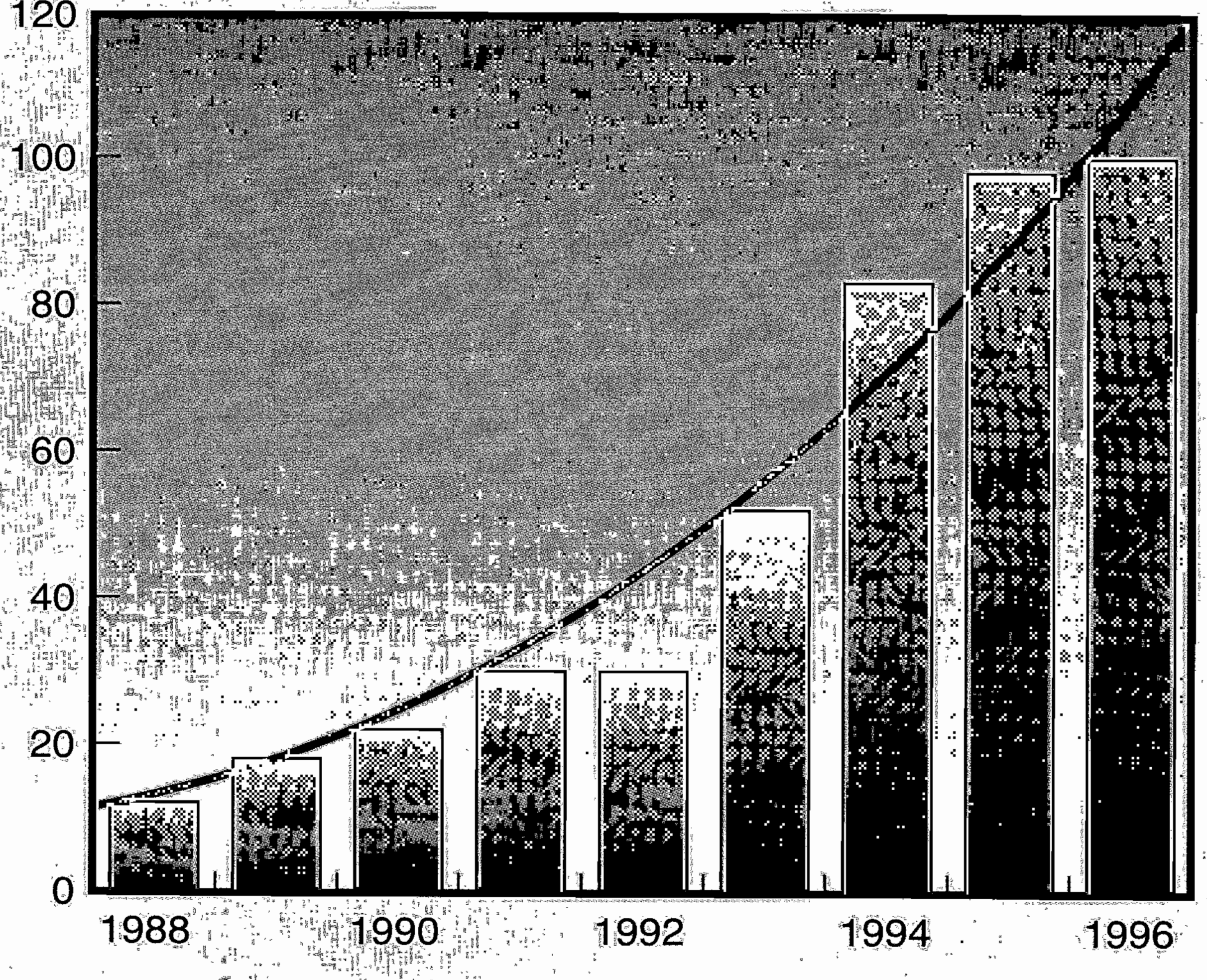
Acoustics Australia



- **Structure-borne Noise Transmission**
- **Gong Acoustics**
- **Occupational Noise**
- **50 Years of Helping People Hear**

Australian Acoustical Society **Vol. 25 No. 3 December 1997**

Acoustics Australia

Acoustics Australia General Business
(subscriptions, extra copies, back issues, advertising, etc.)

Mrs Leigh Wallbank
P O Box 579
CRONULLA NSW 2230
Tel (02) 9528 4362
Fax (02) 9523 9637

Acoustics Australia All Editorial Matters
(articles, reports, news, book reviews, new products, etc)

The Editor, Acoustics Australia
Acoustics & Vibration Unit
Australian Defence Force Academy
CANBERRA ACT 2600
Tel (02) 6268 8241
Fax (02) 6268 8276
email: acoust-aust@adfa.oz.au
http://www.adfa.oz.au/~mxb/aa

Australian Acoustical Society Enquiries see page 124

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: print@cronullaprint.com.au

ISSN 0814-6039

Vol 25 No 3 December 1997

CONTENTS

ARTICLES

NOTES

COVER: *Annual cost of occupational noise – induced compensation claims in NSW. See article by Macrae*

AUSTRALIAN ACOUSTICAL SOCIETY SUSTAINING MEMBERS

ACOUSTIC RESEARCH LABORATORIES
LEVEL 7 BUILDING 2
423 PENNANT HILLS ROAD
PENNANT HILLS 2120

ACU-VIB ELECTRONICS
56A THOMPSON ST
DRUMMOYNE 2047

ADAMSSON ENGINEERING PTY LTD
PO BOX 1294
OSBORNE PARK 6916

ASSOCIATION OF AUSTRALIAN ACOUSTICAL CONSULTANTS
C/- HERRING STORER ACOUSTICS
PO BOX 219
COMO WA 6952

ASSOCIATION OF NOISE CONTROL ENGINEERING
P.O. BOX 1051
BAYSWATER 3153

BARCLAY ENGINEERING
12-16 CATALANO RD
CANNING VALE 6155

BHP STEEL
MILLS & COATING FACILITIES
PO BOX 1854
WOLLONGONG 2500

DALLOZ SAFETY PTY LTD
PO BOX 42
TERREY HILLS 2084

BORAL PLASTERBOARD
676 LORIMER ST
PORT MELBOURNE 3207

BRUEL & KJAER AUSTRALIA PTY LTD
PO BOX 177
TERREY HILLS 2084

CHADWICK TECHNOLOGY
9 COOK STREET
FORESTVILLE 2087

CSR BRADFORD INSULATION
LEVEL 4, 9 HELP STREET
CHATSWOOD 2067

EMONA INSTRUMENTS P/L
86 PARRAMATTA ROAD
CAMPERDOWN 2050

ENVIRONMENTAL NOISE CONTROL P/L
50 RIVERSIDE ROAD
CHIPPING NORTON 2170

G P EMBELTON & CO PTY LTD
147-149 BAKERS ROAD
COBURG 3058

INC CORPORATION PTY LTD
22 CLEELAND ROAD
OAKLEIGH SOUTH 3167

JOYCE FORM PRODUCTS
5-9 BRIDGE ROAD
MOOREBANK 2170

NSW ENVIRONMENT PROTECTION AUTHORITY
LOCKED BAG 1502
BANKSTOWN 2200

NOISE CONTROL AUSTRALIA P/L
70 TENNYSON ROAD
MORTLAKE 2137

PEACE ENGINEERING PTY LTD
P.O. BOX 4160
MILPERRA 1891

PYROTEK PTY LTD
PO BOX 260
TOONGABBIE 2146

ROCLA COMPOSITE PRODUCTS
P.O. BOX 67
SPRINGVALE 3171

SAFETY EQUPMENT AUSTRALIA P/L
PRIVATE BAG 1001
MONA VALE 2103

TONTINE INDUSTRIES
238 MAROONDAH HIGHWAY
MOOROOLBARK 3138

VIPAC ENGINEERS AND SCIENTISTS LTD
275 NORMANBY ROAD
PORT MELBOURNE 3207

WORKCOVER ACOUSTICS
919 LONDONDERRY RD
LONDONDERRY 2753

From the President

A few months ago, all members received and were asked to comment on a draft copy of the proposed new Memorandum, Articles of Association and By-Laws for the Society. About a dozen detailed replies were received, most with very useful viewpoints. After consideration of these responses and obtaining additional legal opinion, a number of relatively minor changes have been made to clarify and enhance the proposed operating rules of the Society. For many years now, Council meetings have been held in association with the Society conference. However, this year Council met a month earlier so that it could discuss the revised draft. A few additional changes have subsequently been incorporated in the document which Council has approved for transmission to Society members and which, hopefully, will be accepted by them at the AGM to be held in Adelaide in December. It is Council's belief that the new Memorandum and Articles of Association do not significantly alter the thrust of the Society but will greatly facilitate its operation, especially as much of the detail is now in the form of By-Laws which future generations can adjust more readily if the need arises.

Another significant step taken by Council was the decision to accept an invitation for the Society to join the International Institute of Acoustics and Vibration (IIAV), which is the organisation behind the 5th International Congress on Sound and Vibration to be held in Adelaide in December. In the past there has been some friction between IIAV and two other major international acoustics groups, I-INCE who run Internoise, and ICA which organises a major Congress every three years, the next being in Seattle in June, 1998. Part of the problem has been the timing of their respective meetings, however, all bodies have now agreed in principle to avoid overlapping conferences. Our Society has long been affiliated with, and wishes to continue having strong links with, both I-INCE and ICA. However, Council decided it was the Society's role to support any significant group involving acoustics. Whether the world needs a third international body involved with acoustics and their associated conferences will be up to future participants to decide by their attendance at the conferences.

On a more personal note, I have enjoyed (at times) working on behalf of the Society as President. As I am about to vacate the chair, I would like to thank the Members of Council and in particular our General Secretary David Watkins for making the task manageable. May your future Presidents have more success than I in getting their Editorial messages in on time to the (very patient and helpful) people organising Acoustics Australia.

Charles Don

The Prediction of Structure-borne Noise Transmission in Ships Using Statistical Energy Analysis

Y. K. Tso
Department of Defence, Aeronautical and Maritime Research Laboratory - DSTO, Fishermens Bend, Victoria 3207, Australia.

and

C. H. Hansen
Department of Mechanical Engineering, The University of Adelaide, South Australia 5005, Australia.

ABSTRACT: This paper is concerned with the transmission of noise and vibration in ship structures, and in particular, naval ship structures. The first part of the paper presents a review of different methods for ship noise prediction. It then introduces the method of Statistical Energy Analysis (SEA) for the investigation of ship noise and vibration in the high frequency regions. Previous studies have shown that SEA is a useful tool for the analysis of vibration transmission in structures which consist mainly of plate elements. However, naval ship structures often involve more complex elements such as shells and beam stiffened plates. In the second part of the paper, two types of structure which are considered to be characteristic of naval ship constructions are identified. They are: a cylindrical shell coupled to an end plate; and a plate with periodic stiffeners coupled at right angles to a uniform plate. The Coupling Loss Factors (CLFs) of these two structures are evaluated, using travelling wave analysis, for SEA studies. Results from an experimental program confirm the validity of the formulation of CLFs.

1. INTRODUCTION

The study of noise and vibration in ships has received considerable attention in the past few decades as a result of stringent ship noise legislation introduced by many countries [1]. Such legislation aims to provide a safe and comfortable working environment for crew members by specifying a maximum allowable sound pressure level in various ship compartments. The ability for the designer to predict noise level in ship compartments at the design stage is therefore highly desirable and several empirical and analytical studies on ship noise prediction have been reported (see, for example, references [2]-[7]).

Naval surface ships and submarines require additional noise and vibration control measures to minimise the risk of detection and the interference with on-board equipment (for example, sonar and weapon systems). Furthermore, the extensive use of periodically stiffened plates and shells in naval ships increases the structural complexity for noise and vibration analysis. It has long been recognised [8] that vibration waves in a periodic structure can only propagate in certain frequency bands (pass bands) and this phenomenon has a significant effect on vibration transmission. Due to the complexity of naval ship structures and the high frequency range of interest (up to 20 kHz for torpedo homing devices), a deterministic analysis of all the resonant modes of vibration is usually impractical. A powerful tool for predicting the high frequency response of complex systems is Statistical Energy Analysis (SEA) which deals with the time-averaged and frequency-averaged (octave or 1/3 octave) flow of vibrational energy between elements of a complex system [9].

In this paper, a number of methods used for the study of noise transmission in ship structures are reviewed. In particular, recent developments in SEA modelling of complex structures characteristic of naval ships are discussed.

2. REVIEW OF SHIP NOISE PREDICTION METHODS

Due to the complexity of ship structures, it is clear that a rigorous analysis based on the 'classical' approach (for example, wave theory) is impractical. Nilsson ([4] and [10]) presented a simplified analytical method based on a grillage model which was made up of two parallel hull frames and the associated plate elements. He considered that the frames would act as wave guides for the transmission of vibration from the hull to the superstructure. The plate elements used in Nilsson's analysis were assumed to be uniform and this approach may not be suitable for the analysis of structures with horizontal stringers between the frames that are typical of naval ship structures. A further restriction of this method is that it is essentially a two-dimensional model and is not readily applicable to the general analysis of vibration transmission in ships.

The Finite Element Method (FEM) may be used to model the response of complex structures. However, in the frequency range of interest for structure-borne noise studies (i.e., up to the several kHz range), the number of elements required is generally too large for the practical analysis of a substantial part of a ship's structure, even with the help of modern computer technology and software packages. Furthermore, at high frequencies where the wavelength is much smaller than the overall dimensions of the structure, FEM would become very sensitive to the system parameters and may lead to incorrect prediction of the structural response. Hence FEM is normally restricted to the vibration analysis of ship structures at low frequencies.

A number of empirical studies (see, for example, references [2], [5] and [11]) have been reported for ship noise predictions. These studies were mainly based on measurements and data taken on board merchant ships. In general, empirical methods are valuable tools in the analysis of a generic type of ship, especially at the design stage where limited information is available. These methods become less attractive in situations where a detailed analysis is required on different types of ships (for example, naval surface ships and submarines).

Statistical Energy Analysis (SEA) is a framework of study for the forced response of systems, and is based on the power balance between individual elements of a system [9]. It provides a basis for the prediction of average vibration and noise levels in complex structures, particularly at high frequencies.

Sawley [12] demonstrated that SEA can be used successfully to investigate the noise transmission paths of a motor vessel. Ødegaard Jensen [3] studied the distribution of vibratory power in a 1:5 scale ship section and also investigated the effects of damping on vibration transmission. Good agreement between calculated and measured results was obtained for the lightly damped case but the agreement was poor for the heavily damped case and Jensen attributed the discrepancy to the effect of in-plane waves acting as flanking transmission paths for the vibratory power.

Other authors ([7], [13] and [14]) also reported on the application of SEA to the study of vibration transmission in ships. A more detailed treatment of this subject was given by Plunt [6] where he investigated the rear section of a cargo ship and found reasonable agreement with experimental results.

A common feature of the SEA studies reviewed so far is that the ship structures were modelled as an assembly of plate elements subjected to bending waves except for Plunt [6] where longitudinal waves were also considered.

Tratch [15] investigated the transmission of vibration in a 1:2.5 scale model of the machinery foundation of a ship bottom structure using SEA. He also modelled the structure as plate elements but considered all the possible wave types generated at the junction (i.e., bending, longitudinal and shear). Good agreement between calculated and experimental results was reported.

Naval ship structures often make use of shell elements coupled to various types of plate element (for example, a submarine hull/bulkhead coupled structure). The transmission of vibration through coupled cylinder/plate structures has been investigated by a number of researchers. Hwang and Pi [16] conducted an experimental investigation on a cylindrical shell welded onto a base plate and concluded that the SEA method was not capable of reaching any intelligent prediction of the coupling loss factor due to the strong interaction at the cylinder/plate interface. Blakemore et al. [17] studied a number of flange-connected cylindrical shells and found considerable discrepancy between measurement and SEA predictions. They attributed the discrepancy to internal acoustic coupling, non-equipartition of energy between modes in a cylindrical shell element and low modal overlap. Pollard [18] also investigated experimentally two cylinder/plate structures (one with a long thin cylinder and the other with a short squat cylinder) and found conflicting results although the short cylinder showed good agreement between the theoretical and experimental results. Recently, Schlesinger [19] presented a theoretical analysis of the transmission of vibration through a cylinder/plate coupled structure based on an arbitrary distribution of the wave energies in the radial, circumferential and longitudinal directions. The theory is supported by a limited amount of experimental data but further work is needed to show that this method satisfies the reciprocity requirement of SEA. Thus the study of cylinder/plate coupled structures using SEA has been less successful compared with plate/plate structures and further research effort is required to address this shortcoming.

Another type of structure often used in naval engineering constructions is a plate or shell element reinforced with periodic stiffeners. The application of conventional SEA through successive elements of this type of structure can significantly overestimate the transmission loss (see, for example, reference [17]). This is a matter of concern and has been the subject of criticism [20]. Clearly, the band pass nature of a periodic structure has to be considered in SEA modelling since it has a strong influence on the transmission of vibratory power.

Keane and Price [21] applied the theory of periodic structures to enhance a one-dimensional SEA model. They investigated a point spring coupled, multi-modal system and compared the results obtained from 'exact' modal analysis with the normal and enhanced SEA model. A significant improvement in results was obtained by using the enhanced SEA model rather than the normal model. However, the model studied by these authors was made up of highly idealised one-dimensional elements and therefore the analysis may not be readily applicable to ship structures such as hull plates and bulkheads. Langley [22] also studied the modal characteristics of periodic structures and derived modal density expressions for one- and two-dimensional structures. He further studied the forced response of a damped one-dimensional periodic structure based on vibratory energy flow and compared the effect of material damping with the effect of damping caused by structural irregularity on vibration attenuation [23]. On the subject of 'near' periodic structures, Langley [24] investigated the wave transmission through a randomly disordered one-dimensional periodic structure and discussed the occurrence of frequencies of perfect transmission.

From the preceding discussion, it can be concluded that SEA is a useful tool for the prediction of vibration transmission through complex built-up structures, especially in situations where the structure can be modelled as an assembly of plate elements. However, a number of areas have to be addressed before this method can be applied successfully to naval ship structures. Notably, the SEA modelling of cylinder/plate structures and coupled periodic structures. The following section outlines some of the recent research activities in SEA modelling conducted in the Aeronautical and Maritime Research Laboratory, Defence Science and Technology Organisation, as part of an effort to control the acoustic signatures of naval vessels.

3. STATISTICAL ENERGY ANALYSIS

In this method, a complex system is considered to be an ensemble average of a set of physically similar systems. The system is then sub-divided into a number of inter-connecting subsystems, usually at locations where the coupling between subsystems may be considered as 'weak' (for example, at structural discontinuities where incident waves are substantially reflected). The subsystems are then modelled as SEA elements, each consisting of a group of resonant modes of the same nature. For example, a uniform plate under bending and in-plane motions may be modelled as two SEA subsystems representing the resonant modes associated with these two types of motion respectively. The mean energy of the subsystems may be related to the input power by SEA parameters, known as modal densities, internal loss factors and coupling loss factors (CLFs), to form a set of linear, power balance equations. Solution of the power balance equations leads to the mean energy level (and hence the response) of the individual elements. The fundamental equations of SEA, as well as the basic theory and assumptions concerning the interaction between multi-mode subsystems, are given by Lyon [9]. In addition, review papers on this subject have been presented by Hodges and Woodhouse [25] and Fahy ([20] and [26]).

3.1 Cylinder/plate Coupled Structure

A number of researchers have modelled a ship structure as coupled plate elements and derived the CLFs on the assumption that the wave field in each plate element is diffuse ([6] and [15]). The concept of a diffuse wave field poses no difficulty for the modelling of isotropic elements like uniform flat plates but is less clear from an SEA point of view for non-isotropic elements like curved plates and cylinders. Langley [27] pointed out that the assumption of a diffuse wave field is equivalent to the equipartition of energy amongst the resonant modes for an isotropic element. He then derived the CLFs for structural junctions between curved plates based on the modal concept of equipartition of energy. The present authors have extended the modal concept to consider the modelling of submarine structures which consist of cylindrical elements.

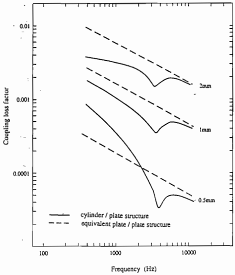


Figure 1. Coupling loss factors of three cylinder / plate structures with different shell thicknesses. The coupling loss factors of their equivalent plate / plate structure are also shown for comparison.

They derived the CLF between a cylindrical shell and an end plate ([28] and [29]) using travelling wave analysis. The derivation consists of the evaluation of transmission efficiency for the cylinder/plate junction (defined as the ratio between the transmitted wave power and the incident wave power) by considering the appropriate boundary conditions (i.e., the compatibility of displacements and the equilibrium of forces and moments at the junction). The transmission efficiency is then related to the CLF between the cylinder and plate elements based on the assumption of equipartition of energy amongst all the resonant modes of the cylinder. In the present study, the plate is assumed to have a hole cut out to accept the cylinder. This arrangement enables the results to be compared with those of an equivalent plate/plate structure in order to confirm the validity of the present formulation of CLF at high frequencies where the cylinder behaves as a flat plate. The equivalent plate/plate structure consists of two flat plates coupled at right angles to each other with the coupling line length equal to that of the cylinder/plate structure. The areas and thicknesses of the flat plates are equal to those of their respective elements of the cylinder/plate structure.

Calculations were performed to evaluate the CLFs of three steel cylinders each coupled to a 2 mm thick steel end plate. The shell thicknesses of the three cylinders are 0.5, 1.0 and 2.0 mm respectively. The length and mean diameter of all cylinders are 0.8 m and 0.45 m respectively. Figure 1 shows the CLFs of the three cylinder/plate structures. The CLFs of their corresponding equivalent plate/plate structure based on a diffuse bending wave field are also plotted in the figure for comparison. It can be seen from Figure 1 that all of the cylinder/plate structures show a dip in the CLF at around the

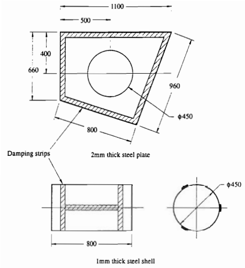


Figure 2. Cylinder and plate elements.

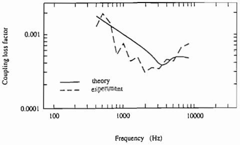


Figure 3. Coupling loss factor of cylinder / plate structure.

ring frequency of 3730 Hz, presumably caused by the increase in modal density of the cylinder around the ring frequency region. Thereafter, the CLFs asymptote to the values of their equivalent plate/plate structures as the frequency increases. This finding is consistent with the well established fact that the response of a cylinder may be approximated by a flat plate at high frequencies. Below the ring frequency, the response of a cylinder is dominated by the membrane effects and as a result, the CLFs of the cylinder/plate structures differ considerably from their equivalent plate/plate structures.

Experiments were conducted to measure the CLF of an example cylinder/plate structure to confirm the validity of the theoretical model [29]. The test structure (see Figure 2) consisted of a thin steel cylinder of 1 mm thickness and 0.45 m diameter coupled to a steel end plate of 2 mm thickness. It can be seen in Figure 3 that the experimental results are fairly well predicted by the theory. The dip in CLF which is predicted in the theoretical analysis can be observed in the experimental data. It occurs at a frequency of around 2500 Hz compared with a predicted value of 3730 Hz which corresponds to the ring frequency of the cylinder. The experimental data also show some discrepancy with the predicted CLF above a frequency 6300 Hz. An attempt to conduct further tests (above 8000 Hz) to confirm the convergence of the experimental results to the theoretical CLF of a plate/plate structure was hampered by the limitation in sampling rate of the data acquisition system. However, further examination of the results reveals that the discrepancy is consistent with previous work on the experimental investigation of CLF (see, for example, references [30] and [31]) and may be partially attributed to the random nature of the experiment and the assumptions involved in the analysis (for example, the equipartition of energy amongst circumferential modes).

3.2 Coupled Periodic Structure

Periodic structures are used extensively in naval ship constructions where relatively lightweight uniform plates or shells are reinforced by the attachment of stiffeners at regular intervals. It is well known [8] that a periodic structure freely transmits vibration waves in certain frequency bands (pass bands) and attenuates waves in other frequency bands (stop bands). This band pass nature has a strong influence on the transmission of noise and vibration through naval ship structures.

The SEA modelling of one-dimensional periodic structures has been considered by Keane and Price [21] by using a probability density function to model the band pass nature of the periodic structure. However, as mentioned earlier in Section 2, this approach is not readily applicable to ship structures since it is based on highly idealised one-dimensional elements. In the present study, the emphasis is focused on the application of wave transmission analysis to evaluate the CLF of coupled periodic structures which consist of two-dimensional elements (such as plates with periodic stiffeners). To this end, the authors have applied the standard travelling wave analysis procedure to evaluate the CLF, with the proviso that wave transmission is not permitted in the attenuation zones [32]. This approach allows the salient characteristic of a two-dimensional periodic structure (i.e., the existence of propagation and attenuation zones) to be incorporated into the standard CLF formulation.

A steel structure which consists of a plate with periodic stiffeners coupled at right angles to a uniform plate (see in Figure 4) is considered here as an example. The coupling line length of the structure is 0.7 m. Both plates are rectangular in shape with overall dimensions of 0.7 m × 1 m and 0.7 m × 1.2 m for the uniform plate and the plate with periodic stiffeners respectively. The thickness of both plates is 2 mm and the stiffeners are 6 mm × 14 mm rectangular sections spaced at 100 mm apart.

The CLF between the uniform plate and the plate with periodic stiffeners was calculated according to the procedures outlined in reference [32]. Experiments were also conducted on this example structure to measure the CLF. Figure 5 shows a comparison between the theoretical and experimental

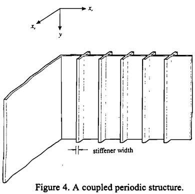


Figure 4. A coupled periodic structure.

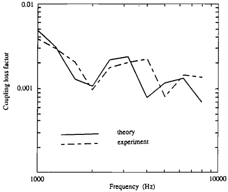


Figure 5. Coupling loss factor between a uniform plate and a plate with periodic stiffeners.

results. The results are in very good agreement in the low frequency bands. At higher frequencies, the theoretical results appear to have shifted by one-third of an octave compared with the experimental data. The apparent shift in frequency may have been caused by the assumptions and simplifications involved in the theoretical analysis. For example, in the analysis of the wave transmission properties of the plate with periodic stiffeners, it is assumed that the boundary conditions can be applied on the plate/stiffener centreline. However, the plate/stiffener attachment point is in fact offset from the stiffener centreline by an amount equal to half the stiffener width and this has an effect on the accuracy of the theoretical model, especially at high frequencies where the cross sectional dimensions of the stiffener are not negligible compared with the bending wavelength. Also, the offset of the attachment point from the stiffener centreline means that the bending wave will travel in the plate elements a distance (in the x_r-direction, see Figure 4) equal to the stiffener spacing minus the stiffener width rather than the stiffener spacing as used in the theoretical model. Overall, the experimental results for the present example are reasonably well predicted by the theoretical model.

4. CONCLUSIONS

Following a review of different methods applicable to the investigation of ship noise, it is concluded that SEA is a useful tool for the high frequency noise and vibration analysis of ship structures. The SEA method has been further developed by the authors, using travelling wave analysis, for naval ship applications and the theoretical values of CLFs of two example structures have been presented here. Experimental results for both structures show good agreement with theoretical predictions and it is therefore suggested that the present formulation of CLFs using travelling wave analysis may be used for SEA studies of structures characteristic of naval ships. However, mention must be made of a number of areas that require further study before the method of SEA can be applied successfully to more realistic naval ship structures. For example, in the analysis of coupled periodic structures, the present work has only considered bending waves to simplify the analysis. In reality, the stiffeners of ship structures are often offset to one side of the plate and thus generate in-plane waves which may have a significant effect on vibration transmission. Also, the coupling between other types of periodic structures typical of naval ship constructions such as ring stiffened cylindrical shells requires further investigation. Finally, the effect of fluid loading on SEA elements has to be considered in the analysis of ship noise.

REFERENCES

1. R. B. Conn, Acoustics and ship design. *Proc. Int. Symp. on Shipboard Acoustics,* Elsevier (1976)
2. J. H. Janssen and J. Buiten, On acoustic designing in naval architecture. *Proceedings, INTER-NOISE 73, Copenhagen* (1973)
3. J. Ødegaard Jensen, Calculation of structure-borne noise transmission in ships, using the statistical energy analysis approach. *Proc. Int. Symp. on Shipboard Acoustics,* Elsevier (1976)
4. A. C. Nilsson, Attenuation of structure-borne sound in superstructures on ships. *J. Sound and Vibration* **55**, 71-91 (1977)
5. J. Plunt, Method for predicting noise levels in ships, part 1: noise level prediction methods for ships, based on empirical data. *Report 80-06, Chalmers Uni. Tech. Göteborg, Sweden* (1980)
6. J. Plunt, Method for predicting noise levels in ships, part 2: prediction of structure-borne sound transmission in complex structures with the SEA method. *Report 80 - 07, Chalmers Uni. Techn. Göteborg, Sweden* (1980)
7. P. Hynnä, P. Klinge and J. Vuoksinen, Prediction of structure-borne sound transmission in large welded ship structures using statistical energy analysis. *J. Sound Vib.* **180**, 583-607 (1995)
8. L. Brillouin, *Wave propagation in periodic structures.* New York: Dover (1946)
9. R. H. Lyon, *Statistical energy analysis of dynamical systems: theory and applications.* MIT Press, Cambridge, Mass. (1975)
10. A. C. Nilssen, Reduction of structure-borne sound in simple ship structures: results of model tests. *J. Sound Vibr.* **61**, 45-60 (1978)
11. J. Buiten, Experiments with structure-borne sound transmission in ships. *Proc. Int. Symp. on Shipboard Acoustics,* Elsevier (1976)

12. R. J. Sawley, The evaluation of a shipboard noise and vibration problem using statistical energy analysis. *ASME Symposium on Stochastic Process in Dynamical Problems, Los Angeles* (1969)
13. Y. Irie and S. Takagi, Structure-borne noise transmission in steel structures like a ship. *Proc. INTER-NOISE 78, San Franisco* (1978)
14. K. Fukuzawa and C. Yasuda, Studies on structure-borne sound in ship. *Proc. INTER-NOISE 79, Warszawa* (1979)
15. J. Tratch, Vibration transmission through machinery foundation and ship bottom structure. *M. S. thesis, M.I.T.* (1985)
16. C. Hwang and W. S. Pi, Investigation of vibration energy transfer in connected structures. *NASA CR - 124456* (1973)
17. M. Blakemore, R. J. M. Myers and J. Woodhouse, The statistical energy analysis of a cylindrical structure. *Proc. Second International Congress on Recent Developments in Air- and Structure-borne Sound and Vibration, Auburn University, U.S.A.* (1992)
18. J. S. Pollard, Measurement of cylinder/plate coupling loss factors and associated problems. *Noise Control Eng.* **39**, 109-117 (1992)
19. A. Schlesinger, Transmission of elastic waves from a cylinder to an attached flat plate. *J. Sound Vibr.* **186**, 761-780 (1995)
20. F. J. Fahy, Statistical energy analysis: a critical overview. *Phil. Trans. Roy. Soc. London, A*, **346**, 431-447 (1994)
21. A. J. Keane and W. G. Price, Statistical energy analysis of periodic structures. *Proc. Roy. Soc. London, A*, **423**, 332-360 (1989)
22. R. S. Langley, On the modal density and energy flow characteristic of periodic structures. *J. Sound Vibr.* **172**, 491-511 (1994)
23. R. S. Langley, On the forced response of one-dimensional periodic structures: vibration localization by damping. *J. Sound Vibr.* **178**, 411-428 (1994)
24. R. S. Langley, Wave transmission through one-dimensional near periodic structures: optimum and random disorder. *J. Sound Vibr.* **188**, 717-743 (1995)
25. C. H. Hodges and J. Woodhouse, Theories of noise and vibration transmission in complex structures. *Rep. Prog. Phys.* **49**, 107-170 (1986)
26. F. J. Fahy, Statistical energy analysis - a critical review. *Shock and Vibration Digest* **6**, 14-33 (1974)
27. R. S. Langley, Elastic wave transmission coefficients and coupling loss factors for structural junctions between curved panels. *J. Sound Vibr.* **169**, 297-317 (1994)
28. Y. K. Tso and C. H. Hansen, Wave propagation through cylinder/plate junctions. *J. Sound Vibr.* **186**, 447-461 (1995)
29. Y. K. Tso and C. H. Hansen, An investigation of the coupling loss factor for a cylinder/plate structure. *J. Sound Vibr.* **199**, 629-643 (1997)
30. D. A. Bies and S. Hamid, In situ determination of loss and coupling loss factors by the power injection method. *J. Sound Vibr.* **70**, 187-204 (1980)
31. B. L. Clarkson and M. F. Ranky, On the measurement of coupling loss factor of structural connections. *J. Sound Vibr.* **94**, 249-261 (1984)
32. Y. K. Tso and C. H. Hansen, The transmission of vibration through a coupled periodic structure. *J. Sound Vibr.* (Submitted 1996)

Finite Element Analysis and Gong Acoustics

Neil McLachlan,
Department of Interior Design, RMIT
Melbourne, VIC 3001, Australia

ABSTRACT: Finite Element Analysis is used to predict the effect of a range of variations of gong geometries on modal frequencies. This data is evaluated in relation to experience in gong manufacture by a variety of methods and its implications for new instruments discussed.

1. INTRODUCTION

During this century advances in the fields of musicology, acoustics and human cognition have created new theoretical contexts in which European musical traditions may be interpreted alongside the musical traditions of many other cultures. In acoustics (now including musical and psycho-acoustics) these advances enable us to re-address questions of the relationships between instrumental timbre, musical form, and the perception of pitch, consonance and harmony.

For example, western orchestras evolved with the exclusion of instruments with non-harmonic overtones since it was thought they would interfere with the harmonic concerns of composers [1]. Through exposure to non-western instrumentation, electronic sound generation and sound recording technology composers are now exploring complex sound sources and instrumentation in compositions no longer structured by eighteenth and nineteenth century European harmonic concerns. While composers such as Harry Partch built entirely new instrument ensembles to explore such interests [2], others have been deeply involved in computer programming and electronics.

Finite element analysis (FEA) modelling has been applied to the design of novel idiophones for use within conventional European musical contexts [3-6]. For example, an entire carillon of bronze bells with major instead of minor third partials has been designed and cast [4-6]. Computer programs which physically model musical instruments through FEA modelling have recently been developed for electronic music synthesis [7,8]. These programs offer a range of models of physical systems such as stretched strings and membranes, wooden and metal bars, resonators and various excitation mechanisms. Novel, virtual instruments may then be generated for use in computer composition.

The instruments described in this paper embrace new musical possibilities by exploring the timbral implications of a range of gong geometries, inspired by instruments from diverse musical traditions, through FEA modelling. This is compared to acoustic spectra for instruments designed and manufactured by the author utilising various contemporary manufacturing technologies, for a range of novel performance, cultural and architectural contexts.

2. INSTRUMENT DESIGN AND ANALYSIS

Very little literature is available on the manufacture and acoustic behaviour of tuned gongs [9-14]. These instruments are features of traditional musical ensembles from Indo-China to Indonesia. They vary greatly in shape and may range in size from about 150 mm to greater than 1 metre in diameter [15]. Throughout South-East Asia musicians and craftspeople have manufactured instruments by whatever means were available, with most of their efforts remaining poorly documented. Manufacturing methods include casting or forging in various copper based alloys [16-18] or more recently (usually for economic reasons) forging in mild steel or fabrication from sheet steel. Metal spinning of sheet steel was successfully used by the author for the manufacture of a range of gongs for a set of outdoor installations.

In order to investigate which elements of shape are essential to producing certain relationships of vibrational overtones, a simple series of FEA experiments were carried out on gong shape models beginning with a flat disk. This data will be discussed with reference to direct experience with the manufacture of tuned gongs.

Acoustic spectra have been measured for gongs from sets of just-tuned cast bronze and spun steel gongs which were made recently in Melbourne without the aid of FEA modelling. Spectra for the bronze gongs vary substantially due to variation in shape and size (the set crosses three octaves), and to dimensional irregularities created during manufacture and whilst tuning by hand grinding. All the gongs had cylindrical rims for ease of manufacture.

Figure 1 shows the acoustic spectra recorded about 100 milliseconds after excitation of three small gongs of less than 300 mm diameter. The first two spectra are of gongs spun from 1.2 mm mild steel sheet, the second of which had a boss beaten into it to raise the fundamental frequency to a specific pitch (a boss is a raised hemispherical dome in the centre of the gong's surface). The third spectrum is of a gong which was cast with a boss in silica bronze. The fundamental frequency was lowered to the required pitch by thinning the gong's surface with a grinder.

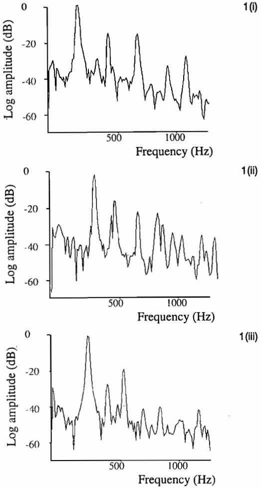


Figure 1. Acoustic spectra recorded 100 ms after excitation of:
i) Spun steel gong (286 mm surface diameter and 100 mm deep rim),
ii) Spun steel gong as above with 50 mm diameter, hemispherical boss,
iii) Cast silica bronze gong (225 mm diameter and approximately 2 mm thick surface, 55 mm deep and 5 mm thick rim, and 65 mm diameter and approximately 5 mm thick hemispherical boss).

The instruments were digitally recorded using the Macromedia Deck version 2.5 sound editing program and a Sennheiser MD 441 dynamic microphone. Excitation was by striking the instruments with padded mallets. The microphone was held above the top surface along the axis of symmetry of the gongs at a distance of about 500 mm. Short time Fourier transforms were performed by the AnnaLies version 4.2PPC program written by David Hirst and Thomas Stainsby for Macintosh computers at La Trobe University [19].

Table 1 includes frequencies of the first six major spectral peaks observed between 50 and 400 milliseconds after excitation with the ratios of these frequencies to the fundamental of each gong expressed numerically or as an octave equivilent just interval. The percentage deviation of the numerical from the just intervals is also given. The instruments were developed for just-tuned ensembles and their partials are described in this way to indicate the degree of consonance of their partials. Carillon bell partials are similarly related to intervals in Western musical scales. For comparison, the tempered major third is 3.5% sharper than the just interval 5/4, which is its closest consonant interval.

Table 1. Modal frequencies and ratios derived from acoustic spectra.

GONG		SPECTRAL PEAK						
		1	2	3	4	5	6	7
Flat	mode*	2,0	0,1	3,0	4,0	1,1	2,1	0,2
Steel	f (Hz)	252	422	498	622-662	738	984	1223
	f/f (1)	1	1.67	1.98	-	2.98	3.91	4.85
	just ratio	1	5/3	2/1	?	3/2	2/1	5/4
	% deviation	-	0	-1.0	-	-0.7	-2.3	+1.2
Steel	f (Hz)	370	540	723	878-925	1080	1380	-
with	f/f (1)	1	1.46	1.95	-	2.92	3.73	-
boss	just ratio	1	3/2	2/1	?	3/2	2/1	-
	% deviation	-	-2.0	-2.5	-	-2.6	-6.7	-
Bronze	mode	2,0/0,1	1,1	3,0	4,0	2,1	0,2	?
	f (Hz)	298	597	891	1110	1190	1404	1699
	f/f (1)	1	2.00	2.99	3.72	3.99	4.71	5.70
	just ratio	1	2/1	3/2	15/8	2/1	7/6	7/5
	% deviation	-	0	-0.3	-0.1	-0.3	+0.9	+1.5

* The assigning of modes is based on FEA modelling data presented later. The first number refers to the number of nodal lines, the second to the number of nodal rings of each mode.

The spectrum of the steel gong with boss was typical of gongs in this set. Their pleasing tonal qualities may be attributed to the closeness of the principal overtones to consonant intervals. The metal thickness and gong geometry was decided upon from experience in fabricating gongs from steel sheet. The cast bronze gong was chosen as an interesting example from a range of gong spectra. In other gongs of similar dimensions in this set the lowest two modal frequencies were close, causing occasional difficulties in pitch definition.

Suprisingly there is little difference between the spectral data for the two steel gongs shown in figure 1. Beating the boss into the gong raised the frequency of all the principal radiating modes by almost the same multiplier. Figure 2 shows plots of data obtained in FEA modelling experiments to explore the effects of adding bosses of various size and thickness to circular plates. In these experiments increasing boss sizes had the greatest effect on the 2,0 mode.

Examination of the data in table 1 does show a greater frequency increase in the 2,0 mode than the 0,1 mode when the boss is added. This results in a smaller just interval between the first two modes of this gong. The next four intervals are not greatly changed by the addition of the boss. Increases in internal tension in the metal surface due to the introduction of the boss were not accounted for in the FEA experiments. This added tension would increase modal frequencies, and comparing the spectral data in table 1 with the FEA data in figure 2 suggests that it is an important factor in the behaviour of these gongs.

FEA modelling was performed using the vibrational analysis package of Pro-Engineering's Mechanica Structures (version 13) program. Since the instruments being modelled have thin walls, the models were constructed as shells of prescribed thicknesses. Models used parameters for phosphor bronze (Young's modulus (Y) of 103 GPa, Poisson ratio (P) of 0.34 and density (D) of 8,900 kg/m³), or low alloy steel (Y=200 GPa , P=0.27 and D=7,800 kg/m³).

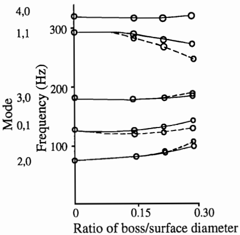


Figure 2. Plots of FEA predicted frequencies for various modes versus the ratios of the diameters of hemispherical bosses to the surface diameter of 1.2 mm thick mild steel circular plate models. Broken lines are plots of data for 2.4 mm thick bosses.

Doubling the thickness of the boss slightly raises the frequencies of modes with nodal diameters, but lowers the frequency of modes with nodal rings, including the 1,1 mode. This may be attributed to increased stiffness for the former modes and increased mass loading for the latter. A similar mass loading effect reported by Rossing [11] was proposed as the mechanism by which a boss could bring the first two modes with nodal rings into an octave relationship. As a boss is beaten into a steel gong the metal being worked thins and work hardens. This will have no effect on mass loadings but the stiffness will be effected in a complex way, since the thinning will reduce stiffness, but work hardening will increase it.

The present data shows that a boss of up to 30% of the surface diameter and twice its thickness has a relatively minor impact on the timbre of cast or spun gongs. In forged gongs beating out the boss pulls out any buckles in the surface and evenly thins it by stretching the metal. This may at first lower the fundamental frequency of the gong until the surface is uniform at which point the pitch will begin to increase with increasing surface tension and stiffness as described above. Bosses are an important feature of sets of tuned gongs in that they assist the maker to tune forged gongs and the player to strike the centre of the gong when playing fast passages.

Most gongs, whether of specific pitch or not, have rims. Data from FEA experiments are used in figure 3 to show the large impact rims have on the timbre of gongs without bosses.

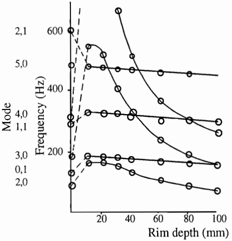


Figure 3. Plots of FEA predicted frequencies (in Hz) for various modes of 1.2 mm thick and 286 mm surface diameter mild steel gong models versus rim depth (in mm.).

The frequencies predicted for modes with nodal diameters only (2,0, 3,0, 4,0 etc.) increase dramatically with the introduction of even a small rim due to increased stiffness in the plane of vibration. As the rim size is increased, these frequencies quickly reach maxima before rapidly decreasing. When the rim depth is about 1/3 the size of the surface diameter (80 mm) they are close to the frequencies predicted for a freely vibrating circular disk. The 5,0 mode also behaves in this manner but is not shown in the figure for reasons of clarity and scale.

The introduction of a rim had less affect on the three modes with nodal circles shown in the figure (even though they may also contain nodal lines). Inspection of the FEA displacement contours for these modes revealed much smaller vibration amplitudes in the gong rims than was predicted for modes without nodal circles. Changes in the rim size therefore did not increase stiffness in regions of the gong which would affect the frequencies of these modes as much as the modes without nodal circles.

The data shown in figure 3 indicates that the two principal types of modes of vibration in gongs may be tuned independently of each other, and suggests rim to surface size ratios worthy of further investigation to produce musically interesting timbral results. Furthermore, modes with nodal diameters only were predicted to have their greatest displacement in the rim, so varying the metal thickness of the surface should have much less effect on them than modes with nodal rings. Figure 4 shows a plot of the FEA predicted frequencies of various modes for three different ratios of metal thickness in the surface compared to the rim.

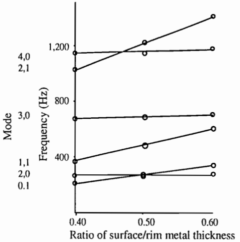


Figure 4. Plots of FEA predicted frequencies for various modes versus the ratio of metal thickness in the surface compared to the rim for phosphor bronze models based on the bronze gong described in figure 1.

Most Indonesian gongs have rims in the shape of inverted, truncated cones (see figure 5). From the preceding discussion it would be expected that increasing the angle of the rim from vertical would affect the modes with nodal diameters only more than those with nodal rings. This is confirmed by the data shown in figure 6. Predicted frequencies for modes with nodal diameters only increase sharply while frequencies for modes with nodal rings remain nearly constant with increasing rim angles.

Two types of rim shapes on gongs in the central Javanese gamelan are shown in figure 5. Rims of the second shape may be up to twice as deep as on comparably pitched gongs of the first. The second shape is found on the highest pitched gongs in the ensemble (in the top octaves of the bonang panerus and barung), which play the more complex elaborations of melodic material. Interestingly, it is also found on the highest pitched gongs usually used for defining rhythmic cycles in the music (kenong) [20]. Kenong are pitched within the same octave as the lower octave bonang barung gongs, and so the rim shape would appear to have an important role in creating timbral distinctions between gongs with the same pitch but differing musical function, and gongs with similar musical functions but tuned an octave apart.

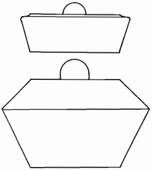


Figure 5. Two rim types found on central Javanese gamelan gongs.

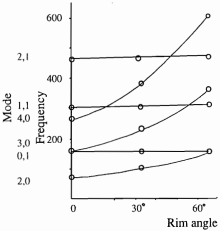


Figure 6. Plots of FEA predicted frequencies for various modes versus the angle from vertical of the rim on 1.2 mm thick, mild steel models with 0.35 rim depth to surface diameter ratio.

3. DISCUSSION

The predicted frequencies of the FEA experiments for gong models based on manufactured gongs did not match the acoustic spectra for these gongs due to various effects of the manufacturing processes which are difficult to accurately model. However when the results are taken in combination they indicate how the near harmonic overtone spectra recorded for these gongs have been produced by the right combination of physical properties. More experiments with actual cast and spun gongs will be necessary to precisely correlate computer models with the behaviour of gongs.

An important aspect of instrument design not addressed by modelling with FEA programs is the radiation efficiency of predicted vibrational modes. Antiphase source distributions will interact to reduce radiation efficiency if the sources are within about half of one wavelength, and such effects will occur to significant degrees for most vibrational modes in gongs. The front and back faces of the surface of a gong contain principal radiating regions emitting in antiphase [16], which are isolated to some extent by the rim. These effects are highly complex to predict and would require more detailed study with prototype instruments to fully understand. Clearly the air volume contained by the rim and the floor when the gongs are suspended horizontally is too large to be an efficient resonator.

The data presented in this paper was the result of personally funded research (generously assisted by a number of universities) aimed at developing a flexible design protocol for instruments to be used in a range of new musical, cultural and architectural contexts. Although significant advances have been made, more work will be necessary to accurately correlate computer predictions with physical instruments. This task will be assisted by the application of highly reproducible, modern manufacturing technologies in metal forming, casting and milling, and more sophisticated analytical methodologies.

4. ACKNOWLEDGMENTS

I would like to thank John Weir at the Department of Manufacturing and Mechanical Engineering, Melbourne University for his advice and assistance and the Design Net for access to the FEA modelling program and computers. Similarly , I would like to thank David Hirst and Jim Sosnin at La Trobe University Music Department for their assistance and access to Fourier transform programs, and RMIT University, Faculty the Constructed Environment for technical support. It is also important to thank the artists and collaborators of GongHouse for their passionate and generous contributions to the evolution of this work.

5. REFERENCES

1. H. Helmholtz, *On the Sensations of Tone as a Physiological Basis for the Theory of Music,* trans. Ellis A.J., Dover Puplications Inc., New York, (1954).
2. H. Partch, *Genesis of a Music,* 2nd ed., Da Capo Press, New York, (1974).
3. N.H. Fletcher, "Tuning a Pentangle-A New Musical Vibrating Element", *Applied Acoustics,* **39**, 145-163 (1993).
4. A. Schoofs, F. Van Aspern, P. Maas and A. Lehr, "A Carillon of Major-Third Bells, I. Computation of Bell Profiles Using Structural Optimization", *Music Perception,* **4** (3), 245-254 (1987).
5. A.J.M. Houtsma and H.J.G.M. Tholen, "A Carillon of Major-Third Bells, II. A Perceptual Evaluation", *Music Perception,* **4** (3), 255-266 (1987).
6. A. Lehr, "A Carillon of Major-Third Bells, III. From Theory to Practice", *Music Perception,* **4** (3), 267-280 (1987).
7. J.D. Morrison and J. Adrien, "MOSAIC; A Framework for Modal Synthesis", *Computer Music Journal,* 17 (1), 45-56 (1993).
8. P. Djoharian, "Generating Models for Modal Synthesis", *Computer Music Journal,* **17** (1), 57-65 (1993).
9. N.H. Fletcher and T.D. Rossing, *The Physics of Musical Instuments,* Springer-Verlag, New York, (1991), and references cited therein.
10. T.D. Rossing and R.W. Peterson, "Vibrations of Plates, Gongs, and Cymbals", *Percussive Notes,* **19** (3), 31-41 (1982).
11. T.D. Rossing and R.B. Shepherd, "Acoustics of Gamelan Instruments", *Percussive Notes,* **19** (3), 73-83 (1982).
12. T.D. Rossing and N.H. Fletcher, "Acoustics of a Tamtam", *Bull. Australian Acoustical Soc.,* **10** (1), 21-25 (1982).
13. M. Campbell and C. Greated, *The Musicians Guide to Acoustics,* J. M. Dent & Sons Ltd., London, (1987) pp. 450-452.
14. P. Adler, "Daniel W. Schmidt: Composer and Instrument Builder", *Percussive Notes,* **23:40** (2), 40-43 (1990-91).
15. S. Sadie, ed., *The New Grove Dictionary of Music and Musicians,* Macmillan Publishers Ltd., London, (1980), Vol. 9, pp. 167-216.
16. P. Croset, trans. J. Olsson, "The Making of Bronze Musical Instruments in Indonesia", *Percussive Notes,* **25** (3), 43-54 (1987).
17. P. Vandermeer, "Gongs and Gong-Making in Java", *Percussive Notes,* **23:40** (2), 48-53 (1990-91).
18. A. Toth, "The Manufacture of Gongs in Semerang", *Indonesia,* 19, 127-172, (1975).
19. D.J.G. Hirst, *Digital Sound Analysis and Synthesis Using Short-Time Fourier Transform,* M.A. Thesis, La Trobe University Music Department, Melbourne, Australia (1985).
20. M.J. Kartomi, *On Concepts and Classifications of Musical Instruments,* The University of Chicago Press, Ltd., London, (1990) pp. 95-99.

A Discussion of the Australian National Standard for Occupational Noise

J. H. Macrae,
Hearing Assessment Research
National Acoustic Laboratories
126 Greville Street, Chatswood NSW 2067, Australia

ABSTRACT: In 1974, the National Health and Medical Research Council warned that "the estimated risks associated with exposure to noise at levels of 85 and 90 dB(A) over a working life-time lead to an incidence of hearing loss in the working community which is unacceptable on medical grounds in the long term". Despite this warning, Australian legislatures set an 8-hour equivalent continuous A-weighted sound pressure level ($L_{Aeq,8h}$) of 90 dB(A) as the occupational noise exposure limit until recent years. Partly as a result of this, occupational noise-induced hearing loss has continued to be a highly prevalent industrial disease in Australia and the associated costs of compensation have escalated since 1988. In response to this situation, the National Occupational Health and Safety Commission in 1992 declared an $L_{Aeq,8h}$ of 85 dB(A) as the Australian National Standard for Occupational Noise and this is gradually being adopted by Australian legislatures. However, given the current magnitude of the problem, a stricter limit seems appropriate. A standard of 80 dB(A) would come much closer to an acceptable solution to the problem of occupational noise-induced hearing loss.

1. INTRODUCTION

It has been widely assumed that the goal of occupational noise management is prevention of noise-induced hearing disability in workers. The underlying assumption of this approach is that it is permissible to damage the hearing of workers as long as that damage does not result in associated disability, where hearing disability is defined, narrowly, as loss of the ability to understand speech for the purposes of everyday life. Impairment of the threshold sensitivity of an initially normal ear of about 20 dB is necessary before hearing disability in this sense begins to occur. If the goal of occupational noise management is only prevention of hearing disability, in the narrowly defined sense of disability, then noise-induced threshold impairment of 20 dB is permissible. However, it is likely that continuing research into hearing will reveal hearing disabilities, in the broad sense of loss of normal abilities to hear for the purposes of everyday life, are associated with impairment of hearing threshold sensitivity of 20 dB or less. For example, every 6 dB loss of hearing threshold sensitivity across frequency can be expected to halve the distance from which sounds can be heard, i.e., to result in contraction of the auditory horizon. A 20 dB loss of sensitivity across frequency can be expected to result in a 10-fold reduction in the distance of the auditory horizon.

The narrow approach to occupational noise management therefore does not go far enough in the direction of protection of the well-being of workers. The basic premise of this article is that the primary goal of occupational noise management should be prevention of noise-induced damage to the inner ears of workers. There is physiological evidence that inner ear damage caused by noise exposure accumulates prior to the onset of hearing threshold impairment [1], which itself accumulates prior to the onset of hearing disability. Objective assessment of the state of the outer hair cells of the inner ear can be made by measurement of otoacoustic emissions [2], which are sounds emitted from the inner ear after stimulation by external sound. Because noise-induced damage to the inner ear may precede the occurrence of threshold impairment, otoacoustic emission testing may provide a more sensitive indication of damage than audiometric thresholds and might eventually replace audiometry as a method of detecting noise-induced damage to the ear in occupational noise management programs [3]. However, further research and standardisation of otoacoustic emission measurement techniques are required before otoacoustic emission testing can be considered for this application [4]. In the meantime, audiometric testing of hearing threshold sensitivity will continue to be the preferred method of monitoring the status of the inner ear in the management of occupational noise exposure. At present, therefore, the practical goal of occupational noise management programs should be to prevent noise-induced hearing threshold impairment.

2. NHMRC MODEL REGULATIONS (1974)

In 1974, the National Health and Medical Research Council (NHMRC) published its Model Regulations for Hearing Conservation [5]. The NHMRC recommended that the daily 8-hour equivalent continuous A-weighted sound pressure level ($L_{Aeq,8h}$): (1) should not exceed 90 dB(A) for existing premises; (2) should not exceed 85 dB(A) for any premises at and after a period of 5 years from the time that the regulations were brought into effect; and (3) should not exceed 85 dB(A)

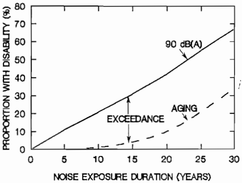


Figure 1. Estimated proportion of a population of otologically screened males with hearing disability, when the population is not exposed to harmful noise (effects of aging alone) and when the population is exposed to occupational noise with an $L_{Aeq,8h}$ of 90 dB(A), as a function of duration of noise exposure, in years. The difference between the two curves at any noise exposure duration is described as hearing disability exceedance. For the purposes of this graph, occupational noise exposure is assumed to begin at the age of 20 years.

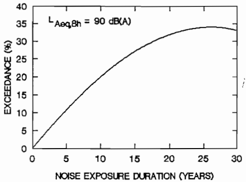


Figure 2. Hearing disability exceedance (as defined in the caption to Figure 1) for a population of otologically screened males exposed to an $L_{Aeq,8h}$ of 90 dB(A), as a function of duration of noise exposure, in years.

for any new premises after the time the regulations were brought into effect. The NHMRC warned that "the estimated risks associated with exposure to noise at levels of 85 and 90 dB(A) over a working life-time lead to an incidence of hearing loss in the working community which is unacceptable on medical grounds in the long term". The publication of these model regulations stimulated the development of actual hearing conservation regulations in the various Australian legislatures but, despite the warning concerning estimated risks, only recommendation (1) was brought into effect. By 1986, no Australian legislature had adopted recommendations (2) and (3) [6].

3. DISABILITY EXCEEDANCE

Partly because the maximum permissible 8-hour equivalent continuous A-weighted sound pressure level was set at 90 dB(A) in all jurisdictions, noise-induced hearing loss has continued to be a highly prevalent industrial disease in Australia [7], as would be expected from the NHMRC warning. The two curves presented in Figure 1 were derived from values given in a published table of the estimated prevalence of hearing disability in otologically screened, noise-exposed male populations [8], where otologically screened means free from all signs and symptoms of ear disease other than the effect of occupational noise exposure. The values in the table were calculated by means of equations given in International Standard ISO1999 [9] and the National Acoustic Laboratories (NAL) procedure for determining percentage loss of hearing [10]. In Australia, hearing disability for compensation purposes is quantified in terms of percentage loss of hearing, as determined by the NAL procedure. Hearing disability exists if the percentage loss of hearing is greater than zero. Some disability can be expected to occur in some workers not exposed to harmful levels of noise, as a result of the process of aging. This is represented by the lower of the two curves in the graph. For the purposes of the table and the graph, occupational noise exposure is assumed to begin at the age of 20 years. Thus, at the age of 50 years, about one-third of workers not exposed to harmful noise can be expected to have some hearing disability.

The higher of the two curves shows the proportion of workers who can be expected to have some hearing disability when they are exposed to noise with an $L_{Aeq,8h}$ of 90 dB(A). The difference between the two curves can be described as exceedance, where exceedance refers, in this context, to the amount by which the proportion of noise-exposed workers with hearing disability exceeds the proportion of workers who have hearing disability purely as a result of aging. Subtracting the curve for aging from the curve for 90 dB(A), the exceedance curve shown in Figure 2 is obtained. Reading this graph, after 25 years of exposure to noise with an $L_{Aeq,8h}$ of 90 dB(A), 34% of the exposed workers will have a hearing disability who would otherwise not have had any hearing disability. In view of the exceedance associated with an $L_{Aeq,8h}$ of 90 dB(A), it is not surprising that noise-induced hearing loss has continued to be a highly prevalent industrial disease in Australia.

Figure 3 shows that the cost of compensation claims for occupational noise-induced hearing loss in NSW grew from about 12 million dollars in 1988 to about 101 million dollars in 1996. Faced with escalating costs of this kind, the response of some rel(.ant statutory authorities and legislators has been to introduce thresholds of hearing loss, of the order of 5 - 7%, that must be exceeded in order for claimants to be eligible for compensation. Since a large proportion of compensation claims for noise-induced hearing loss are for losses of 5% or less, this means that the costs of these claims and the associated administrative costs are eliminated. However, although this eases the financial burden of compensation, it does nothing to solve the problem of occupational noise-induced loss of hearing among workers.

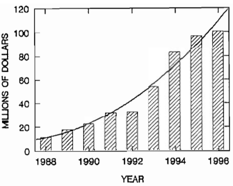


Figure 3. Cost, in millions of dollars per annum, of compensation claims for occupational noise-induced hearing loss in New South Wales from 1988 to 1996 (Source: Workcover Authority of NSW).

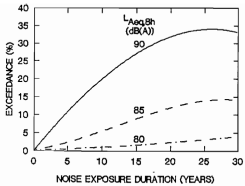


Figure 4. Hearing disability exceedance for a population of otologically screened males exposed to occupational noise with $L_{Aeq,8h}$ values of 80, 85 and 90 dB(A), as a function of duration of noise exposure, in years.

4. CURRENT NATIONAL STANDARD FOR OCCUPATIONAL NOISE

When the National Occupational Health and Safety Commission (NOHSC) was established in 1985, responsibility for setting occupational health standards passed from the NHMRC to the NOHSC. In response to its concern about the prevalence of occupational noise-induced hearing loss, the NOHSC formally declared the current Australian National Standard for Occupational Noise in 1992 and the National Code of Practice for Noise Management and Protection of Hearing at Work in 1993 [7]. The standard is an $L_{Aeq,8h}$ of 85 dB(A) and an unweighted (linear) peak sound pressure level, L_{peak}, of 140 dB. Like the original NHMRC model regulations, the National Standard and National Code of Practice are advisory documents but can be expected to affect regulations in the various Australian jurisdictions, as did the NHMRC model. By the end of 1996, the Commonwealth and most State and Territory governments had incorporated the National Standard in regulations and had either adopted the National Code of Practice verbatim or incorporated its principles in their own codes of practice [11].

However, does this National Standard for Occupational Noise go far enough in limiting the permissible noise exposure of workers? In 1974, the NHMRC warned that the estimated risks associated with exposure over a working life-time to noise at levels of 85 dB(A), as well as 90 dB(A), lead to an incidence of hearing loss in the working community which is unacceptable in the long term. In 1987, Macrae [6] pointed out, in an article which presented a table concerning the estimated incidence of hearing threshold impairment in noise-exposed populations, that an $L_{Aeq,8h}$ of 80 dB(A) comes closer to meeting the occupational noise management goal of preventing noise-induced hearing threshold impairment in the workforce than an $L_{Aeq,8h}$ of 85 dB(A). The table showed that, if noise-induced hearing threshold impairment at the most affected frequency, 4 kHz, is not to exceed 10 dB over a working life-time for 95% of the noise-exposed population, then noise exposure levels must be kept to not greater than 85 dB(A) but if noise-induced threshold impairment at 4 kHz is not to exceed 2 dB over a working life-time for 95% of the noise-exposed population, then noise exposure levels must be kept to not greater than 80 dB(A). The table also showed that, in order to obtain no noise-induced threshold impairment at any frequency, an $L_{Aeq,8h}$ of 75 dB(A) or less is necessary.

5. CONCLUDING REMARKS

The relative effectiveness of different noise exposure limits can also be evaluated in terms of hearing disability exceedance, as defined earlier in this article. Figure 4 shows the exceedance for noise exposure levels of 80, 85 and 90 dB(A). It is apparent that a noise exposure limit of 85 dB(A) will do little better than halve the problem. Given the current magnitude of the problem, a stricter limit seems appropriate. A standard of 80 dB(A) would come much closer to an acceptable solution to the problem of occupational noise-induced hearing loss. When data concerning occupational noise-induced damage to the inner ear obtained by means of otoacoustic emission testing become available, an even stricter noise exposure standard may seem appropriate. In the meantime, industries would be well advised to aim for a noise exposure limit, $L_{Aeq,8h}$, of 80 dB(A) rather than the National Standard value of 85 dB(A) and serious consideration should be given to reducing the National Standard noise exposure limit to an $L_{Aeq,8h}$ of 80 dB(A).

REFERENCES

1. R.A. Altschuler, R. Yehoash, C.A. Prosen, D.F. Dolan and D.B. Moody, "Acoustic stimulation and overstimulation in the cochlea: a comparison between basal and apical turns of the cochlea". In: A.L. Dancer, D. Henderson, R.J. Salvi and R.P. Hamernik (eds), *Noise-Induced Hearing Loss.* Mosby Year Book, St. Louis, 1992.

2. D.T. Kemp, "Cochlear echoes: implications for noise- induced hearing loss". In: R.P. Hamernik, D. Henderson, and R. Salvi (eds), *New Perspectives in Noise-Induced Hearing Loss.* Raven Press, New York, 1982.
3. E.L. LePage and N.M. Murray, "Click-evoked otoacoustic emissions: comparing emission strengths with pure tone audiometric thresholds", *Aust. J. Audiol.* **15**, 9-22 (1993).
4. J.H. Macrae, "The role of audiometry in occupational noise management", *J. Occup. Health Safety - Aust NZ* **12**, 571-576 (1996).
5. National Health and Medical Research Council. *Model Regulations for Hearing Conservation.* Australian Government Publishing Service, Canberra, 1974.
6. J.H. Macrae, "Hearing impairment and hearing conservation: an update", *J. Occup. Health Safety - Aust NZ* 3, 282-285 (1987).
7. National Occupational Health and Safety Commission, *Occupational Noise: National Standard and National Code of Practice.* Australian Government Publishing Service, Canberra, 1993.
8. J.H. Macrae, "Occupational hearing loss", *J. Occup. Health Safety - Aust NZ* **2**, 204-209 (1986).
9. International Standard ISO 1999, *Acoustics - Determination of occupational noise exposure and estimation of noise-induced hearing impairment.* International Organization for Standardization, Geneva, 1990.
10. J.H. Macrae, *Improved procedure for determining percentage loss of hearing.* NAL Report No. 118, National Acoustic Laboratories, Sydney, 1988.
11. R.L. Waugh, "Hearing is believing: Recent developments in Australian occupational noise policy", *Complete Safety Australia* 1: 28-31 (1997).

An Innovative Use of Hay Bales to Provide Ventilation Fan Noise Control

R T Benbow, Dick Benbow & Assoc,
Member firm Aust. Assoc. Acoustical Consultants

This article discusses an unusual method that was successfully used to provide a low cost effective means to reduce noise. The source of noise was emitted from ventilation fans during the construction of the sewerage tunnel through the Blue Mountains west of Sydney. The article is presented to demonstrate the use of an unusual solution which solves a short term environmental problem at significant cost savings to the community.

BACKGROUND

During the early 1990's a sewerage tunnel was constructed from Warrimoo through to Katoomba. The tunnel enabled sewerage from townships scattered through the upper Blue Mountains to be treated in a modern sewerage treatment plant with significant environmental advantages. The City of the Blue Mountains is unusual in that it is a city within a National Park.

The construction of the tunnel required the short term use of sites within close proximity to residences (30 - 150m). Ambient noise levels at night in the Blue Mountains are free of the traffic disturbances experienced in most urban areas and typically have background noise levels, L_{A90} of 30-35 dB(A).

The tunnel construction required centrifugal type ventilation fans to operate continuously. No excessive noise was being generated at the construction site near Faulconbridge and project engineers for the construction authority requested urgent technical assistance. An immediate solution was needed.

ACOUSTIC INVESTIGATION

Statistical noise level analysis was undertaken during the early hours of the morning to establish the background noise level in a similar residential area located away from the construction site. An L_{A90} of 35.5 dB(A) was measured. The fan outlet noise level at 7 metres was measured at 92 dB(A) with predominant octave band noise levels at 500 Hz. A combination of distance and directivity losses reduced the fan noise level at the worst affected residence to 44 dB(A). The fan noise was clearly audible and sufficiently tonal to cause extreme annoyance.

A solution was required before the following night otherwise construction would be forced to cease.

THE SOLUTION

It was clear that an attenuator was needed, but where do you obtain one on such short notice, deliver it to a site 80 kms from Sydney and have it installed before night fall?

An absorptive silencer would provide sufficient sound insertion loss. This triggered the idea of using hay bales. By early afternoon, a 5m long absorptive silencer was constructed using the bales as blocks to form a tunnel. The solution could be extended if further noise reduction was needed.

The solu'`on worked adequately achieving a 10 - 12 dB(A) noise reducuon and satisfying the residents concerns.

The next construction site was located at Woodford with the ventilation fan located within 30 metres of a residence. A shipping container was used to house the fan and a labyrinth was constructed, again from hay bales placed within the container so that discharge air passed through a series of bends. The outlet of the container was pointed away from the residence to gain noise reduction through directivity effects. Significant cost savings were achieved.

Australian Hearing Services

50 years of helping people hear

National Acoustic Laboratories

Australian Hearing Services (AHS) is celebrating 50 years of operation under various names (Commonwealth Acoustic Laboratories, National Acoustic Laboratories, Australian Hearing Services). With the most recent name change in 1992, the name National Acoustic Laboratories (NAL) was retained for the Research Division of AHS which includes the engineering, prevention services, and Acoustic Test Facilities sections as well as the research staff. The main areas of research concern hearing and hearing aids, and noise and hearing loss prevention. The Managing Director of AHS is Philip Bert and the Research Director of NAL is Denis Byrne. The following information, from a brochure prepared for 50th anniversary celebrations, summarises the history of AHS in "milestone" form:-

1947 Commonwealth Acoustic Laboratory (CAL) established by the Health Department in Sydney for hearing and noise research and to provide hearing services to school children and war veterans. CAL emerged from the Acoustic Research Laboratory, set up in 1943 to investigate problems in noise and communication in the defence services. First CAL Centre has a staff of four.

1949 Branch centres established in every Australian Capital City. The first hearing aid developed and produced by CAL was issued (simply known as the CALAID).

1954 1,120 hearing aids fitted. First visit to a NT aboriginal community (from the Adelaide Laboratory).

1955 Transistor hearing aids, the body-worn Calaid T, replace the cumbersome valve-type aids, requiring only one-tenth of the battery supply.

1960 Minister for Health approves extension of hearing services scheme to include all children up to the age of 21.

1961 Induction coils incorporated into Calaid T model allowing clients to use loop systems, and CAL assists many schools to install loops. All Commonwealth Hearing Compensation cases now seen by CAL.

1963 CAL Research now at Hickson Road at The Rocks, and the Hearing Centre to Grace Building, Sydney. CAL develops individually moulded ear protectors extensively used by the Defence Forces and industry.

1964 CAL now has 17 hearing centres, including 10 country centres. A regular visiting service commenced to the Northern Territory.

1965 Introduction of the Calaid E, an in-the-ear hearing aid suitable for children and adults with mild losses.

1967 CAL now has 115 full-time staff. 1,250 Calaid T's and 2,500 Calaid E's fitted this year.

1968 Commonwealth hearing services scheme extended to pensioners. 6,500 hearing aids fitted this year.

1972 CAL becomes National Acoustic Laboratories (NAL). A new testing procedure for infants (COR audiometry) introduced into the clinics using sophisticated equipment designed by NAL.

1973 100,000th Calaid issued and 250,000th new client tested this year. Binaural (two) hearing aids now fitted routinely to children after NAL research demonstrates the benefits.

1974 First BTE hearing aid developed by NAL, the Calaid H. Mobile Noise Evaluation Unit is acquired for major noise work in the field. NAL researchers develop procedure for selecting hearing aids which maximise intelligibility of speech. 17,700 hearing aids issued this year.

1975 High powered behind-the-ear hearing aids purchased to enable severely and profoundly deaf clients to benefit from the ear level model. NAL the only hearing service to children in the world fitting BTE aids routinely. 24,500 hearing aid fitted this year.

1978 NAL introduces the high powered behind-the-ear Calaid P, suitable for children and adults with severe and profound losses.

1982 New improved behind-the-ear range of hearing aids, designed and produced by NAL - the Calaid V. New Paediatric specialist audiologist positions improve services to children. Investigations into community reaction to aircraft noise results in ANEF criteria which are later adopted by the Australian Government.

1983 Number of Hearing Centres is now 26. 42,000 hearing aids fitted this year.

1985 Program to routinely fit vibrotactile aids (which convert sounds into vibrations) was introduced for profoundly deaf clients. Rehabilitation specialist positions created to strengthen adult rehabilitation services.

1986 NAL moved from The Rocks to new purpose-built premises at Chatswood. New BTE output compression hearing aid, the VLK, offers greatly improved sound quality. Unique Calaid FM wireless system now available for children. NAL aid selection procedure further modified and improved.

1988 New technique known as 'insertion gain measurement' introduced to improve hearing aid fitting evaluation. The number of Hearing Centres climbed to 35, with 65 visiting sites.

1990 Unique Sound Field Amplification system trialed in an Alice Springs Aboriginal community school, resulting in improved listening conditions for children with mild hearing loss.

1991 NAL aid selection procedure incorporates adjustments for severe and profound losses. Joint venture with Bernafon commenced to design programmable hearing aids.

1992 AHS becomes a Statutory Authority. NAL becomes Australian Hearing Services (AHS), with "NAL" retained for Research Division. AHS enters agreement to supply repairs and spare parts for clients with Cochlear Implants. NAL becomes a core participant in the Co-operative Research Centre for Cochlear Implant, Speech and Hearing Research.

1993 Programmable in-the-ear hearing aid, the IT312, and remote controls are introduced, and AHS becomes the only Government service offering such sophisticated technology as part of a standard service, anywhere in the world. AHS services extended to part-pensioners.

1994 Medium powered BTE, the SB13, added to the range of programmable products. As a result from investigation into community response to impulse noise from large calibre weapons and explosions, NAL develops criterion for Department of Defence.

1995 High powered programmable hearing aid, the PB675, introduced and large numbers exported by Bernafon. Memorandum of Understanding signed with Office of Aboriginal and Torres Strait Islander Health Services to train health workers and provide audiometric equipment.

1996 AHS, in collaboration with Macquarie University, establishes the first School of Audiology in Beijing, China. With several million hearing impaired people, there is a great need for audiologists in China.

1997 AHS celebrates 50 years of commitment and service to the Australian community. Number of Hearing Centres is now 60, with over 200 visiting sites, and 700 staff. 100,000 hearing aids fitted this year.

AHS is still undergoing changes. From 1 November 1997 adults who are eligible for government Hearing Services may choose to obtain those services from any of a number of accredited providers, including AHS. AHS will remain the sole provider of services to children and some other specified groups of clients. NAL's research activities are not directly affected by the change; they are funded as a community obligation. To celebrate 50 years of research (in fact a few years more, counting the work of the Acoustic Research Laboratory) NAL has compiled a complete set of research publications (about 620) into sets of bound volumes which have been presented to the National Library and to universities providing courses in acoustics or audiology.

News...

AAS Web Pages

Thanks to the hard work of Carl Howard from South Australia, the www pages for the Society and its various activities have been improved in presentation style. Carl has introduced a number of smart features. There is still some work to be done to bring all the pages to the same quality but Carl has provided the impetus to initiate that additional work. The Web pages should be updated regularly to correct errors and ensure that they are current so please let us know if there are any corrections etc needed. We also need more information on links to other appropriate pages. You can see the main page for the AAS at http://www.adfa.oz.au/~mxb/aas

Noise Effects 98

Noise Effects 98 will take place in Sydney on 22 to 26 November 1998. It is the 7th International Congress in the series on Noise as a Public Health Problem, often referred to as ICBEN. These conferences are only held every five years and this is the first to be held in the Southern Hemisphere. It offers a unique opportunity to participate in a conference that will deal with a range of the effects of noise on people and animals. The key speakers will be internationally acknowledged experts in their fields. This conference will be of interest to all those involved with any aspect of the effects of noise.

The scientific program will include invited and submitted oral presentations, posters and workshops in the following topic areas:

- Noise-induced hearing loss
- Noise and communication
- Non-auditory physiological & health effects induced by noise
- Influence of noise on performance and behaviour
- Effects of noise on sleep
- Community response to noise
- Noise and animals
- Combined agents
- Implications for regulations and standards

The call for papers is now available.

Further Information:
http://www.acay.com.au/~dstuckey/noise-effects98 or from Noise Effects '98, GPO Box 128, Sydney NSW 2001, tel +61 2 9262 2277, fax +61 2 9262 2277, or noise98@tourhosts.com.au

Internoise 98

Internoise 98, the 1998 International Congress on Noise Control Engineering, will be held in Christchurch, New Zealand November 16 - 18, 1998. The theme of INTER-NOISE 98 is "SOUND AND SILENCE: SETTING THE BALANCE". The conference is sponsored by I/INCE, the International Institute of Noise Control Engineering, and is being organised by the New Zealand Acoustical Society. The technical programme will provide for the presentation of posters and both invited and contributed papers with as many sessions in parallel as needed to accommodate the topics offered. Distinguished Lectures will be given by Dr Leo L Beranek, and Professors Jeremy Astley, Christopher Rice and Colin Hansen as plenary sessions. Topics will be grouped with a Keynote Paper invited for each session.

The call for papers is now available. Technical Papers in all areas of noise control engineering will be considered for presentation at the Congress.

A Satellite Symposium on "RECREATIONAL NOISE" linking the themes of these two conferences is planned for 20 November in Queenstown, New Zealand. Structured sessions are planned on: Noise Control in National Parks, Hearing Conservation for Tour Operators, Watercraft and other Recreational Vehicles, and Noise in and from Entertainment Centers. Other topics, such as the effect of sound in aerobics classes and noise in and from gun clubs, may be included.

Further Information:
http://www.auckland.ac.nz/internoise98 or from INTER-NOISE 98, NZ Acoustical Society, PO Box 1181, Auckland 1001, New Zealand, Tel: +64 9 623 3147, Fax: +64 9 623 3248, internoise98@auckland.ac.nz

VIPAC Award

The 1997 Excellence Gold Award, organised by the Australasian Society of Automotive Engineers, has been won by Vipac engineers and Scientists for their submission 'AutoSEA - Computer Aided Engineering for the Prediction of Noise & Vibration"

This high quality software was developed by Vipac and its subsidiary Acoustics Sciences Ltd, uses statistical energy analysis to approach noise and vibration reduction in design. The potential application of this product crosses many industry boundaries and has been sold to organisations in 16 countries.

Stolen Equipment

On Wednesday 3 September 1997 a metal equipment case was stolen from the concourse of Perth Domestic Airport. The case contained a Bruel & Kjaer Sound Level Meter Type 2231 S/N 1323299, a Bruel & Kjaer Field Calibrator Type 4230 S/N 830130, a Sony DAT tape recorder and sundry other items. If these items are presented for sale, valuation etc it would be appreciated if you would contact Erik Fry at Pierce Calibration Laboratory, Tel: 0412 945 777 Fax: 08 9313 2924.

Catching Illegal Copying

Early in 1998 the Copyright Agency Limited, (CAL) will be launching a campaign to catch illegal copying in the corporate and business environment. It is the first time in Australia that such a campaign will target the copying of printed materials in the workplace.

A recent survey of copying practices in the workplace suggests that over 70% of people who copy print material at work never obtain the copyright owners' permission. What this means for author and publisher members of CAL is that, potentially, they could be missing out on receiving payment of the copying of their works if these businesses are not licensed. The experiences of overseas copyright collecting societies suggest that such payments could amount to millions of dollars.

Copyright in the corporate environment is becoming a hot topic. The Copyright Act 1968 does not provide Australian companies with any specific entitlement to reproduce copyright material without the permission of the copyright owner, their agent or exclusive licensee. The fact that a company purchases a book or journal does not usually give the company a right to reproduce it. Because copyright works gain their value from being made public, consumers have a tendency to assume no one owns them. If it's on the Internet , on the air, on the library bookshelf, in a magazine or in a newspaper in someone's office - it must be there to use as we like. This is such a common trespass that many people or corporations have ceased to consider it a breach of the law.

The 'CopyCatch' campaign aims to inform, educate and send a clear message to businesses that illegal copying requires urgent consideration within the legal framework of corporate Australia. The main aim, is to encourage corporations and businesses to obtain a CAL licence without having to resort to litigation. Initially, CAL will target corporations and businesses located in Sydney and Melbourne.

Further information can be obtain from CAL on 02 9394 7600.

From CALender September 1997.

Copyright and Databases

The World Intellectual Property Organisation (WIPO) has proposed a treaty on the copyright of databases. Scientists fear that the proposed treaty would restrict their access to information. The International Council of Scientific Unions has become increasingly alarmed by the potential adverse effects of the new laws on the conduct of science and education.

At issue is the protection of databases that require a substantial amount of time effort and money to produce but that lack creativity in the selection, arrangement or presentation of the information. The creative elements of databases are already protected under copyright.

The Federal Attorney-Generals Legal Practice, supported by the Academy of Science, held a workshop on the proposed treaty and implications at the Academy on 18 April 1997. Excerpts from two talks by scientists are in the Academy of Science Newsletter No 37, 1997. WIPO is due to consider the proposed treaty again in September 1997.

Impact Meeting

The Victoria Division AGM for 1997 was held on 1 October at the RMIT, in conjunction with a Technical meeting to inspect the Physics Dept reverberation rooms, and to hear Ken Cook discuss the method for testing the impact sound insulation of partitions as described in the National Building Code of Australia (BCA).

While AS 1191 describes a method of test in which the impact sounds generated at the partition are made with a standard tapping machine impacting directly on the partition, the BCA has included an extension of this test in which the standard tapping machine is placed, and made to operate, on a horizontal plate attached to the vertical partition.

This use of an addition plate attached to the partition introduced several possible inconsistencies in the testing procedure not accounted for in the Code (which is more a guide than a code): the impact plate may be either close to or away from studs; the impact region in relation to the partition is insufficiently described; a number of tapping locations on the impact plate, rather than only one, may well be required for consistent test results; insufficient detail is given in the code of the method for sampling the receiving room noise levels (including the sampling period); questions could be raised as to the merits of the testing rooms, which, even if they comply generally with AS 1191, may not necessarily satisfy the conditions for testing impact noise. ISO 140, Part 1 provides additional help with this.

Inconsistencies can also arise in interpreting the test results obtained using the Code impact testing method (assuming there are no measurement abnormalities). Interpretations involving comparison with TABLE F5.5 partitions raise questions of: allowing for the presence of one or more deficiencies in the partitions under test' the usefulness of a single number rating system (the IIC is only for floors); whether, for this impact noise testing, a frequency band of 50 to 1000 Hz would be preferable to 125 to 4000 Hz. In addition, the test results must take account of the type of impact, and of the subjective effect of intrusive impact noise in the receiving room (to ensure that acceptable standards of amenity are maintained for the benefit of the community).

Other impact testing procedures considered more representative of impacts found in practice were then suggested. The impact from the fall of a poly-carbonate rod on to the partition under test can be standarised because the energy of the fall is calculable, and the test results don't require normalisation. Other possible sources of impact noise on a partition derive from a plastic-headed hammer, from a standarised rubber ball, or from the impact of running water. Proper account, also needs to be taken of the characteristics of the two reverberation rooms of their separation, and of the presence of possible flanking paths.

In conclusion it was suggested that, with a suitable testing method, the field testing of partition constructed similarly to a specimen used in a laboratory test is definitely possible.

Louis Fouvy

◆ ◆ ◆

STANDARDS

From Standards Australia

Book Publishing

Consensus Books is a new service for authors of technical, business, quality and environmental works which might be of interest to a niche market sector. It has recently been launched by Standards Australia. It offers short-term, royalty-based contracts for authors to be published under the imprint of Consensus Books, and may help authors gain contracts with mainstream publishers. According to Howard Paul, General Manager of Standards Australia Publishing, the new service has been established to cater for books and papers covering topics which relate to Standards issues, but which have been authored outside the usual technical committee process.

Consensus Books offers all the usual publishing services including editing proofing and design and take advantage of the latest in electronic printing. The first books to be published under the Consensus Publications imprint is The Consequences of Quality by Dr Neil Hardie.

Details: Howard Paul Tel: 02 97464803 or Garry Lock Tel: 07 38318142.

Standards Australia - 75 years

The first meeting of Standards Australia was held on the 2nd and 3rd November 1922 and this year, the organisation celebrates its 75th Anniversary. From its beginnings as the Australian Commonwealth Engineering Standards Association, Standards Australia has grown to become one of the largest independent technical infrastructure organisations in Australia today. The real history of this organisation is made up of people. Not only staff but the vast army of experts serving as technical committee members. It is the ideals, vision, struggles, endeavours and commitment of these people, which over the past 75 years, have created this organisation and contributed to its legacy. **Professor Anita Lawrence** made history when she became the first woman appointed to the Standards Australia Council in 1980, and later, to the Executive Board in 1985.

Closer to heart are her memories of the AK/5 technical committee - the acoustics committee she helped form in the early 1970's now known as AV/5. It was here she made her mark as the first woman ever appointed Chairman of a technical committee.

While on the committee she worked with a gentleman named **H Viven Taylor,** who many considered the 'father' of acoustics in Australia. He was instrumental in forming the Committee and in bringing together acoustic experts who had, until that time, worked in scattered isolation throughout Australia.

SA and ABCB Cement Relations

Standards Australia and the Australia Building Codes Board (ABCB) have cemented relations in a Memorandum of Understanding which will result in improved integration between the Building Code of Australia and the Australian Standards it references.

It provides for greater coordination between the release of Standards and their call up in amendments to the BCA, which are issued twice a year. The MoU also allows for greater cross-representation in matters of common interest and the freer exchange of information between both organizations.

Standards Writing

Standards Australia has embarked on an ambitious project of reengineering its core business of Standards writing.

It will be carried out by four project teams:

- Executive Committee - project steering and sponsorship.
- Reengineeering - made up of staff and customers to develop new processes.
- Risk and benefits management team - to assess and refine progress.
- Technical support team - financial and technical support; change management.

Global Standards

In a perfect world the ultimate goal of a standardizer would be to have all the national Standards identical to the international Standards published by the International Organization for Standardisation (ISO) and the International Electrotechnical Commission (IEC). It is obvious that we do not live in such a world, nor is there a country that would be able to satisfy the above.

Nevertheless, the policy of Standards Australia and Standards New Zealand is to have Standards based on International Standards to the maximum extent feasible and to use the World Trade Organisation (WTO) Agreement on Technical Barriers to Trade (TBT Code, formerly known as the GATT Agreement) as a benchmark.

The immediate consequence is that Australian, New Zealand and Joint Australian/New Zealand Standards should be direct adoptions of International Standards as a matter of course, unless there are good reasons to the contrary.

ISO/IEC have identified three levels of international equivalence:

IDT (Identical with) - The expression "identical with" may be used when a national Standard is identical in technical content and fully corresponding in presentation to the international Standard.

EQV (Equivalent to) - The expression "equivalent to" may be used when the national Standard is equivalent in technical content to an international Standard but with minor editorial deviations.

NEQ (Based but not equivalent to) - The expression "based but not equivalent to" is used when he Standard is not equivalent in technical content to an international Standard although it is based on that Standard.

American Standards

The following American National Standards on Acoustics have recently been released: ANSI S1.15-1997/Part 1 Measurement Microphones Part 1: Specifications for Laboratory Standard Microphones.

ANSI S3.5-1997 Methods for Calculation of the Speech Intelligibility Index.

ANSI S3.46-1997 Methods of Measurement of Real-Ear Performance Characteristics of Hearing Aids.

ANSI S12.6-1997 Methods for Measuring the Real-Ear Attenuation of Hearing Protectors.

ANSI S12.43-1997 Methods for Measurement of Sound Emitted by Machinery and Equipment at Workstations and Other Specified Positions

ANSI S12.44-1997 Methods for Calculation of Sound Emitted by Machinery and Equipment at Workstations and other specified Positions from Sound Power Level.

Further Information: A. Brenig, Standards Manager, Acoustical Society of America, 120 Wall St, 32nd Floor New York, NY 10005-3993, USA, Tel: +1 212 2480373 Fax: +1 212 2480146 asastds@aip.org

Books...

Vibration Control of Active Structures

Andre Preumont

Kluwer Academic Publishers, 1997, pp 259, Hard cover, ISBN 0 7923 4392 1, Australian Distributor:DA Information Services, PO Box 163, Mitcham Vic 3132, tel 1800 338863, fax 03 9873 5679. Price $171

Vibration control of active structures is an emerging new field of research which comes under the general umbrella of mechatronics. It involves a number of sub-disciplines such as mechanical engineering, structural mechanics, control engineering, material science, electronic engineering and computer science. This book has been written by an established expert to introduce structural engineers to active vibration control. An active structure here refers to a structure in which a set of actuators is coupled to sensors by a controller.

The book is divided into eleven chapters. The reader is assumed to be familiar with the basic principles of structural dynamics and linear system theory including classical control methods. In chapter 1, the motivation for active control, the definition of smart structures and general control strategies in terms of feedback and feedforward control techniques are outlined. Some basic concepts of structural dynamics are given in chapter 2. This is then followed by the modelling of actuators, sensors and some smart structures in chapter 3. Chapters 4 to 8 are basically aimed at designing single input single output (SISO) compensators both in the frequency and time domains. Specifically, in chapter 4, control systems using collocated actuator/sensor pairs are discussed and their advantage over non-collocated systems in terms of stability is highlighted. The concept of active damping is introduced in chapter 5 and its implementation using displacement, velocity, acceleration or integral force feedback is discussed. The state space approach of modern control methods which is convenient to apply to multiple input multiple output (MIMO) systems is presented in chapter 6. The analysis and synthesis of SISO systems in the frequency domain is conducted in chapter 7. Chapter 8 introduces the concept of optimal control for MIMO systems. Following the discussions of controllability and observability in chapter 9 and stability in chapter 10, chapter 11 discusses the digital implementation of active control systems supplemented by a number of practical examples and experimental results. There is a list of good references at the end of each chapter. I have found the list of problems at the end of each chapter especially helpful in furthering the understanding of the material presented in the text.

The book is well structured and the author has managed to convey the essence of the topic in a methodical approach. Unfortunately it contains quite a number of typographical mistakes. This is a specialised text which will be useful for postgraduate students, mechanical and structural engineers, researchers and practitioners who are invovled in the control of active structures.

Joseph Lai

Active Control of Noise and Vibration

C Hansen and S Snyder

E & FN Spon, 1997, pp 1267, Hard cover, ISBN 0 419 19390 1, Australian Distributor: Jackaranda Wiley, PO Box 174, Nth Ryde NSW 2113 Tel 1800 022 852 Fax 02 9805 1597 Price A$205.

Research into the active control of sound and vibration has been hotly pursued by both academics and engineers in the past decade and has generated a lot of excitement about its potential applications. It is a multi-disciplinary topic which requires fundamental understanding of the principles of acoustics, vibration, signal processing, modern control theory and implementation with electronic hardware and software. While there are a number of books dealing with either active control of sound or vibration, this is the first book that treats active control of both sound and vibration together under a single framework. The authors are acknowledged experts in the field.

The book is divided into fifteen chapters covering almost 1300 pages. The first half of the book is devoted to the fundamental principles of acoustics, vibration and control theory while the second half is concerned with the practical implementation of these concepts for applications in various areas of interest.

A brief overview of the historical development of active control of sound and vibration and applications is given in Chapter 1. Chapter 2, which consists of almost 180 pages, introduces the fundamental principles of acoustics and vibration with an emphasis on structural acoustics. Spectral analysis using digital filtering and FFT are described in Chapter 3. Chapter 4 provides a good description of the theoretical and experimental aspects of modal analysis. In just over 100 pages, the essence of modern feedback control theory is covered in Chapter 5. The principles of feedforward control system design including FIR and IIR filters and various algorithms are discussed in Chapter 6.

The next 3 chapters, which consist of over 300 pages, deal with various applications of active control of sound. In Chapter 7, the active control of acoustic plane waves propagating in ducts including exhaust outlets is described. The control of higher order modes is also treated. This chapter concludes with applications to active headsets and hearing protectors. A rather thorough treatment of active control of sound radiation from vibrating surfaces is given in Chapter 8. The principles of active control of sound in enclosed spaces are introduced in Chapter 9. Various control strategies and mechanisms are discussed together with the influence of the distribution of control sources and error sensors. Practical examples of control of interior noise in aircraft and automobiles are given.

Chapters 10, 11 and 12 are concerned with the applications of active control of vibration. In Chapter 10, feedforward control of vibration in beams and semi-infinite plates is discussed while feedback control of structural vibration is treated in Chapter 11. Both feedforward and feedback control of vibration isolation of one structure from another are described in Chapter 12.

The last 3 chapters are concerned with the hardware implementation of a physical control system with a good discussion of various actuators (sound/vibration sources) and sensors. Ample references are included at the end of each chapter. There is also an appendix which provides a quick reference to some results of linear algebra used in the book.

While the field of active control of sound and vibration is changing rapidly, this book contains some of the very latest results in the literature. Both the theoretical and practical aspects have been well covered. The authors have treated each topic with care and the book is a delight to read. The authors have done an excellent job in bringing together the various sub-disciplines in active control of sound and vibration. The only drawback of the book is that the publisher should have paid more attention to the quality of the reproduction of some of the diagrams. This is an outstanding book which I would highly recommend to postgraduate students, researchers, engineers and practitioners in the field of active control of sound and vibration. It is an essential addition to any physics and engineering library.

Joseph Lai

Joseph Lai is an Associate Professor in the School of Aerospace and Mechanical Engineering at Australian Defence Force Academy, Canberra. In his work associated with the activities of the Acoustics and Vibration Unit he has considerable experience in many aspects of active control.

Dictionary of Acoustics, Noise and Vibration Terminology

David Eager

University of Technology Sydney, 1997, pp105, soft cover, ISBN 1 86365 407 0. Distributor University Co-operative Bookshops throughout Australia. Price $8.95

This Dictionary evolved from information compiled by the author while he was undertaking postgraduate studies in acoustics. His studies were on noise and vibration control with the aim of reducing occupational noise exposure. Although he has subsequently built upon this information, the bulk of the definitions of acoustical terms are in these areas. The layout is clear with each of the terms in upper case followed by the explanation, which is mostly contained within 2 or 3 sentences. Cross referencing for pertinent terms is indicated with italics.

The first thing to note is that there are very few symbols and equations are rare. The author has attempted to explain many complex concepts using as much non-technical jargon as possible. This approach of course has limitations but then a dictionary should only be an explanation of terms with the fuller implications of the terms covered in text and reference books. The dictionary is sure to be appealing to students and those embarking on work in the acoustics area. The title is perhaps a little misleading as acoustics covers such a broad range and there are few terms from areas such as architectural and building acoustics.

This dictionary is certainly a worthwhile reference for those involved with engineering noise control and the low cost should encourage personal purchase.

Marion Burgess

Marion Burgess is a research officer with the Acoustics and Vibration Unit at the Australian Defence Force Academy in Canberra and has been involved in teaching, research and consulting for many aspects of acoustics.

Letters...

Noise Exposure for Construction Workers

We are currently undertaking a research project, sponsored by Workcover NSW, into improved noise management programs for workers in the construction industry.

Much of the work on the noise from construction sites has been driven by the reduction of the noise for the surrounding communities. There is little published data on the noise exposure of the workers themselves. In order to determine the extent of the problem it is necessary to have such data for the variety of workers on construction sites.

While measurements will be made as part of this study it would be of great benefit to obtain data from as many sources as possible. If there are any readers of this journal who have noise exposure data for workers on construction sites that they could make available for this study, would they please contact me at the address below. Any such data would be used in a "anonymous" manner to preserve confidentiality.

M Burgess, Acoustics and Vibration Unit, ADFA, Canberra ACT 2600,
Tel 02 6268 8241, Fax 02 6268 8276,
m-burgess@adfa.oz.au

Seeking Job

I am writing to you in the hope that one of your readers in the Adelaide area would know of a postition for an acoustical engineer. My educational background comprises degrees in Physics and Mechanical Engineeering and I have worked in the acoustics and vibration area as a consultant in Canada, Australia and the Asian region since my graduation in 1985. I have just completed three and a half years as Manager of R&D with a leading hi fi loudspeaker company in Adelaide. Before that I was involved with architectural and building acoustics, computer modelling of complex vibroacoustic systems and a great range of acoustical consulting jobs in South Australia, Interstate and internationally. Any interested parties are encouraged to contact me at the following address and telephone number.

John (Jack) Davis, 525 Kensington Road, Wattle Park S.A. 5066 Tel: 08-8332-0485.
scamp@senet.com.au

Seeking Job

I am pursuing PhD on the sound quality of vehicle engine noise and I will defense my PhD before summer next year. I have extensive knowledge in conducting research work in the field of sound quality or psychoacoustics. I have joined several conferences abroad and have published several papers in the international journals.

I am looking for a challenging and responsible position where my triangle educational knowledge (Psychology, Ergonomics and Acoustics) and interpersonal skills can be fully utilized from the perspective of human comfort, in a stable organization that will allow me to maximize my personal and professional growth.

If you need a hard working and an amicable person, please contact me at the following address.

M. Shafiquzzaman Khan, Acoustics Group, Division of Environment Technology Lulea University of Technology, S-971 87 Lulea, Sweden. Fax.+46 920 91030: saka@arb.luth.se
http://www.ludd.luth.se/users/saka

New Members...

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

NSW

Member	Mr R Hartog, Dr Burgemeister, Dr Madry, Mr J Baxter
Subscriber	Mr R Crandon, Mr C Field, Mr L Clark
Student	Mr P Conroy

SA

Member	Dr G Furnell
Subscriber	Mr R Koehler, Mr S Winn,

VIC

Member	Mr J Upton
Subscriber	Mr I Robinson, Mr P Pirozek, Mr P Kwong, Mr Debevc

QLD

Member	Mr D Dang

New Products...

VIBRO ACOUSTIC SERVICES

AutoSEA-X

AutoSEA, a high quality software program developed by Vibro-Acoustic Sciences uses Statistical Energy Analysis (SEA) to approach vibration problems. AutoSEA-X, a software package offering new ways to process test data and solve broadband noise and vibration problems, and offers three important new capabilities to the noise and vibration engineer: experimental modeling, analytical model validation and hybrid modeling. When used in conjunction with AutoSEA, the prediction of broadband noise and vibration is faster, simpler and more reliable. Time and money are saved and better products designed. AutoSEA and AutoSEA-X are complementary, bringing together analytical modelling and experimental modelling techniques.

Further information: Vipac Engineers & Scientists Ltd, 275 Normanby Road Port Melbourne VIC 3207 Tel: 03 96479700 Fax: 03 96464370 http://www.vasci.com

BRUEL & KJAER

Controller

Bruel & Kjaer have released a new 2237 Controller™ which measures sound and hand-arm vibration, providing a cost-effective easy-to-use solution for occupational health issues. To measure hand-arm vibration, the microphone and preamplifier are removed and the accelerometer and charge amplifier mounted. Once the power is switched on, the instrument becomes a Hand-Arm Vibration Meter complying with ISO 8041 and ISO5349. Measurements can be made with the standardised hand-arm frequency weighting (6.3-1250Hz) or unweighted up to 5 kHz. All relevant parameters are measured simultaneously. These include the A_{eq1} the maximum Peak and minimum and maximum levels. As a sound level meter, the 2237 Controller measures the L_{eq1} the maximum peak and the minimum and maximum levels. The A-weighted RMS levels and the C-weighted peak are measured simultaneously.

With its simple, concise control panel, pre-set measurement times and automatic storage, the 2237 Controller offers unrivalled ease-of-use, even for people lacking acoustic training. Software for downloading results to a spreadsheet is included.

Further information: Bruel & Kjaer on 1800 802 852

LMS

SEADS

SEADS (Statistical Energy Analysis Design System) is a state-of-the-art module that helps the engineer perform a predictive SEA analysis more easily, more robustly, and more productively than ever before.

Statistical Energy Analysis is the missing link in the vibro-acoustic design chain. It uses a statistically based model that is easier to use, simpler to understand and is much more adapted for high frequency applications. SEA predicts the flow of energy between the various components of a multi-assembly system. It smoothly handles the complex interactions between the structure and the fluid. Because it is so computationally efficient, it is ideal for interactive feasibility studies, sensitivity analysis and design optimisation.

The analysis is implemented in a three-step procedure. During the preprocessing phase, the model is constructed by means of dedicated preprocessing tools: a network display and menus. Next, the vibration or sound pressure levels are solved. Finally, in the post-processing phase, the results are visualized and the model optimised with respect to its vibro-acoustic behaviour.

Further Information: Compumod
Tel: 02 92832577 Fax: 02 92832585 http://www.compumod.com.au

ACEL

Engineering Yearbook

Australia's leading reference to the engineering industry has a new name. Engineering Yearbook is now called Australian Engineering Yearbook. The 27th Edition is now available for anyone who requires a fast and efficient way of accessing engineering product information. Changes have happened over the year, and alongside the new name is a new look and a new publisher.

Fully revised, the Yearbook remains a vital, time-saving buying guide to all engineering disciplines.

Further Information: Sally Lloyd IHS Australia Pty Ltd, Central Green, Como Centre, 660 Chapel St. South Yarra, 3124. Tel: 03 98266099, Fax: 03 98266886

RACAL AUSTRALIA

DiSC-16 Front-End

The Racal-Heim DiSC-16 is a data acquisition and signal conditioning front-end with advanced data transfer to PC. It uses the well accepted input signal modules fitted in the DATaRec-A series recorders, thereby allowing operations where direct computer access to the test is possible. The DiSC-16 mainframe is a DC or AC powered unit capable of accepting up to 8 twin channel analog or digital signal modules. Integral to the input modules are conditioning amplifiers for ICP, Microphone, Straingauge and Charge coupled signals.

A Data and Clock or Bi-Phase digital input module is also available with up to a total rate of 4Mbit/s. AES/EBU Data can be input through a special module accepting 20 bit data and extracting the relevant 16 bits for onward transmission to the analysis system. Acquired data output to a computer via an industry standard SCSI interface in DATaRec-Digital-2 format. The transfer software provides formats compatible with DATS, DaDisp, nSoft, FAMOS, SDF, WAV, ASCII, Binary and Hex. Total control of the acquisition and signal conditioning is achieved through comprehensive remote control and monitoring interfaces.

Data Recorder

For applications requiring high total data rate and high data volume, Racal Australia supplies the Storeplex range digital helical scan S-VHS recorders. Analog performance features up to 32 channels at 45.5kHz with 96 dB dynamic range. The high bandwidth analog channel (1.28Mhz, 16 bit sampling or 2.56Mhz 8 bit sampling) is capable of outstanding signal to noise ratio and phase accuracy.

For digital recording the maximum user single serial bit stream is 51.2Mbit/s. The electronic buffering and control allows the tape speed to be automatically adjusted to suit bandwidth and number of channels in use. This provides a variable recording duration from 30 minutes to 256 hours. The tape transport unit can be separated by 50m from the signal processing unit. The Control unit is either integrated with the Tape drive unit or for remote control connected by cable to Tape drive or Signal Processing unit, whatever the application.

Further Information: Mr Ko Oosterhuis, Racal-Heim Recorders, Racal Australia Pty Ltd, 3 Powells Rd, Brookvale NSW 2100 Tel: 02 9936 7000 Fax: 02 9936 7036

VIBROSOUND

Monitors

Huson & Associates have been appointed as agents in Australia and the Asia Pacific region for the innovative Vibrosound range of noise and vibration instruments. These state-of-the-art instruments are Type 1 approved to IEC 651 and ANSI S1.4 and fully comply with the requirements of DIN 4150, BS 6472, BS6955 and BS7385.

Vibrosound CM1 Compliance Monitor: for simultaneous monitoring of vibration and noise levels, thus providing legal and tamper proof evidence of compliance. This unit is ideally suited to monitoring at construction sites to check compliance with environmental noise and vibration requirements.

Vibrosound QB1 Quarry Blast Monitor: for monitoring of vibration and air over pressure levels from quarry blasting in accordance with the QLD Environmental Protection (Noise) Policy and IEC 651.

Vibrosound SP1 Mecro Seismic Mine/Tunnel Monitor: for monitoring micro-seismic activity together with larger vibrations to predict location of tunnel collapse and/or rock outbursts. Applications include monitoring of underground mines and structures including car parks.

Vibrosound AV1 Advanced Vibration Monitor: designed to monitor and record vibration over a large dynamic range, thus enabling detailed technical analysis of vibration and potential effects on structures.

Key features of the instruments include: event or continuous mode; up to six channels per unit; 5 and 15 day battery life options; mains powered and battery charger built-in, options include GPS and modem transmission; expandable memory and flash card options; internal pen and ink printer/plotter; and user friendly software.

Further Information: Huson & Associates, PO Box 1016, Browns Plains QLD 4118, Tel: 07 38083177 Fax: 07 38083134

INDEX
Volume 25, 1997

A. ARTICLES

B. NOTES

AUSTRALIAN ACOUSTICAL SOCIETY ENQUIRIES

NATIONAL MATTERS

* Notification of change of address
* Payment of annual subscription
* Proceedings of annual conferences

General Secretary
AAS– Professional Centre of Australia
Private Bag 1, Darlinghurst 2010
Tel/Fax (03) 9887 9400
email: watkinsd@melbpc.org.au
http://www.adfa.oz.au/~mxb/aa

SOCIETY SUBSCRIPTION RATES

From 1 APRIL 1998 membership subscriptions are:

Fellow and Member	$92
Associate and Subscriber	$74
Student	$20

DIVISIONAL MATTERS

Enquiries regarding membership and sustaining membership should be directed to the appropriate State Division Secretary

AAS - NSW Division
Professional Centre of Australia
Private Bag 1,
DARLINGHURST 2010
Sec: Mr D Eager
Tel (02) 9514 2687
Fax (02) 9514 2665
D.Eager@uts.edu.au

AAS - Queensland Division
PO Box 150,
OMMANEY 4074
Sec: Mr J Carter
Tel (07) 3806 7522
Fax (07) 3806 7999

AAS - SA Division
C/–Department of Mech Eng
University of Adelaide
SOUTH AUSTRALIA 5005
Sec: Carl Howard
Tel (08) 8303 5460
Fax (08) 8303 4367
cqhoward@mecheng.
adelaide.edu.au

AAS - Victoria Division
PO Box 417 Market St PO
MELBOURNE 3000
Sec: Mr D Dolly
Tel (03) 9859 9447
Fax (03) 9859 5552

AAS–W A Division
PO Box 1090
WEST PERTH 6872
Sec: Mr J Macpherson
Tel (08) 9222 7119
Fax (08) 9222 7157
john_macpherson@environ.gov.au

ACOUSTICS AUSTRALIA INFORMATION

GENERAL BUSINESS

Advertising Subscriptions

Mrs Leigh Wallbank
PO Box 579, CRONULLA 2230
Tel (02) 9528 4362
Fax (02) 9523 9637

ARTICLES & REPORTS NEWS, BOOK REVIEWS NEW PRODUCTS

The Editor
Acoustics Australia
Acoustics & Vibration Centre
ADFA
CANBERRA ACT 2600
Tel (02) 6268 8241
Fax (02) 6268 8276
email: acoust-aust@adfa.oz.au

PRINTING, ARTWORK

Scott Williams
16 Cronulla Plaza
CRONULLA 2230
Tel (02) 9523 5954 Fax (02) 9523 9637
email: print@cronullaprint.com.au

SUBSCRIPTION RATES 1998

	Aust	Overseas
1 year	A$ 48	A$ 60
2 year	A$ 82	A$106
3 year	A$115	A$151

Overseas subscriptions go by airmail

New Subscription 20% Discount

20% Discount for extra copies

Agents discount 15% of surface (Aust) rate

ADVERTISING RATES

B&W	Non-members	Sus Mem
1/1 Page	$530	$480
1/2 Page	345	310
1/3 Page	265	240
1/4 Page	225	210

Spot colour: $90 per colour
Prepared insert: $250 (additional postage may apply)
Column rate: $15 per cm (1/3 p width)

Discounted rates for 3 consecutive ads in advance

Special rates available for 4-colour printing

All enquiries to: Mrs Leigh Wallbank
Tel (02) 9528 4362 Fax (02) 9523 9637

ACOUSTICS AUSTRALIA ADVERTISER INDEX - VOL 25 No 3

Diary...

* Indicates an Australian Activity

1998

***February 24-26, SYDNEY**
UDT Pacific 98
Details: Nexus Information Technology, Nexus House, Swanley, Kent BR8 8HY UK, Tel: +44 1322660070, Fax: +44 1322661257, UDT.Pacific@nexusmedia.co.uk

March 4-5, SINGAPORE
Society of Acoustics (Singapore) Annual Meeting
Details: W. Gan c/o Acoustical Services, 209-212 Innovation Centre, NTU Nanyang Avenue, Signapore 639798, Republic of Singapore. Fax: +65 791 3665, wsgan@signet.com.sg

March 22-25 BOSTON
American Inst. Ultrasound in Medicine 42nd Annual Conven.
Details: AIUM, 14750 Sweitzer Lane, Suite 100, Laurel, MD 20707-5906. Tel: +1 301 4984100 Fax: +1 301 4984450

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

April 5-8, MICHIGAN
NOISE-CON'98
Details: INCE, PO Box 3206 Arlington Branch, Poughkeepsie NY 12603, Fax: +1 9144624006, inceusa@aol.com, http://users.aol.com.noisecon98/nc98_cfp.html

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

June 2-4, KRYNICA
Noise Control '98
Details: Noise Control '98, Katedra Mechaniki I Wibroakustyki AGH, al. Mickiewicza 30, 30-059 Krakow, Poland. Tel: + 048 12 173620, Fax: +048 12 332314, kmiw@uci.agh.edu.pl, http://www.wibro.agh.edu.pl

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, krylspb@sovam.com

June 9-12, SWEDEN
8th Int Conf. on Hand-Arm Vibration
Details: National Institute for Working Life, Conf. Secretariat HAV98, PO Box 7654, 90713 Umeå, Sweden, Fax: +46 90165027, hav98@niwl.se

June 20-28, SEATTLE
16th Int Congress on Acoustics
Details: 16th ICA Secretariat, Applied Physics Laboraty, Uni of Washington, 1013 NE 40th St. Seattle, WA 98105-6698, USA. http://www.apl.washington.edu/asa

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

June 26 - July 1, LEAVENWORTH
Tone and Technology in Musical Acoustics
Details: Catgut Acoustical Society, 112 Essex Avenue, Montclair, NJ 07042 USA, Fax: +1 2017449197, catgutas@msn.com, http://www.boystown.org/isma98/

September 14-18, CZECH
35th Int. Conf. Ultrasonics and Acous. Emission.
Detail: H. Kotschova, Geophysical Inst AS CR Bocni II/401, 14131 Prague 4 Czech Republic, Fax: +42 2 761549, hko@ig.cas.cz, http://www.ig.cas.cz

September 16-18, BELGUIM
Int. Conf. on Noise and Vibration Engineering
Details: Ms L Notré, KU Leuven, Division PMA, Celestijnenlaan 300B, 3001 Leuven, Belgium, Fax: +32 16322987, lieve.notre@mech.kuleuven.ac.be, http://www.mech.kuleuven.ac.be/pma/events/isma/isma.html

October 4-7, GERMANY
EURO-Noise 98
Details: CSM, Industriestrabe 35, D-82194 Grobrnzell, Tel: +49 8142 570183, Fax:49 8142 54735, csm_congress@compuserve.com

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 11-13, SINGAPORE
APAV 98
Details: APAV 98, 1 Selegie Rd #09-01, Paradiz Centre, Singapore 188306, Tel: +65 3399129, Fax: +65 334 7891, apavcon@singnet.co.sg

November 16-20, CHRISTCHURCH
INTER-NOISE 98
Details: NZAS. P.O. Box 1181, Auckland, NZ, Fax +64 9 309 3540
http://www.auckland.ac.nz/internoise98/

November 20, QUEENSTOWN
Recreational Noise
Details: P. Dickenson, Ministry Health, PO Box 5013, Wellington, New Zealand, Fax +64 4 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 Tel: 02 92622277 Fax: 02 92622323, tourhosts@tourhosts.com.au http://www.acay.com.au/~dstuckey/noise-effects98/

***November 30 - 4 December, SYDNEY**
5th Int. Conf. on Spoken Language Processing
Details: Tour Hosts, GPO Box 128, Sydney NSW 2001 Australia, Fax: 02 92623135, tourhosts@tourhosts.com.au, http://cslab.anu.edu.au/icslp98

***December 8-11, TASMANIA**
COMADEM 98
Details: Centre of Machine Condition Monitoring, Monash Uni. Dept. of Mechanical Engineering, Wellington Rd, Clayton VIC 3168, Tel: 03 99055699, Fax: 03 99055726, maltezos@eng2.eng.monash.edu.au, http://www.monash.edu.au/cmcm/

December 15-17, INDIA
"Designing for Quietness" an Int. Symp.
Details: Prof. ML Munjal, Center of Excellence for Technical Acoustics, Dept. of Mechanical Engineering, Indian Institute of Science, Bangalore 560 012, India, munjai@mecheng.iisc.ernet.in

1999

March 15-19, BERLIN
Forum Acusticum & ASA Meeting
Details: ASA, 500 Sunnyside Blvd., Woodbury, NY 11797 USA. Fax +1 516 576 2377, asa@aip.org

June 28-30, RUSSIA
EEAA Congress - 1st Int. Cong. East European Acoustical Society
Details: EEAA, Moskovskoe Shosse 44, St Petersburg 196158, Russia, Fax: +7 812 1279323, krylspb@sovam.com

July 5-8 DENMARK
6th Int. Congress on Sound & Vibration
Details: Dept Acoustic Technology, Technical Uni of Denmark, Bldg 352, DK-2800 Lyngby, Denmark. Tel: +45 45 881622
Fax: +45 45 880577 icsv6@dat.dtu.dk, http://www.icsv6.dat.dtu.dk

September 1-4 GERMANY
15th Int. Symp. Nonlinear Acosutics (ISNA-15)
Details: W. Lauterborn, Drittes Physikalisches Inst., Universitat Gottingen, Burgerstr 42-44, 37073 Gottingen, Germany, Fax: +49 551 39 7720, lb@physik3.gwdg.de

November 1-5, COLUMBUS
138th Meeting of ASA
Details: ASA, 500 Sunnyside Blvd., Woodbury, NY 11797 USA. Fax +1 516 576 2377, asa@aip.org

2000

October 3-5 KUMAMOTO
WESTPRAC VII
Details: Dept Computer Science, Kumamoto Uni. 2-39-1 Kurokami, Kumamoto, Japan 860-0862. Tel: +81 96 3423622 Fax: +81 96 3423630 westprac7@cogfni.eecs.kumamoto-u.ac.jp http://cogni.eecs.kumamoto-u.ac.jp/others/westprac7

December 4-8, NEWPORT BEACH
Meeting of the ASA
Details: ASA, 500 Sunnyside Blvd., Woodbury, NY 11797 USA. Fax +1 516 576 2377, asa@aip.org