECE
Intonation: Theory, Models and Applications
Details: ESCA Workship, Dept. of Informatics,
Uni of Athens, Fax: +30 17228981,
tonesca@di.uoa.gr

September 21-23, USA
Product Sound Quality 1997
Details: PSQ'97, RH Lyon Corp, 691 Concord Av, Cambridge, MA 02138, USA,
Fax: +1 6178640779, rhlyon@mit.edu

*** October 1-3, QUEENSLAND**
(CM)2 Forum - for more effective condition monitoring
Details: Centre for Machine Condition Monitoring, Monash University, Wellington Road,
Clayton, Victoria 3168 Australia.
Ph: +61-3 9905 5699, Fax: +61 3 9905 5726
maltezos@eng2.eng.monash.edu.au,
http://www.monash.edu.au/cmcm

October 8-10, WINDSOR
Acoustics Week in Canada 1997
Details: R Ramakrishnan, Vibron Ltd, 1720 Meyerside Drv. Mississauga, Ontario, L5T 1A3
Canada. Fax: +1 905 670 1698

October 23-26, UNITED KINGDOM
Reproduced Sound 13
Details: Inst. of Acoustics, Agriculture House,
5 Holywell Hill, St Albans, Herts AL1 1EU, UK
Fax: +44 1727850533, acoustics@clusl.ulcc.ac.uk

November 9-13, KYONGJU
Asia Pacific Vibration Conf 97
Details: APVC97, Intercom Services, 4Fl Jisung Bldg, 645-20 Yoksam 1-dong, Kangnam-gu,
Seoul 135-081, Korea. Fax +82 2 3452 7292
intercom@sobak.kornet.nm.kr

November 19-20, HONG KONG
WESTPRAC '97
Details: S Tang, WESTPRAC Secretary, Dept. of Building Services Engineering, The Hong Kong Polytechnics Uni., Hong Kong.
Fax +852 27746146 besktang@polyu.edu.hk
http://www.polyu.edu.hk/~westprac

December 1-5, SAN DIEGO
Meeting of the ASA
Details: ASA, 500 Sunnyside Blvd., Woodbury, NY 11797 USA. Fax +1 516 576 2377, asa@aip.org

December 15-18, ADELAIDE
* 5th Int Conf on Sound & Vibration
Details: Dept Mech Eng, University Adelaide, SA 5005, Australia. Tel +61 8 303 5698, Fax +61 8 303 4367, icsv5@mecheng.adelaide.edu.au

1998

March 23-27, ZURICH
DAGA 98 - German Acoustical Society Meeting
Details: DEGA, Physics/Acoustics Dept.,
Universität Oldenburg, 26111 Oldenburg,
Germany. Fax: +49 441 798 3698,
dega@aku.physik.uni-oldenburg.de

May 12-15, SEATTLE
IEEE Conf. on Acous, Speed & Signal Processing
Details: L. Atlas, Dept. EE (FT 10),
University of Washington, Seattle, WA, USA.
Fax +1 206 543 3842, atlas@ee.washington.edu

May 25-27, ITALY
Noise and Planning 98
Details: Noise & Planning, via Bragadino 2,
20144 Milano, Italy, Fax: +39 248018839,
mdl467@mclink.it

1998 continued

June 8-10, TALLINN
Transport Noise and Vibration
Details: East-European Acosutical Assoc., Moskovskoe Shosse 44, 196158 St. - Petersburg, Russia. Fax: +7 812 127 9323, kryspb@sovam.com

June 20-28, SEATTLE
16th International Congress on Acoustics
Details: 16th ICA Secretariat, Applied Physics laboraty, Uni of Washington, 1013 NE 40th St. Seattle, WA 98105-6698, USA.

June 21-26, USA
13th U.S. National Congress of Theoretical and Applied Mechanics.
Details: M. Eisenberg, AeMES Dept., Uni of Florida, PO Box 116250, Gainesville, FL 32611-6250, USA, Fax: +1 3523927303, meise@eng.ufl.edu

October 12-16, AMERICA
Meeting of ASA
Details: ASA, 500 Sunnyside Blvd., Woodbury, NY 11797 USA. Fax +1 516 576 2377, asa@aip.org

November 16-20, CHRISTCHURCH
INTER-NOISE 98
Details: NZAS. P.O. Box 1181, Auckland, NZ, Fax +64 9 309 3540

November 20, QUEENSTOWN
Recreational Noise
Details: P. Dickenson, NZ Ministry Health, PO Box 5013, Wellington, NZ Fax +644 4962340, philip.dickenson@mohwn.synet.net.nz

November 22-27, SYDNEY
Noise Effects 98
ICBEN Congress
Details: Noise Effects '98, GPO Box 128, Sydney NSW 2001 Australia

COURSES

In accordance with the recognition of the importance of continuing education, details of courses held in Australia are included in this section at no charge. Additional details can be given in an advertisement at normal rates.

1997

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Details: Jenni Metcalfe, Econnect, PO Box 464, Paddington QLD 4064. Tel 07 3367 2646 or 014 91 6372, Fax 07 3217 6376.

PRACTICAL NOISE MANAGEMENT PROGRAMS - various times
Details: Dick Benbow & Assoc., Unit 4, 5-9 Hunter St. Parramatta NSW 2150
Tel: 02 6355099 Fax: 02 6891385

BASIS OF NOISE & VIBRATION CONTROL
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Details: Acoustics & Vibration Unit, ADFA, Canberra, ACT 2600
Tel: 06 268 8241 Fax: 06 268 8276
avunit@adfa.oz.au

SIGNAL ANALYSIS, NOISE & ACOUSTICS
- various locations and dates
Details: MB & KJ Davidson, 17 Roberna St Moorabbin Vic 3189,
Tel: 03 9555 7277 Fax: 03 9555 7956

Information for Authors...

Acoustics Australia is the journal of the Australian Acoustical Society. It publishes general technical articles in all areas of acoustics of interest to members of the Society, together with relevant news and views. Review papers, covering particular fields of acoustics and addressed to a non-specialist acoustics readership, as well as papers of a "tutorial" nature dealing with important acoustical principles or techniques are most welcome. Acoustics Australia does not aim to be a primary scientific journal, and therefore does not normally publish primary research papers, with the exception of those that apply specifically to Australia.

Submission of an article carries the implication that it has not been published in the same form elsewhere, and its not currently under consideration by another publication. Upon acceptance, authors will be asked to transfer copyright to Acoustics Australia (to enable us to deal easily with requests for reprint permissions) but the author will retain rights to use the material freely.

Articles should generally not exceed five journal pages in length. This implies a maximum of about 5000 words, with each square single-column diagram being counted as 300 words, and pro-rata for diagrams of other shapes. Authors submitting longer articles may be asked to bear the extra publication costs involved. Shorter "Technical Notes", not exceeding one journal page in length, are also welcome. All articles will be submitted to independent review before being accepted for publication.

Three copies of text and diagrams, together with originals of all drawings, should be submitted to the Editor. The drawings must be of publication quality and lettering must be of such a size that it will be no less than 2mm high when the figure is reduced to single-column size (width 85mm). The equivalent of either Times Roman or Helvetica type may be used in diagram lettering. The journal cannot undertake to re-draw unsatisfactory diagrams. Half-tone photographs of good quality may also be included.

Full instructions for authors can be found from http://www.adfa.oz.au/~mxb/aa.

All material for publication should be submitted to The Editor, Acoustics Australia, Acoustics and Vibration Unit, Australian Defence Force Academy, CANBERRA ACT 2600 email: acoust-aust@adfa.oz.au.

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cqhoward@mecheng.adelaide.edu.au

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