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- **_Termites and vibrations_**
- **_Uncertain vibration characteristics_**
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## PAPERS



*Cover illustration: Heidi Hereth.*

# Message from the President

At the Annual General Meeting in November, the Society recognized Marion Burgess with the "Outstanding Contribution to Acoustics" award. Marion has been a tireless researcher and academic in the field and has made considerable contributions in the workings of the society. Congratulations Marion, your contribution over many years to the betterment of the society and acoustics in general has been outstanding.

The AGM also passed a number of resolutions from the Setting of Annual Subscriptions, Indexing Subscriptions, Graduate Membership Annual Subscriptions and Amendment of a number of By-Laws. Other business included awarding the Educations Grants and questions with notice. See elsewhere in this publication for more information.

The Victorian Division has planning well underway for the 2008 Acoustics and Sustainability conference in Geelong, November 2008. This will be the first national conference since 2005, because 2006 was the combined Australia/New Zealand conference and 2007 was the combined Australia and International Congress of Sound and Vibration congress. Conferences present valuable opportunities to network with others in the industry, the equipment distributors, researchers and presenters.

Christmas and New Year usually give the opportunity for some personal time out from very busy work schedules. Take some time to review your work patterns and seek more efficient ways to achieve the same goals. On reflection, you may find improving/learning new skills can lead to greater efficiency, which in turn may have the potential to reduce stress.

I wish all readers a happy new year. Take time to reflect on the year that was and plan your goals.

*Terrance McMinn*

# From the Editors

Several of the editors, and no doubt many readers, have been at conferences lately. The large and successful international conference ICSV14 was held very successfully at a large conference centre in Cairns. ICA2007 was held at a similarly large conference centre in Madrid. A rather smaller conference, also attended by some of the editors, was the International Conference on Music Communication Science, held at the University of New South Wales in Sydney.

The conference centre in Madrid is a very imposing building, whose arched roof produces some remarkable echos if one claps at a point on the centre line. As is the case for many conference centres, it has one very large auditorium, with good acoustics, audiovisual equipment and sightlines.

Conference centres also have a number of smaller rooms. Sometimes the small rooms have moveable partitions. They often have loud air conditioning. Many have low ceilings, which mean that sightlines to the screens are poor.

I expect that almost every reader has had the experience of groups of say 30 to 100 or so acousticians sitting in rooms in which communication required amplification (and thus also required technicians and microphone deliverers for the question sessions) and in which people leaned and stretched to see the screens. What are your thoughts? Do you dream of consulting on the acoustics of the next conference centre? Contemplate the ducting required for quiet air conditioning? Or remember that public address systems cost less than functional acoustics? Do you think of the theatres of ancient Greece, in which audiences over a thousand listened to unamplified actors, who may have been masked. Do you wonder about what we call progress?

A conference held at a university has the advantage that universities have a large numbers of rooms, seating from dozens to several hundred. In nearly all of these spaces, one may comfortably speak and be understood without amplification, even in large venues. They also usually have high quality audiovisual equipment and good sightlines.

Of course the universities are not always available. However, when they are, they may even have cheap accommodation available. Scheduling a conference to coincide with a university break may also increase the attendance, if academics are thus able to attend. Usually universities have good public transport and cheap food available nearby.

In case you were wondering, this editorial is not sponsored by a campus conference company. Rather it is the result of sitting with a few dozen colleagues in a noisy, dead room, struggling to hear and to see a presentation, while knowing that, on a campus a few km away, many fine auditoria with excellent facilities sat vacant.

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# WHAT DO VIBRATIONS HAVE TO DO WITH TERMITES' FOOD CHOICE?

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ABSTRACT. It has been shown previously that termites are sensitive to vibrations, using them as a communication channel. However, their ability to use vibrations in assessment of food structures is little understood. Here we present timber of differing quantities to two drywood termite species, *Cryptotermes domesticus* and *Cr. secundus*. We also expose the termites to vibration signals produced as a by-product of their feeding, and to food sources with altered effective material properties. We show here that both species have a food size preference, which is determined by vibrations. We also show that *Cr. secundus* is able to discriminate material properties. Although the exact characteristics in the vibration signals they utilise are yet to be fully identified, these observations reveal previously unexplored aspects of termite foraging decision-making, which might help to minimise their economic impact.

## 1. BACKGROUND

Termites are important insects, both economically and ecologically. It has been estimated that roughly one in three houses in Australia would be attacked by termites at some time during their life [1]. The total cost due to termite damage in Australia was estimated recently to be A$780 million per annum [2]. According to another estimate [3], the cost of structural damage caused by termites amounted, in the USA alone, to US$11 billion. Termites are also among the most important herbivores in Australia [4], and play an important role in engineering soil properties [5]; it has been estimated that termites contribute up to 5% of the annual global atmospheric methane [6].

Observations indicating that termites use vibrations in conspecific communication were made quite some time ago, both naturally [7, 8], and artificially, induced [9-12]. Vibratory signals are well known to be a useful means of communication amongst animals. It has been estimated that 80% of arthropod species use substrate vibrations in some way [13]. There are many reasons why termites specifically might use vibratory signals. Non-reproductive castes of termites are blind and have a number of mechanoreceptors. The subgenual organ, located in their tibiae, is the most sensitive to vibrations, being able to detect displacements down to 0.2 nm [14, 15]. Also, considering their highly social and complex societies, the use of vibratory communication would be highly beneficial for rapid transmission of information among termites.

Perhaps the most easily observed vibratory communication is that of the alarm signals produced by the termite soldiers. They repeatedly and forcefully strike the substrate with their heads in response to an intrusion [8, 11, 12, 16, 17] or detection of a potentially toxic pathogen [18]. If there is sufficient soldier alarm activity, the termite workers will retreat to a more central region of their nest [8].

More recently, vibratory signals have been associated with foraging activity of termites, e.g. [19, 20]. The vibratory signals produced during feeding have been partly characterised (e.g. [19]) and could be used for termite detection [21-23]. Patterns of development of reproductives (kings and queens) and survival in termite colonies, for a range of species, vary with changes in resource availability [24, 25]. This demonstrates a complex allocation of biological strategies in response to access to resources, despite the fact that the termites were not able to pace out the wood and did not tunnel to the wood surface. This prompted speculation that the mechanism behind this food assessment might be vibratory in origin [24]. If this speculation proves correct, what key measures do they use to assess the food type and size? Termites have a relatively simple nervous system, with the entire cerebral ganglia of most termites occupying a volume of the order of 0.1 $mm^3$ [26]. Thus it might be expected that they rely on relatively simple features of the vibratory signals.

In this paper, evidence is presented firstly to establish that termites can assess their food using vibrations, and then some key measures they might apply in this assessment will be explored. This is demonstrated by way of bioassays, whereby the foraging termites are given a choice between two potential food sources, one of which is of a standard size (160 mm X 20 mm X 20 mm) and the other is varied appropriately. The first series of assays was designed to test the hypothesis that vibrations are the mechanism behind food size assessment of termites, by exploiting their innate preferences for food size. The second was designed to further investigate possible key features in the vibratory signals that the termites use in their assessment by attempting to manipulate their behaviour using blocks of composite materials.

## 2. MATERIALS AND METHODS

**Termite and wood species:** Two drywood termite species were used in the bioassays and the recordings: *Cryptotermes domesticus* and *Cr. secundus*, both obtained from mangrove trees near Darwin, NT (012°31' S, 130°55' E) respectively. All bioassays were conducted in a controlled environment at 28°C and 80% RH. During signal recordings, the temperature was maintained at 28°C but the humidity was not controlled. All wood used in the studies were of seasoned untreated *Pinus* *radiata*, fashioned into rectangular blocks such that the grain was aligned parallel to the main axis.

**Vibratory recording:** Fifteen pseudergate (worker) termites were housed in a small chamber drilled into blocks 160 mm in length, held loosely in the middle by a clamp with foam rubber jaws (Figure 1). An accelerometer (Brüel & Kjær 4370, Nærum, Denmark. S/N:1360490) was stud-mounted to the base, which was connected via a charge amplifier (Brüel & Kjær 2635, Nærum, Denmark) to the sound card of a desktop computer running the software package CoolEdit (Syntrillium Software Corporation, Phoenix, AZ). The recordings were performed inside an anechoic chamber.

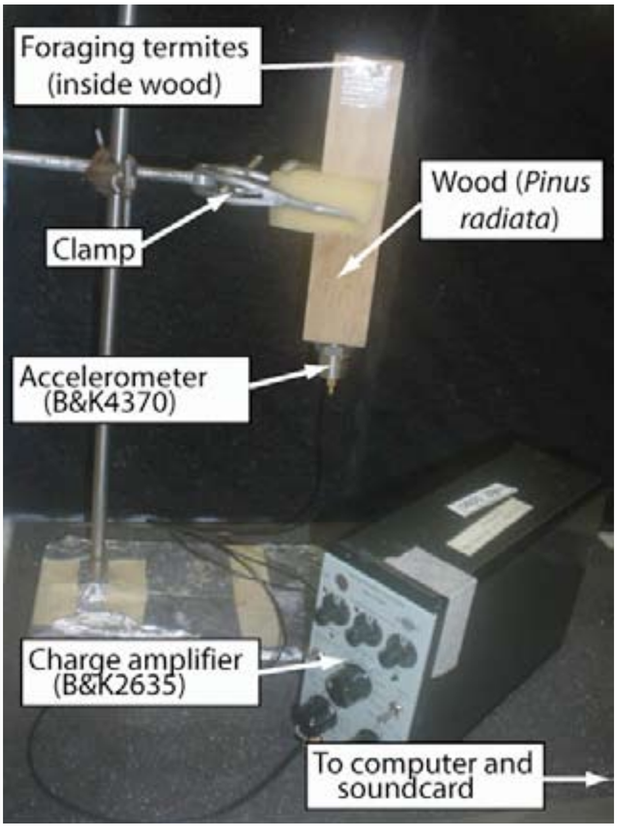


**Figure 1:** The set-up used to record the vibratory emissions given by drywood termite workers (subgenus: *Cryptotermes*) foraging at one end of a block of wood. Fifteen drywood worker termites fed on one end of a wooden block, while an accelerometer recorded the vibratory signals at the opposite end.

**Feeding choice bioassays:** For both studies, we needed to determine the feeding preference of the termites. As an extension of previous termite foraging experiments (e.g. [24]), blocks of wood having a constant, square, 20 mm X 20 mm cross-section were cut contiguously from the same timber source, and were presented directly opposite each other, separated by a 20 mm X 20 mm X 20 mm sealed cubic cell with walls of clear thin plastic (LDPE) (after [27-29]). Fifteen termites of either species were inserted into the central chamber. They fed on the wood for a period of fourteen days. As drywood termites tend to make relatively constant cylindrical tunnels, the total length and number of tunnels was recorded, to obtain a measure of tunnelling activity, and the position of individual termites (i.e. on either block) was noted daily for the first five days of the experiment. Together these observations were taken as a gauge of feeding preference [27]. The blocks were placed on vibration damping foam rubber, alternated spatially, to reduce broad environmental effects, and the orientation of the blocks was rotated by 180º after observation of the termites on each of the first five days.

**Food size preferences:** There were a total of four treatments (Figure 2), with number of replicates as specified in the Results section. All treatments had a 160 mm long block of wood on one side as a reference, and both species, *Cr. domesticus* and *Cr. secundus*, were used. Treatment 1, two 160 mm blocks (160:160), was a control. Treatment 2, having a 20 mm 'test block' opposite the 160 mm 'reference block' (20:160), was designed to test the natural food size preference of the termites [27-29]. Treatments 3 and 4 involved playback of vibratory signals through 20 mm blocks, which were fixed onto aluminium bars with a single wood screw, and driven by a shaker (Phillip Harris vibrator shaker, Leicester, England) *via* a CD player (Sony D-EJ100, Tokyo, Japan). The signal played in Treatment 3 (pink) was pink noise, in the band 0-20 kHz, synthesised using MATLAB. This was designed to act as treatment for a non-specific source of vibrations. Pink noise was chosen in order to emulate the noise profile of the instrumentation used in the vibratory recordings, which was largely due to instrumentation noise resulting from the high levels of amplification required. Treatment 4 (160 natural) played back the recordings made of the particular termite species feeding on a 160 mm block of wood [27, 29]. Although the signals used in playback were recorded from 160 mm lengths of wood, the signals perceived by the termites during playback will be modified by the properties of the playback system.

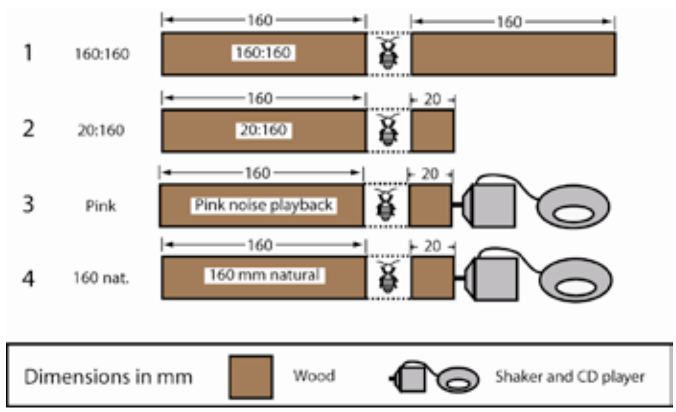


**Figure 2:** Schematic of set-up to test food size preferences in the drywood termite species *Cryptotermes secundus* and *Cryptotermes domesticus*. All treatments were opposite blocks, having a constant, square, 20 mm X 20 mm cross-section, with a 160 mm long block of wood on one side as a reference. The two playback experiments (pink noise and 160 mm natural foraging signals) were played back through a 20 mm block.

**Ability to discriminate materials:** This study comprised five treatments (Figure 3). Again, all treatments had a 160 mm long block of wood on one side as a reference. However, in this study, only the species *Cr. secundus* was used. The test block in Treatment 5 (discontinuity) consisted of a 20 mm and 140 mm contiguous block glued together to provide an artificial impedance boundary in the block. Treatments 5-9 were designed to test key measures in the vibratory signals the termites might use: that of the fundamental frequency, the mass or possibly the damping (or impedance) properties of the block [28]. This was done by glueing lengths of aluminium (having a high speed of sound and very low damping) or EPDM rubber (very low speed of sound, very high damping), each with constant 20 mm X 20 mm square cross-sections, on to a 20 mm wooden block.. The test blocks used in Treatments 6 (aluminium frequency) and 8 (rubber frequency) were designed such that they had approximately the same fundamental frequency as the reference block, using, respectively, lengths of aluminium and rubber. Treatments 7 (aluminium mass) and 9 (rubber mass) were designed to have the same mass as the reference block, again with aluminium or EPDM rubber. The accelerance spectra of the beams were measured to ensure these properties were, in fact, altered accordingly [28].

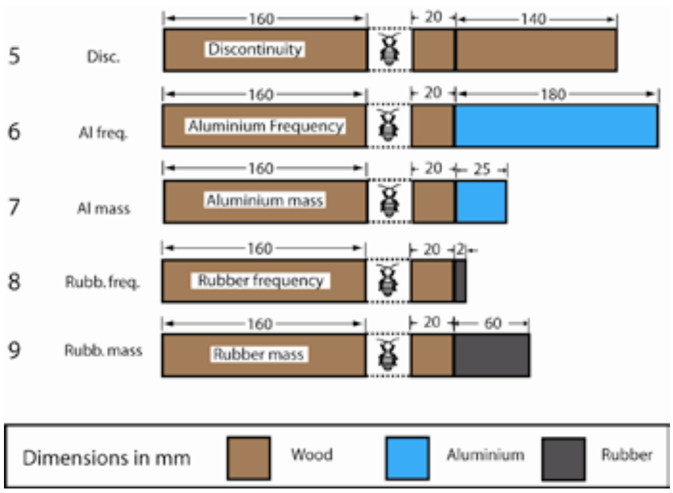


**Figure 3:** Schematic of set-up to assess the ability for discriminating materials of potential food structures for the species *Cryptotermes secundus*. All treatments were opposite blocks, having a constant, square, 20 mm X 20 mm cross-section, with a 160 mm long block of wood on one side as a reference opposite a composite test block with a 20 mm long block of wood with some other material attached to alter the effective vibratory characteristic of that food structure.

## 3. RESULTS

**Recordings:** the measured foraging signals (Figure 4) are similar to those of mechanical impulses; the dominant frequencies of the acceleration spectra, over time-averaged data taken from a series of peaks, are very similar to those obtained by striking the wood with a pair of tweezers, suggesting that the feeding signals are excitations of the substrate structure, and not produced by the termites themselves. However, this does not rule out vibratory communication using transient vibrations. The peak force, obtained from the measured acceleration levels, was *ca.* 20 mN. The motion of the beam, deduced from the measured acceleration spectra, appears to be that of a free-free beam mass loaded at one of the antinodes.

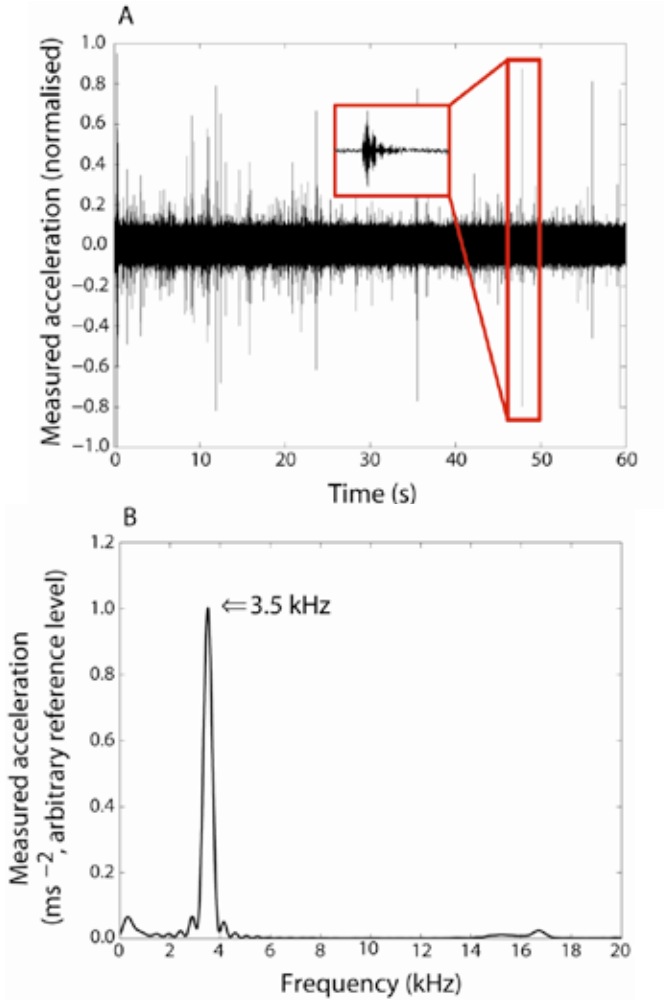


**Figure 4:** Measured acceleration of *Cr. secundus* feeding on a 160 mm length of wood. A shows the acceleration varying over time as a series of impulses, including a time-enlargement of a single pulse (window size: 10 ms). B shows the power spectrum average of the pulses, with a peak at approximately 3.5 kHz for this size of wood.

**Feeding bioassays:** Because the tunnel length data were not normally distributed, the bias introduced by outliers between replicates was reduced by examining the proportion of the amount of tunnelling in the 160 mm wooden reference block, taken, in each replicate, as the amount of tunnelling in the reference block divided by the total amount of tunnelling. The mean of the observed number of termites on the 160 mm block, for each replicate, on each day, was taken as a single measure of position for each treatment.

**Food size preferences:** For both species, there was no significant preference for blocks of wood of the same size (Treatment 1 (160:160)). For *Cr. secundus* (as a mean proportion in the 160 mm block, total tunnel length ± s.e., number of tunnels ± s.e., position ± s.e. (N = number of replicates)) (0.510 ± 0.111, 0.514 ± 0.088, 0.520 ± 0.041 (N = 12)), Cr. domesticus, (0.439

± 0.086, 0.432 ± 0.068, 0.489 ± 0.028 (N = 16)) Figure 5). However there was a significant effect of food size (Treatment 2 (20:160), *Cr. secundus* (0.805 ± 0.072, 0.812 ± 0.058, 0.759 ± 0.034 (N = 11)), *Cr. domesticus* (0.402 ± 0.053, 0.329 ± 0.038, 0.368 ± 0.015 (N = 44))). Most interestingly, despite this significance, the two species had *opposite* food size preferences.

In testing for a response to the playback of vibratory signals, the preference of *Cr. secundus* in Treatment 3 (pink) was altered to show no significant preference (0.544 ± 0.109, 0.583 ± 0.088, 0.473 ± 0.033 (N = 12)), as for *Cr. domesticus* (0.370 ± 0.068, 0.357 ± 0.061, 0.395 ± 0.022(N = 32)). Playback of the respective species feeding on 160 mm long blocks (Treatment 4) had the effect of swapping the preference for both species: for *Cr. secundus* (0.425 ± 0.070, 0.417 ± 0.052, 0.401 ± 0.014 (N = 24)), and for *Cr. domesticus* (0.698 ± 0.064, 0.656 ± 0.056, 0.539 ± 0.029 (N = 32)).

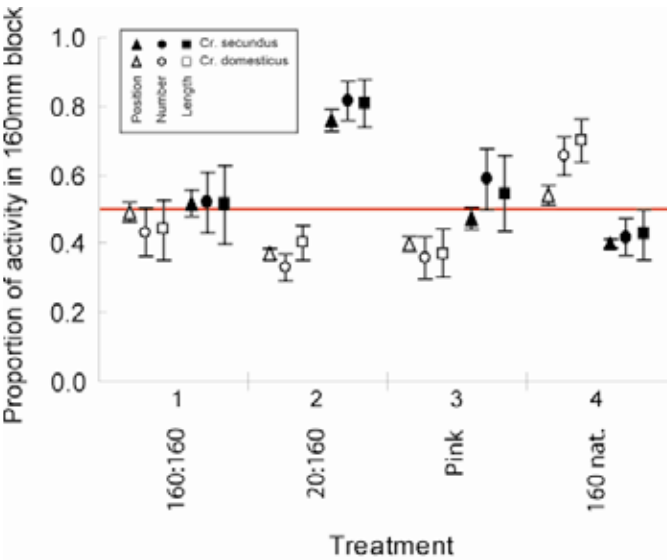


**Figure 5:** Proportion of total tunnelling activity (mean ± standard error) in 160 mm long reference wooden blocks for the two species of drywood termite to determine their food size preferences. Triangles denote the observed position of the termites, circles the number, and squares the total length, of tunnels. Treatments 1 (160:160mm control) and 2 (20:160 mm) were designed to test the natural foraging preference of the termites. Treatments 3 (pink noise playback) and 4 (recorded 160 mm playback) were designed to test the effect of vibrations, played back to the termites, on their preference. The red line at 0.5 is a reference indicating the ordinate position for no preference for either block.

**Ability to discriminate materials:** A discontinuity in the wood (Treatment 5) appeared to have no effect on the feeding preference of *Cr. secundus* (0.406 ± 0.031, 0.406 ± 0.031, 0.406 ± 0.031(N = 12)). However *Cr. secundus* preferred the 160 mm reference block for the aluminium frequency treatment (Treatment 6, (0.806 ± 0.019, 0.806 ± 0.019, 0.806 ± 0.019 (N = 12))) and the aluminium mass treatment (Treatment 7, (0.878 ± 0.011, 0.878 ± 0.011, 0.878 ± 0.011 (N = 12))) as well as for the rubber frequency treatment (Treatment 8, (0.806 ± 0.044, 0.806 ± 0.044, 0.806 ± 0.044 (N = 12))) and the rubber mass treatment (Treatment 9, (0.642 ± 0.060, 0.642 ± 0.060, 0.642 ± 0.060 (N = 12))).

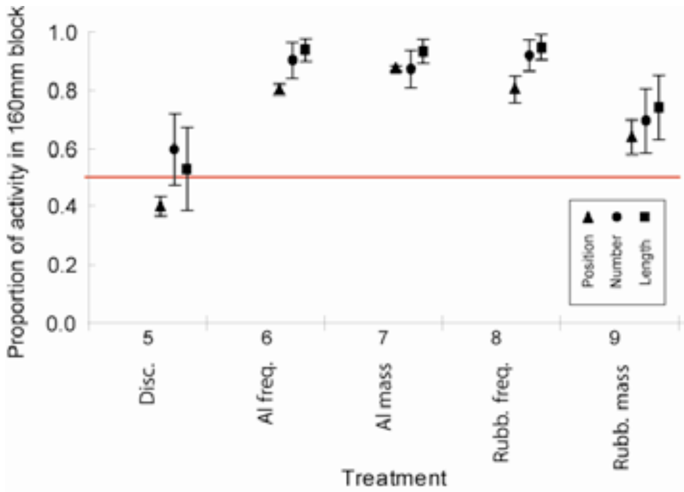


**Figure 6:** Proportion of total tunnelling activity (mean ± standard error) in 160 mm long reference wooden blocks to assess the ability to discriminate materials for the drywood termite species *Cr. secundus*. Triangles denote the observed position of the termites, circles the number, and squares the total length, of tunnels.

## 4. DISCUSSION

The studies presented here are meant as illustrative examples only, and more precise tests of the role of vibrations in food assessment by termites would be very useful. However, the results in the study on food size preferences (Figure 5) show definitively that termites use vibratory signals to make foraging choices of potential food structures. The results in the follow up study, to attempt to determine key measures in vibratory signals (Figure 6), show a perhaps surprising degree of sophistication, considering the simplicity of the nervous system of these organisms.

The two species used here, *Cr. secundus* and *Cr. domesticus*, were chosen to illustrate their contrasting foraging strategies. The former had a preference for larger, while the latter preferred smaller, blocks of wood. This may be because of their difference in life history: *Cr. domesticus* is an invasive cosmopolitan pest adapted to utilising small to large timber items, while *Cr. secundus* is a non-invasive native that occupies trees of variable size. It is plausible that the social nature of termites is a factor in the choice of potential food, especially under the experimental conditions presented here; the worker termites are in an unfamiliar environment, separated from the majority of their nest-mates [29].

The reaction of both termite species to the playback signals (Treatments 3-6) was similar. Playback of pink noise and foraging signals obtained in 20 mm wood block through the 20 mm wood block appeared to have no effect on their natural food choice preference (i.e. *Cr. secundus* preferred larger, while *Cr. domesticus* preferred smaller, wood blocks) or slightly shifting their preference towards the 20 mm block. However, playback of foraging signals obtained in 160 mm wood block through the 20 mm block reversed their natural food choice preference (i.e. *Cr. secundus* now prefers the smaller, while *Cr.*

*domesticus* prefers the larger, block). These results suggest that both termite species perceived the 20 mm wood block that was driven by the foraging signals obtained in 160 mm wood block as larger than 20 mm.

The feeding preferences of *Cr. secundus* indicate that they are able to determine some aspect of the material properties of their food structures in Treatments 7-11. These termites do not appear to use the fundamental frequency, or total mass, as the only measures in determining their preference for a potential food structure. However, the key characteristics in the vibratory signals they use have not yet been fully studied and identified. By applying similar methods described here, it would be possible to test if the termites make use of information obtained from, for example, the damping or impedance mismatch properties of the food that they eat [28].

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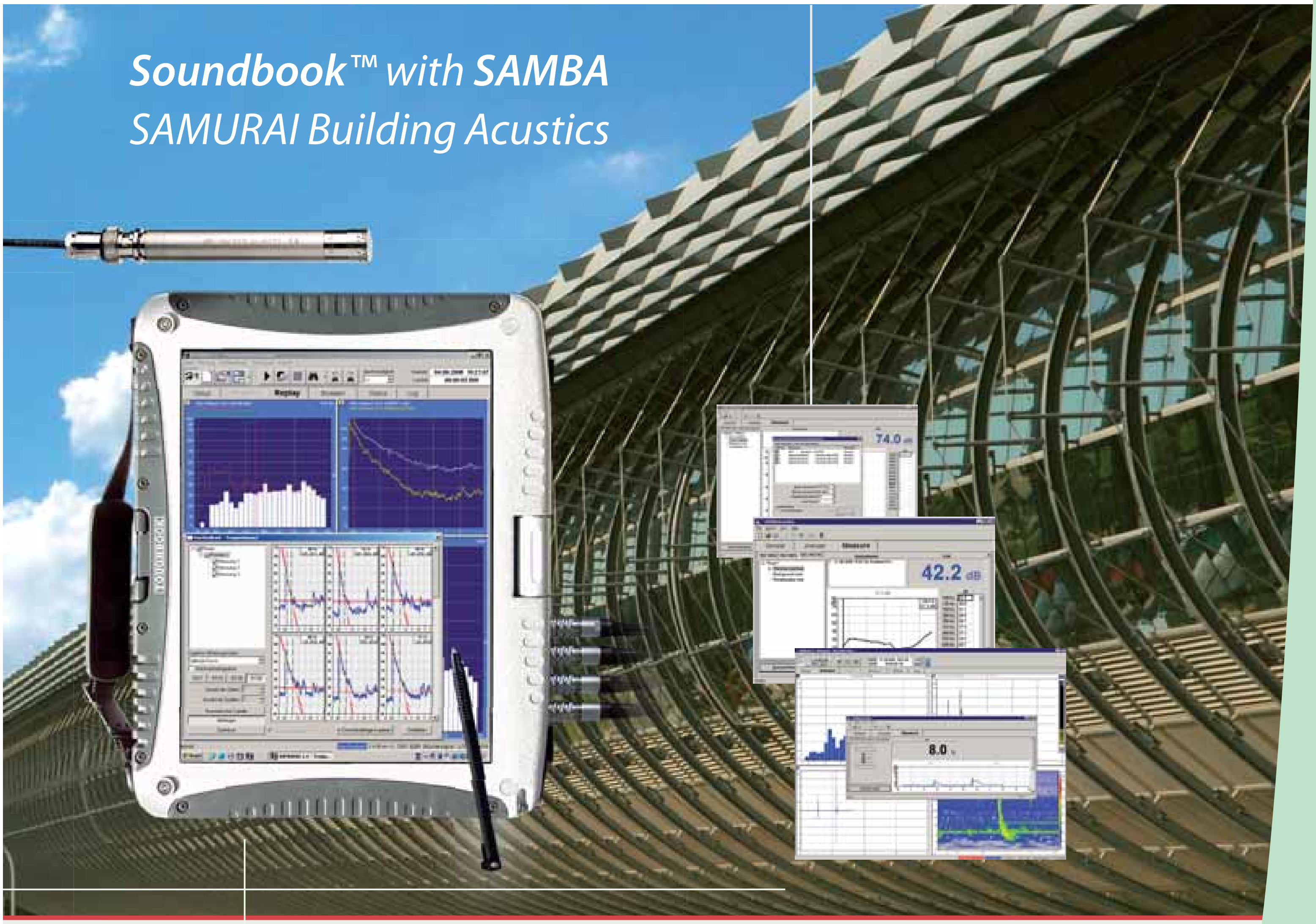


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# STATISTICAL MEASURES TO DESCRIBE THE VIBRATIONAL CHARACTERISTICS OF STRUCTURES WITH UNCERTAINTY

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Predicting the vibro-acoustic responses of structures with uncertainty can be difficult. For example, the dynamic response of body panels in a vehicle can differ greatly across an ensemble of vehicles due to small variations in spot welds from the assembly process. In this paper, several statistical measures including the statistical overlap factor, distribution of modal spacings and an ergodic hypothesis are used to examine the natural frequencies and responses for a range of dynamic systems. Uncertainty across an ensemble of nominally identical structures has been generated by adding small masses and/or springs at random locations. A measure of the uncertainty is obtained by observing the variation in the natural frequencies of an ensemble member from their mean value across the ensemble. Using an ergodic hypothesis, a comparison between the mean vibrational response of an ensemble of nominally identical structures at each frequency is made with the frequency averaged response of a single ensemble member.

## 1. INTRODUCTION

For most practical engineering systems, there are degrees of uncertainty arising from variation in material properties, geometry and manufacturing tolerances [1]. Uncertainties are also observed in the variation in the natural frequencies and dynamic responses across an ensemble of nominally identical structures. For example, the interior noise levels have been measured for successive vehicles from a production line for which the frequency responses were found to differ by at least 10 dB [2,3]. The variation is in part attributed to the manufacturing and assembly process resulting in local material property changes. Furthermore, as the frequency increases, so does the sensitivity of dynamic responses to uncertainties.

A number of techniques have been developed to account for uncertainty in the dynamic models of structures; some are briefly mentioned in what follows. The stochastic finite element method using Monte-Carlo simulations can account for structural uncertainty [4,5], but this method is restricted by the amount of information required to model joints between subsystems [6] and the significant computational expense required. An improved finite element method which only uses the mean, variance and covariance of the properties of the uncertainty has been developed to reduce the computation time and increase accuracy of the results [7]. Perturbation methods based on the finite element method have also been used to investigate the dynamic response of structures with uncertain parameters [8]. Including the second and higher order terms in the perturbation analysis is sometimes necessary, however this increases the time required to obtain a solution. Interval analysis has been utilised to examine the effect of uncertainty in the material parameters and dimensions on the eigenvalues and dynamic responses of structures [9,10]. In this method, the lower bound, upper bound and mean values of parameters with uncertainty were allowed to vary within a predefined band. A disadvantage of this method is that it produces very conservative results. A review of non-probabilistic approaches for uncertainty treatment in finite element analysis including both interval and fuzzy theory is given by Moens and Vandepitte [11].

In a pioneering paper, Weaver [12] transferred the study of the eigenvalues of random matrices in quantum mechanics to examine modal statistics in linear acoustics and vibrations. He showed that breaking physical symmetry by cutting slits into aluminium blocks results in the probability density function of the modal spacings being described by a Rayleigh distribution. These findings have been numerically and experimentally validated by examining the modal spacing distributions for vibrating plates with deformed boundaries [13] and for mass-loaded plates [14]. A motivation for this work is to investigate the modal spacing statistics of more complex structures which are a combination of both rigid body and flexural components, as well as structures coupled by joints with uncertain parameters.

In this paper, several statistical methodologies are used to investigate the natural frequencies and dynamic responses of structures with uncertainty across a wide frequency range. Attempting to predict and model all the various causes of uncertainty would be a very time consuming and difficult task. However, if the uncertainty becomes large enough, the response of a system becomes independent of the details of the uncertainty [1,15]. This paper attempts to address this statement by examining a range of dynamic systems corresponding to a mass-and-spring-loaded plate, two plates coupled by springs and a frame-plate structure. In each case, uncertainty is generated by adding point masses and/or springs at random locations on the structures. A measure of the uncertainty across an ensemble of nominally identical structures is obtained using a non-dimensional parameter called the statistical overlap factor [16]. Statistical overlap occurs when there is sufficient random variation in an individual natural frequency of a system from its mean value across the ensemble. A measure of the uncertainty occurring in a single ensemble member is obtained by observing the distribution of the spacings between successive natural frequencies. A statistical measure of the dynamic response is examined using an ergodic hypothesis, in which the frequency averaged response of a single system in the ensemble is compared with the mean response averaged across the ensemble at each frequency.

## 2. STATISTICAL METHODOLOGIES

### Modal spacing distributions

The earliest work on examining the distribution of the spacings between successive natural frequencies was conducted by Bolt [17] and Lyon [18]. They showed that for a perfectly rectangular room, the probability density function of the modal spacings followed an exponential distribution, which is given by

$$p(s) = ae^{-as}, \quad a = 1/\mu, \quad s \geq 0 \tag{1}$$

where $\mu$ is the mean spacing between neighbouring natural frequencies. However, later work has shown that an exponential distribution of the modal spacings only applies for simple and physically symmetrical structures and acoustic volumes such as a perfectly rectangular plate or box-shaped room. A Rayleigh distribution, which is given by [19]

$$p(s) = \frac{s}{c^2} e^{-s^2/2c^2}, \quad c = \mu\sqrt{2/\pi}, \quad s \geq 0 \tag{2}$$

of the modal spacings of a structure indicates that there is sufficient uncertainty in a structure such that its dynamic response is independent of the details of its uncertainty.

### Statistical overlap factor

The statistical overlap factor is a useful parameter to obtain a measure of the amount of variation in the position of the modes across an ensemble of nominally identical systems with uncertainties and is defined by [16,20]

$$S = \frac{2\{\mathrm{var}[\Delta\omega_n]\}^{1/2}}{\langle \omega_{n+1} - \omega_n \rangle} = \frac{2\sigma}{\mu} \tag{3}$$

where $\sigma$ is the standard deviation of any particular natural frequency $\omega_n$ from its mean value due to uncertainties in the system and is measured across an ensemble of random structures. Statistical overlap occurs when the random variation in an individual natural frequency of a system exceeds the mean frequency spacing.

### Ergodic hypothesis

The natural frequencies of structures with uncertainty can be considered to be ergodic in the sense that the statistical response of an ensemble are contained within one member of that ensemble [21]. Application of the ergodic hypothesis to dynamic systems states that the mean response is ergodic such that the frequency averaged response is equal to the ensemble average and can be expressed by [20]

$$\langle e_i(\mathbf{x},\omega,\mathbf{p})\rangle_{\mathbf{p}} = \left(\frac{1}{\Delta\omega}\right)\int_{\Delta\omega} e_i(\mathbf{x},\omega,\mathbf{p})\,d\omega \tag{4}$$

where $e_i$ is the kinetic energy density for subsystem $i$ as a function of location $\mathbf{x}$ and frequency $\omega$. $\mathbf{p}$ corresponds to a set of random parameters that describe the uncertainty in the system properties, $\langle\ \rangle_{\mathbf{p}}$ represents the ensemble average and $\Delta\omega$ is the frequency averaging bandwidth. The ergodic hypothesis requires the averaging bandwidth $\Delta\omega$ to be sufficiently wide such that frequency averaging the response of one member of the ensemble will be the same as averaging at a single frequency across the ensemble. A sufficiently wide averaging band would include at least 3 modes [20].

To implement the various aforementioned statistical methodologies, a range of dynamic systems are examined corresponding to a mass-and-spring-loaded plate, two plates coupled by springs and a frame-plate structure. In each case, uncertainty is generated by varying the location of the added masses and/or springs.

## 3. DYNAMIC MODELS OF STRUCTURES WITH UNCERTAINTY

### Lagrange-Rayleigh-Ritz method

The equations of motion of a dynamic system in modal space can be derived using the Lagrange-Rayleigh-Ritz technique in what follows. The flexural displacement of a bare rectangular plate in modal space is given by [22]

$$w(x,y,t) = \sum_{mn} q_{mn}(t)\phi_{mn}(\mathbf{x}) \tag{5}$$

$q_{mn}$ is the modal coordinate, and $m, n$ are the mode numbers of the shape functions in the $x$ and $y$ directions respectively. $\phi_{mn}(\mathbf{x}) = \phi_m(x)\,\phi_n(y)$ are the mass-normalised eigenfunctions which satisfy the following orthogonality condition [23]

$$\int_0^{L_x}\int_0^{L_y} \rho h \phi_{mn}\phi_{m'n'}\,dxdy = \begin{cases} 1 & mn = m'n' \\ 0 & mn \neq m'n' \end{cases} \tag{6}$$

where $L_x$, $L_y$ are respectively the lengths of the plate in the $x$ and $y$ directions, $h$ is the plate thickness and $\rho$ is the density. For a plate simply supported on all four sides, the mass-normalised eigenfunctions are given by

$$\phi_{mn} = \frac{1}{M_n}\sin\left(\frac{m\pi x}{L_x}\right)\sin\left(\frac{n\pi y}{L_y}\right) \tag{7}$$

where $M_n = \rho h L_x L_y/4$ is the modal mass. Using the orthogonality condition, an expression for the kinetic energy of a bare plate becomes

$$\begin{aligned} T &= \frac{\rho h}{2}\int_0^{L_x}\int_0^{L_y} \dot{w}^2(\mathbf{x})\,dxdy = \frac{\rho h}{2}\sum_{mn}\sum_{jk}\dot{q}_{mn}\dot{q}_{jk}\phi_{mn}(\mathbf{x})\phi_{jk}(\mathbf{x}) \\ &= \frac{1}{2}\sum_{mn}\dot{q}_{mn}^2 \end{aligned} \tag{8}$$

where $\dot{w}$ denotes the derivative of $w$ with respect to time. Similarly, an expression for the potential energy of the plate can be obtained as

$$V = \frac{1}{2}\sum_{mn}\omega_{mn}^2 q_{mn}^2 \tag{9}$$

where $\omega_{mn} = \sqrt{\frac{D}{\rho h}}\left(\left(\frac{m\pi}{L_x}\right)^2 + \left(\frac{n\pi}{L_y}\right)^2\right)$ corresponds to the natural frequencies of the bare plate,

$D = \frac{Eh^3}{12(1-v^2)}$ is the plate flexural rigidity, and $E$, $v$ are respectively Young's modulus and Poisson's ratio.

Lagrange's equation for a particular modal coordinate $j$ is given by [22]

$$\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{q}_j}\right) - \frac{\partial T}{\partial q_j} + \frac{\partial V}{\partial q_j} = 0, \qquad j = 1,2,...,N \tag{10}$$

Differentiating the kinetic and potential energies given by Eqs. (8) and (9) with respect to the modal coordinate $pq$ and substituting into Lagrange's equation results in the equation of motion of the bare plate.

$$\ddot{q}_{pq} + \omega_{pq}^2 q_{pq} = 0 \tag{11}$$

The natural frequencies can then be obtained by eigenvalue analysis $\left(\mathbf{K} - \omega^2\mathbf{M}\right)\Phi = 0$. This was performed in Matlab using the command *eig*, which returns a diagonal matrix of eigenvalues and a matrix of corresponding eigenvectors.

**Mass-and-spring-loaded plate**

Now consider a mass-and-spring-loaded plate as shown in Fig. 1. For the simply supported plate in free vibration with $N_m$ number of point masses (of size $m$) and $N_k$ springs to ground (of stiffness $k$), the equation of motion for a particular modal coordinate $pq$ of the bare plate has been previously developed and is given by [24]

$$\ddot{q}_{pq} + \sum_{mn}\sum_{N_m} m\ddot{q}_{mn}\phi_{mn}(\mathbf{x}_m)\phi_{pq}(\mathbf{x}_m) + \sum_{mn}\sum_{N_k} kq_{mn}\phi_{mn}(\mathbf{x}_k)\phi_{pq}(\mathbf{x}_k) + \omega_{pq}^2 q_{pq} = 0 \tag{12}$$

$\mathbf{x}_m$ and $\mathbf{x}_k$ correspond to the random locations of the added masses and springs, respectively. In the absence of the added masses and springs, Eq. (12) simply reduces to that of the bare plate given by Eq. (11). The natural frequencies of the mass-spring loaded plate were obtained by eigenvalue analysis using Matlab.

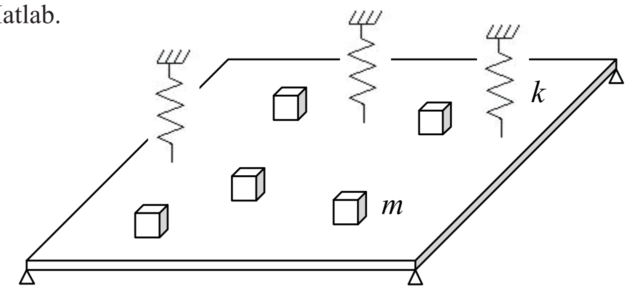



**Figure 1.** A simply supported plate with randomly located point masses and springs to ground.

**Plates coupled by springs**

Consider two simply supported plates coupled by a linear spring of stiffness $k$ at a random location $\mathbf{x}_i$ on each plate ($i$=1,2 is the plate number). The total kinetic energy of the system is

$$T = \frac{\rho_1 h_1}{2}\int_0^{L_{x1}}\int_0^{L_{y1}}\dot{w}_1^2(\mathbf{x})dxdy + \frac{\rho_2 h_2}{2}\int_0^{L_{x2}}\int_0^{L_{y2}}\dot{w}_2^2(\mathbf{x})dxdy = \frac{1}{2}\sum_{mn}\dot{q}_{1,mn}^2 + \frac{1}{2}\sum_{mn}\dot{q}_{2,mn}^2 \tag{13}$$

Making use of the eigenfunction orthogonality conditions, the potential energy of the coupled plate system is given by

$$V = \frac{1}{2}\sum_{mn}\omega_{1,mn}^2 q_{1,mn}^2 + \frac{1}{2}\sum_{mn}\omega_{2,mn}^2 q_{2,mn}^2 + \frac{k}{2}\left(w_1(\mathbf{x}_1) - w_2(\mathbf{x}_2)\right)^2 \tag{14}$$

where $\omega_{1,mn}$ and $\omega_{2,mn}$ are the natural frequencies of each uncoupled plate. The last term on the right hand side of Eq. (14) describes the coupling dynamics due to the randomly located spring. Differentiating the kinetic and potential energies with respect to the modal coordinate $pq$ of the uncoupled plates and substituting into Lagrange's equation, the equations of motion for the spring-coupled plates are given by

$$\ddot{q}_{1,pq} + k\phi_{1,pq}(\mathbf{x}_1)\sum_{mn} q_{1,mn}\phi_{1,mn}(\mathbf{x}_1) - k\phi_{1,pq}(\mathbf{x}_1)\sum_{jk} q_{2,jk}\phi_{2,jk}(\mathbf{x}_2) + \omega_{1,pq}^2 q_{1,pq} = 0 \tag{15}$$

$$\ddot{q}_{2,pq} + k\phi_{2,pq}(\mathbf{x}_2)\sum_{jk} q_{2,jk}\phi_{2,jk}(\mathbf{x}_2) - k\phi_{2,pq}(\mathbf{x}_2)\sum_{mn} q_{1,mn}\phi_{1,mn}(\mathbf{x}_1) + \omega_{2,pq}^2 q_{2,pq} = 0 \tag{16}$$

Equations (15) and (16) can easily be expanded to account for $N$ number of randomly located springs, as shown in Fig. 2. The natural frequencies of the spring-coupled plates were obtained by eigenvalue analysis using Matlab.

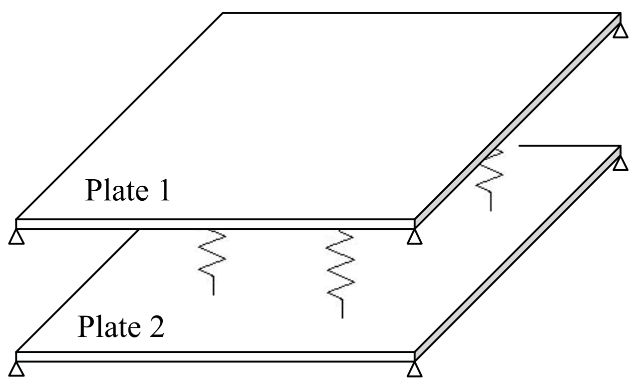



**Figure 2**. Simply supported plates coupled by randomly located springs.

**Frame-plate structure**

Finally, it is of interest to examine the dynamic characteristics of a structure with both stiff and flexible components, where the dynamic responses of the stiff components (of low modal density) are not sensitive to uncertainties but the flexible components (of high modal density) are sensitive to uncertainties. The dynamic characteristics of a frame-plate structure were obtained both computationally and experimentally. The frame-plate structure was modelled using finite element analysis, where the flexible plates were represented by quad 4 elements and the frame was modelled using bar elements. Damping was included in the model using a structural loss factor of 0.1%. The frame was constructed from 19 mm square hollow section aluminium tubes with a wall thickness of 1.2 mm. The flexible plates were made from 1.6 mm thick aluminium plate. The overall dimensions of the structure were 1000 mm long, 600 mm high and 600 mm wide. Twenty 3 gram masses were attached to the structure (7 masses on the two side plates and 6 masses on the base plate). Using Monte-Carlo simulations, 50 different configurations of the randomly located masses have been solved. Figure 3 shows a computational model of an ensemble member of the frame-plate structure.

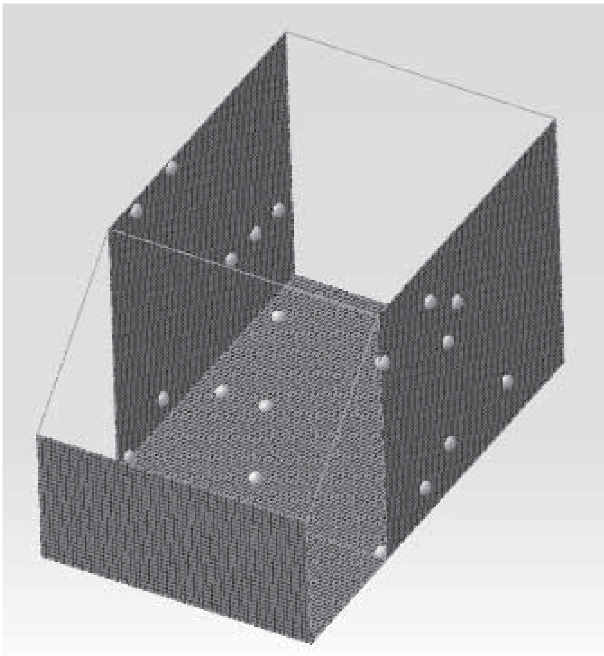


**Figure 3.** FEA model of the frame-plate structure with randomly located masses attached to the plates.

The same dimensions and material parameters were used in the construction of the experimental rig of the frame-plate structure. The tubes for the frame were welded together and the plates were attached using an epoxy adhesive. The frame was hung on soft springs to simulate free-free boundary conditions. Twenty 3 gram masses were attached at random locations across three of the plates. The structure was excited by a shaker mounted horizontally to the front, lower left hand corner of the frame, as shown in Fig. 4. The responses were measured in the horizontal plane at the rear, top, right hand corner. The measured signals from both the excitation and the response were passed though charge conditioning amplifiers before being sampled by an FFT analyser. 50 different configurations have been measured by randomising the locations of the added masses.

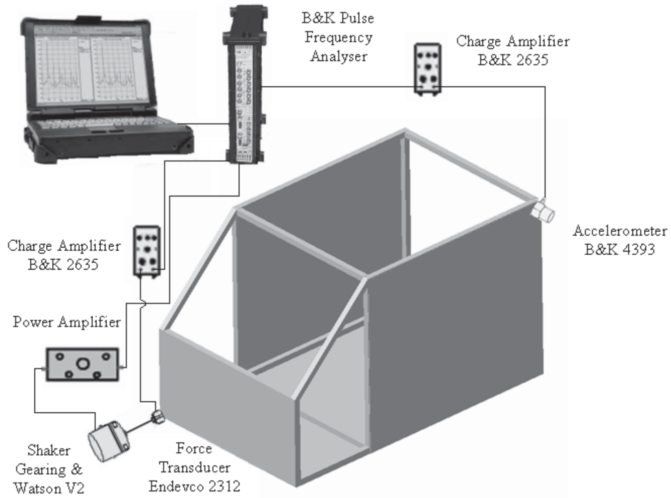



**Figure 4**. Schematic diagram of the experimental set-up for the frame-plate structure.

## 4. RESULTS

**Natural frequency statistics**

The natural frequencies for the bare plate, mass-and-spring-loaded plate and plates coupled by linear springs were obtained using Matlab. In each case, the plates were of dimensions $L_x$=899 mm, $L_y$=600 mm and thickness $h$=2 mm, with material properties of aluminium ($\rho$ =2800 kg/m$^3$, $E$=70 GPa, $v$ = 0.3). Damping was included in the analysis by using a complex Young's modulus E(1+j $\eta$) where $\eta$=0.1% is the structural loss factor. A probability density function (PDF) of the modal spacings was achieved by conducting the following steps: the natural frequencies for each dynamic system were arranged in ascending order, the spacings between successive frequencies were obtained, a histogram of the frequency spacings was generated and then converted to a PDF by scaling to unit area. The mean frequency spacing for each ensemble member was also calculated for comparison of the PDFs with the Rayleigh and exponential distributions. It is worth noting that the mean frequency spacing for a given dynamic system does not significantly vary between each ensemble member.

Figure 5 presents the frequency spacing distribution of the bare plate. The frequency spacings were obtained for a frequency range up to 4 kHz and the mean frequency spacing is approximately 12 Hz. Figure 5 shows that the modal spacing distribution of a structure with physical symmetries clearly follows an exponential distribution, which is given by Eq. (1) and is a function of the mean frequency spacing of the bare plate.

The PDF of the modal spacings and statistical overlap factor results for the mass-and-spring-loaded plate are given by Figs. 6 and 7. Fifty masses and springs were added at random locations, where each mass represents 0.2% of the mass of the bare plate and the springs each have stiffness 2 x $10^5$ N/m. The added masses and springs were considered to be both non-collocated (Fig. 6) and collocated (Fig. 7). Using Eq. (3), the statistical overlap factor (SOF) was calculated by examining the variation of each natural frequency across an ensemble of 50 spring-mass plates. The trend line for the SOF curves is also shown. The curves of the statistical overlap factor tend to level off with increasing frequency. This indicates that the results for the SOF have ‘saturated’ such that no further increase in statistical overlap will be observed with increasing frequency. The saturation in the SOF is attributed to the fact that the dynamic characteristics of the system have become independent of the details of the uncertainty, thereby indicating the frequency range beyond which a Rayleigh distribution of the modal spacings is expected to apply. An interesting observation in Fig. 7 for the plate ensemble with collocated masses and springs is the distinct dip in the statistical overlap factor, at which the value for S is zero. The frequency at which this dip occurs corresponds to the natural frequency for an equivalent single degree of freedom spring-mass system in terms of the added masses and springs, that is, $\omega_n = \sqrt{kN_k / mN_m}$ (which in this case occurs at approximately 916 Hz). At this frequency, the impedance of the added masses and springs is zero and hence are not generating any uncertainty on the plate. The low frequency range corresponds to the stiffness controlled region in which the springs dominate the structural response. As the frequency increases, the dynamic response of the structure becomes more sensitive to the inertial effects of the added point masses. The PDFs were obtained for the modal ranges beyond which the SOF begins to level off (4 kHz to 12 kHz). For both the collocated and non-collocated masses and springs, the PDF of the modal spacings clearly follows a Rayleigh distribution, which was calculated using Eq. (2) and the mean frequency spacing of an ensemble member for the same frequency range.

A PDF of the frequency spacings (for a single ensemble member) and statistical overlap factor (for the ensemble) are given in Figs. 8 and 9 for the spring-coupled plates and frame-plate structure respectively. For the spring-coupled plates, 10 springs of linear stiffness 5e6 N/m were randomly located on each plate. Similar results are observed for each system where the SOF initially increases with increasing frequency and then tends to level off as the statistical overlap saturates. The PDFs of the modal spacings are given for the frequency ranges 2 kHz to 10 kHz (spring-coupled plates) and 1 kHz to 4 kHz (frame-plate structure).

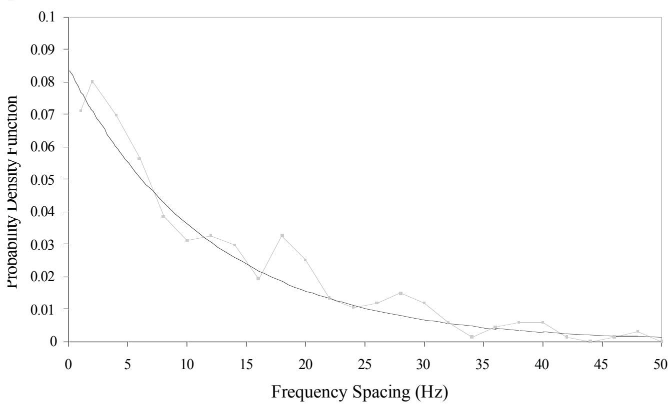



**Figure 5**. Probability density function of the natural frequency spacings for the bare plate: simulation results (grey line); exponential distribution (black line).

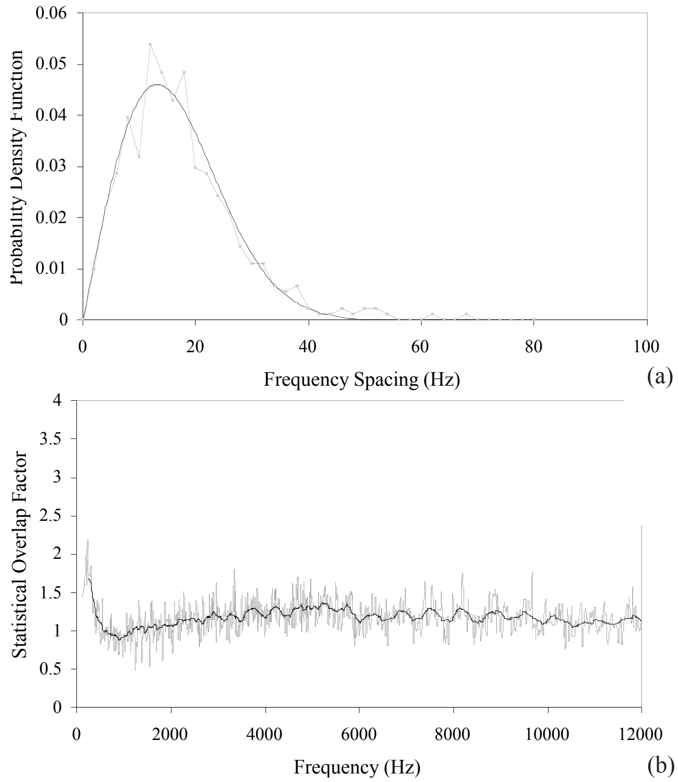



**Figure 6.** Results for the mass-and-spring-loaded plate (non-collocated masses and springs). (a) Probability density function of the modal spacings: simulation results (grey line); Rayleigh distribution (black line). (b) Statistical overlap factor (grey line) and trend curve (black line).

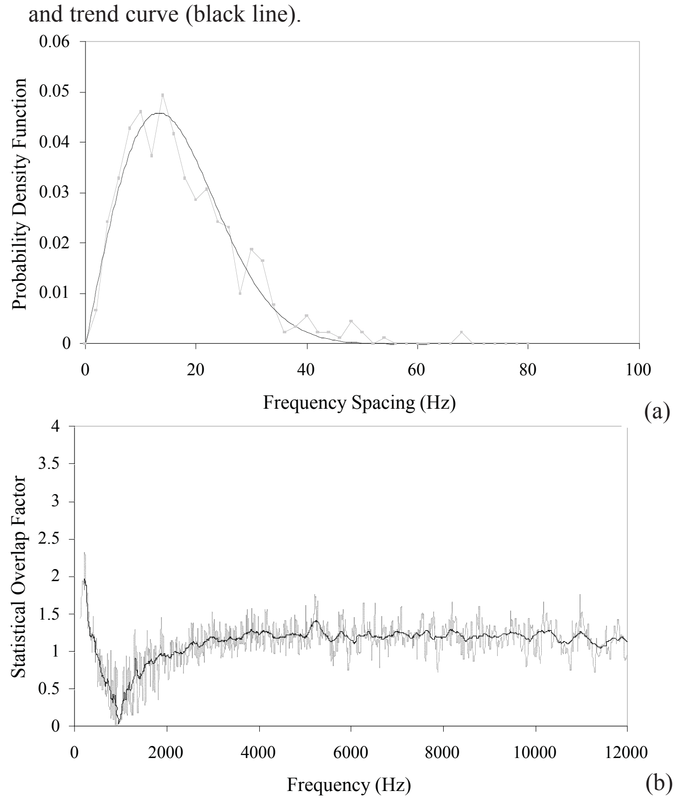
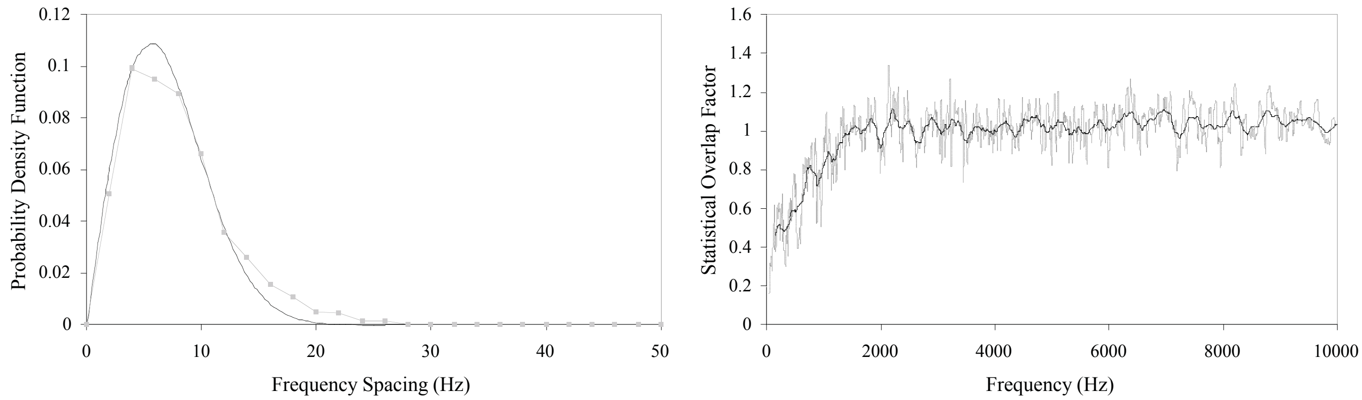
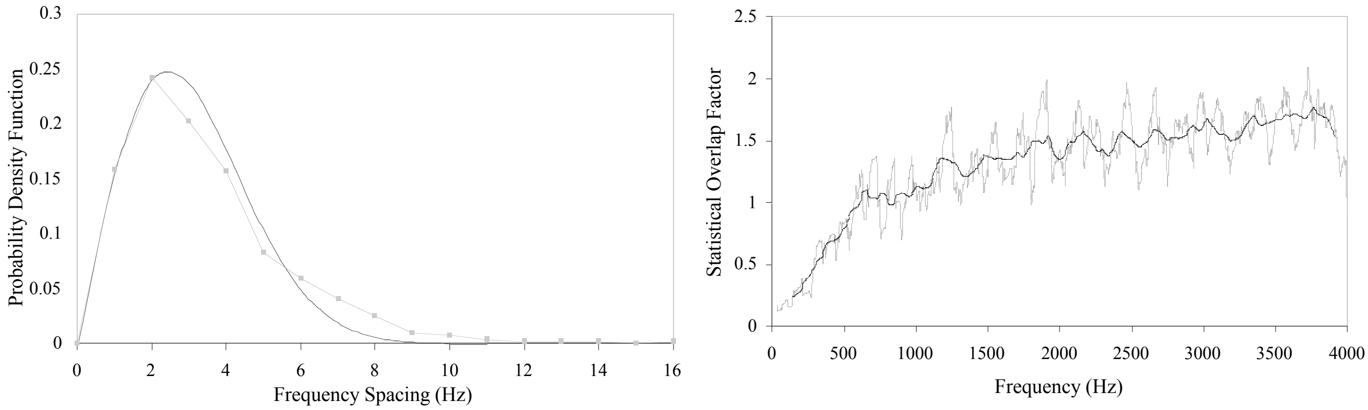



**Figure 7**. Results for the mass-and-spring-loaded plate (collocated masses and springs). (a) Probability density function of the modal spacings: simulation results (grey line); Rayleigh distribution (black line). (b) Statistical overlap factor (grey line) and trend curve (black line).




(a) (b)

**Figure 8.** Results for the spring-coupled plates. (a) Probability density function of the modal spacings: simulation results (grey line); Rayleigh distribution (black line). (b) Statistical overlap factor (grey line) and trend curve (black line).




(a) (b)

**Figure 9**. Results for the frame-plate structure. (a) Probability density function of the modal spacings: simulation results (grey line); Rayleigh distribution (black line). (b) Statistical overlap factor (grey line) and trend curve (black line).

**Frequency and ensemble averaged responses**

In this section, the ergodic hypothesis has been utilised to compare the frequency averaged and ensemble averaged responses for the spring-coupled plates and the frame-plate structure. The frequency averaged response of a single ensemble member was obtained by averaging the response using a proportional bandwidth of 5% of the frequency range for the spring-coupled plates, and 2% of the frequency range for the frame-plate structure. The ensemble averaging was achieved by averaging the responses across the ensemble at each discrete frequency. A measure of the quality of the match between the ensemble and frequency averaged results is observed using the z-score, which tests whether the residuals (corresponding to the error between the frequency and ensemble averaged results) has a mean of zero. The z-score is given by $z = x/\sigma_r$ , where $x$ is the mean of the residuals and $\sigma_r$ is the standard deviation of the residuals.

The frequency and ensemble averaged energy levels of the spring-coupled plates are shown in Fig. 10. Very good agreement between the ensemble averaged and frequency averaged results is observed. The corresponding z-score is calculated to be $|z|$=0.0091 with a standard deviation of the residuals, $\sigma_r$ = 4 dB. Similarly, Figs. 11 and 12 present the frequency averaged and ensemble averaged responses for the frame-plate structure obtained computationally (Fig. 11) and experimentally (Fig. 12). The z-scores are $|z|$=0.0668 and $\sigma_r$ =3.2 dB (computational result) and $|z|$=0.0343, $\sigma_r$ =5.2 dB (experimental result). In order to observe the effect of the frequency averaging bandwidth on the ergodic hypothesis results, the *z*-score and standard deviation for a range of averaging bandwidths are presented in Table 1, for the spring-coupled plates and frame-plate structure. It can be seen that increasing the averaging bandwidth results in a decrease in the *z*-score and standard deviation and hence an increase in the similarity between the ensemble and frequency averaged results. This decrease in the *z*-score and standard deviation will occur until an optimum frequency averaging bandwidth is reached. Beyond this bandwidth, the *z*-score and standard deviation will increase again due to loss of detail in the responses. The ergodic hypothesis shows the potential of obtaining the statistical responses of an ensemble of nominally identical structures from just one member of the ensemble.

Table 1. The *z*-score and standard deviations for a range of frequency averaging bandwidths for the spring-coupled plates and frame-plate structure.

Spring-coupled plates

| | | | | | | |
|---|---|---|---|---|---|---|
| Bandwidth (%) | 1 | 2 | 3 | 4 | 5 | 10 |
| z-score | 0.2730 | 0.1799 | 0.1071 | 0.0515 | 0.0091 | 0.1215 |
| Std deviation | 6.6070 | 5.3469 | 4.5975 | 4.1723 | 3.9827 | 3.9211 |

Frame-plate structure (computational results)

| | | | | | | |
|---|---|---|---|---|---|---|
| Bandwidth (%) | 0.1 | 0.25 | 0.5 | 1 | 2 | 5 |
| z-score | 0.2129 | 0.1769 | 0.0934 | 0.0668 | 0.2142 | 0.3361 |
| Std deviation | 4.4364 | 3.9901 | 3.4342 | 3.1548 | 3.7253 | 4.7828 |

Frame-plate structure (experimental results)

| | | | | | | |
|---|---|---|---|---|---|---|
| Bandwidth (%) | 0.1 | 0.25 | 0.5 | 1 | 2 | 5 |
| z-score | 0.1763 | 0.1544 | 0.1112 | 0.0343 | 0.0631 | 0.1807 |
| Std deviation | 6.1878 | 5.9180 | 5.5609 | 5.2406 | 5.3509 | 6.0253 |

## 5. CONCLUSIONS

The statistical responses of a mass-and-spring-loaded plate, two plates coupled by springs and a frame-plate structure have been investigated. For complex structures, it was found that the spacings between successive natural frequencies of a structure follow a Rayleigh distribution, indicating that the response of the structure is independent of the properties of the uncertainty. An ergodic hypothesis was employed which showed that the statistical responses of an ensemble can be predicted by frequency averaging the response of one member of the ensemble, as long as the averaging bandwidth is sufficiently wide. The results presented in this paper demonstrate that there are universal statistical descriptors for the modal spacings and dynamic responses of structures with sufficient uncertainty. This can serve to reduce the computational difficulties involved in the study of complex systems.

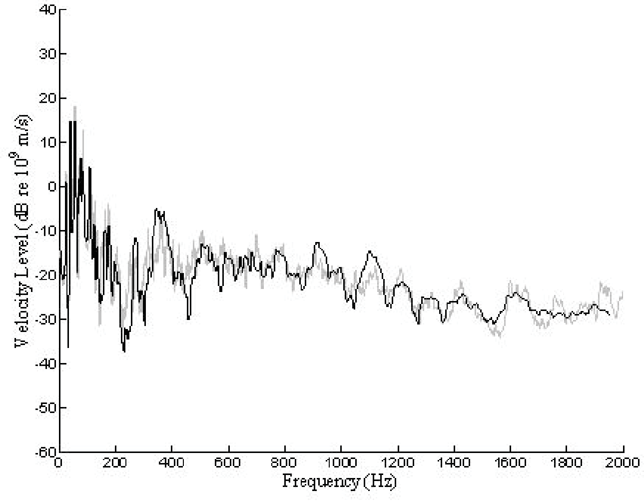



**Figure 10.** The frequency averaged response (black line) and ensemble averaged response (grey line) for the spring-coupled plates.

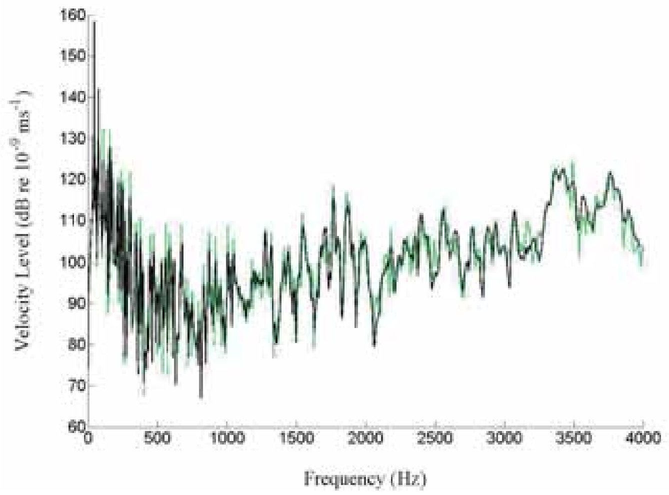



**Figure 11.** The frequency averaged response (black line) and ensemble averaged response (grey line) for the frame-plate structure (computational result).

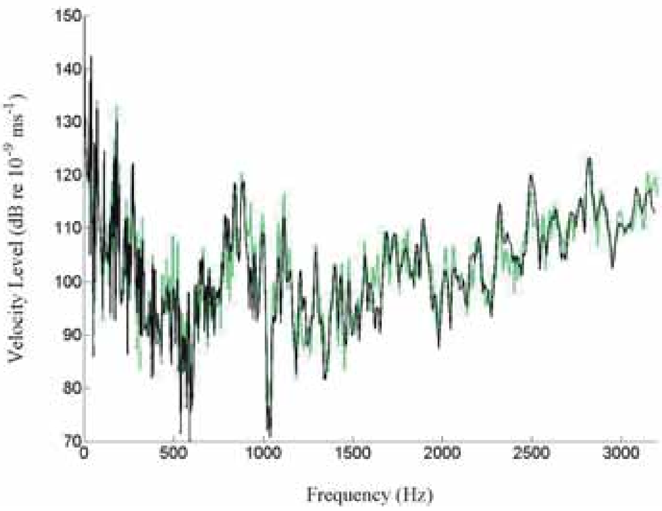



**Figure 12.** The frequency averaged response (black line) and ensemble averaged response (grey line) for the frame-plate structure (experimental result).

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# GRAVITATIONAL OSCILLATORS: BOUNCING BALLS, ROCKING BEAMS, AND SPINNING DISCS

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When an object is constrained to lie in a half-space bounded by a rigid horizontal wall, it will fall against this wall under the influence of gravity and then rebound in some way so as to execute oscillations which will gradually decay as energy is lost in the collisions. Common cases are bouncing balls, rocking beams, and discs allowed to spin obliquely onto the surface. In this last case, exemplified by coins and saucepan lids, the resulting radiated sound has interesting properties.

## 1. INTRODUCTION

Upon reflection, we are all familiar with the behaviour of a disc such as a coin or a saucepan lid dropped obliquely upon a table. After a brief unstable wobble it settles into a controlled motion in which its tilt rotates rapidly in either a clockwise or anticlockwise direction and any pattern on the surface is seen to rotate very slowly in the same direction. As the mechanical energy dies away over a few seconds the rotation rate of the slope angle speeds up but that of the surface pattern slows down. This behaviour is coupled to the sound radiated from the disc, so that we hear a rather broadband sound, the frequency of which rises at first slowly and then rapidly as the disc settles to the table surface.

The physical principles underlying the behaviour of such an object, known as “Euler’s disc”, have been treated in several classical texts [1,2] but still attract attention even in journals as prominent as Nature [4,5]. My aim in the present short paper is to explain the behaviour of such a disc, to link it to that of other simpler constrained oscillators such as bounding balls and rocking beams, and then to examine the resulting acoustic excitation and radiation.

## 2. BOUNCING BALLS

The simplest case to consider is that of an elastic ball dropped onto a flat surface. The ball rebounds, but its rebound velocity is less than its impact velocity by a factor $\alpha<1$ known as the coefficient of restitution. The rebound energy, and thus the rebound height, is therefore reduced by a factor $\alpha^2$. If the rebound velocity is $v$, then cycle time until the next impact is $2v/g$, where $g$ is the acceleration due to gravity. After $n$ bounces, the bounce height has been reduced by a factor $\alpha^{2n}$ and the cycle time by a factor $\alpha^n$. The impact frequency $f_n$ after $n$ rebounds has the value

$$f_n = \frac{g}{2v_0\alpha^n} = \left(\frac{g}{8h_n}\right)^{1/2} \tag{1}$$

where $v_0$ is the initial release velocity and $h_n$ is the bounce height after the $n$th bounce. From this it can be deduced, after a little algebra, that the oscillation actually ceases at a time $t_\infty$ after the first impact, where

$$t_\infty = \frac{2v_0\alpha}{g(1-\alpha)} = \left(\frac{8h_0}{g}\right)^{1/2}\left(\frac{\alpha}{1-\alpha}\right) \tag{2}$$

in which $h_0$ is the initial release height. More importantly from our present viewpoint, the impact frequency varies as a function of time according to the equation

$$f(t) = \frac{\alpha g}{2\alpha v_0 - gt(1-\alpha)} \ . \tag{3}$$

This result is shown in Figure 1, from which it is clear that the impact frequency rises at an increasing rate and actually becomes formally infinite just before the oscillation ceases at time $t_\infty$. We shall see that some of these behavioural features apply at least qualitatively to all the gravitational oscillators we discuss. A simple experiment with a super-elastic ball verifies the predictions of the model.

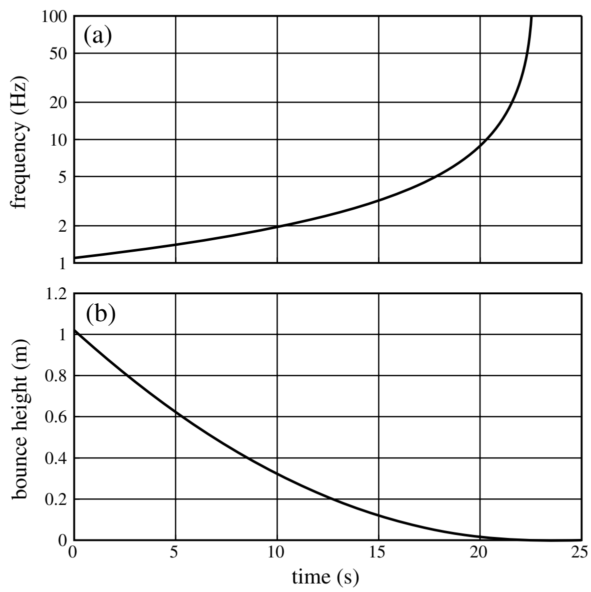


**Figure 1:** Calculated behaviour of (a) bounce frequency and (b) bounce height as functions of time after initial release for an elastic ball released from a height of 1m. Assumed coefficient of restitution $\alpha$ is 0.96 and the value of $t_\infty$ is then 21.7s.

## 3. ROCKING BEAMS

We turn now to a rather more complex system consisting of a beam rocking on two symmetrically placed supporting ridges, as shown in Figure 2(c). To simplify the analysis the solid beam can be replaced by two point masses located symmetrically on a light beam as in (a) and (b), the beam also being symmetrically located relative to the two fulcra. As an initial condition we suppose that the beam is in contact with just one fulcrum A, as in Figure 2(a), and is released from a stationary state inclined at an angle. It is assumed that the beam does not slide on the fulcrum and remains in contact until the beam impacts on the other fulcrum B. There are now impulsive vertical forces acting on the beam at its two points of contact and these serve both to reverse the sign of the motion of the centre of mass and also to change the speed of rotation of the bar. Just what occurs then depends upon the ratio of the distance between the two fulcra and the distance between the two masses.

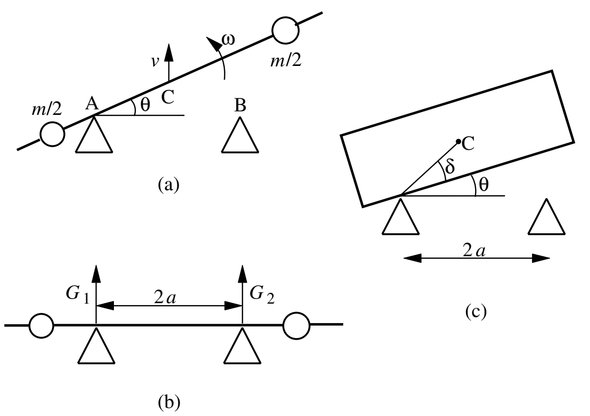


**Figure 2**: Simplified geometry of a beam rocking upon two symmetrically placed supports. When the beam is in contact with only one support, as in (a), there is a non-impulsive force acting upon it, but when it contacts both supports, as in (b), there are impulsive forces $G_1$ and $G_2$ acting as well. Panel (c) shows a more realistic situation for a beam of finite thickness.

Analysis of the behaviour is straightforward but the results are simple and interesting. Suppose that the mass of the beam and its attached masses is $m$, that its moment of inertia about the centre of mass is $I$, and that the distance between the two supporting fulcra is $2a$. Suppose also that the impacts are ideally elastic so that there is no loss of energy and let $v_1$ and $\omega_1$ be the centre of mass velocity and the rotation velocity respectively just before impact, with $v_2$ and $\omega_2$ being the same quantities just after impact. Then the impact equations can be solved to give

$$v_2 = \left(\frac{ma^2 - 3I}{I + ma^2}\right) v_1 \tag{4}$$

$$\omega_2 = \left(\frac{I - 3ma^2}{I + ma^2}\right) \omega_1 . \tag{5}$$

Several simple cases arise. The first occurs if $I = ma^2$, which is the case for two simple masses on a light rod and separated by the same distance as separates the two fulcra, or for a uniform rod of length $a\sqrt{12} \approx 3.46a$. For these parameter values the motion simply reverses with $v_2 = -v_1$ and $\omega_2 = -\omega_1$ and there is no impulse on fulcrum A. This is the equivalent of finding the "sweet spot" for hitting a ball with a cricket bat or tennis racquet. The other simple case occurs if $I = 3ma^2$, which can be achieved by a uniform beam of length $6a$, for then $v_2 = -2v_1$ and $\omega_2 = 0$ so that the beam loses contact with both fulcra and simply bounces vertically, maintaining its horizontal orientation. The beam can then continue to bounce up and down, remaining horizontal. The only other really simple behaviour is that which occurs for the limiting case in which the beam length, or separation between the two masses in the simple case, is almost infinitely long compared with the fulcrum separation $2a$. In this case support of the beam is simply transferred from one fulcrum to the other and the beam continues to rotate in the same direction at the same angular speed until gravity causes the motion to reverse. In all other cases, and particularly if $ma^2 < I < 3ma^2$, the motion involves bouncing contact on both the fulcra immediately after the initial impact. Allowing a certain amount of energy loss upon impact blurs the distinctions a little, so that there is a small region around each parameter value in which its distinctive behaviour can be expressed.

There is one other significant result that emerges from the analysis, and this relates to the frequency of the rocking oscillations. The equations are complicated but, for the long-beam case in which contact is transferred repeatedly from one fulcrum to the other, the result is that for oscillations of small amplitude $\theta$

$$f \approx \frac{1}{2}\left[\frac{mga^2}{2h(I + ma^2)}\right]^{1/2}, \tag{6}$$

where $h = a \sin\theta$ is the maximum height reached by the centre of mass. This is qualitatively similar to the result found for a bouncing ball in equation (1), with $f \propto h^{-1/2}$, so that the evolution of the oscillation frequency will follow essentially the same path as that shown in Figure 1(a) if there are fractional energy losses at each impact.

## 4. SPINNING DISCS

After this preamble we come now to the main topic of this paper, the behaviour of a disc released at an angle onto a horizontal plane in such a way that it begins a spinning motion. This could also be achieved either by initially rolling the disc along on its edge or by spinning it with its plane vertical. The disc will then collapse into the behaviour to be considered here. As noted in Section 1, the angle at which the disc is inclined to the support plane, and hence the point of contact, is observed to precess rapidly, while the disc itself rotates only slowly in the same angular sense. We will now examine the processes by which this comes about and the consequences for vibration and sound radiation from the structures involved.

The physical situation and coordinates involved are both shown in Figure 3. We take the radius of the disc to be $a$ and that of the contact circle to be $r$, the angular speed of rotation of the disc about its axis to be $\omega$ and the rate of precession of the contact point on the plane to be $\Omega$. Then the condition that there be no slipping at the contact point requires that $r\Omega = -a\omega$. The apparent rate of rotation of the pattern on the upper face of the disc is $\Omega+\omega = \Omega(1-r/a)$, and this is observed to have the same direction as that of the rotation of the contact point, which implies that $r < a$.

Understandably, analysis of the motion of the disc is rather complex, though essentially straightforward, since two coupled rotating motions must be considered as well as the stability of the height of the centre of mass above the plane. There are two ways of approaching this calculation. The first, and more usually adopted, is based upon moments and rotational inertia, while the second splits the motion into two sinusoidal oscillations that are geometrically at right angles and temporally 90 degrees out of phase. The final result is the same in both cases and gives the rotational angular velocity of the point of contact of the disc on the supporting plane as

$$\Omega = \left[\frac{4g\cos\theta}{(6\delta + a\cos\theta)\sin\theta}\right]^{1/2} \tag{7}$$

where $\delta = r - a\cos\theta$ is the radius of the circle traced out by the centre of mass of the disc. This is the result given by Ramsey [1], while Olsson [3] adds the assumption that $\delta = 0$. In reality it seems that $\delta$ may vary with the initial conditions, since for a disc spinning in a vertical plane $\delta = 0$ initially, while for a vertical rolling disc $\delta = \infty$. When the disc motion collapses to its inclined-plane state, however, it seems that $\delta$ converges towards a standard value near zero. Details have not yet been worked out.

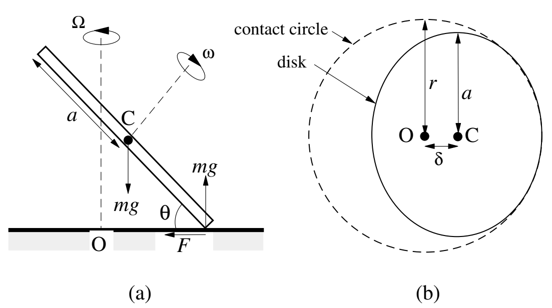


**Figure 3:** (a) Elevation view of the precessing disc, showing the forces acting upon it, $F$ being a frictional force. (b) Plan view. C is the centre of the disc and O the centre of its circular contact path with the supporting plane, shown as a broken line. $F$ is a frictional force that prevents the disc from slipping.

If $\delta$ is very much less than the disc radius $a$, then equation (7) shows that the angular rate of rotation $\Omega$ of the contact point between the disc and the plane becomes

$$\Omega \approx \left(\frac{4g}{h}\right)^{1/2}, \tag{8}$$

where $h$ is the height of the disc centre above the plane, while the visual rotation rate of the pattern on the disc becomes

$$\Omega + \omega \approx \frac{h^2}{2a^2}\left(\frac{4g}{h}\right)^{1/2}. \tag{9}$$

If energy is lost, then from (8) the rotation rate of the disc increases as $h^{-1/2}$, which is just the same behaviour found before in the case of bouncing balls or rocking beams and illustrated in Figure 1. The visual rotation rate of the pattern, however, does not increase but rather, as shown in (9), decreases as $h^{3/2}$. Both these effects can be easily observed for a spinning coin or saucepan lid.

## 5. ACOUSTIC EFFECTS

We come now to examine the vibrational interaction between the disc and the supporting plane and the way in which sound is generated and radiated. These matters are simple in the case of a bouncing ball or a rocking beam, since the excitation consists of a series of impulses delivered to both the moving object and the supporting plane at a rate that increases with time in the manner illustrated in Figure 1. For a spinning disc, however, there is simply a constant force $mg$ applied to the disc rim at a point, the position of which rotates at a speed given by equation (8).

If the supporting plane is very large so that reflections from its edges can be neglected, then radiated sound comes simply from the vibrational waves propagating away from the rotating source. This gives a wave equation on the plane of the form

$$\rho\frac{\partial^2 z}{\partial t^2} + S\nabla^4 z = mg\delta(x - r\cos\omega t)\delta(y - r\sin\omega t), \tag{10}$$

where $z$ is the displacement normal to the supporting plane, $S$ is the elastic stiffness of that plane, and $\delta(x-x_0)$ is a Dirac delta function. Formal solution of this equation is not simple, but it clearly leads to elastic waves propagating outwards from the place where the rotating disc is located. These waves will not be exactly sinusoidal because of an effective Doppler shift as the contact point moves around the circle.

In the case of a coin, there is another matter to consider and this is the effect of the milled edge, which adds a high harmonic to the excitation. A coin typically has about two milled grooves per millimetre on its circumference, which typically measures between 50 and 100 mm, so that we are dealing with an excitation frequency that is 100 to 200 times the coin rotation frequency, which brings it into the low kilohertz range. However, since the duration of the rotation phase for a coin is brief, it is quite likely that it will spend much of this time in a transient state in which there may even be rebounds from the surface, which will add impulsive excitations. A much larger disc, such as a saucepan lid, can be manipulated more easily into a stable rotating state and also maintains this state for much longer, so that it is easier to observe and hear.

Excitation of the disc itself is quite different, since it is a confined structure and generally has well-defined normal modes of the form $\psi_{mn} = R_{mn}(r)\cos n\phi$, or the sine equivalent, where $\psi_{mn}$ has $m$ nodal circles and $n$ nodal diameters. Since the excitation point applies a force $mg$ and is moving round the free circumference of the disc with angular speed $\Omega$, it effectively applies an excitation force of frequency $n\Omega$ to mode $(m,n)$, so that the excitation is effective at quite a high frequency. While it would be possible to analyse this behaviour in detail, the problem is hardly of sufficient importance to warrant this. This excitation signal sweeps upwards with time as the disc loses rotational energy and its centre of mass moves closer to the plane, as described by equation (8), and this is clearly audible.

Sound radiation from the rotating disc is complicated for several reasons. The oscillating disc is itself a set of multipole sources, each of distinct order $mn$, which are all correlated in phase at the contact point. The lower side of the disc, however, is shielded by the supporting plane and its oscillations are imposed upon the wedge of air between the disc and the plane, from where they radiate preferentially in a direction opposite to the contact point. This imposes a rapid fluctuation on the sound in any given direction.

There is one other interesting resonance phenomenon that influences the sound radiation from a disc. This comes from the fact that the air volume enclosed under the disc, and vented by the opening between it and the supporting plane, acts as a resonator, which may impose a sort of "vocal formant" on the radiated sound. In the case of a domed disc, such as a saucepan lid, the resonance frequency $f^*$ is determined by the enclosed air volume $V$ and the area $S$ of the vent. In such cases we can easily deduce the approximate relation

$$f^* \approx \frac{c}{2\pi}\left(\frac{S}{Vd}\right)^{1/2} \tag{11}$$

where $c$ is the velocity of sound and $d$ is the radiation "end correction" applicable to the opening. This Helmholtz approximation is, however, valid only if the sound wavelength at frequency $f^*$ is large compared with the diameter of the disc. For a domed disc such as found in a saucepan lid, the enclosed volume has the form $V_0 + \alpha h$ where $V_0$ is the volume of the dome itself and $\alpha$ is a constant. If $\alpha h << V_0$, as will be the case when the disc has nearly settled, then since $S$ is also proportional to $h$ while $d$ tends to a small constant value, (11) predicts that $f$ will be about proportional to $h^{1/2}$ so that the formant frequency will decrease as the disc sinks towards the plane.

The situation for a plane disc is rather different, since the geometry of the air volume is simply reduced in the vertical direction as the disc settles, and this is rather analogous to changing the divergence angle of a conical horn, which has almost no effect on its resonance frequency. We should expect this frequency to be a little less than $c/2D$ where $D$ is the diameter of the disc.

The behaviour of a domed lid rotating and settling over a period of about 5 seconds is illustrated in the sonogram of Figure 4. The lid itself was 13 cm in diameter and about 30 mm in height at its centre, giving an enclosed volume of about 300 $cm^3$. Its material was steel about 0.3 mm in thickness. The sound intensity occurs as pulses as the raised side of the disk rotates, initially at a frequency of about 7 Hz but then increasingly rapidly. The lower dark band at about 800 Hz is the formant described above, and it is clear that its frequency reduces slowly over most of the time before plunging fairly abruptly to zero in the final few tenths of a second. The frequency of this band is close to what would be expected if the effective value of $d$, determined in this case by the aperture and the overhanging rim, is a few millimetres. Some higher bands, particularly those at about 2.2 and 2.6 kHz, are presumably resonances of the lid itself, and their frequency does not change with time. Sonograms of the sound from a spinning plane disc show the same sort of time structure for disc revolution, but lack the low formant band, and all the frequency bands are constant with time.

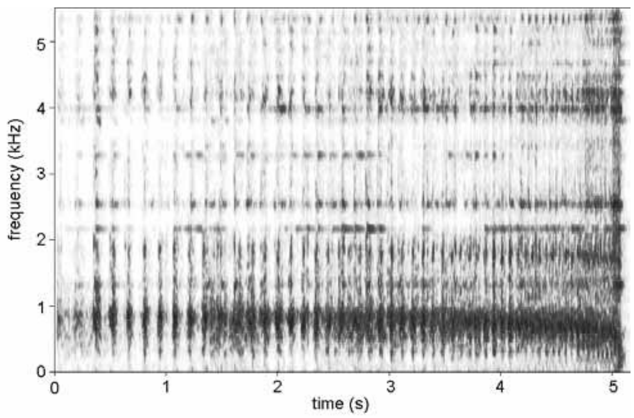


**Figure 4:** Time-resolved spectral analysis of the sound from a domed metal lid rotating and settling upon a smooth hard plane.

## 6. CONCLUSIONS

This small study cannot claim any fundamental importance, but was an interesting diversion. One striking thing to emerge was the uniformity of behaviour of the settling rate for bouncing balls, rocking beams, and spinning discs, all leading to a formal infinity in the oscillation rate after a finite time. Because these objects are all things that we encounter from time to time in ordinary life, it is interesting to have some insight into their behaviour.


## ACKNOWLEDGMENTS

I am grateful to Colin Sholl for comments on an original more technical version of this paper which gave details of the analysis, and also for his experimental confirmation of the predictions for the behaviour of a bouncing ball shown in Figure 1. I would also like to thank Glen Torr for recording the sound of the spinning discs and lids that formed the basis of Figure 4.


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# THE COMBINATION OF WORKPLACE AND RECREATIONAL NOISE EXPOSURE

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ABSTRACT: There are many noisy recreational activities undertaken by individuals during their leisure activities. How significant is noise exposure during recreational activities compared to noise exposure in the workplace? This paper reviews noise levels from common recreation activities. Comparisons are then made between possible noise exposures arising from work situations in combination with noise exposure from recreation activities. The findings indicate that the care taken to reduce noise exposure in the workplace can be swiftly negated with recreation noise dominating the overall exposure when recreation noise levels continue unchecked. If individuals are to maintain their hearing health they need to be more aware of the problems from exposure to excessive noise and to take preventative action similar to that used in the workplace.

## 1.0 INTRODUCTION

There are two criteria for occupation noise exposure applicable in Australia and New Zealand (NOHSC (1007): 2000: HSER: 1995); one for continuous noise and the other for impulse noise. For continuous noise, the eight hour A weighted equivalent, continuous sound exposure, $L_{Aeq,8h}$, must not exceed 85 dB[1]. This is the steady noise level that would, in the course of an eight-hour period, represent the same sound energy as that due to typical workplace noise, which usually varies over time. Noise exposures for shorter or longer periods must be normalised to an 8 hour period for the assessment and an equal energy concept is assumed where for an increase in level of 3 dB a halving of the exposure time must be applied and *vice versa*. For any impulse noise exposure, the C weighted peak sound pressure level, $L_{Cpeak}$, must not exceed 140 dB. This criterion is usually only exceeded during exposure to high impulse noise such as that from firearms, explosives or high powered impact tools.

It is important to understand that the exposure criteria values are not set at values that represent a “safe” exposure, at which no one would be expected to suffer harmful effects. Rather, they are set at values that represent a level of ‘acceptable risk’ for the general working community. For example, it is estimated (ISO 1999:1990; AS/NZS 1269.4:2005) that when noise exposure in terms of $L_{Aeq,8h}$ is limited to 85 dB for a working life of 40 years, 74% of an exposed otologically normal male population would on average suffer a 6% hearing loss – sufficient to lodge a successful hearing compensation claim in many jurisdictions.

The exposure criterion for $L_{Aeq,8h}$ is based on the assumption that, after the working day, the remainder of the 24 hours and the weekend are spent in a quiet environment (less than 75 dB). In order to compensate for any reduction in recovery time for long work shifts the assessment method (AS/NZS 1269.1:2005) includes an adjustment (shift loading) which is added to the worker’s $L_{Aeq,8h}$ before comparison with the criterion, i.e: for a shift length of between 10 to 14 hours the adjustment is +1 dB; for 14 to 20 hours, +2 dB; and for 20 to 24 hours, +3 dB. 1.

While noise is conventionally defined as ‘unwanted sound’, it is generally accepted that excessive ‘wanted sound’, such as music or sporty cars, will also cause hearing loss (Chassin: 1996). With this in mind no distinction in this paper is made between what can be considered as the psychological difference between noise and sound. It is also assumed that the sound energy associated with recreation activity noise has the same effect on hearing as does the sound energy produced by workplace noise.

## 2.0 NOISE LEVELS FOR RECREATION ACTIVITIES

While there may be a system in place for managing excess noise in the workplace, many people inadvertently (or deliberately) expose themselves to high levels of noise during recreational activities. The noise levels experienced during some common recreational activities are discussed in the following sections. This is not a comprehensive review but rather aims to provide an indication of the range of noise levels possible from various recreational activities.

### 2.1 Amplified music in clubs, concerts

Concerns have been expressed about the high levels of noise experienced in clubs, pubs, concerts and other venues with music. While there has been some discussion about the effect on the patrons, most research has been directed toward assessing the risk for the workers at such venues (Sadhra, Jackson, Ryder & Brown: 2002; Groothoff: 1999; Guo & Gunn: 2005).

As part of their ‘*Don’t lose the music*’ campaign, the Royal National Institute for the Deaf (RNID) in the UK published noise level data from three nightclubs in each of five UK cities. The clubs in each city were chosen on the basis of music style to ensure the samples included one house style, one pop style and one drum, bass, dance style. In terms of $L_{Aeq}$, the average noise level on the dance floor ranged between 90 and 110 dB.

1. The convention adopted here will be not to duplicate the A or the C after the unit dB to represent the weighting when it is included in the descriptor, ie $L_{Aeq}$ and $L_{Cpeak}$.

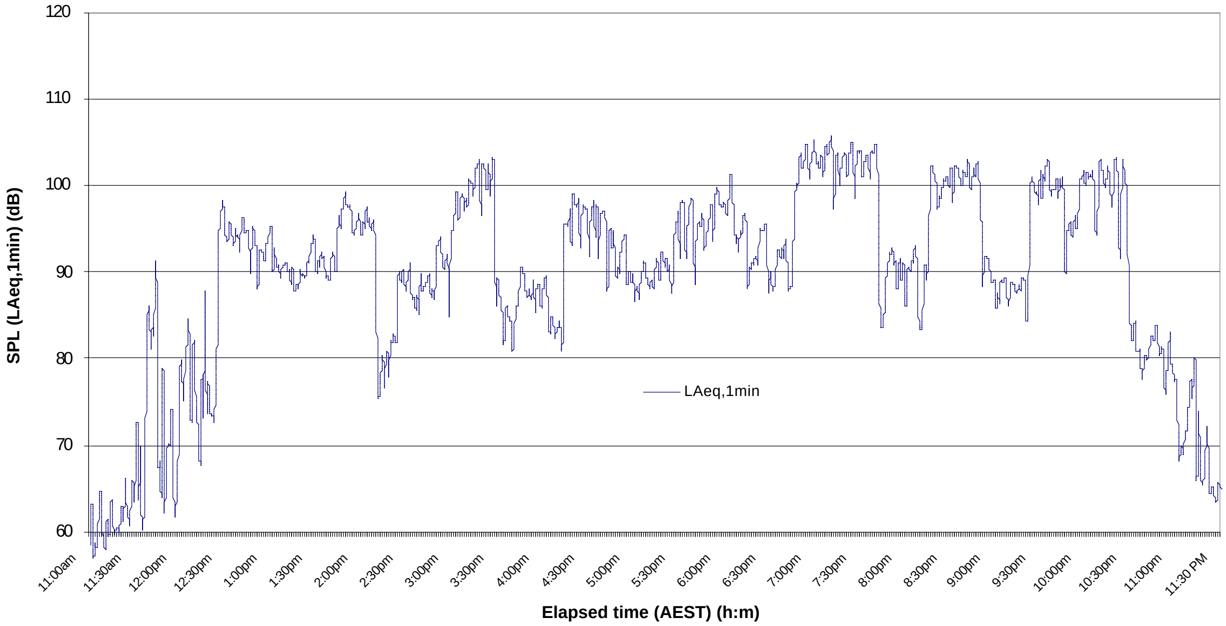


**Figure 1:** Measured noise levels at a typical outdoor concert over a twelve and a half hour period (Hall: 2007)

Even in 'chill out' areas the average noise level was found to be 92 dB. A recent study by Guo and Gunn (2006) in Western Australia that focused on the noise exposure levels ($L_{Aeq,8h}$) for a range of employees in clubs and pubs found that, in general, noise levels at music entertainment venues are "*excessively high*". They found that the exposure levels for workers ranged from 85 dB for a security person in a bar with only recorded music to 98 dB for a glass collector and manager in a venue with a live band.

Noise level measurements taken at a Sydney 'pub' venue on a typical week night with a live band showed that the $L_{Aeq}$ amongst the audience typically ranged from 102 to 107 dB and hovered around 94 dB on the outside footpath (Williams: 2006). With recorded music as the background between live performances the level was maintained around 83 dB inside the venue. *Figure 1* (Hall: 2007) plots the noise level in the audience for an annual, large outdoor concert. This particular event usually lasts for about four days and the sound systems are set up so that the level is fairly constant over the main audience area.

### 2.2 Amplified sound in cinemas

Concerns have been expressed in the media about increasing noise levels in cinemas. Movies which rely on special effects are more likely to have the higher noise levels with average levels of 78 dB(A) over three hours being reported for such movies (Hear-it: 2007). With the increase in availability of home cinema and associated high quality sound systems it is possible that there could be more regular exposure to these or higher noise levels during recreation times at home.

### 2.3 Personal music systems

There has been considerable media coverage of the potential damage to hearing from long term use of personal music systems such as MP3 players, tape players, etc. Typical of these is the warning by the RNID, UK about potential hearing loss from use of personal players, including comments from users about use over long hours and at high sound levels (RNID: 2006). However, much of the concern focuses on the maximum output level and there have been few studies of the noise exposure for typical users. A study by Williams (2004) measured the exposure levels of 55 randomly selected subjects who were using their personal players in noisy public areas in central Melbourne and Sydney. These devices were mainly being used during commuting where the range of background ($L_{Aeq}$) noise was 71 - 76 dB. The equivalent free field "at-ear" noise level from the player was measured over a two minute sampling period using the level that each subject was listening to immediately before selection. The sound levels ranged from 74 to 110 dB with a mean of 86 dB and reported listening times ranging from 40 minutes to 13 hours per day. From these values the $L_{Aeq,8h}$ were calculated to range from 66 dB to 104 dB while the mean exposure level was 79.8 dB. Twenty five percent of listeners exceeded the 85 dB workplace noise criterion.

### 2.4 Motor Sports

The noise from motor sport activities often draws considerable media attention, usually related to the noise emanating from the venue or race track into the surrounding area, *i.e.* concern about community/environmental noise. The patrons at the venue can be exposed to noise from general revving, racing, specialist high power vehicles, dynamometer testing and amplified music. Drivers and support crew may well have modern communication helmets that sometimes include hearing protection. On the other hand, patrons are subject to the noise from the output of the vehicle and are often located close to the track to ensure best views.

Specialised high performance vehicles currently have no output noise limits. For other motor sports vehicles the

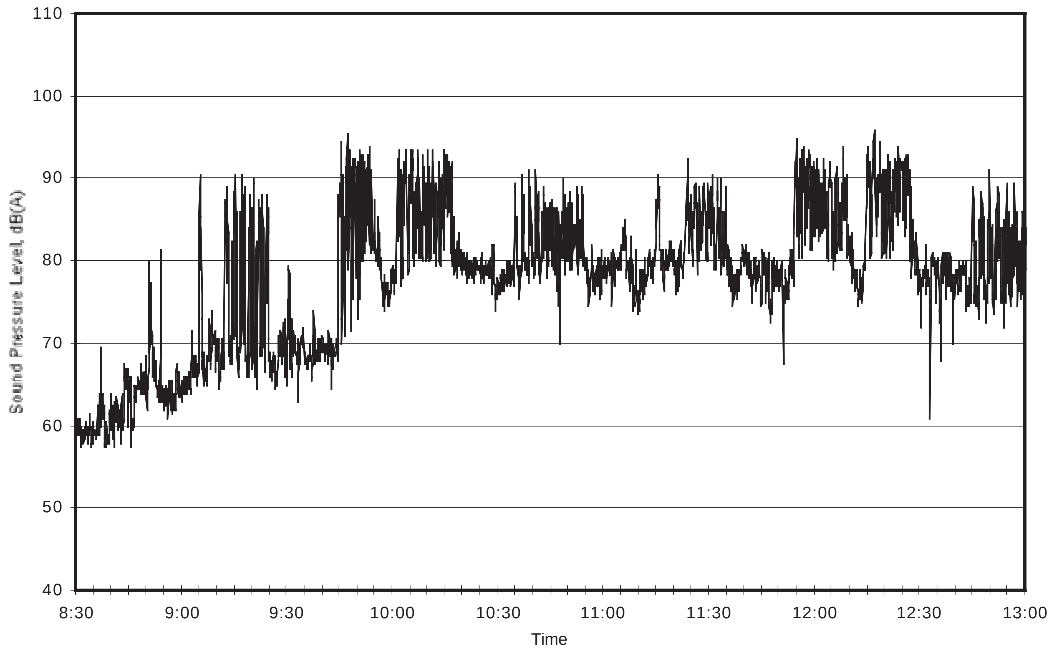


**Figure 2:** Measured noise levels at the trackside spectator area at a V8 Supercar event over a four and a half hour period (Burgess: 2002)

limiting values have been set with a view to minimising environmental noise impact. In Australia these limits are 95 dB for cars (CAMS: 2004) at 30m, *ie* at the edge of the spectator areas, and 102 dB for bikes at 0.5m from the exhaust (Motorcycling Australia: 2007) under full acceleration. *Figure 2* (Burgess: 2002) shows the noise level variation at the edge of the track during a major event for V8 Supercars. The levels, in terms of $L_{Aeq,5sec}$, were in the 80 to 90 dB range for much of the day.

While acknowledging that some spectators may be exposed to higher levels, it is reasonable to assume that a spectator at a range of motor sports activities could be exposed to an $L_{Aeq}$ of around 90 dB over the time of the event.

### 2.5 Car Stereo

While the engine noise level inside modern cars has been considerably reduced, custom built stereo systems are becoming common in many vehicles. These usually have very high power and have been found to produce $L_{Aeq}$ up to 104 dB (driver's window open) and are often set well above 80 dB when travelling. $L_{Cpeak}$ levels easily exceed 132 dB with many of the low frequency enhancements in use (Williams: 2006).

### 2.6 Home Workshop/Garden

Many power tools available for use in the home workshop and garden produce high noise levels and can often be used for long periods. Tools such as portable saws, routers, belt sanders, rotary hammer drills, grinders, chain saws and leaf blowers typically produce noise levels ($L_{Aeq}$) around 100 dB at the operator ear, while more specialised devices such as staplers and nail guns, utilising impulsive forces, can produce impulse noise levels with a peak ($L_{Cpeak}$) in excess of 140 dB.

It is very difficult to estimate an typical noise level for home workshop exposure as it is dependent on the tool, the material and the task but it would not be unreasonable to assume an exposure level of at least 85 dB during a couple of hours of activity. For example, the use of a circular saw with an $L_{Aeq}$ of 100 dB for only 15 minutes is equivalent to an exposure level ($L_{Aeq,8h}$) of 85 dB.

## 3.0 OVERALL EXPOSURE FROM A COMBINATION OF WORK AND RECREATION NOISE

In the previous section, common recreation activities have been shown to have high noise levels. The length of time people are exposed to these recreation noises varies significantly. If the approach as for occupational noise exposure assessment is used to assess the recreation noise exposure, in many cases the $L_{Aeq,8h}$ would be in excess of the recommended 85 dB. If the total noise exposure from the combined workplace noise and subsequent recreation activity noise were assessed, the overall exposure for the individual could be well in excess of the occupational noise exposure criterion.

Two models are presented to investigate the effect of the combination of noisy recreation activities with workplace noise exposure. As there is a requirement to manage the noise in the workplace, these models are based on the noise exposure during the work day not exceeding the exposure standard for $L_{Aeq,8h}$ of 85 dB. The models consider the overall noise exposure from a combination of eight hours of noise exposure below this limiting level plus varying hours for recreational noise at several noise levels. The adjustments from AS/NZS 1269.1:2005 for extended workshifts have been included and hence the steps which occur in each curve when the total exposure time exceeds 10 and 14 hours.

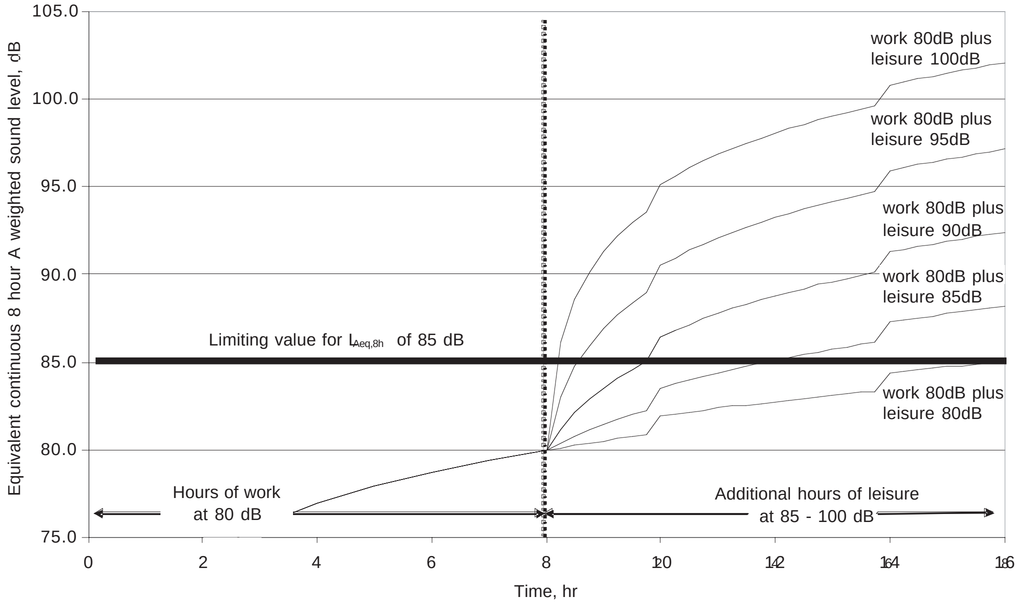


**Figure 3:** Example of the effect of regularly combining noise from recreation activities after a work day for which the $L_{Aeq,8h}$ was 80 dB.

*Figure 3* shows the increasing total exposure over time with various noise levels for recreation activities combined with an eight hour work day where the $L_{Aeq,8h}$ is limited to 80 dB. Hence at the end of the work period the noise exposure level of 80 dB is below the exposure criterion. If this person then spent time each day in recreation pursuits for which the noise levels were less than 80 dB, there would be little concern about exceeding the exposure criterion. If the noise from the recreation activity were equal to the noise level during the work day, *i.e.* 80 dB, a total 16 hour exposure which was a combination of eight hours work plus eight hours of noisy recreation at 80 dB would lead to an $L_{Aeq,8h}$ noise exposure level of 85 dB which is at the exposure criterion. However if the recreation were in a club where the level was 100 dB, the combined noise exposure would exceed the 85 dB criterion after only approximately ten minutes in the club. If a person regularly spends eight hours recreation time in the club at 100 dB, their noise exposure level for the combination of the work day plus the recreation noise would be 102 dB – well above the criterion and with considerable risk of hearing damage.

*Figure 4* presents overall exposure where the worker is exposed to a recreational noise of 95 dB and varying controlled levels of exposure at work. Any benefit of controlling the work exposure to 75, 80 or 85 dB is negated by a relatively short recreational exposure. For example, using the lower curve where the work exposure is controlled to an $L_{Aeq,8h}$ of 75 dB followed by the activity with an $L_{Aeq}$ of 95 dB a combined work and leisure $L_{Aeq,8h}$ of 90 dB is achieved after only two hours. It is the exposure to the dominating higher noise level during recreation that takes the noise exposure above the recommended criterion thus negating any benefit from reducing the workplace noise to 75 dB rather than 85 dB.

The strategies for reducing noise exposure during recreation are the same as in the workplace. It is a process of risk management with a hierarchy of controls commencing with the elimination of the hazard as the preferred process, through to the use of personal protective equipment (hearing protectors) as the least preferred.

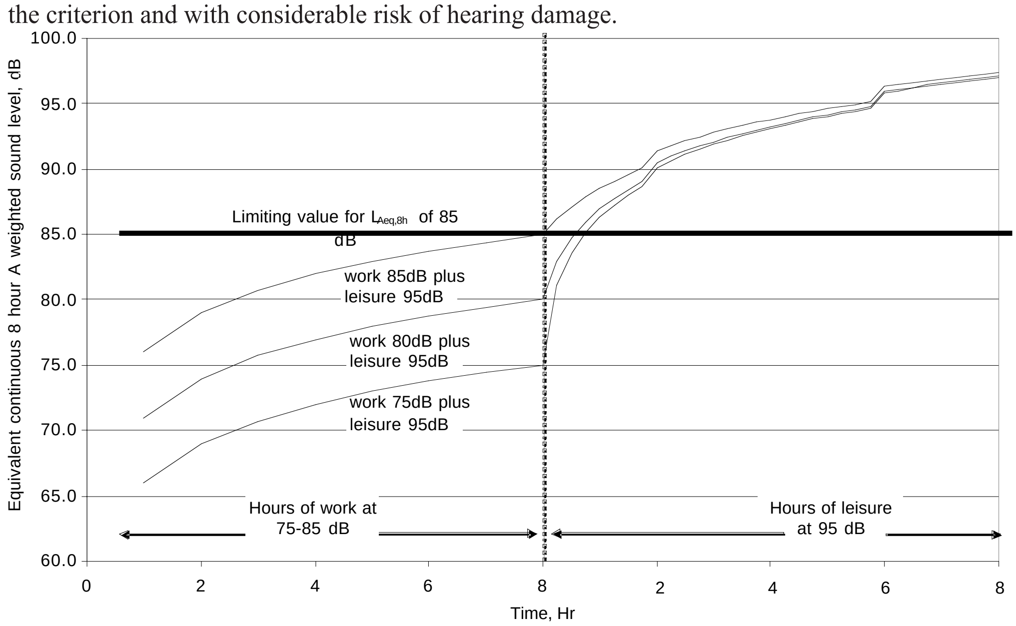


**Figure 4:** Example showing how the effect of regular exposure to recreational noise with $L_{Aeq}$ 95 dB dominates the combined exposure when workplace noise exposure is below the criterion.

Applying this principle to noisy recreational activities leads to simple actions such as:

- limit the number of visits to noisy venues;
- reduce the volume;
- avoid excessively noisy areas;
- limit the time spent in excessively noisy areas;
- make use of low noise ‘chill out’ spaces;
- move away from the sources of noise (loud speakers, revving engines, etc.);
- use appropriate, less noisy tools;
- relocate so as not to be within the line of sight of the noise source;
- use appropriate hearing protectors (plugs, muffs, communication devices);
- try to mix less noisy activities with quieter activities.

While individuals are at work, they come under the jurisdiction of various workplace occupational health and safety legislation and codes of practice where responsibilities are well defined for both individuals and organisations. However, when the individuals are away from the workplace, they must be responsible for their own health and safety. In relation to immediate physical dangers this responsibility is usually obvious, but with respect to future health difficulties individuals often do not necessarily know they must act or how to act in their own best interest. This may be due to a number of factors including optimistic bias (“it won’t happen to me”), ignorance of the health consequences of their actions or through generally unsafe practices.

## 4.0 CONCLUSION

In the workplace, regulations require the implementation of noise management strategies with the goal to ensure no workers have noise exposure levels, $L_{Aeq,8hr}$, greater than 85 dB or peak levels, $L_{Cpeak}$, greater than 140 dB. For recreational noise exposure, there are no legally binding noise exposure criteria. However, popular recreational activities have been shown to produce noise levels, $L_{Aeq}$, well in excess of 85 dB.

The combination of time spent in a controlled workplace, where the exposure does not exceed the regulation, plus time spent in a noisy recreational activity can lead to an overall noise exposure that may be considered as posing a risk to hearing health. To minimise the risk of hearing loss from such activities, individuals must take responsibility and minimise or control their exposure to excessive recreational noise. Simple strategies, similar to those for mitigating workplace noise exposure, can be applied in recreational pursuits and in many situations this will require a change in both attitude and behaviour of all those involved with the recreational activity.

## 5.0 ACKNOWLEDGEMENTS

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AAS 2008
# Australian Acoustical Society National Conference

‘Acoustics and Sustainability’
How Should Acoustics Adapt
to Meet Future Demands?’

to be held at the
Deakin Management Centre
Geelong, Victoria

24 – 26 November 2008

Abstracts are sought by March 2008
See www.acoustics.asn,au for details

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# News

## AAS Award for Marion Burgess

The award for "Outstanding Contribution to Acoustics" was presented to Marion Burgess at the 2007 AGM held in Sydney. The award is made to a member of the Society to recognize their extensive contribution to the advancement of acoustics and significant service to the Society.

*Marion Burgess with AAS President, Terence McMinn*

Marion has been working in the field of acoustics for over 30 years and has made an excellent contribution to building acoustics, occupational and environmental noise. She has over 60 publications in journals and conference proceeding and 200 consulting and research reports to her name. Over the last 15 years she has been involved with conducting courses on occupational and environmental noise, providing input to proposed government noise regulations and providing quality consultancies to both industry and government organizations.

Over the same period of time, Marion has been a consistent supporter of the activities of the Society. She has served in many capacities including Chairman of the NSW Division, Council member and organizer of a number of conferences. She represents the Society on a number of committees including the I-INCE and ICA. She is presently chairing the committee organizing the ICA 2010 conference in Sydney.

Marion has made huge contributions to the Society over a period of 35 years. Her contribution to the Society's journal, Acoustics Australia, are without parallel. She has been involved with the journal longer than any other member of the Society, starting as a member of the editorial committee in 1975, then managing editor in 1982, associate editor from 1985 to 1993 and a member of the editorial team from 1994. In 1987 she moved to Canberra and started the ACT group of the NSW Division organizing regular technical meetings.

## Guide to acoustic rating of apartments and townhouses

A new edition of the Apartment and Townhouse Star Rating Guide has been issued recently by the Association of Australian Acoustical Consultants (AAAC).

The new guide incorporates many changes and sets objective guidelines of amenity appropriate for a given standard of construction. It is intended to be used by developers, builders, architects, engineers, occupants and purchasers, with the aim of protecting residential amenity and acoustic privacy in many new apartment projects throughout Australia.

The Association Chairman, Tim Marks., says "Compliance with the Building Code of Australia (BCA) regulations does not ensure acceptable acoustic standards in all situations. For example, the current BCA standard for inter-tenancy walls is equal to a 4 star rating in the AAAC guide. Many developers, purchasers or occupiers may find this level of amenity unacceptable and require a 5, or even 6 star rating, for their apartments or townhouses. Major acoustic problems such as activities, music or voices from adjacent apartments being disturbing or audible, plumbing noise, chair or footfall impacts, or even traffic noise, can be distressing to experience and costly to fix. Furthermore, the actual level of acoustic insulation or treatment in a building cannot be easily established by visual inspection, unless an expert is involved".

The AAAC believes that the Star Rating Guide is an invaluable and essential guide to protecting well being, privacy and property values.

For further details contact your local AAAC member firm or visit www.aaac.org.au for a free download of the new edition of the star rating guide.

## Soundproofing to reduce skateboard noise

One half of the large Sports Hall within Sydney's Olympic Park is used by skateboarders, whereas the other half is used for team sports such as football and volley ball. The skateboarders create considerable noise when their very hard wheels drop from a great height onto toughened floor and ramps, and also when they hit the metal transitions between the wooden floor and ramps. Their preference for loud music was also a problem.

The solution proposed by Pyrotek involved hanging a massive 7-tonne curtain of Wavebar, filling in the entire space from wall to wall and floor to roof, to separate, acoustically and physically, the two halves of the cavernous space. This barrier could be removed to turn the two spaces into one great exhibition hall during Royal Easter Shows.

It was also proposed to reduce the noise transmission from the ramps via the concrete floor by isolating the ramps using Sylomer under their bases.

Finally Sorbertex 2D — a long lasting, non-woven, fine-fibre polyester — was to be placed under the ramps to take the echo out of the space, and Soundpaint under the steel fairings between floor and ramp to reduce the impact and wheel click noise.

A dramatic improvement reported by occupants of the Sports Hall remains to be confirmed objectively.

## FASTS

The AAS has continued to support the Federation of Scientific and Technological Societies (FASTS). Ken Baldwyn's first actions as incoming FASTS President were to meet with the relevant new Ministers and Shadow Ministers. The following highlights of FASTS's activities in 2007 (also listed on the FASTS website - www.fasts.org) were extracted from a letter by the new President.

- The importance of FASTS has been recognised by the Federal Government which has committed $200,000 per annum to FASTS activities over the next four years (commencing in 2007). FASTS will use these funds to build additional capacity to develop new policies to more effectively engage with the key issues confronting science in the coming years.
- FASTS organised the 8th annual "Science meets Parliament" event, where approximately 200 scientists had face-to-face meetings with members of Parliament. (Note that we plan the 9th SmP event for March 18/19 in 2008).
- FASTS raised policy issues with the Prime Minister and cabinet colleagues through its seat on the Prime Minister's Science, Engineering and Innovation Council (PMSEIC).
- We continued the advocacy of 'preparedness' as a key argument for broad-based support for science, which was adopted by the Productivity Commission in their major study, Public Support for Science and Innovation (March 2007), and on which we are preparing a subsequent policy document.
- FASTS has contributed to the development of the ABS's new classification of fields of research (FoR) and socio-economic objectives (SEOs) after successfully lobbying for a review of the system in 2006.
- Following up on our successful lobbying for the ARC grant announcements to be made in the first two weeks of October, we encouraged the Minister to release the outcomes this year in time to beat the pre-election caretaker period.
- At its AGM in November, FASTS presented a new Strategic Plan to focus our activities until the end of the decade. "

In relation to important issues Ken advises that "Most of our major competitors will be significantly increasing their funding for science and education over the coming five years, not the least of which will be the "America Competes" Act which amongst many initiatives will almost double the funding committed to scientific research by 2011. Australia, by contrast, has our science funding flat-lined until 2011 at which point the Backing Australia's Ability II programme will end. Therefore, during the early phase of the new Government, a plan for Australia's scientific and technological future beyond BAA II must be established."

"FASTS will highlight the need for a continuous, overarching assessment of the capacity of Australian science to prepare the nation for future challenges and opportunities…... While communication of science is now a well-developed function in society with many organisations playing an important national role, there is an ongoing need to provide sound scientific perspectives that represent the consensus scientific view. …… We need to look at the way in which science is governed – both at a strategic level to ensure that our national mix of scientific capacity adequately prepares us for the future, but also at the level of discipline-based or problem-based groupings to establish whether existing scientific society structures represent the best model for effective self-organisation."

## Future Meetings

### AAS2008

The next AAS Annual Acoustics Conference will be hosted by the Victorian Division and promises to be not only a very futuristic conference but also will provide a very lush surrounding and access to Victoria's prime tourist attractions along the Great Ocean Road and in Geelong, the second largest city in Victoria.

The Conference will be held at the Deakin Management Centre just out of Geelong from 24th - 26th November, 2008. The theme of the Conference will be "acoustics and sustainability" and papers are invited in all aspects of acoustics with particular emphasis on the role of acoustics in achieving sustainability. Prime tourist attractions apart from the fine historic buildings in the area are the Twelve Apostles, the Otway Fly, Port Campbell and Otway National Parks, fine beaches, waterfalls and walks.

Registration will commence Sunday afternoon in conjunction with a casual "happy hour" and there will be a BBQ on Monday evening. The Conference banquet will be held on Tuesday evening. The Deakin Management Centre has world class conference facilities and is only about 10 minutes from the city of Geelong and 20 minutes from Avalon airport. The Conference will finish at Wednesday lunchtime, allowing delegates to either return home or enjoy the local wineries and tourist attractions.

There will be a large exhibition area with some 20 exhibitors committed to being there and sponsorships for the Conference BBQ, Banquet etc Contact Norm Broner at nbroner@skm.com.au to confirm your place or Sponsorship.

We look forward to seeing you there. More information from the conference link on www.acoustics.asn.au or from aas2008@acoustics.asn.au.

*Norm Brone*

# Conference reports

## ICSV14
## 14th International Congress on Sound and Vibration

**Cairns 9-12 July 2007.**

This Congress was organised by IIAV (International Institute of Acoustics and Vibration), AAS (Australian Acoustical Society) and UNSW (University of New South Wales).

A total of 600 papers from 43 countries were published in the Congress Proceedings (on CD-ROM), from which 520 papers were presented during the congress. The scientific program consisted of 7 keynote lectures, 37 structured sessions and 24 regular sessions. The technical sessions were divided into 11 parallel sessions. In total, there were over 600 Congress delegates and accompanying persons at ICSV14. All abstracts and papers submitted to ICSV14 were reviewed by an international scientific committee, while full papers submitted by Australian and New Zealand delegates were peer-reviewed by a technical review committee.

ICSV14 received substantial financial support through sponsorship and exhibition sales. There were 11 sponsors and 25 exhibitors at the Congress. The sponsors were divided into categories corresponding to 3 Gold sponsors, 1 Silver sponsor, 3 Bronze sponsors, 2 Corporate sponsors and 2 Associate sponsors. A list of the various sponsors and exhibitors can be found in the Congress Proceedings.

Seven distinguished keynote presentations were presented during the congress by prominent researchers in the field of acoustics and vibration. The keynote speakers and title of their presentation are as follows (in their order of delivery):

- Jeong-Guon Ih (KAIST, Korea) “Acoustic holography based on the inverse-BEM for the source identification of machinery noise”
- David Thompson (ISVR, UK) “But are the trains getting any quieter?”
- Svante Finnveden (KTH, Sweden) “Two observations on the wave approach to SEA”
- Ilene Busch-Vishniac (Johns Hopkins University, USA) “The challenges of noise control in hospitals”
- Jeremy Astley (ISVR, UK) “Predicting and reducing aircraft noise”
- Colin Hansen (Adelaide University, Australia) “Optimisation of active and semi-active noise and vibration systems”
- Kimihiro Sakagami (Kobe University, Japan) “Recent developments in applications of microperforated panel absorbers”.

On the afternoon of Sunday 8 July, the Congress registration opened at the Cairns Convention Centre. That evening there was a chairpersons dinner in the Palm Court Terrace of the Hilton Cairns.

On Monday 9 July, the morning opening ceremony started with a welcome from the Congress Chair, Dr Nicole Kessissoglou (UNSW), following by a brief history of IIAV from Professor Malcolm Crocker (Executive Director, IIAV). This was followed by a few words on the technical program by Professor Bob Randall (UNSW) and a welcome from Mr Terry McMinn (AAS President). A traditional aboriginal welcome was then performed by a group of aboriginal dancers and didgeridoo players. In the evening there was a welcome reception held in the exhibition area of the Cairns Convention Centre where the 3 Gold sponsors were able to introduce their products.

On Tuesday afternoon, the technical sessions finished early and all delegates were ushered into buses for a trip up to the Rainforestation Nature Park in the tropical rainforest mountains surrounding Cairns. The various activities involved a trip in the army ducks with an educational commentary on the various tropical plants, the opportunity to try throwing a boomerang and a spear, and watching several traditional aboriginal tribal dances. The delegates also met the locals of the nature park – koalas, kangaroos, crocodiles, snakes, wombats and cockatoos!

On Wednesday evening the congress banquet was held at the Cairns Convention Centre with an underwater theme. After a welcome drink, delegates walked through a shark’s jaw into a huge hall transformed into a shimmering display of blue light and floating fish. During the dinner, the keynote speakers were thanked for their excellent presentations and contribution to the congress. Two IIAV Fellow Awards were presented to Professor Jeremy Astley (ISVR, University of Southampton, UK) and Professor Colin Hansen (University of Adelaide, Australia). After dinner, groups of delegates from various countries sang a traditional song from their country, and then a local band played. The band was excellent and the dance floor was packed with delegates dancing and enjoying themselves immensely.

Following the last technical sessions on Thursday morning, the closing ceremony was held. Dr Chong-Won Lee, Congress Co-Chair of the ICSV15 gave a presentation on the ICSV15 Congress in 2008 to be held in Daejeon, Korea. Terry McMinn (AAS President) said a few words about the next Australian Acoustical Society conference. Nicole Kessissoglou (Congress Chair) thanked all the delegates for attending as well as all the people who had worked so hard to make ICSV14 a great success. She thanked Malcolm Crocker and the IIAV Executive for their guidance, Nickolay Ivanov and his team (for the promotional material), Bob Randall (for coordinating the abstract and paper reviews and the technical program), Norm Broner (for his fantastic effort as the sponsorship and exhibition manager), Terry McMinn (AAS President), Jodie Doyle (the Congress secretary) and her team for their dedicated efforts, Max Stanton for his on-going help and support on all aspects of the congress, as well as members of the international scientific committee, chairpersons of the regular and structured sessions, and organisers of the structured sessions who worked hard preparing their sessions and inviting many colleagues to present their work at ICSV14. She also thanked the companies who sponsored and exhibited at ICSV14. Malcolm Crocker (IIAV Executive Director) also thanked the various people involved in making ICSV14 a huge success. The Congress was then officially declared as closed.

*Nicole Kessissoglou, Chair ICSV14*

# ICA2007

## 19th International Congress on Acoustics,

**Madrid 2-7 September 2007**

The International Congress on Acoustics ICA 2007, was held in Madrid over the week of 3-7 September. The venue was very spacious and served the congress well. This modern convention centre is to the east of the city and the excellent Metro system made travel very easy. As expected with a European location this was a very successful conference in terms of attendance with 1,475 registrations and 1,295 papers. Outside the time for the five plenary presentations, there was frequently a challenge to set priorities on where to go. The high number of papers and the time schedule limits led to 16 parallel session of contributed papers plus poster sessions plus workshops! There were two cultural events in the early evening and the social program included welcome and farewell cocktail parties and the conference dinner in a spectacular location outside Madrid. The full conference proceedings are available on line from http://www.sea-acustica.es/ where you can download any papers of interest. There is also a photo gallery in which you can find many of the Australian and New Zealand attendees. The organisers of ICA 2007 achieved an outstanding success, in particular Antonio Peres Lopez (the man who never sleeps). A very high standard has now been set for the organisers of the next ICA in Sydney in 2010.

*Marion Burgess*

## ICA 2007 Member reports

Two AAS members were awarded travel grants to assist them to attend.

Here, Chris Schulten and Chen Jer Ming, report on the meeting.

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Attending my first international acoustics conference at Madrid in Spain was something I wanted to do since seeing the promo for ICA2007 in Acoustics Australia (Vol 34 Dec 2006). Even though I wasn't presenting a paper, I thought it would still be an invaluable experience to hear first-hand about how others are working in acoustics around the world.

With a Travelling Grant kindly provided by the AAS to contribute towards the travelling costs, I excitedly arranged the flights and accommodation.

When I arrived at the conference venue in Madrid to register, I realised ICA2007 was much larger than any Australian acoustics conferences I had attended previously! There were people from many nations - around 1400 delegates in total – and the presentations were given in 16 concurrent sessions. The opening ceremony gave us an amazing display of Spanish flamenco dance and music, and the opening plenary lecture was a thought-provoking presentation on 'acoustics for the 21st century'. Until hearing this lecture, I hadn't really thought too much about how useful acoustics can be in space exploration!

Throughout the five days of ICA2007, there were many excellent presentations. Some of the ones most memorable for me included:

- bubble acoustics used by whales and dolphins
- active noise control in ear muffs using piezoelectric films and
- children's response to environmental noise

ICA2007 also gave me an insight into all the hard work that goes into organising an international conference. As part of the organising committee for ICA2010, which will be held in Sydney from 23-27 August 2010, I greatly appreciated the opportunity to observe the workings of ICA2007. As a practitioner of acoustics in Australia, attending ICA2007 was a great way for me to meet some of the presenters and talk to them in relaxed surroundings about their work and experiences.

I'd suggest to anyone in the field of acoustics that they consider attending an international acoustic conference.

*Chris Schulten, Senior Noise Officer, Noise Policy Section, Department of Environment and Climate Change (NSW)*

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The dates of the 2007 International Congress of Acoustics (ICA) and International Symposium of Musical Acoustics (ISMA) (held in September in Madrid and Barcelona respectively) fell in the second year of my PhD candidature with the Musical Acoustics group, School of Physics UNSW. Held triennially, this would be the sole opportunity for me to attend these two international acoustics events as a student, and this was made possible with the support of the AAS travelling grant and a conference travel grant from the UNSW Graduate Research School.

Besides meeting other researchers, I had the opportunity to present different sets of findings from my research on vocal tract acoustics at each conference. Although one expects only weak vocal tract coupling in recorder playing, some players teach the involvement of the vocal tract to elicit musical effects. To investigate, we measured the vocal tract impedances of a recorder virtuoso during playing and presented the results at ICA (details in proceedings). In contrast one might expect the vocal tract to play a more significant role in reed woodwind performance, especially if vocal tract resonances were large compared to the instrument. Direct measurements of vocal tract impedances of saxophonists during performance yielded results that were presented at ISMA; a paper detailing these findings has also recently been accepted by the journal Science.

ICA 2007 was held at the imposing Palacio Municipal de Congresos in the modern-looking Campo De Las Naciones area of Madrid. With over 1300 papers from more than 50 countries, ICA 2007 is one of the largest gatherings in the acoustics community, with colloquia in 16 parallel specialised streams running over 5 days. Over the week I was introduced to prominent researchers and authors. I also met fellow Australian acousticians; in particular, the ICA 2010 Sydney booth at the exhibition area provided a steady supply of familiar faces, vegemite and koalas! ICA 2007 culminated with the congress dinner hosted at the Palacio del Negralejo, a majestic hacienda located outside Madrid. My evening at the congress dinner was spent with fellow musical acoustics researchers from the GIPSA-Lab (CNRS, France) who turned out to be collegial, fun-loving folk, with a penchant for table games and breaking out into singing between courses!

ISMA 2007 (a satellite conference of ICA 2007) was held the following week in a more intimate setting in Barcelona, where some 100 participants presented papers and posters in musical acoustics over three days. Additionally, ISMA 2007 coincided with the National Day of Catalonia and we were treated to a special demonstration of the Sadarna – a stately Catalan circle dance – by the conference organizers during the conference dinner, while the cobla (accompanying music) was provided by delegates playing variously on the mouth-bow, didjeridu, trumpet, flute, overtone-singing, guitar and hang (a new pitched percussion) in addition to the obligatory tenora (Catalonian shawm).

Participating in ICA 2007 and ISMA 2007 was a valuable and memorable experience. It was an eye-opener into the broader world of acoustics research and researchers, and I thank the AAS for the opportunity.

*Jer-Ming Chen, Music Acoustics, School of Physics, UNSW. 2007*

# Meeting Reports

## NSW DIVISION

### Changes in Aircraft Noise Assessment - Sydney Third Runway to Brisbane

On Tuesday 22 May, Doctor Rob Bullen from Wilkinson Murray gave a presentation at NAL entitled "Changes in Aircraft Noise Assessment - Sydney Third Runway to Brisbane". This talk was well attended with approximately 35 interested listeners. Whilst the focus of the talk was on aircraft noise, the real message related to the way we assess noise in either a "descriptive" or "prescriptive" manner. For aspects of development such as land use planning, assessment really needs to be as prescriptive as possible, to provide definitive guidance for planners - a line can be drawn on a map and the noise receiver is either inside or outside.

On the other hand, when assessing the impact of a new airport, road, rail line or any other noise source, the assessment really ought to be more descriptive so that those potentially affected can understand the likely degree of affectation given their own personal sensitivity to noise. The presenter considered that (unfortunately) assessments of noise are often undertaken in a prescriptive manner when they really should give information in a more descriptive way.

This is the main reason that noise assessment of the Sydney Airport Third Runway proved inadequate in predicting the scale of noise reaction following the opening of the runway. The assessment was fundamentally based on the ANEF system, which is a reasonably good tool for land use planning, but was inappropriate for assessing or describing the potential impacts of this newly-introduced noise.

The use of descriptors such as N70 (the number of events per ? above 70dBA, where ? can be day, night, or even hour) plus a much more detailed presentation of arrival and departure routes allows a far more sophisticated but readily understandable description of potential noise impacts. The development of the Transparent Noise Information Package (TNIP) by DoTARS now provides a relatively straightforward tool for anyone impacted by aircraft noise to be able to better understand the future impacts. Of course, setting up this program still requires a comprehensive understanding of the proposed airport operations and the predicted noise levels from all aircraft types in all modes on all arrival and departure routes.

*Neil Gross*

## VICTORIA DIVISION

### The Sounds of the Sea

The final AAS Victoria Division technical meeting for 2007 was a dinner meeting attended by 31 members and friends, held on Nov 27 at "The Great Provider", Marine Pde, Elwood. The guest speaker was Dr Doug Cato of the Defence Science and Technology Organization, Marine Operations Division, and Adjunct Professor and Director, University of Sydney Institute of Marine Science.

His talk was entitled "The sounds of the sea : from songs of whales to clicks of shrimps", and was illustrated by recordings of some of the more interesting sounds. It addressed numerous questions such as – why do some whales have such complicated songs? what do they mean (are they about the meaning of life, or a form of culture)? why are tiny shrimps so noisy? why do fish produce such a variety of sounds, and aggregate in large numbers to produce choruses? how are breaking waves like fish and shrimps in the way they generate sound? why do we and the marine animals rely so much on sound in the sea, and how are we affecting each other in this respect? – and even provided some answers, though mysteries remain.

The medium, water, in which these sounds occur and are transmitted, while in some respects like air, is different in that the velocity of sound is greater (by a factor of around 5), but absorption is much less, allowing sound to travel further, so that underwater background sound comes from very great distances. In an ocean environment the ambient noise comes from many sources :surface noise from falling rain and breaking waves, biological noise from invertebrates, fish and whales, and traffic noise from ships and submarines.

Under water, an air bubble is a simple source of sound and thus a very efficient radiator. As a wave breaks, it curls over and entraps air as bubbles, compressing them so that they oscillate, radiating sound. The noise produced, known as wind dependent noise has wide frequency band and spectrum levels around 1kHz, varying from about 50 to 70 dB [in dB re 1 $(\mu Pa)^2$ / Hz] at wind speeds from 2 to 10 m/s (4 to 20 knots). Underwater background noise includes the cumulative noise from all ships at great distances, but excluding that from nearby ships. A diagram showing underwater traffic noise at 50 Hz showed the region off Australia's east coast to be amongst the noisiest.

Of the biological sounds, snapping shrimps make a clicking sound, sounding like a barbecue sizzle and containing frequencies up to 300 kHz. Their high speed claw movements form cavitation bubbles, which then collapse, producing the sharp clicks. Many fish have a swim bladder which acts acoustically like air bubbles and resonates. Large numbers of fish form a chorus with spectrum level around 20 dB above the background. Dolphins have a forehead bump (a "melon") which focuses narrow directed beams of sound. They use whistles for communication, and clicks for echo location, and are good at locating buried objects. There is evidence that the jaws and teeth of toothed whales play a part in sound reception and localization. Toothed whales include killer and pilot whales, and dolphins. Baleen whales [having a "filter" instead of teeth] use low frequency sound with most energy from 20 to 4000 Hz.

Humpback whales sing – a complex song incorporating a collection of themes comprising a sequence such as moan, whistle and sigh [as illustrated by several recorded examples]. In any particular region, all sing the same song, but the song changes with time. The songs are stereotyped, with apparently little information conveyed. Only male humpback whales sing, the evidence being that it is associated with breeding. Humpback whale song has been referred to as "culture", because changes in song are passed on by learning. Some recent work with whales tracked visually and acoustically off eastern Australia has shown that their sounds become louder as the background noise level increases.

After his talk, Doug Cato answered several audience questions. In conclusion, Andrew Rogers [as acting chairman] thanked Doug for his most interesting illustrated talk, with the thanks carried by acclamation.

*Louis Fouvy*

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

The International Congress on Acoustics for 2010 (ICA2010) was formally launched during the closing ceremony of ICA 2007 in Madrid in September. On behalf of the Organising Committee, Marion Burgess, both congratulated the ICA2007 committee on achieving such an outstanding success and invited all the registrants to meet together again in Sydney in 2010. Her presentation highlighted the conference location right at Sydney Harbour and featured images of the tourist opportunities associated with travel to Australia for this event. It was also the official launch of the logo for ICA 2010.

The ICA2007 organisers kindly provided a booth for the promotion of ICA2010. ICMS, the conference management company appointed to provide services for ICA2010, had obtained sponsorship to attend for promotion and Jodie Doyle was present at the booth for the duration of the congress. With great teamwork, the committee prepared promotional material for the booth in a very cost effective manner. This included Dave Anderson bargaining at Paddys Market for a bulk buy of clip-on koalas and Chris Schulten and Marion Burgess tagging some 1500 koalas. These clip-on koalas were a most successful promotional aid encouraging many repeat visits to our booth. The vegemite tasting had varied responses but all went to making our booth a welcome stopping place for the conference-weary participants. The Australian and New Zealand registrants all helped with promotion at the booth. In particular special thanks should go to the Rumbles – Ron surprised many passing by with his skills at tossing the koalas right into their hands.

The committee is very aware of the challenge in organizing such a large event but has been encouraged by the willingness of members of the Society to undertake tasks. The ICA 2010 provides the opportunity to both showcase the high standard in Australia for the wide range of topics within acoustics as well as to learn from international colleagues.

For more information on the ICA2010 go to **www.ica2010sydney.org**

*Marion Burgess, Chair ICA 2010*

# Book Reviews

## Handbook of Noise and Vibration Control

**Malcolm Crocker editor**

John Wiley 2007, 1569 pp (hard-cover), ISBN-978 0 471 39599 7.
Approx Price A$250

This Handbook aims *"to fill the need for a comprehensive resource on noise and vibration control"* (Preface). This grand goal has produced a thick book comprising 130 chapters divided into 11 main sections. Malcolm Crocker has written 14 of the chapters while the others have authors from over 18 countries. The review panel numbered over 250 from 30 countries. It has truly been a challenging task but Crocker has had experience with similar books; "Encyclopedia of Acoustics" (1997) and "Handbook of Acoustics" (1998).

I need to declare an interest being a joint author for one small chapter, for which I have been provided with one copy of the handbook. This has given me a little insight into the amount of work undertaken by the editor and the inevitable slipping of the deadlines when there are so many authors involved: contributor invitations commenced back in 2003. The authors provided their chapters independently, leaving the editor the responsibility for consistency among chapters and to check that no vital topics were omitted. A number of Australians appear in the author listing and also in the long reviewers list.

So what does the Handbook cover? The first 10 chapters introduce the fundamentals of acoustics and noise. These are followed by 8 chapters on fundamentals of vibration; 4 on hearing and speech, 12 on effects on people; 19 on transducers and measurement techniques; 14 on principle of noise control; 15 on machinery prediction and control; 11 on transportation noise and vibration; 8 on interior transportation noise and vibration; 17 on control in buildings; 12 on community noise prediction and control.

A "Handbook" needs to be a reference book, with explanations of terms and concepts, plus provide guidance on the solution to a problem. The reader should be able to find the information easily. This book achieves this in a number of ways. The grouping of the chapters allows the reader to go first to the part of the book dealing with the general topic. The clear titles identify the subject matter of each chapter, for instance there are three chapters dealing with classification of sound insulation; one for the ISO ratings, one for US codes and one for European codes. Then there is a comprehensive index. The strict page limit applied by the editor has ensured concise chapters, each of which has a reference list for further information.

With so many authors some inconsistencies in terminology have inevitably slipped past the editor's eye, but I found none that were likely to lead to confusion. A critical reader could find a few omissions; for example the effects of shock on humans is covered but I could not find anything on shock isolation seating. Nor could I find anything on the Star rating for building insulation which is promoted by the AAAC in Australia. However any small omissions are far outweighed by the advantage of having such a comprehensive coverage of noise and vibration and its control in one volume.

This handbook is highly recommended for the library of any organisation or company that has any dealing with assessment and control of noise and vibration. It would be of great assistance to anyone commencing work for an acoustic consultancy and also a valuable reference for more experienced acousticians who encounter a new challenge. *Marion Burgess*

*Marion Burgess is a research officer with the Acoustics and Vibration Unit of UNSW@ADFA who has some experience with measuring and assessing effects of vibration in the workplace.*

## Sound System Engineering: Third Edition

**Don Davis and Eugene Patronis Jr.**

Elsevier/Focal Press, 2006, 489pp
(Hardcover edition)
ISBN-13: 978-0-240-80830-7
ISBN-10: 0-240-80830-4
Approximately $AUD136.50
(dymocks.com.au)

*Sound System Engineering* is a single textbook dedicated to component design and detailed analysis of audio systems.

Don and Carolyn Davis are well respected in the audio community and are probably best known for the lecture circuit tutorials and workshops, mainly in the United States, presented under the SynAudCon label. Synergetic Audio Concepts provides practical, in-depth training on the principles of audio and acoustics. The experience obtained in teaching others about audio has carried through to this publication and provides a solid understanding of the design, installation, commissioning and operation of sound system components.

Eugene Patronis Jr. replaces Carolyn Davies in this third edition and brings a technical writing style with added detail that enhances explanation of topic areas. This complements the chapters by Don Davis to provide a different presentation of similar topical information.

The information contained in the third edition of SSE is very similar in content to previous editions. Key headings have been identified which provides easy referencing, but other than minor formatting and removal of worked examples, the majority of text written by Don Davis remains unchanged. Chapters include Loudspeaker Directivity and Coverage, Large and Small Room Acoustics, Designing for Acoustic Gain and Speech Intelligibility.

Chapters re-authored by Eugene Patronis Jr have been reworked and in some cases, provide a more in-depth presentation of the topic. These include "Microphones", "Loudspeakers and Loudspeaker Arrays", "Signal Processing" and the final chapter "Putting It All Together."

In this third edition, Don Davis credits Glen Ballou (*Editor of Handbook for Sound Engineers*) in the Preface, for "transcribing our material into a publishable format". The table of contents at the beginning of each chapter follows the format familiar to readers of the *Handbook for Sound Engineers* (reviewed *Acoustics Australia* Vol 34 December 2006).

The third edition has almost 180 fewer pages than its predecessor. Chapters specifically dedicated to more technical aspects including worked examples have been edited or removed with some of this information being absorbed in other chapters. The new format provides better sectional overview with clear headings.

The familiar yellow cover has remained and will continue to be an easily identified text on the bookshelf.

*Sound System Engineering* is an American text. As a result, formulas and worked examples use American terminology and care should be taken in reading and applying examples where formulas are presented for different markets. The first chapter "Mathematics for Audio Systems" provides comment on conversion factors between SI and U.S. base units.

*Sound System Engineering* is a stand-alone reference text that can be used by professionals and those looking to learn more about sound reinforcement design and analysis. The text provides comprehensive information on all represented topics and provide references to related texts for further reading. This is an excellent text covering all the key topics for sound system design.

Some useful links:
www.soundsystemengineering.com
www.synaudcon.com

*Tim Kuschel*

*Tim Kuschel is an acoustic consultant and architectural projects coordinator. With a background in architecture and music, his business, GUZ BOX design + audio specialises in architectural acoustics and sound reinforcement design.*

## Noise of Polyphase Electric Motors

**Jacek F. Gieras, Chong Wang and Joseph Cho Lai**

Marcel Dekker Inc 2005, 392 pp (hard-cover), ISBN-9780824723811, ISBN-10: 0824723813. available from Palgrave Macmillan, www.palgravemacmillan.com.au (special offer of free delivery via enquiry to Caroline.Bruckner@macmillan.com.au )
Approx Price A$210

The year 2006 appears to be a new milestone in the understanding of the vibro-acoustic behaviour of electric motors. This book by Gieras, Wang and Lai represents the latest step in the series of the famous text books; the first analysis of the noise of the electrical machines was authored by Jordan (1950); followed by Heller and Hamata (1977) dealing with the magnetic field analysis in details; Yang (1981) and Timar et al. (1989). As time has progressed, one may find a serious improvement in the scientific and theoretical explanations.

The book comprises 11 chapters mixing the fundamentals with the theoretical explanations and practical considerations. The authors aim to summarise principles, the existing knowledge and then introduce new ideas and methods. What is really novel in this book is that it deals with the permanent magnet synchronous motors and the Statistical Energy Analysis (SEA). To clarify this; polyphase electrical motors are electrical rotating machines where polyphase winding (usually three-phase winding) is located in the stator core slots. In general, this polyphase winding is fed with a balanced system of sinusoidal phase currents. The permanent magnet synchronous motors are a subset of the polyphase electric motors. In relation to SEA the authors say very honestly (on p 258) – and it is correct to do so – that the use of the SEA for electrical motors' noise calculation is still difficult and there is not available a well-established procedure.

Showing the strong relation between the mechanical vibration and the acoustics is particularly a great value of this book. Sometimes the reader may have the feeling that the basics of the acoustics and vibration theory is too long but still it can be accepted as the technical experts on the field of electrotechnics, the real consumers of this book, may need a deeper refreshment of these topics.

This book represents the highest level of the theoretical explanation across many relevant areas including the electromagnetic field theory, the physics and the acoustics. The visualisation of the magnetic field analysis performed by using the Finite Element Method provides a good picture on the electromagnetic field distribution along the stator bore. The great set of figures, diagrams, pictures is a help to the reader to understand and "feel" the theory. The theory is fortunately mixed with practical solutions based on the wide-ranging experience of the authors and detailed introduction may be found into the world of the relevant standards.

The enormous set of the rich Bibliography with 256 titles will be a great help for technicians dealing with noise control of electrical motors in industrial practice. While this book is not for the novice engineer it is certainly a valuable reference book for those working with the noise of motors and with noise control in general.

*Laszlo Timar-Peregrin*

*Prof. em. Dr. Laszlo Timar-Peregrin (RMIT) is the author of 8 books and 240 scientific papers – the majority on the topic of electric motors.*

# *Diary*

## 2008

**31 March - 2 April, Leuven**
LSAME.08 - Leuven Symposium on Applied Mechanics in Engineering
http://www.mech.kuleuven.be/lsame08/

**12 – 15 May, Sopot, Poland**
10th School on Acousto-Optics and Applications
http://univ.gda.pl/~school/

**20 - 23rd May, Canberra**
Audiological Society Australia Annual Conference
www.audiology.asn.au

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

**6 – 10 July, Daejon, Korea**
15th International Conference on Sound and Vibration
http://www.icsv15.org/

**7 – 10 July, Stockholm**
18th International Symposium on Nonlinear Acoustics (ISNA18)
http://www.congrex.com/18th_isna/

**21 – 25 July, Mashantucket**
Noise Effects 2008.
http://www.icben.org

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

**26 - 29 October, Shanghai**
Internoise 2008
www.internoise2008.org

**24 - 26 November, Geelong**
Australian Acoustics Society National Conference
'Acoustics and Sustainability'
http://www.acoustics.asn.au/conference-link.shtml

## 2009

**6 - 10 September, Brighton**
Interspeech 2009
www.interspeech2009.org

## 2010

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

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

# *New Members*

**Member**

Granger Bennett (WA)

Craig Beyers (QLD)

Rhys Brown (QLD)

Mathew Bryce (VIC)

Sandro Ghiotto (WA)

**Graduate**

Mathew Vance (VIC)



Acoustics Australia

## Annual Index
## Vol 35 2007

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* Payment of annual subscription
* Proceedings of annual conferences

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# Just *Showing Off*

**Type 2250**

You've heard about it from colleagues, or glanced at it in an advertisement. The look is perfect Brüel & Kjær - ***simply unmistakable***.

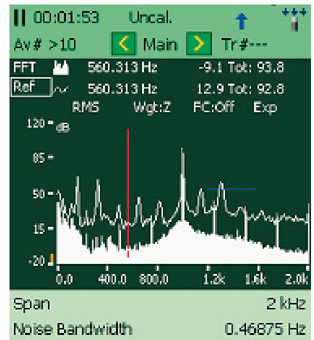
Unparalleled performance and more integrated applications. This powerful tool was built just for you to help do your work better and faster!

Like the great green sound level meters of the past, Type 2250 has become an industry standard, setting sales records worldwide.

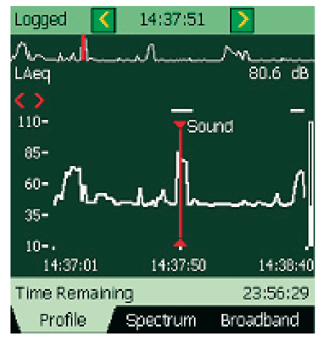
Don't take our word for it, just listen to what our customers have to say.

- "…after spending a few days with my new 2250 all I can say is "wow." This is a product done right. Not only is it extremely easy to use (and has a great long-lasting battery) the documentation and software is excellent. The 2250 represents Brüel & Kjær at its best."

*For more details please contact your local sales representative or go to* ***bksv.com***




**6400 Line FFT**




**Noise Profiles**


**Reverberation Time**

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**Type 2250:** *Easy, Safe, Clever!*

**Brüel & Kjær**