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• Transient vibrations in pipes
• Trailing edge noise from airfoils
• Acoustic coupling in vocal tract

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PAPERS

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Message from the President

This column gives me the opportunity to raise three important issues. The first concernts some apparent confusion about the status of this Society. The Australian Acoustics Society is a 'Learned' Society rather than a 'Professional' Society. Typically, 'Professional' societies have accreditation and monitor professional conduct. The Australian Acoustics Society does not accredit its members or monitor their professional work. Admission to the Society is open to all people interested in acoustics and companies and other organisations who may wish to support the Society. The grade of membership is determined by qualifications and experience (see http://www.acoustics.asn.au/sql/membership.php). Members are obliged to uphold the 'Code of Ethics' (http://www.acoustics.asn.au/code-ethics.shtml).

As members will be aware, the organisation for this year's national conference in Geelong is well under way. The conference presents valuable opportunities not only to learn but to network with others in the industry, the equipment distributors, researchers and presenters. I urge readers to support the conference by submitting papers and attending the conference. The deadline for abstract submissions has been extended to April 30, 2008.

At the November 2007 Federal Council meeting, our long standing General Secretary David Watkins, tendered his intent to resign from his post. David agreed to serve as General Secretary in 2008 to allow time to select a new secretary and to provide the opportunity to train the new person into the position. The position is a part time salaried position – see http://www.acoustics.asn.au/general/current-jobs.shtml for more details. Please email applications to president@acoustics.asn.au by the end of May. I take this opportunity to thank David for his hard work over many years in his role of General Secretary.

Terrance Mc Minn

From the Editors

For at least a little while longer, one of the essential facilities of a modern city is an airport. Some editors of this journal work at the University of New South Wales and are consequently only 5 km from the international airport. Although it is sometimes most convenient to be only 10 to 15 minutes away from an airport, there are some disadvantages.

The physical dangers of living close to an airport are well known. There is a finite risk of an incident that airlines prefer to call 'aircraft making premature contact with the ground'. There is also the risk, especially in these days of reduced maintenance budgets, that some component part of an aircraft unexpectedly separates and thereafter follows an independent flightplan. The noise associated with living under a flightpath is often thought of as merely an annoyance and an issue connected with quality of life, rather than a direct health threat.

However, a recent study* reports the effects of night-time noise exposure on blood pressure monitored for 140 subjects living around four major airports, including Heathrow. (Indeed even just thinking about Heathrow can probably cause the blood pressure of airline passengers to rise.) The study found that subjects experienced significant increases in blood pressure in response to noise events greater than 35 dBA, even if the subjects did not wake up in response to the noise. The effect appeared to be independent of whether the noise source were due to aircraft, road traffic, or even loud snoring. Such increases in blood pressure due to noise are thought to play a role in the development of hypertension and cardiovascular disease.

Many, if not the majority, of our readers are involved in some aspect of noise reduction. This can range from techniques for reducing noise from aircraft, road traffic, appliances and machinery to devices designed to reduce sleep apnea and snoring. Other readers work on improving sound insulation. So, all you valued readers and colleagues who work to make life quieter, you now have reason to think proudly of yourselves as 'health-care workers', too.

John Smith

*European Heart Journal doi:10.1093/eurheartj/ehn013

A REVIEW OF AIRFOIL TRAILING EDGE NOISE AND ITS PREDICTION

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ABSTRACT: If technology such as aircraft, submarines and wind turbines are to further reduce their noise emissions, then a better understanding of airfoil trailing edge noise is required. This paper will discuss the physical causes of trailing edge noise and then review the methodologies used over the past couple of decades to model and to estimate trailing edge noise. A comprehensive reference list is given for readers wishing to learn more about this important area of aeroacoutsics. It is shown that one of the major restrictions to further development of prediction methods is a lack of suitable experimental data for validation purposes. Additionally, new turbulence models are needed to improve noise prediction, especially at high frequency.

1. INTRODUCTION

In recent years, engineers and scientists have been able to reduce aeroacoustic and vibroacoustic noise to such an extent that broadband sources are now limiting further noise reduction. This is particularly true for technology that utilise airfoils and airfoil-like shapes that generate broadband noise at the trailing edge (TE). For example, TE noise is an important component of civil aircraft airframe noise during approach and landing. In fact, the long-term goal of the aviation industry is to reduce aircraft noise by 20 dB [1] and the control of TE noise has been identified as critical to achieving this. In naval applications, TE noise from hydrofoils and propellers must be controlled in order to increase the stealthiness of underwater and surface craft [2]. TE noise is also a major noise generation mechanism for the rotor blades of wind turbines [3] and helicopters [4], and limits their use in urban areas. Indeed, the list of applications for which TE noise is significant is extensive and illustrates the universal need for quiet airfoil designs. However, the design of quiet airfoils requires accurate methods to predict TE noise in the far-field.

Predicting TE noise has been an on-going challenge for engineers over the last 30 years. Calculating TE noise is made difficult due to the complexity of the noise source, which is turbulent fluid flow. The complex and stochastic nature of turbulence has forced the development of methods that use simplified turbulence models to calculate noise. Unfortunately, these assumptions hinder the design of new, quiet airfoils due to their limited accuracy. Recent advances in computing power now provide a much better representation of turbulent flow and therefore open the possibility of radically new, quiet airfoil shapes.

It is the goal of this paper to review the various methods of modelling and estimating TE noise and to discuss what challenges remain to develop accurate prediction methods. It is hoped that the paper can be used as a starting point for those wishing to understand and compute TE noise. Key references are provided that can be used to obtain more detail.

2. THE MECHANISM OF TRAILING EDGE NOISE GENERATION

Airfoil self noise occurs when an airfoil shape is placed in an otherwise uniform and steady fluid flow. As in most aeroacoustic noise generation situations, noise is generated by flow unsteadiness. In the case of airfoil self noise, it is the interaction of flow unsteadiness (usually in the form of fluid turbulence) with the surfaces of the airfoil that generates sound. There are a variety of specific noise-generating flows associated with airfoil self noise that are concisely summarised in Ref. [5]. These are: (1) laminar boundary layer – vortex shedding noise; (2) separation stall noise; (3) tip vortex formation noise and (4) TE noise.

Trailing edge noise is caused from the interaction of turbulence with the TE. Fluid turbulence is a term that characterises the irregular flow of air and other fluids past (airfoils) or through objects (pipes, engines) and it is the usual condition of airflow considered in engineering applications. Turbulent flow can be thought of as a continuous series of randomly orientated eddies of various sizes and intensity that are linked in a form of energy cascade. This energy cascade is the physical mechanism that dissipates the energy that the immersed object imparts to the flow (i.e. the fluid reaction to drag and lift). Hence, turbulent flow is unsteady and contains fluctuating eddies with a large range of sizes (or scales). Fluctuating eddies by themselves are a source of noise, the most familiar form being caused by airline jet engines. The addition of a close boundary, such as an airfoil, will amplify the noise generated by fluid turbulence.

Figure 1 is a diagram illustrating TE noise. On the left of the figure are some technologies where TE noise needs to be considered in their design. On the right of the figure, the major flow processes that occur over an airfoil placed in an otherwise uniform, steady and quiet fluid stream are shown. The flow encounters the leading edge of the airfoil and forms a boundary layer due to fluid shear that normally transitions to a turbulent state on the surface of the airfoil. Figure 1 illustrates the growth of this boundary layer over the airfoil surface and defines its thickness at the TE as δ. Turbulent eddies are formed within the boundary layer and it is the interaction of these eddies with the TE that generates broadband aerodynamic noise.

In acoustic terms, the edge presents itself as a sharp impedance discontinuity. This discontinuity scatters acoustic waves generated by fluid turbulence (considered to be quadrupoles) and creates an intensified radiated acoustic field [6]. When the dimensions of the airfoil are small compared with the radiated acoustic wavelength (chord = $C << \lambda$ = acoustic wavelength), then the fluctuating flow causes surface pressure fluctuations that are (effectively instantaneously) transmitted across the airfoil in the hydrodynamic near field. In this case, the radiated sound is of dipole character with strength proportional to the fluctuating total force amplitude. This type of noise amplitude scales with the sixth power of the Mach number (M^6).

When the airfoil dimensions are large compared with the radiated acoustic wavelength ($C >> \lambda$), the TE will diffract turbulence induced quadrupole noise. In this case, the intensified radiated noise is still of a multipole nature (sometimes known as a 3/2 pole) with an amplitude governed by the intensity and spatial distribution of the turbulent field. Diffracted turbulence scales with M^5, hence for a subsonic flow ($M < 1$) this noise is more intense than the dipole case described above. More detailed descriptions of TE noise generation processes can be found in Ref. [2] and theoretical descriptions of acoustic scattering and diffraction mechanisms can be found in Ref.[7].

The airfoil in Fig. 1 also includes TE bluntness of thickness h. The effect of bluntness is to create vortex shedding in the wake of the airfoil. This creates a stream of counter-rotating vortices with a higher span-wise (z-direction) coherency than the turbulent eddies in the turbulent boundary layer.

This results in tonal noise, sometimes of dipole nature if the wavelength is smaller than the chord. The diffraction of boundary layer turbulence, on the other hand, creates broadband noise up to high frequencies.

Figure 2 illustrates the general features of the noise spectrum created by these two sound generation mechanisms at the TE.

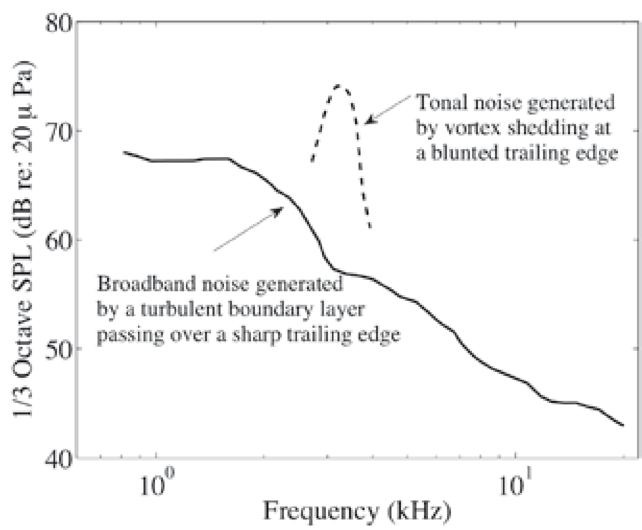

Figure 2. Illustration of flow induced TE noise spectrum. This figure was constructed using data from Ref. [5]. The broadband noise spectrum was measured for a NACA 0012 airfoil with a sharp trailing edge operating at a Reynolds number of 7.2×10^5 and a Mach number of 0.2. The tonal noise spectrum was measured for a NACA 0012 airfoil with trailing edge bluntness ($h/C = 0.06$) operating with a Reynolds number of 2.8×10^6 and a Mach number of 0.2.

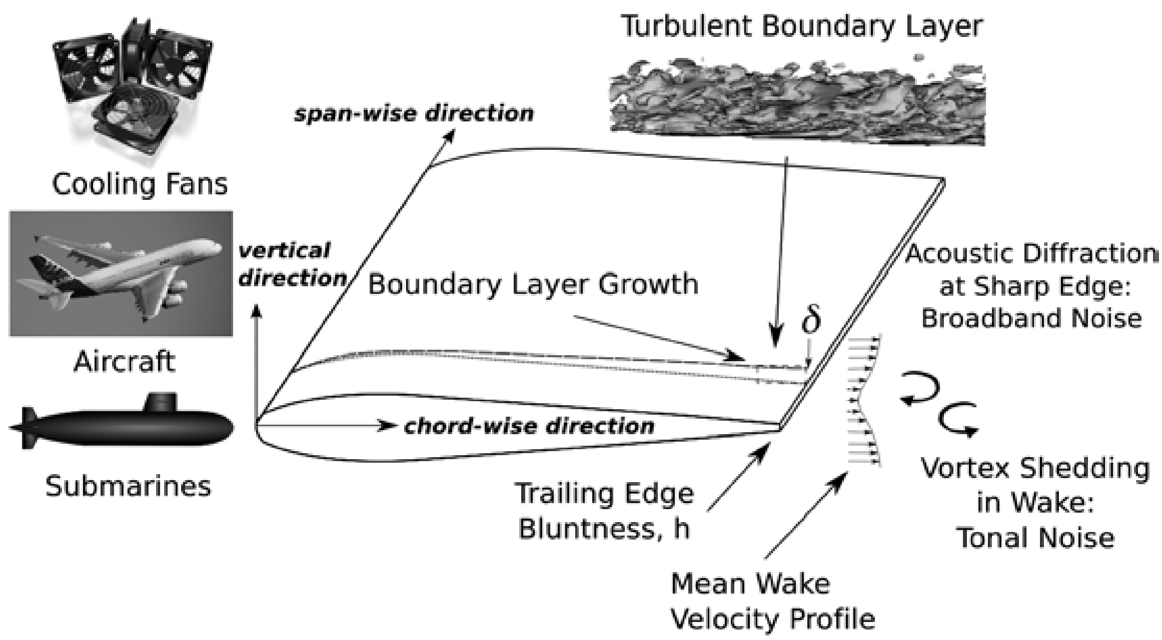

Figure 1. The generation of TE noise. On the left are some technologies where TE noise limits their operation. On the right is a schematic of an airfoil, showing boundary layer growth over the surface and aerodynamic noise generation mechanisms at the TE. The turbulent boundary layer (upper right) is a computer simulation showing iso-vorticity contours of the boundary layer structure.

3. THEORETICAL BACKGROUND

To illustrate the challenges involved in modelling and predicting TE noise, consider Lighthill's acoustic analogy [8]

$$\frac{\partial^2 \rho'}{\partial t^2} - c_0^2 \frac{\partial^2 \rho'}{\partial x_i^2} = \frac{\partial^2 T_{ij}}{\partial x_i \partial x_j} \tag{1}$$

which expresses the wave equation for fluctuating density (ρ') with a source term proportional to Lighthill's stress tensor (T_{ij})

$$T_{ij} = \rho u_i u_j - \tau_{ij} + p' - c_0^2 \rho' \delta_{ij} \tag{2}$$

where u_i, is the velocity of the flow in the *i*-direction, τ_{ij} represents the viscous forces, p' is the fluctuating pressure and δ_{ij} is the Kronecker delta (see Ref. [9] for a complete derivation). What is interesting about this formulation is that no assumptions have been made during its derivation, which implies that all fluid flow can be described as an acoustic field. The final three terms are usually neglected when performing an aeroacoustic prediction. Hence, the source of noise is related to the fluid motion (first term), which is the Reynolds stress tensor, a familiar quantity to those involved in turbulent flow research.

What these methods tells us is if the turbulent flow about the TE were perfectly known, then calculating far field noise would be accurate and the design of quiet airfoils would be a trivial matter. However, this is not the case and describing TE turbulent flow remains a great scientific challenge. Therefore, the development and design of quiet airfoils is intricately linked to the development of turbulence simulation techniques.

4. TRAILING EDGE NOISE COMPUTATION

Figure 3 is a schematic 'road map' that shows the various methods of computing TE noise. Given the required inputs of airfoil geometry and flow condition (e.g. Mach number, Reynolds number, etc), different methods can be chosen in a left-to-right manner to arrive at an estimation of TE noise (the output).

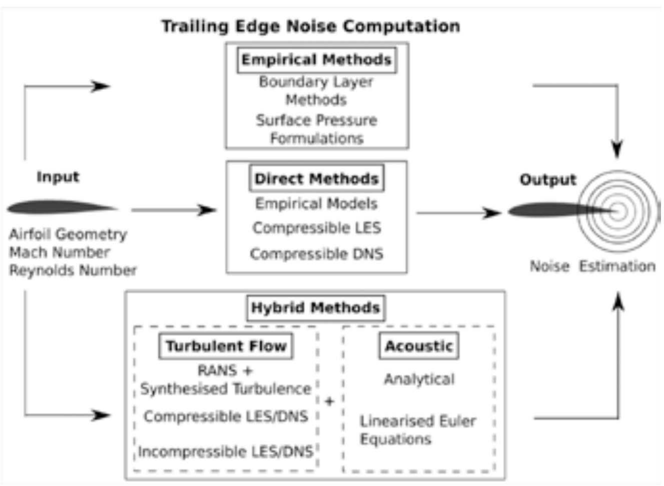

Figure 3. TE noise computation road map.

The techniques have been classified into three broad areas that have been termed as the empirical, direct and hybrid methods. The empirical methods were derived from anechoic wind tunnel results. In the direct method, an estimation of the noise can be made in a single computational step that calculates both turbulent flow and noise. The hybrid method assumes that the flow and noise are decoupled and can be calculated in two steps. An estimation of the turbulent flow field is obtained in the first step and the sound field is then computed using the turbulent flow field as the source in the second step. The various methods are discussed below.

4.1 Empirical Methods

4.1.1 Boundary Layer Methods

Approximately 25-30 years ago, engineers and scientists did not have the ability to perform complex computer simulations of turbulent flows. Therefore, they relied upon models derived from experimental measurements. Two well known empirical models have been developed by NASA and are based on boundary layer height at the TE and airfoil Reynolds number. The most straightforward was developed by Schinkler and Amiet [10] for helicopter rotors, which was subsequently used to good effect in Ref. [11] to predict the noise of wind turbine blades. This took the form of a scaling law prediction for 1/3 octave noise

$$\mathbf{SPL}_{1/3} = 10\log_{10}\left\{3.5U_\infty^5 D\frac{\delta s}{r^2}\left(\frac{S}{0.1}\right)^4\left[\left(\frac{S}{0.1}\right)^{1.5} + 0.5\right]^{-4}\right\} \tag{3}$$

where U_∞ is the free-stream velocity, D is a user defined directivity function, s is the span, r is the distance to the observer and $S = f\delta/U_\infty$ is the Strouhal number. A more detailed empirical model was developed by Brooks, Pope and Marcolini [5], known as the BPM model, which incorporates more types of airfoil self-noise (i.e. bluntness, separation, etc). While more comprehensive, the BPM model can be easily programmed and quickly give an estimate of TE noise.

Hence, the simplest TE noise predictive scheme would consist of a method of computing the boundary layer thickness at the TE and substituting this into Eq. (3) or the BPM model. However, as these models are empirical, they are limited to the range of experimental parameters that was used to develop them. These models also have a limited ability to incorporate changes in the turbulence field induced by geometrical changes. Therefore, their applicability is restricted to common airfoil shapes that have no span-wise variation in geometry or other modification to the TE (e.g. brushes, serrations, etc).

4.1.2 Surface Pressure Formulations

An alternative to the empirical models based on boundary layer thickness are the formulations based on fluctuating surface pressure. The advantage of these methods is that they eliminate the need for estimates of fluctuating velocity about the TE. It does this by re-casting the problem in terms of fluctuating surface pressure and the diffraction of evanescent hydrodynamic waves at a knife-edge. At the time of the development of these models, there was much progress in surface pressure measurement techniques (e.g. Ref. [12]) and this method enabled the use of these empirical measurements or a modelled turbulent wall pressure spectrum to predict noise.

Using this method, the peak radiated sound spectrum can be estimated by [2]

$$\Phi_{p,\mathrm{rad}}\left(r,\omega\right)_{\mathrm{PEAK}} \approx \frac{1}{2\pi} M_c \frac{s\Lambda_3^2}{r^2} \phi_{pp}\left(x,k_3,\omega\right) \tag{4}$$

where M_c is the average convective Mach number of turbulence past the TE (i.e. the velocity of the turbulent eddies, not the flow, divided by the speed of sound), Λ_3 is the span-wise turbulent length scale, ω is the radial frequency, ϕ_{pp} is the transverse surface pressure fluctuation spectrum, with a spatial Fourier decomposition across the span (into wavenumbers, k_3) and across time (into frequencies, $\omega = 2\pi f$). Note that this method is limited to turbulence fields that are both spatially and temporally homogeneous, a situation that may not occur for new low noise airfoil designs.

The key to an accurate TE noise prediction is to estimate the turbulence properties correctly. Good noise predictions were made by Brooks and Hodgson [12] using data obtained from simultaneous noise and surface pressure measurements. For cases where the exact surface pressure spectrum (i.e. the turbulent field) is not known, estimates are used [2] and predictions are poor at high frequencies. Recently, Ref. [13] has used a surface pressure formulation with a boundary layer numerical flow simulation to improve the estimate of the fluctuating surface pressure spectrum. While an improvement in predicting the overall shape of the noise spectrum has been made, noise predictions are still 10 dB below what is measured experimentally. This can be attributed to inaccuracies in the modelling of turbulent flow properties. Much better agreement between theory and experiment was obtained by Casper and Farassat [14] using their so-called 'Formulation 1-B' surface pressure formulation technique.

5. COMPUTATIONAL AEROACOUSTICS

Since the development of the empirical models summarised above, there has been a rapid advance in the ability to compute complex turbulent flow fields. This has been driven by enormous increases in computational power. This development has resulted in the emergence of computational aeroacoutsics (CAA) that uses computational fluid dynamics (CFD) solutions to calculate the properties of the noise sources and, in some cases, propagate acoustic disturbances to the far-field.

In this paper, CAA encompasses both direct and hybrid methods. In the direct method, a single simulation calculates the turbulent flow field and propagation of acoustic waves. This form of CAA is used for compressible turbulent jet noise [15] however; there are limited applications of the direct method to TE noise in the literature. Only two examples could be found at the time of writing. One is the numerical study of Ref. [16] where low Reynolds number flow was simulated over an infinitely thin, two-dimensional TE and compared with a theoretical model. The other is the high Reynolds number application of Ref. [17] that shows promising results but is not validated against theory or experiment. It is unlikely that the direct approach will be used routinely for high Reynolds number TE studies in the near future. This is because of the extreme computational cost required to simulate the fine turbulent structures in the boundary layer as well as resolve high frequency acoustic waves at large distances into the far field.

Hybrid methods represent the second main area of CAA and is the most popular technique for simulating TE noise. The basis of hybrid methods is to split the noise prediction into an aerodynamic/turbulence part to calculate the Reynolds stresses, then to use these as source terms in a second acoustic computation. Splitting the fluid dynamics and acoustic propagation is a sensible approach due to the very large separation of velocity and density scales. For example, the ratio of gas velocity at a listener's ear in the far-field to that at the source (i.e. the TE) is of the order of 10^{-3}-10^{-4}. Resolving such a wide variation in scales over a large computational domain is still too challenging for most desktop computers. This is because a large number of cells and costly, high-order discretisation methods are required. Excellent reviews of CAA numerical methods can be found in Refs. [18, 19, 20, 21] where these issues are discussed in more detail.

In the remainder of the paper, hybrid methods relating to TE noise estimation will be reviewed. Procedures that model the turbulent sources will be discussed first followed by techniques to predict far-field sound.

5.1 Prediction of Turbulent Flow

5.1.1 Synthesised Turbulence Methods

The most common method to simulate turbulent flow is to solve the Reynolds Averaged Navier Stokes (RANS) equations. This method produces a time-averaged flow field using a model to estimate the effects of turbulence. It produces mean turbulence quantities, such turbulent kinetic energy and dissipation rate, which can be used to calculate the integral scales of turbulence at every position within the flow. As the solution is time-averaged, spectral information about the Reynolds stresses is lost. It is not the intention to review RANS based CFD methods in this paper, therefore the reader is referred to Ref. [22] which provides one of the most thorough descriptions of RANS models and their implementation. What we are interested in here is how to use time averaged turbulence quantities to recreate a stochastic turbulence field.

Recently, methods have been devised to synthesise a turbulence field based on these time-averaged turbulence quantities [23, 24]. There has been a number of model turbulence spectrums developed over the years from experimental velocity correlations. Commonly used models are the Leipmann or Von Kármán spectra (see for example the appendix of Ref. [25]). These model spectra use the turbulence length scales calculated by the RANS model. Once the spectral information has been assumed, a deconvolution procedure is used to synthesise the transient velocity field at each point required by the noise prediction model. This information is subsequently used as the source terms in a separate acoustic prediction method.

5.1.2 Simulated Turbulence using Large Eddy Simulation

Large eddy simulation (LES) is more accurate than RANS for modelling a turbulent flow field however, until recently, LES has been limited to flows of academic interest due to its high computational cost. Projected performance increases in computational power and the rise of massively parallel computing, makes LES possible for engineering flows, such as turbulent TE noise.

The basis of LES is spatial filtering, rather than time-averaging, as is the case in RANS modelling. Spatial filtering has the consequence that turbulence scales larger than the grid size are directly resolved with no modelling assumptions. For turbulence scales smaller than the grid size, a special turbulence model is used, known as a sub-grid-scale (SGS) model. The model is transient, simulating the fluctuations directly above the grid scale, therefore requiring significantly more memory than RANS, which is steady state (as the turbulence is averaged over infinite time). As the grid is refined, LES will progressively resolve smaller turbulence scales, until eventually all turbulent scales are reproduced. In this case, the simulation is known as a Direct Numerical Simulation (DNS) and no SGS model is required. Current computers are not capable of performing a DNS of TE flows at realistic Reynolds numbers, hence LES will be the computational tool of choice for engineers and scientists wishing to calculate airfoil noise in the coming years. Reference [26] is an excellent textbook that describes the use of LES in acoustic calculation and can be used to learn more about these methods.

Recent attempts at coupling turbulence and TE noise calculations [27, 28, 29] reduce computational expense by modelling aspects of turbulence using an eddy viscosity model. Models of this type poorly predict noise levels at high frequencies [20]. It is important to correctly account for high frequency noise components as they are annoying to the human ear and are heavily weighted in aircraft and other noise regulations. Therefore, future research using LES for TE noise will need to develop new 'acoustical' SGS models to account for missing turbulence scales.

5.2 Prediction of Noise

5.2.1 Analytical Noise Prediction

Our attention now focuses on how to estimate noise from transient turbulent flow data (synthesised or simulated). Traditionally, analytical solutions are used. Ffowcs Willams and Hall [30] provided one of the first analytical solutions of Lighthill's acoustic analogy for turbulent diffraction about a semi-infinite half plane (knife-edged TE). This method derived an analytical Green's function for an idealised TE. Further theoretical development of the TE scattering and diffraction problem has been performed by Refs. [2, 6, 31, 32, 33].

The Ffowcs Williams Hall method has been successfully used by a number of researchers to calculate TE noise from incompressible LES simulation data [28, 29]. An incompressible LES assumes infinite sound speed in the fluid, hence no coupling is permitted between the fluid dynamics and acoustics. Reference [27] used a hybrid incompressible LES and Ffowcs-Williams Hall technique in a numerical optimisation routine to design a quiet airfoil. They found that by changing the shape of the TE, turbulent energy could be redistributed over smaller scales resulting in lower overall noise levels.

If a compressible LES can be performed, then analytical estimates of noise can theoretically be obtained using a free-space Green's function. This procedure uses the theory of Curle [34]. However, there appears to be no published study that couples a compressible LES with Curle's formulation. The Ffowcs William Hawkings [35] equation can be considered an extension of Curle's formulation that takes into account moving noise sources (such as a rotor blade) with respect to the listener. Reference [36] used the Ffowcs William Hawkings equation with a compressible LES to compute TE noise. While excellent results were obtained, they are yet to be validated. In fact, there are very few validated CAA TE noise results in the literature. In order to develop more accurate techniques, detailed comparison between computation and experiment is required.

5.2.2 Numerical Noise Prediction

Numerical methods can also be used to estimate TE noise. Here, the turbulent source terms from a CFD solution are used as a source for the propagation of acoustic disturbances. Recently, methods that solve the Linearised Euler Equations (LEE) have been developed. LEE methods were developed for jet flow [23] and have been in continuous development since [37, 38]. There has, however, been limited application to TE noise, with the work of Ewert and Schroder [39] being the only example. Ewert and Scroder developed a special variant of LEE known as the Acoustic Perturbation Equations (APE) where numerical errors were minimised.

LEE methods use the usual acoustic decomposition of flow variables into mean and perturbed parts. For example, a two-dimensional decomposition is

$$\begin{aligned} p &= \bar{p} + p' \\ u &= \bar{u} + u' \\ v &= \bar{v} + v' \\ \rho &= \bar{\rho} + \rho' \end{aligned} \tag{5}$$

where p is the pressure, u is the velocity in the x-direction, v is the velocity in the y-direction and ρ is the fluid density. The overbar denotes mean quantities and the prime denotes the perturbed part.

The linearised Euler equations are then found by substitution and linearisation of the Navier Stokes equations. e two-dimensional system of equations [37] is

$$\frac{\partial \mathrm{V}}{\partial t} + \frac{\partial \mathrm{E}}{\partial x} + \frac{\partial \mathrm{F}}{\partial y} + \mathrm{H} = \mathrm{S} \tag{6}$$

where the solution vector is

$$\mathbf{V} = \begin{bmatrix} p' \\ \bar{\rho} u' \\ \bar{\rho} v' \\ \rho' \end{bmatrix} \tag{7}$$

The flux terms are

$$\mathbf{E} = \begin{bmatrix} \bar{u}p' + \gamma\bar{p}u' \\ \bar{\rho}.\bar{u}u' + p' \\ \bar{\rho}.\bar{u}v' \\ \bar{\rho}u' + \rho'.\bar{u} \end{bmatrix}, \qquad \mathbf{F} = \begin{bmatrix} \bar{v}p' + \gamma\bar{p}v' \\ \bar{\rho}.\bar{v}u' \\ \bar{\rho}.\bar{v}v' + p' \\ \bar{\rho}v' + \rho'.\bar{v} \end{bmatrix} \tag{8}$$

where the ratio of specific heats for air is $\gamma = 1.4$. The vector **H** includes terms that depend on the derivative of mean source terms from a CFD solution.

$$\mathrm{H} = \begin{bmatrix} (1-\gamma)\left(u'\frac{\partial\bar{p}}{\partial x} - p'\frac{\partial\bar{u}}{\partial x} + v'\frac{\partial\bar{p}}{\partial y} - p'\frac{\partial\bar{v}}{\partial y}\right) \\ \left(\bar{\rho}u' + \rho'\bar{u}\right)\frac{\partial\bar{u}}{\partial x} + \left(\bar{\rho}v' + \rho'\bar{v}\right)\frac{\partial\bar{u}}{\partial y} \\ \left(\bar{\rho}u' + \rho'\bar{u}\right)\frac{\partial\bar{v}}{\partial x} + \left(\bar{\rho}v' + \rho'\bar{v}\right)\frac{\partial\bar{v}}{\partial y} \\ 0 \end{bmatrix} \tag{9}$$

The source terms are included in the following vector

$$\mathbf{S} = \begin{bmatrix} 0 \\ S_u - \bar{S}_u \\ S_v - \bar{S}_v \\ 0 \end{bmatrix} \tag{10}$$

$$S_u = -\frac{\partial\bar{\rho}u'v'}{\partial x},\ S_v = -\frac{\partial\bar{\rho}v'u'}{\partial y},\ \bar{S}_u = -\frac{\partial\overline{\rho uv}}{\partial x},\quad \bar{S}_v = -\frac{\partial\overline{\rho vu}}{\partial y} \tag{11}$$

Hence CFD determined noise sources (velocity fluctuations) are coupled to the acoustic computation through the assembly of the vector, **S**.

The use of LEE techniques is so far showing great promise, but there has been only one application to TE noise. In the single work that has applied LEE to TE noise [39], no comparisons have been made with experimental results.

Another numerical method for use in hybrid CAA schemes is the approach developed in Ref. [40]. Here, a variational formulation of Lighthill's acoustic analogy was derived and implemented using a finite element method. Large eddy simulation was used to determine the acoustic source terms. While the method was shown to work well, to date, there have been no comparisons made with experimental results.

CONCLUDING REMARKS

This paper has reviewed TE noise, which is one particular aspect of the overall phenomena known as airfoil self noise. It is a peculiar and academically interesting type of noise generating flow that also has wide practical application. A succinct overview of the latest work in this field has been presented, with a focus on airfoil TE noise prediction methods.

In order to design new, quiet airfoils, the computational techniques described above need further development. The largest issue that remains is experimental validation. In order to develop accurate and credible prediction methods, detailed experimental results are required. Surprisingly, there are few studies or datasets available that have detailed simultaneous TE turbulent flow and far field noise measurements. Future research must begin with turbulent flow and noise measurements about simple and complex geometry airfoils in a controlled environment such as an anechoic wind tunnel.

The modelling of the turbulent flow field needs improvement for better representation of turbulent scales. As LES will become the turbulent flow simulation technique of choice for engineers over the next 10-20 years, better SGS models are required to describe the finer scales of turbulence that affect high frequency noise components. There is a need to develop an 'acoustical' SGS model [20]. This, however, can only be done by thorough experimental validation and dedicated model development.

Numerical acoustic solvers, such as LEE solvers, have developed rapidly over the past few years and will play an increasingly important role in the design of quiet airfoils. However, critical aspects such as numerical stability and accuracy still need to be resolved through rigorous validation against analytical and experimental results.

The continued development of TE noise prediction methods brings about the possibility of numerically optimising TE shapes for quiet operation. This, in fact, has been attempted [27] at Stanford University using a hybrid LES/analytical noise prediction method. While only a preliminary study, the results were dramatic, showing a 10 dB reduction in sound power through the elimination of vortex shedding. Future research in this area will need to focus on improving model accuracy and efficiency as well as the development of more efficient optimisation routines.

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TRANSIENT VIBRATION IN A SIMPLE FLUID CARRYING PIPE SYSTEM

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ABSTRACT: This study aimed to investigate the behaviour of coupled transient acoustic and structural waves travelling within an L-shaped, statically pressurised, water filled pipe system consisting of two pipe straights separated by a pipe bend. Specifically, theoretical models were utilised to predict the time domain response of the system subject to a single, impulse-like excitation applied to a boundary modelled as an end cap. Two models of the bend were used: one utilised a simple discrete model and the other a more complicated continuous model. Moreover, an experimental rig was designed and built to test the theory. The designed ring frequency and ratio of bend radius to pipe radius were respectively 24 kHz and 4.4. The results show that for a broad impulse consisting of significant frequencies up to 1 kHz, the discrete bend model is superior to the continuous model due to computational efficiency.

1. INTRODUCTION

Despite many decades of research, transient vibration within fluid carrying pipe systems still presents vibration and dynamic fatigue problems in industry. In order to move forward, the limitations of current methods and models need to be more rigorously assessed. This is the background motivation for this study.

Preliminary research into 'water-hammer' dates back to pre-1900s. After the 1960s, much of the work in this particular field of fluid-structure interaction ("FSI") focused on modelling low frequency response in conjunction with the various forms of coupling. D'Souza and Oldenburger's work [1] was one of the first studies to analyse pressure waves in a straight pipe interacting with a closed end. The theoretical work utilised Laplace transformations, concentrated on frequencies less than 100 Hz and took account of frictional coupling. Davidson and Smith [2] were the first to model bends in a Timoshenko-like manner but without Poisson and frictional coupling. They analysed an L-shaped system in the frequency domain. They solved the partial differential equations analytically by a series approximation to obtain a bend transfer matrix. They validated their analysis with mobility experiments. Davidson and Samsury [3] extended the analysis of Davidson & Smith and analysed a system incorporating one in-plane and one out-of-plane bend. The results of these two papers are further discussed in the comparison paper by Hatfield et al. [4]. In their paper the configurations from both of the previous Davidson papers are solved using a component-synthesis method and the former PDE series approximation method.

With regard to the modelling of pipe bends, Wood and Chao [5] conducted time domain experimental work on various bend set-ups and showed that rigidly supported bends had little influence on pressure waves, but unsupported bends affected the fluid considerably. Their experimental work was thorough and consisted of a series of tests involving 30˚, 60˚, 90˚, 120˚ and 150˚ mitred bends. Wilkinson [6] in his frequency domain analysis was the first to present a complete fourteen equation pipe straight model which accounted for all the important wave families (pressure, axial, flexural and torsional) in low frequency systems. He utilised the transfer matrix method in which pipe bends were modelled as a series of two point discontinuities and mitred straight pipe segments. All straight segments were modelled by the Bernoulli-Euler model and did not take into account Poisson coupling. Like Davidson and his co-authors, Valentin *et al.* [7] also analysed a Timoshenko pipe bend model in an L-shaped system, but unlike the former, they took into account Poisson coupling. Lesmez *et al*. [8] used Wilkinson's transfer matrix method except that they use the fourteen equation model.

The work of Tijsseling *et al*. [9] provided many useful benefits to this study. They use the method-of-characteristics to solve a subset of the fourteen equation model for an L-shaped pipe in the time domain. They present simple boundary conditions and a discrete model of the pipe neglecting both the size and mass of the bend. Additionally, they attempt to model cavitation and in doing so provide a useful insight into the effects and potential occurrence of the phenomenon. Experimentally, they excite the system by impacting one end of the pipe with a large, pendulum-like rod. They also experimentally investigate the effect of statically pressurizing the fluid prior to impact in order to prevent the occurrence of cavitation. The study by Vardy *et al.* [10] is similar to that above but analyses a T-piece pipe system instead.

In this study, a subset of the fourteen-equation model (eight equations) is used to model the pipe straights and two models are used to model the pipe bend: (1) a simple discrete model from Tijsseling *et al.* [9] which neglects the geometry and mass of the bend and (2) a more complicated model from Valentin *et al.* [7]. The former was tested without consideration to the input frequency while the latter has not yet been tested experimentally. It is the aim of this study to compare predictions by both bend models in the time domain with time domain histories taken from a well controlled experiment.

1. THEORY

Theoretical Models

The fluid in the system is assumed to be homogeneous, isotropic and perfectly elastic. No dissipative effects occur and density changes are small. There is also assumed to be an absence of liquid column separation. Lastly and most importantly, the fluid wave is assumed to be planar or one-dimensional in the direction of the pipe axial axis.

In terms of the structure, it is assumed that the pipe material is linearly elastic and damping is negligible. The pipe's axial response is based on the membrane model and because of the coupling with the internal fluid, the radial pipe motion is quasi-static and a biaxial stress state results in which the radial stress is zero. Furthermore, the transverse response is modelled as a beam. For the continuous pipe bend, the equations for these two models are coupled. For the pipe straight, they are not.

The lateral pipe motion is governed by the Timoshenko model. Unlike the simpler Euler-Bernoulli beam model, this accounts for both rotary inertia and deformation due to shear forces. Like the fluid, the structural waves are assumed to be planar. Additionally, changes in the angle of the bend and their associated effects on the wave dynamics are assumed negligible; elbow ovalization and the associated increase in flexibility and stress intensification are ignored.

In terms of the discrete model, it neglects the mass and the dimensions of the bend as well as the forces exerted on the bend due to changes in fluid momentum. This model is valid if the length of the bend is considerably smaller than that of the straight pipes connected to it. Moreover, like the continuous model, the bend angle is assumed to remain constant. To account for the loss of the pipe length, half of the actual centerline length of the bend is added onto each pipe straight.

The continuous pipe bend model consists of Equations (1) – (8). The pipe straight model can be obtained from these equations by setting $R_b \approx \infty$ (with a change in coordinate system also, see Figure 2). Note that V is the centerline fluid velocity, P the fluid pressure, ρ_f the fluid density, (s, n, r) is the coordinate system, R_b the bend radius, R_p the internal pipe radius, K the fluid bulk modulus, υ the Poisson's ratio, E the Young's modulus for the pipe material, T the pipe wall thickness, t the time, $\dot{u}_s$ the axial pipe velocity, $\dot{u}_r$ the transverse pipe velocity, ρ_p the pipe material density, σ_s the axial pipe stress ($N_s = A_p\sigma_s$), A_p the pipe cross-sectional area, Q_r the pipe shear force, κ the Timoshenko shear coefficient, G the shear modulus, $\dot{\theta}_n$ the rotation of the pipe element cross-sectional face, A_f the fluid cross-sectional area, I_p the second moment of area of the pipe and M_n the pipe moment.

Note that $\phi = \partial u_r / \partial s$ and is the rotation of the pipe element cross-sectional face $s + ds$, ξ is the pipe element centre-line rotation loss at $s + ds$ due to shear, $\xi = Q_r / \kappa G A_p$ and that the rotation of the centre line of the element is $\theta_n = -(\phi + \xi - u_s / R_b)$.

$$\frac{\partial V}{\partial t} + \frac{1}{\rho_f}\frac{\partial P}{\partial s} = 0 \tag{1}$$

$$2\frac{R_b^{\,2}}{R_p^{\,2}}\left[1-\left(1-\frac{R_p^{\,2}}{R_b^{\,2}}\right)^{\frac{1}{2}}\right]\frac{\partial V}{\partial s} + \left(\frac{1}{K} + \frac{2R_p(1-\nu^2)}{ET}\right)\frac{\partial P}{\partial t} - \frac{2\upsilon}{E}\frac{\partial \dot{u}_s}{\partial s} + (1-2\upsilon)\frac{\dot{u}_r}{R_p} = 0 \tag{2}$$

$$\frac{\partial \dot{u}_s}{\partial t} - \frac{1}{\rho_p}\frac{\partial \sigma_s}{\partial s} - \frac{1}{A_p R_b} Q_r = 0 \tag{3}$$

$$\frac{\partial \dot{u}_s}{\partial s} - \frac{1}{E}\frac{\partial \sigma_s}{\partial t} + \frac{\upsilon R_p}{ET}\frac{\partial P}{\partial t} + \frac{\dot{u}_r}{R_b} = 0 \tag{4}$$

$$\frac{\partial \dot{u}_r}{\partial s} + \frac{1}{\kappa G A_p}\frac{\partial Q_r}{\partial t} - \frac{\dot{u}_s}{R_b} = -\dot{\theta}_n \tag{5}$$

$$\frac{\partial \dot{u}_r}{\partial t} + \frac{1}{\rho_p A_p + \rho_f A_f}\left[\frac{\partial Q_r}{\partial s} + \frac{A_p}{R_b}\sigma_s - \frac{A_f}{R_b}P\right] = 0 \tag{6}$$

$$\frac{\partial \dot{\theta}_n}{\partial s} + \frac{1}{EI_p}\frac{\partial M_n}{\partial t} = 0 \tag{7}$$

$$\frac{\partial \dot{\theta}_n}{\partial t} + \frac{1}{\rho_p I_p}\frac{\partial M_n}{\partial s} = \frac{Q_r}{\rho_p I_p} \tag{8}$$

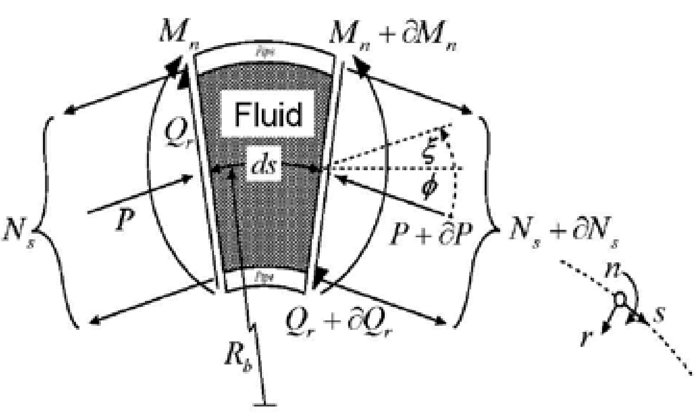

Figure 1 Sign convention for pipe element

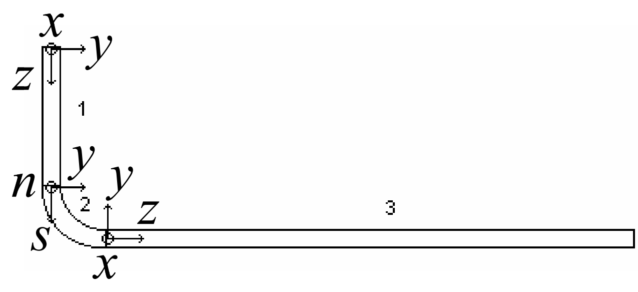

Figure 2 Pipe component numbering and coordinate systems

The discrete pipe bend model is given by Equations (9) – (18). The superscripts refer to the pipe component ("component") numbering given in Figure 2 below. The variables shown are equivalent to their continuous pipe bend counterparts, except for a coordinate system change.

$$A_f(V^1 - \dot{u}_z^1) = A_f(V^3 - \dot{u}_z^3) \qquad P^1 = P^3 \qquad (9),(10)$$

$$\dot{u}_z^1 = \dot{u}_y^3 \qquad A_f P^1 - A_p \sigma_z^1 = Q_y^3 \qquad (11),(12)$$

$$\dot{u}_y^1 = -\dot{u}_z^3 \qquad -Q_y^1 = A_f P^3 - A_p \sigma_z^3 \qquad (13),(14)$$

$$\dot{\theta}_z^1 = \dot{\theta}_z^3 \qquad M_z^1 = M_z^3 \qquad (15),(16)$$

The boundary condition for the excitation end (component 1) is given by Equations (17) – (20). Note that Equation (18) is the source of the system excitation. The boundary condition for the opposite end also consists of Equations (17) – (20) bar the $F_{excitation}$ term in Equation (18). The component joint conditions simply consisted of equating the respective state variables.

$$V^1 = \dot{u}_z^1 \qquad A_f P^1 + F_{excitation} = A_p \sigma_z^1 - m\ddot{u}_z^1 \qquad (17),(18)$$

$$Q_y^1 = 0 \qquad M_x^1 = 0 \qquad (19),(20)$$

For experimental reasons, this study is concerned with the following state variables: P, $a_{z,s}$ (axial acceleration) and $a_{y,r}$ (transverse acceleration).

Solving for the unknowns

In order to solve the system of PDEs above, the spectral method is used. This allows for frequency domain information to be extracted in addition to providing a time domain solution. The method involves (1) obtaining the frequency or "spectral" representation of the force input, $\hat{F}(\omega)$, by applying the forward Fourier Transform (FFT), (2) obtaining the system transfer function, $\hat{H}(\omega)$ from theory, (3) multiplying the two, and (4) applying the inverse Fourier Transform (IFT) to the result. In other words,

$$\text{solution at } (z,t) = \text{IFT}\left[\hat{H}(\omega)\cdot\left\{\text{FT}\left[F(z,t)\right]\right\}\right] \qquad (21)$$

In order to obtain $\hat{H}(\omega)$, each state variable is expressed in a similar form to that shown below, in which $\hat{u}_n(\omega_n)$ is the amplitude spectrum:

$$\dot{u}(z,t) = \sum_n \hat{\dot{u}}_n(\omega_n) e^{i\omega_n t} \qquad (22)$$

The benefit of this representation is that time derivatives of differential equations can be replaced by quasi-algebraic spectral expressions:

$$\frac{\partial \dot{u}}{\partial t} = \sum_n i\omega \hat{\dot{u}}_n e^{i\omega t} \qquad \frac{\partial^m \dot{u}}{\partial t^m} = \sum_n i^m \omega^m \hat{\dot{u}}_n e^{i\omega t} \qquad (23,\ 24)$$

If we also express $\hat{\dot{u}}(z,\omega) = \dot{u}_0 e^{-ikz}$, and substitute this equation and the forms given by Equations (23) and (24) for every state variable into the PDEs above, then a set of simultaneous equations is obtained in which $\boldsymbol{MX} = 0$, where $\boldsymbol{X}$ is a vector of the amplitude coefficients (e.g. $\dot{u}_0$) and $\boldsymbol{M}$ is a matrix of the coefficients of the simultaneous equations (i.e. constants, i,ω,k). Hence, $\boldsymbol{MX} = 0$. This is only possible if

$$\det[\boldsymbol{M}] = 0 \qquad (25)$$

The above equation will yield a polynomial which enables us to determine the quantity and form of the wavenumbers, k.

In order to solve for the amplitude coefficients, one needs to look at the boundary and component joint conditions of the system. But first, the equations which represent the response of each state variable need to be set-up for each pipe component in the system. For example, for component 1, the full response for the fluid pressure is given by:

$$\hat{P}^1(z,\omega) = P_1^1 e^{-ik_1 z} + P_2^1 e^{-ik_2 z} + P_3^1 e^{ik_1 z} + P_4^1 e^{ik_2 z} \qquad (26)$$

The number of modes or waves is determined by the quantity of wavenumbers solved above. For the pressure here, there are 4 waves in total: 2 propagating waves in both the forward and backward directions. Once these state variable equations are input into the boundary and joint component conditions, the algebraic relationships between the various amplitude coefficients for a particular mode (e.g. $V_1 = f(P_1)$) together with the result enable these coefficients to be determined. That is, if $\boldsymbol{B}$ is the boundary and component joint condition matrix, $\hat{\boldsymbol{F}}$ is the spectral form of the force history and $\boldsymbol{C}$ is the coefficient matrix, then $C = B^{-1}\hat{F}$ and $B^{-1} = \hat{H}(\omega)$. This then permits a time domain solution through use of the IFT.

2. EXPERIMENT DESIGN

An impact hammer was chosen as the means to excite the system experimentally. This was for a number of reasons:

- an impact hammer produces a "pulse" excitation which is often clearly identifiable in time domain histories;
- use of an impact hammer meant that the force could be measured directly and easily;
- this method of excitation meant that the end of the pipe which the hammer struck was simple. The associated modelling of this end meant that discrepancies between theory and experiment would be minimised;
- mounting the large impact hammer in a pendulum-like manner meant that the experiment was reproducible and was not overly complicated.

In order to prevent cavitation occurring within the pipe, the fluid was statically pressurised to 500 kPa.

Because this study attempts to compare theory with experiment, 3 state variables which adequately capture the response of the fluid, pipe axial motion and pipe transverse motion were deemed to be sufficient. Hence, only the following measurements were accounted for: P, $a_{z,s}$ and $a_{y,r}$. These variables allowed a suitable comparison between the predictions given by

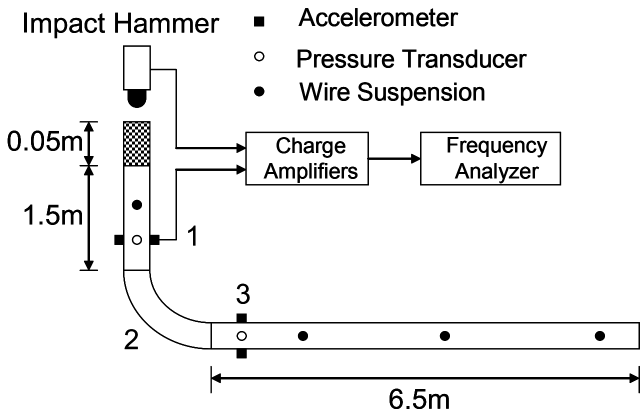

Figure 3 Experimental set-up

the theory and the actuality of the experimental results.

A schematic of the rig can be seen in Figure 3. A force measurement taken from the transducer in the impact hammer is used to input into the theoretical models via Equation (18). The physical properties of the materials involved and the experimental design parameters are listed in Table 1. Details of the pipe components, pendulum impact-hammer and a cross section where the pressure and acceleration sensors are mounted are shown in Figure 4.

Pipe (black steel)		Liquid (water)
$L_1 = 1.5$ m	$E = 210$ Gpa	$K = 2.2$ Gpa
$L_3 = 6.5$ m	$\rho_p = 7850$ kg/m^3	$P_f = 1000$kg/m^3
$R_b = 0.152$ m	$\upsilon = 0.29$	$P_0 = 500$ kPa
$R_p = 0.03485$ m	$m_1 = 2.06$ kg	
$T = 0.0032$ m	$m_3 = 0.1114$ kg	

Table 1 Physical properties and some design parameters

Measurements were taken from two positions located 0.725 m from either side of the pipe bend joints. This distance ensured that all higher-order evanescent modes were negligible.

Dimensional sizing

In studies such as this one, non-dimensional frequency values are often quoted because of the inherent connection between frequency and system lengths. That is, increasing the excitation frequency is equivalent to decreasing the characteristic length of the system and vice versa. Non-dimensional frequencies are not used here. However, we do quote two important values that fulfil the same function. The ring frequency, f_{ring}, is approximately 24 kHz. Additionally, the ratio of the bend radius to pipe radius is approximately 4.4. Above the ring frequency, the response of the pipe is similar to that of a flat thin plate, while below it the response is more complex due to the shell wall curvature [11]. The R_b/R_p ratio characterises the size of the pipe bend: a large value indicates the pipe bend is 'sweeping' while a small value indicates a 'stubby' pipe bend.

3. RESULTS AND DISCUSSION

A power spectrum of a typical strike from the impact hammer is shown in Figure 5. Note the narrow band of significant frequencies (< 1 kHz).

The theoretical time domain responses predicted by both bend models are shown in Figure 6 and Figure 8. Experimental time histories and the theoretical predictions using the continuous bend model are shown in Figure 7 and Figure 9. The results are plotted separately to improve the clarity of the results. Note that the time window is only very short in both sets of figures. This is also to increase the clarity of the results.

In general, there is not a significant difference between the predictions of the two models. The fact that the discrete model does not adequately account for the size and mass of the bend does not seem to have a dramatic effect on the quality of the prediction in the frequency range of interests. There are, however, some minor

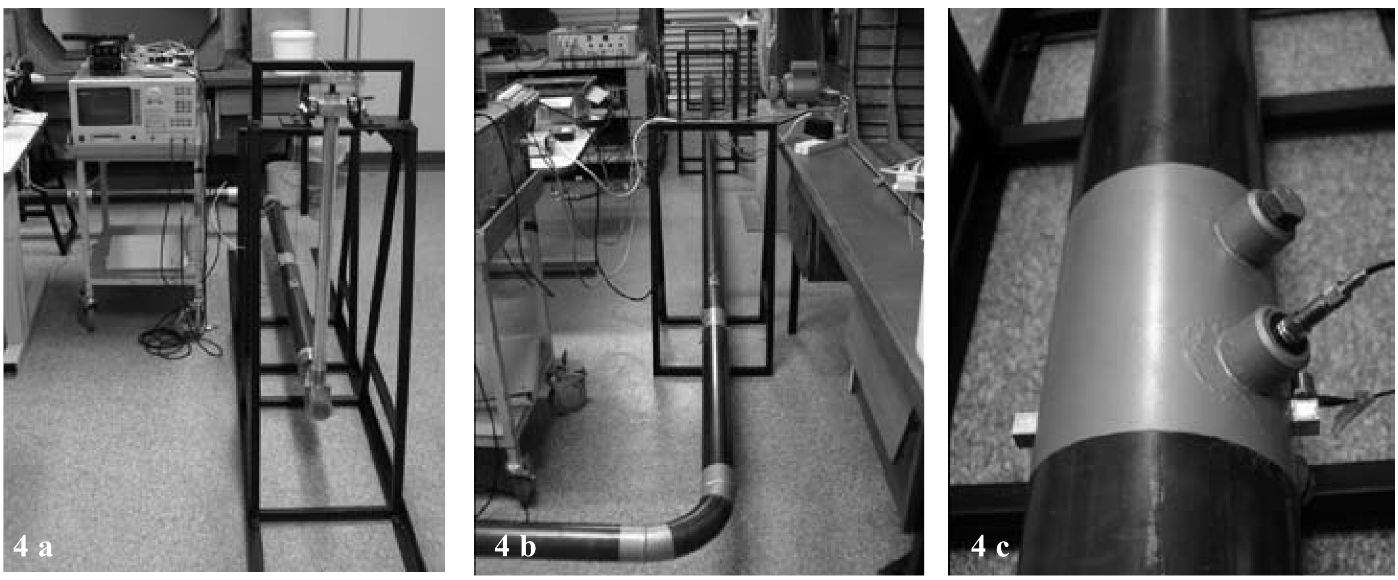

Figure 4 Photographs of experimental set-up (a) data acquisition equipment, pendulum impact-hammer and pipe component 1; (b) pipe component 2 and supports; (c) cross-section of sensor positions, pressure transducer and cube-mounted accelerometer shown

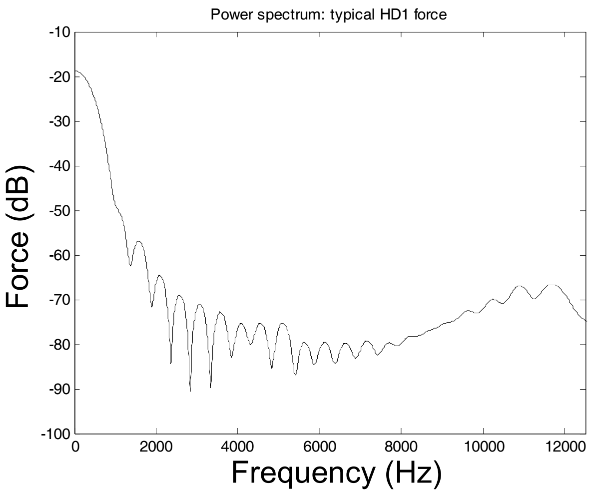

Figure 5 Power spectrum of typical strike from the impact hammer

discrepancies with the transverse accelerations given in Figure 6 and Figure 8. It is interesting to note that such discrepancies are not found with the pressure and axial acceleration. The power spectrum of the transverse accelerations (not shown) actually shows appreciable magnitudes past 5 kHz. Because of this fact, one expects there to be discrepancies between the two models as they are only equivalent in the low frequency ranges.

Furthermore, the theory predicts the response of the system somewhat accurately. For both pressure and axial acceleration measurements, the theory performs reasonably well. However, for the transverse acceleration, there are discrepancies that are different in nature to those shown for the theoretical comparisons. Here the continuous model consistently leads the experimental measurement. This indicates that the waves in the real system are actually travelling slower. A reasonable explanation for this is the fact that the pipe is subject to a complicated pre-stress as a result of the static pressure and fluid loading. A component of this stress might be axial pre-compression, which decreases the speed of travelling waves. Although the effect of axial pre-stress on the speed of transverse wave in beams has been studied [12], further study on this effect in pipeline systems will still have some practical value.

Discrepancy other than the time lags may be due to the limitation of the modelling of the experimental rig by using the in-plane components. The circumferential distribution of transverse waves in practical pipes may be 'polarised' [13] in directions other than in parallel or perpendicular to the (y, z) plane of the pipe coordinates (as defined in Figure 2). As a result, the measured transverse wave response in the (y, z) plane may be contributed by both in-plane and out-of-plane transverse wave components. For this case, both components have two nodes in the circumferential direction. Modelling of the interaction between the in-plane pressure/axial waves with in-plane and out-of-plane transverse waves at pipe bends requires the use of a 3-dimensional pipeline model and experimental determination of the polarization angle of the circumferential modes of the transverse waves.

Although this work concentrates on the measurement and modelling of transient pressure and vibration waves in fluid-filled pipes with bends, the methods can be used to evaluate the acoustical energy transmission and energy exchange between

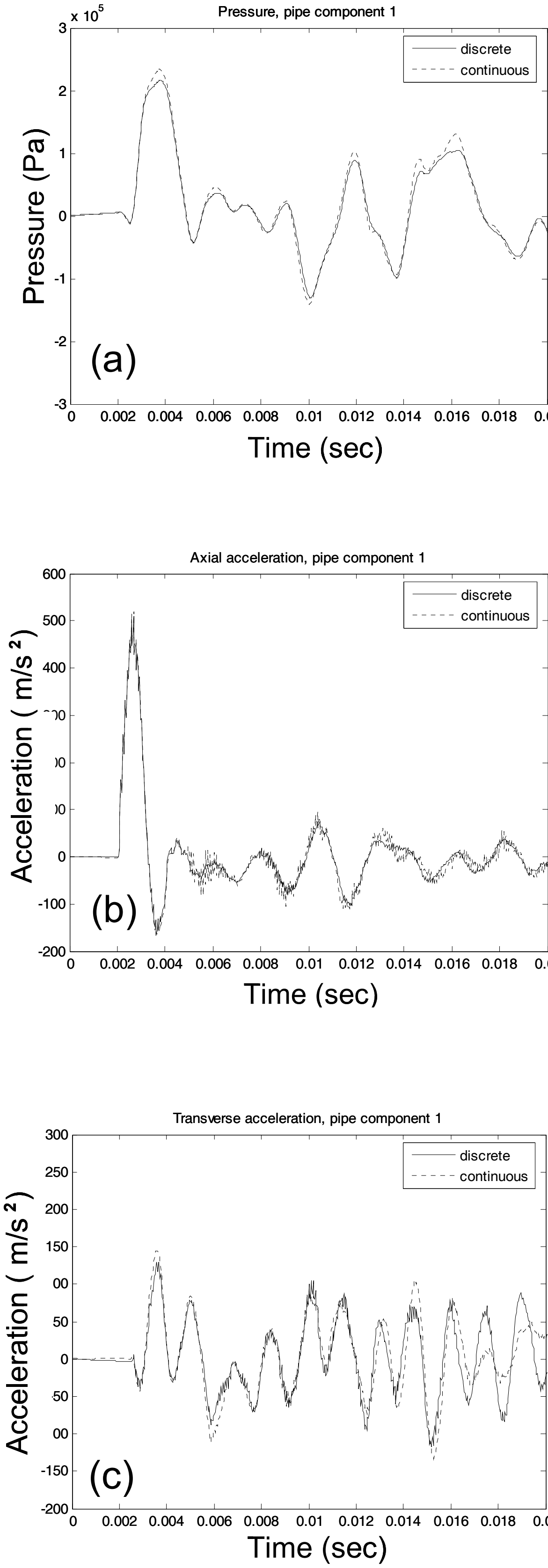

Figure 6 Time domain responses, discrete versus continuous bend models, HD1 typical impact (top to bottom): (a) pressure, pipe component 1; (b) axial acceleration, pipe component 1; (c) transverse acceleration, pipe component 1

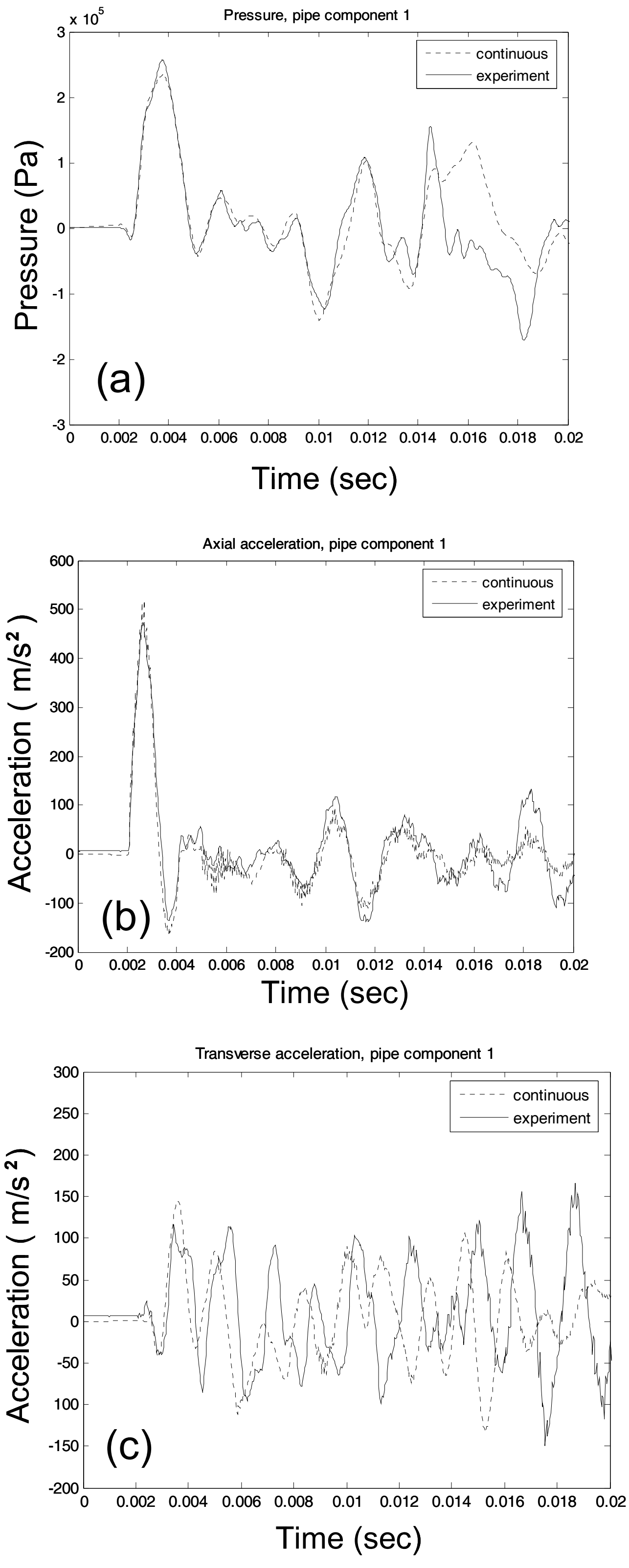

Figure 7 Time domain responses, experimental versus continuous bend models, HD1 typical impact (top to bottom): (a) pressure, pipe component 1; (b) axial acceleration, pipe component 1; (c) transverse acceleration, pipe component 1. All acceleration measurements have been filtered by an analog low-pass filter with a cut-off frequency of 10 kHz

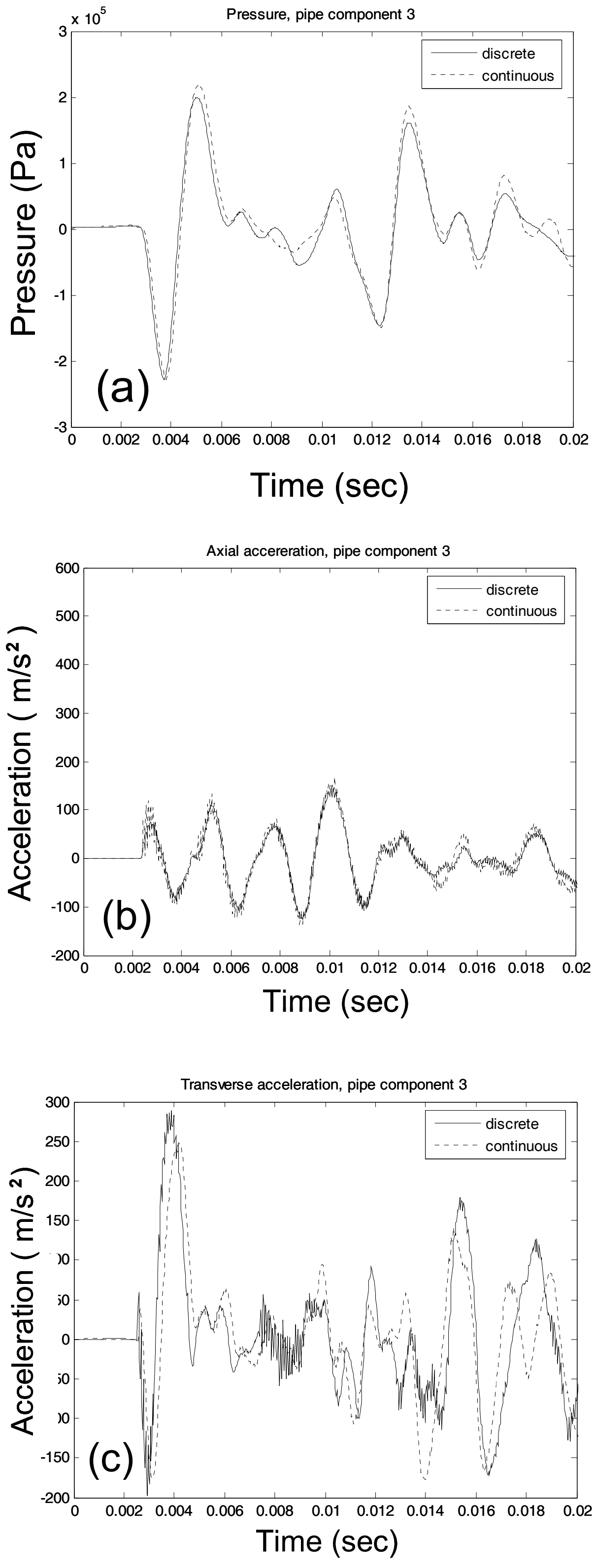

Figure 8 Time domain responses, discrete versus continuous bend models, HD1 typical impact (top to bottom): (a) pressure, pipe component 3; (b) axial acceleration, pipe component 3; (c) transverse acceleration, pipe component 3

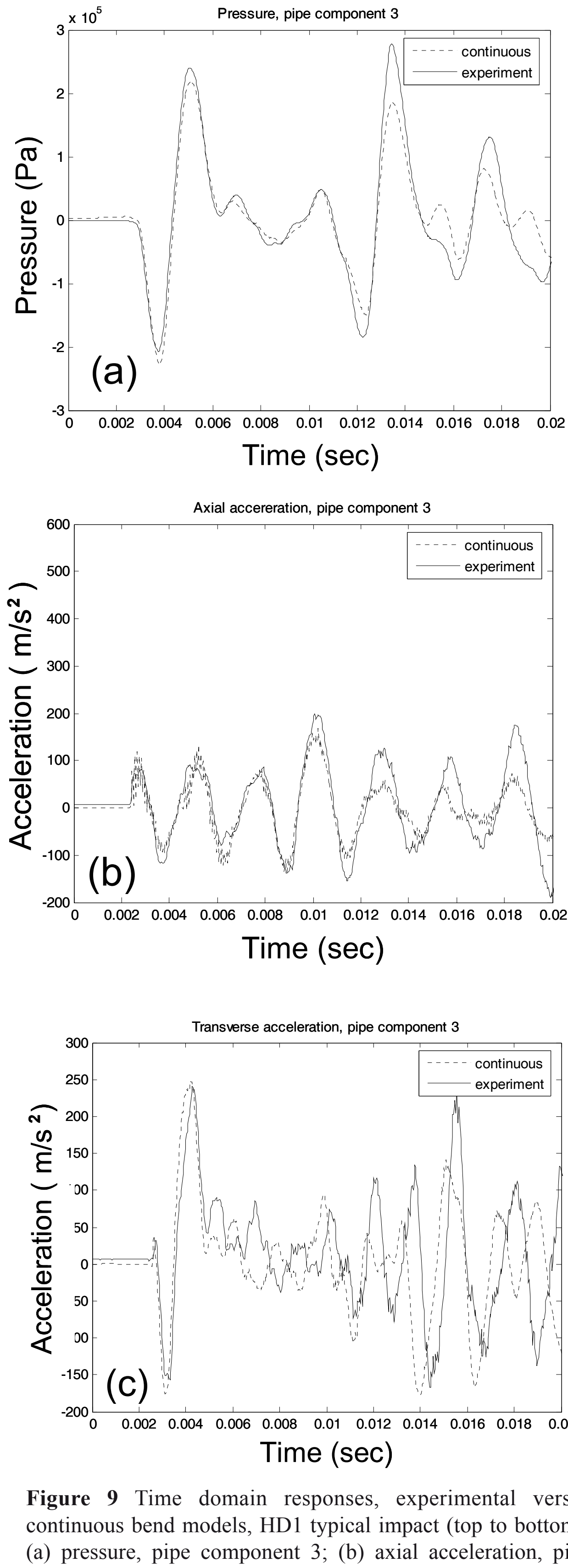

Figure 9 Time domain responses, experimental versus continuous bend models, HD1 typical impact (top to bottom): (a) pressure, pipe component 3; (b) axial acceleration, pipe component 3; (c) transverse acceleration, pipe component 3; All acceleration measurements have been filtered by an analog low-pass filter with a cut-off frequency of 10 kHz

different wave types at the pipe bends. Ultimately, the energy distributions in different part of pipelines and carried by different wave types are responsible to the noise emission from and dynamic stress concentration in pipes, which are important information for pipeline design and maintenance.

4. CONCLUSIONS

This study compared the theoretical predictions of two different bend models of an L-shaped, water-filled pipe system with measurements taken from an experimental rig. The results show that for a broad impulse consisting of significant frequencies up to 1 kHz ($f_{\text{ring}} \approx 24$ kHz, $R_b/R_p \approx 4.4$), the discrete bend model is superior to the continuous model due to computational efficiency. Future work will consist of repeating the analysis here but for higher excitation frequencies. It is important to determine when the accuracy of the two models disagrees.

ACKNOWLEDGEMENTS

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ACOUSTICAL COUPLING BETWEEN LIP VALVES AND VOCAL FOLDS

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ABSTRACT: In normal speech and singing, the standing waves produced in the upper vocal tract are thought to have relatively little effect on vocal fold vibration. We demonstrate the effect of acoustic waves on vocal fold motion. The waves, whose magnitudes are comparable with those produced by vocalisation, are produced by playing a pipe in the manner of a didjeridu. We monitor vocal fold and lip motion by measuring their electrical admittance through the skin, and compare them with the radiated sound. In the presence of deliberate vocalisation, interesting heterodyne effects are produced, which are visible in all three signals. When the folds are relaxed, or in the configuration used for whispering, standing waves produced by playing the pipe produce vibrations in the vocal folds whose magnitudes are comparable with those associated with active vocalisation.

1. INTRODUCTION

In the widely used source-filter model for speech, the vocal tract is regarded as a filter whose frequency dependence modifies the spectrum produced by the vibration of the vocal folds [1, 2]. The frequency of the vibration of the folds (typically 100 to 300 Hz) usually lies below those of the resonances in the vocal tract (the first is typically 300 to 800 Hz). The frequency of vocal fold vibration is determined by mechanical parameters of the folds, the average subglottal pressure applied across them and aero-acoustic effects. Of course, the acoustic pressure waves in the vocal tract also exert force on the vocal folds, and thus may affect their motion [3,4] but this is usually thought to be a rather smaller effect. In brief, the vocal folds ‘drive’ acoustic waves in the vocal tract.

In lip-reed musical instruments, such as trumpet, tuba, etc, the lips of the player regulate air flow into the instrument which, like the vocal tract, has a number of resonances [5]. The lips of the player are therefore somewhat analogous to the vocal folds in the case of the voice. However, in such instruments, various instrument and lip parameters are usually chosen so that the standing waves in the bore of the instrument ‘drive’ the player’s lips, which thus oscillate at a frequency close to one of the resonances of the bore. In brief, acoustic standing waves in the instrument bore ‘drive’ the lips.

Is it possible for pressure waves in the vocal tract to control the motion of the vocal folds in an analogous manner? In normal singing, this seems unlikely: most forms of singing require singers to control pitch and phonemes independently: in other words, they must control the frequency of the fold vibration and the tract resonances independently. The tract resonances have a relatively low Q factor and, for male voices especially, lie at frequencies well above that of the fold vibration.

Large amplitude pressure waves are produced in the vocal tract when playing musical wind instruments. However, vocalisation (the periodic opening of the vocal folds whilst playing) is not usually used in performance on most wind instruments. On the didjeridu, however, vocalisations are an important performance technique. So a study of vocalisation in didjeridu performance may therefore give information about the extent to which the vocal folds may be influenced by pressure waves.

Here we report briefly the results of a study in which vocal fold motion is partly or completely controlled by acoustic waves in the tract produced by a ‘didjeridu’ – in this case a plastic pipe being used in the manner of a didjeridu. The vocal fold and lip motion were studied simultaneously by measuring the electrical admittance between one pair of electrodes placed either side of the neck, at the level of the vocal folds, and that between another pair placed either side of the lips. The output sound and the sound pressure in the player’s mouth were also measured, simultaneously.

One of the features of didjeridu performance is the production of heterodyne components when the player vocalises at a pitch different from that of the instrument [6]. Here we show the lip, vocal fold and sound signals involved in such heterodyne production, and show that the magnitude of vibrations of the folds due to standing waves in the vocal tract may be comparable with the magnitude of vibrations produced by the voice itself. We also show the effect of the waves in driving passive vocal folds.

2. MATERIALS AND METHODS

Two disc electrodes (33 mm diameter) of an electroglottograph (EGG) (model EG-2, Glottal enterprises, Syracuse, NY) were coated with conducting gel and positioned conventionally on either side of the throat, at the level of the vocal folds. They monitored the aperture of the glottis, the gap between the vocal folds. Another pair was positioned on either side of the lips, as shown in Fig. 1, to monitor the contact between the lips. All electrodes were held firmly in place with Velcro bands about the neck and head. The EGG supplied a current at 2 MHz and the output signals from each channel correspond to the admittance between the electrodes. Closing the glottis or closing the lip aperture in each case increases the admittance, so the trace rises as the contact between the folds or lips is increased. Is there significant electrical crosstalk between the two pairs of electrodes? When the mouth is open, the admittance signal across the lips produced by vocalisation is too small to measure. In contrast, the admittance signal across the relaxed vocal folds produced by lip vibration can be measured: it is typically 5% of that measured simultaneously at the lips. However, this vocal fold signal is not noticeably changed by disconnecting the electrodes from the lips: consequently the coupling is acoustical, not electrical. Nevertheless, the lip electrodes were disconnected when not required to eliminate the possibility of any crosstalk.

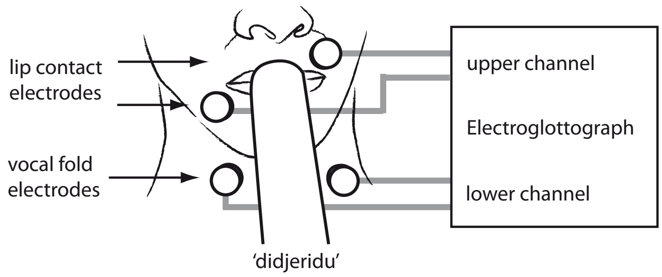

Figure 1. A schematic diagram showing the approximate positions of the electroglottograph (EGG) electrodes.

The aim of the experiment was to investigate the possible effects of heterodyne production and to observe the effects of large amplitude waves on the vocal folds. This only required a player who could vocalise reliably at the desired pitch whilst playing; consequently the experiment was conducted using one of the authors (JW). We were able to detect reliably the features of interest. An experienced player could presumably produce louder vocalisations and consequently stronger heterodyne components. However, the aim did not involve determining any parameters typical of didjeridu playing; and consequently there was little to be gained by measuring additional subjects.

A study involving didjeridu playing styles would normally use traditional instruments, however in this project we were only concerned with acoustical properties rather than musical significance. Furthermore, traditional didjeridus can carry significant spiritual significance that cannot be known by the investigators, and this can be problematical. Consequently, two simple cylindrical plastic pipes were used as substitutes. One had a length of 121 cm and an internal diameter of 34 mm – values typical of a didjeridu. The other had a length of 52 cm and an internal diameter of 26 mm. The musical quality of such pipes is not ranked highly by players [7]. However, this is not significant in this experiment. The sound was measured with an electret microphone positioned 10 cm from the bell, on the axis of the instrument. Sound pressures in the mouth were measured using a calibrated microphone (Bruel and Kjær Deltatron 1/4" type 4944A) with a Nexus conditioning amplifier.

3. RESULTS AND DISCUSSION

Heterodyne components

Figs. 2 to 4 show spectra of the radiated sound, the electrical admittance measured across the lips and the electrical admittance measured across the vocal folds, all measured simultaneously. In each case the subject vocalised at a consonant musical interval above the fundamental of the pipe, so as to produce simple heterodyne tones.

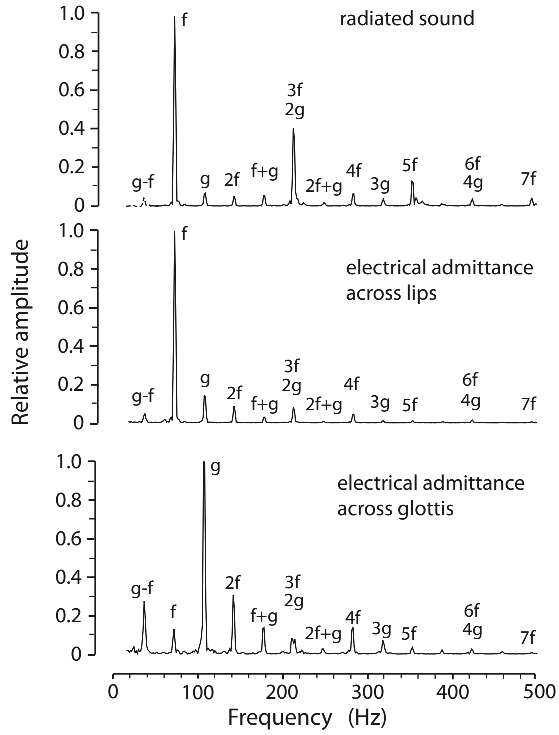

Figure 2. The spectra of the radiated sound, the electrical admittance across the lips and that across the glottis. The subject vocalises at a frequency g (≈106 Hz) that is 3/2 times (i.e. a perfect fifth above) the fundamental frequency of the long pipe f (≈71 Hz). The spectra were calculated from a series of 33072 samples lasting 750 ms using a Hann window. For clarity the sound spectrum only has been increased by a factor of 10 for frequencies below f (indicated by a dashed line).

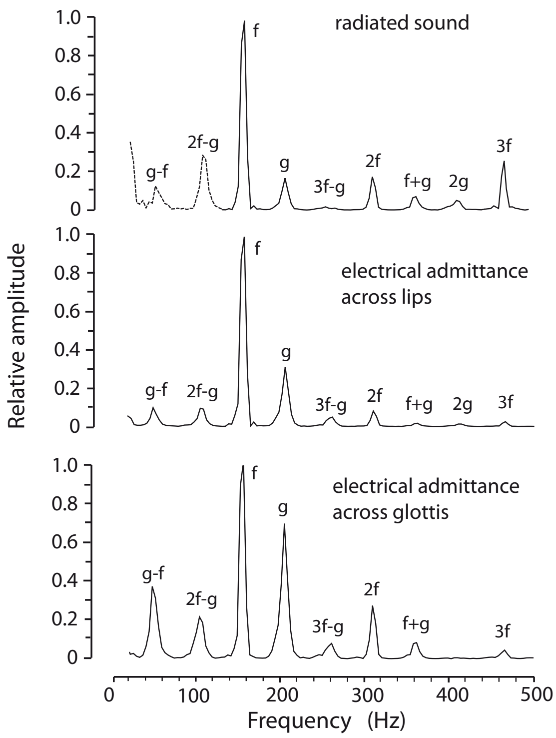

Figure 3. The spectra when the subject vocalises at a frequency g (≈206 Hz) that is 4/3 times (i.e. a perfect fourth above) the fundamental frequency of the short pipe f (≈155 Hz). The spectra were calculated from a series of 12000 samples lasting 272 ms using a Hann window. See caption for figure 2 for more details.

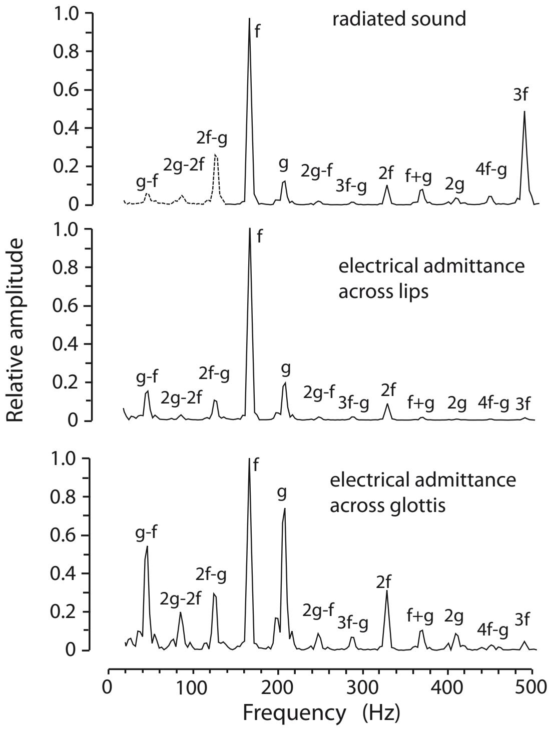

Figure 4. The spectra when the subject vocalises at a frequency g (≈205 Hz) that is 5/4 times (i.e. a major third above) the fundamental frequency of the short pipe f (≈164 Hz). The spectra were calculated from a series of 15870 samples lasting 360 ms using a Hann window. See caption for figure 2 for more details.

In Fig. 2, the longer pipe is used and the subject 'sings' a note (with fundamental frequency g) a musical fifth above that of the instrument (fundamental frequency f): *i.e.* $g = 3f/2$. In this case, the difference frequency $g - f = f/2$, and consequently occurs an octave below f. In Fig. 3, the shorter pipe is used and the subject vocalises at a perfect fourth so $g = 4f/3$, giving a difference frequency $g - f = f/3$, corresponding to a frequency one octave plus a fifth below the fundamental of the instrument. In Fig. 4, the subject vocalises at a major third so $g = 5f/4$, giving a difference frequency $g - f = f/4$, corresponding to two octaves below the fundamental of the instrument.

In these three cases, the air flow into the instrument is modulated by the periodic, but non-sinusoidal, motion of both the lips and vocal folds. The open areas of the lips (L) and glottis (G) can be modelled as $S_L = \Sigma a(n) \sin(2\pi nft)$ and $S_G = \Sigma b(m) \sin(2\pi mgt)$ respectively, where $a(n)$ and $b(m)$ are the amplitudes of the Fourier components, n and m integers. In a very simple model that neglects the effects of the vocal tract impedance, the flow through the lips is proportional to the product $S_G S_L$ and so has components at all frequencies $nf \pm mg$ [6,8,9]. For the simple, musically consonant cases shown here, these terms are all harmonics of the difference frequency $g - f$, and virtually all these are present in all the spectra shown in Figs. 2 to 4. The difference frequency $g - f$ itself is rather weak in the sound signal, in part because lower frequencies are less well radiated from a pipe than are higher frequencies.

In the spectra, there are some similarities between the lip signal and the radiated sound. Although the fundamental frequency of the lips is largely determined by the lowest resonance of the pipe, the motion of the lips determines the flow of air into the pipe and thus strongly influences the sound that is produced. The oscillating air flow through the lips produces sound waves, with comparable amplitude, that travel in both directions: into both the pipe and the vocal tract. While the spectrum of the sound inside the mouth was not measured in this set of measurements, one would expect its spectrum also to share features with that of the lip motion, as modified by the resonances in the vocal tract itself [8,10]. However, there is one systematic difference between lip and sound spectra. The pipe is nearly closed by the lips, so the resonances of the pipe fall at frequencies that are close to odd harmonics of the fundamental. Consequently, at the frequencies shown in these figures, the odd harmonics of the lip motion lie close to resonances of the pipe, and so are well matched to the radiation field. Hence the first few odd harmonics in the output sound (f, $3f$, $5f$) are stronger than the even harmonics ($2f$, $4f$).

In Fig. 2, the vocal fold signal shows, as expected, a strong component at the vocalisation frequency $g = 3f/2$. In this example, the ratios of the amplitudes of components at frequencies f and g are inverted between the lip signal and the fold signal. The folds thus influence the fundamental of the lip motion in approximately the same proportion that the lip motion influences the fold motion.

The experiments represented in Figs. 3 and 4 show frequency ratios g/f = 4/3 and 5/4. A shorter pipe was used because the subject found it difficult to produce a powerful vocal signal at $5f/4$ for the longer pipe (about F#2). In part, this is because it is close to the lower end of his vocal range, and in part it was because of the difficulty of controlling the vocal folds in the presence of low frequency interference from the instrument sound in the tract. On the short pipe, $5f/4$ (A3) fell in a range in which he could sing loudly. Nevertheless, it is interesting to see that the influence of the lip signal (at frequency f) on the vocal signal (at frequency g) is much stronger than the reverse. For no combination of pipes, notes and blowing pressure was the subject able to produce a vocalisation signal whose influence on the lip signal was much stronger than the converse. This is perhaps not surprising: although the subject had the impression of singing loudly at frequency g, the lips are driven by a high Q resonator at f. The lips are strongly coupled to the mouth as well as to the pipe, and so produce a large amplitude pressure wave inside the mouth.

Figs 5 and 6 show (in the time domain) the sound pressure inside the mouth and the admittance at the vocal folds while the subject plays the shorter pipe. Lip electrodes were not used for these measurements to eliminate the possibility of crosstalk. For comfortable (neither loud nor soft) playing levels, the sound pressure level inside the mouth varied between about 130 and 145 dB with respect to 20 μPa and varied only slightly with position in the mouth, which is not surprising for large wavelengths. (The sound level in such experiments was also a few dB larger for the longer pipe than for the shorter.) How do these sound levels compare with those measured inside the mouth while singing? The subject was asked to sing and to produce a sound level 10 cm outside the mouth similar to that produced by the played pipe, measured 10 cm from its end. Singing "oo" (mouth nearly closed) produced a sound level in the mouth of 136 to 140 dB. Humming produced similar sound levels inside the mouth (around

140-145 dB), but the sound level measured outside the mouth and nose was 10 to 12 dB lower than for singing.

So figs. 2 to 4 show clear examples of how the motion of the vocal folds, even during deliberate vocalisation, can be strongly influenced by independently generated pressure waves in the tract. This observation has potential importance in understanding source-filter interactions in speech, singing and other contexts. What happens when the vocal folds are subject to large amplitude pressure waves when they are relaxed rather than vocalising?

Standing waves and 'passive' vocal folds

Fig. 5 shows how the vocal folds can be affected even when the player is not actively vocalising. In this typical example, the subject initially plays the pipe at a comfortable level while "whispering into the instrument" i.e. positioning his vocal folds in the position used for whispering. Although the subject is not actively vocalising, the amplitude of vibration recorded from the vocal folds is similar to that measured for active vocalisation. The vocal folds are then moved to a position that the subject reports as relaxed position, without consciously changing the other articulators. The vocal fold signal decreases, but does not become negligible, indicating the vocal fold contact area is still varying. Acoustically driven vibration of the vocal folds has been measured previously via laryngeal endoscopy [11].

Fig. 6 shows an example in which the subject vocalised deliberately, while playing a note on the pipe at comparable amplitude, so as to produce clear interference beats. All scales are the same as in Fig. 5, and there were no changes in the apparatus between the measurements. In the first part of the trace shown in Fig. 6, the two frequencies differ by a few Hz, producing the interference beats that appear as amplitude fluctuations in both the sound pressure and the vocal fold admittance. The subject then tunes the vocal fold vibration to match that of the lips, while maintaining a similar vocal effort. The beats seen in the first part of the trace suggest that the two signals have comparable size.

Now compare the first part of the traces in Fig. 5 with those

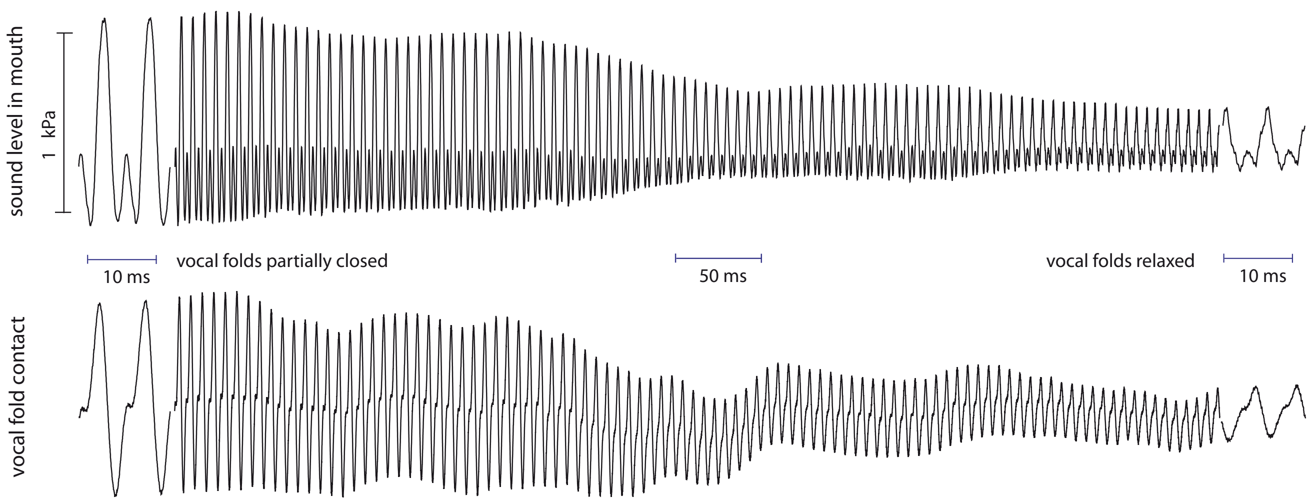

Figure 5. Oscillograms showing the sound pressure in the mouth (upper trace) and the admittance measured across the vocal folds (lower). The initial and final two cycles are shown on an expanded time scale. During this segment, the subject moves his vocal folds from the whispering configuration (vocal folds partially closed) to the relaxed configuration.

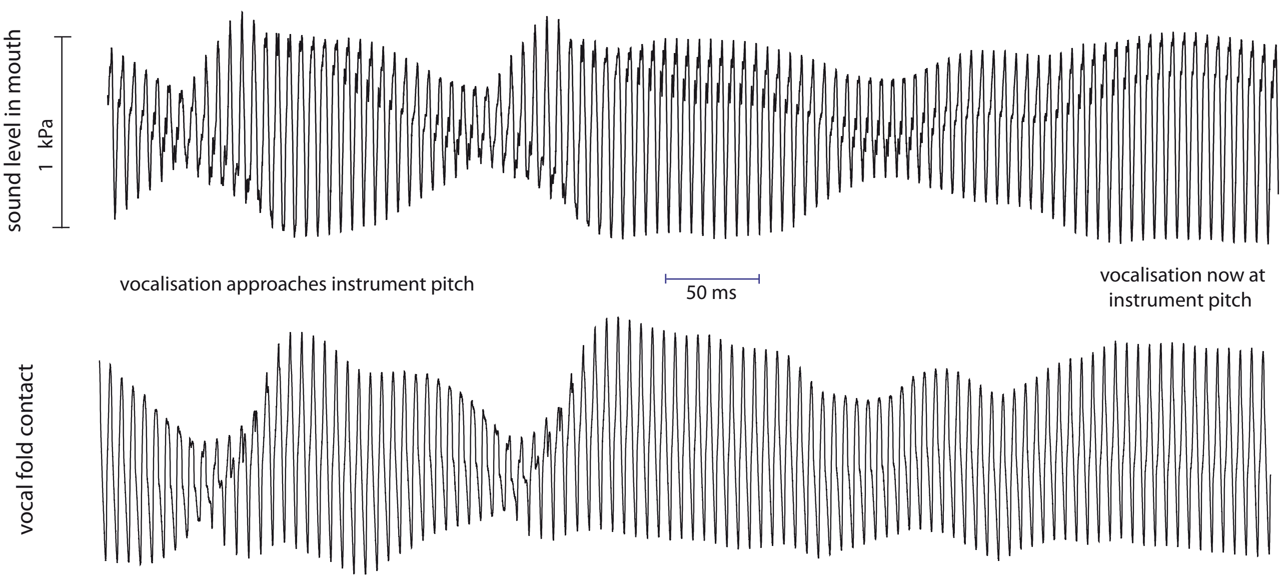

Figure 6. Oscillograms showing the sound pressure in the mouth (upper trace) and the admittance measured across the vocal folds (lower). During this segment, the subject adjusts the pitch of his vocalisation to match that of the instrument. The scales for vocal fold contact and pressure are identical to those of Fig. 5.

in the first part of Fig. 6. In both these cases, the acoustic waves in the mouth have similar amplitude. The variation in vocal fold contact also has similar magnitude: in the presence of these acoustical waves, simply putting the vocal folds in the whispering position produces a vocal fold signal comparable in size with that produced by deliberate vocalisation. The vocal fold signal effectively measures contact area and it is difficult to relate this quantitatively to glottis size. Nevertheless, this observation suggests that the vibration of the vocal folds when 'driven' by an acoustic signal in the whispering position can be comparable in magnitude with those produced by deliberate vocalisation.

In simple models, Rothenberg [3] and Titze [4] have considered the acoustic load of the vocal tract on the motion of the folds. The current study indicates that the effect of acoustic waves in the tract on vocal fold motion is considerable. One would expect this effect to be greatest when a harmonic of the fold motion lay near a resonance of the vocal tract. We have reported consistent examples of this vocal tract tuning in two classes of singing [12-14, see also 15]. The impedance of the tract 'seen' by the vocal folds has a large imaginary component (i.e. pressure and flow out of phase). This component changes sign as the frequency passes through the resonance, going from inertive below the resonance to compliant above. The observation that strong pressure waves can drive the vocal folds suggests a mechanism whereby singers learn the technique of vocal tract tuning: perhaps it is physically easier to sing when the resonance is close to the fold vibration frequency.

CONCLUSIONS

Electrical admittance measured across the lips is an effective way to monitor lip motion in lip-valve wind instruments. Measurements of vocalisations at harmonic intervals show the expected heterodyne effects, not only in the sound, but also in the lip motion and especially in the vocal fold motion. We demonstrate that pressure waves with amplitudes around one kPa in the mouth can drive 'passive' vocal folds at amplitudes comparable with those used in deliberate vocalisation.

ACKNOWLEDGMENT

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SOUND SCIENCE
http://www.soundscience.com.au

VIPAC ENGINEERS AND SCIENTISTS
http://www.vipac.com.au

News

CSR Bradford Insulation Excellence in Acoustics Award

This award aims at fostering and rewarding excellence in acoustics. The entries will be judged on demonstrated innovation or creativity from within any field of acoustics. Entries may be a piece of scientific research, professional work, a design, a monograph or a works of sonic art, for instance. It is important that submissions are representative of up to date technology, creativity and relevancy. Thus entries should describe work that is recent - normally no older than three years at the time of submission. For projects that began before this time important developments of the last three years should be demonstrated.

The prizes include a trophy and a gift to the value of $2,500 to the winner, and a certificate and gift to the value of $500 to the runner up.

Any member of the Australian Acoustical Society at an appropriate grade is eligible to enter. Entries are limited to one entry per person or group.

The judging panel will comprise 2 representatives from the AAS and 2 from CSR Bradford Insulation.

Entries close on 30 July 2008. The winners will be announced at the Annual Conference of the Australian Acoustical Society in November 2008. For more information see

http://www.acoustics.asn.au/sql/csr-award.php

Australian Acoustical Society Education Grant

The Australian Acoustical Society (AAS) Education Grant has been established to encourage and enhance the study of acoustics in Australia and in particular to encourage research in acoustics. The main criterion is the likely benefit that the successful completion of the proposed project will provide for acoustics, for the AAS and for its members. Features of projects likely to be viewed favourably include:

- outcomes that would enable a significant number of AAS members to resolve community or environmental problems more satisfactorily;
- collaboration between education/research institutions and industry based organisations;
- good publicity for the AAS and for the science and practice of acoustics in Australia;
- enhancement of the quality or quantity of educational offerings in acoustics in Australia.

Projects will be judged on their originality and their likelihood of successful completion within reasonable timeframes. The multiplier effect of having partial funding from other sources will also appeal to the judges. However, the AAS remains interested in funding projects that may not proceed without AAS funding. If the proposal involves funds from a number of sources, it should be made clear how the AAS contribution would be acknowledged. The potential for the outcomes to be published in Acoustics Australia and/or presented at the annual conferences and technical meetings of the Society would be viewed favourably.

The grant is open to educational institutions, companies and organisations and individuals for the purpose of financing special acoustic projects, providing scholarships, assisting projects with the purchase of software and equipment or any other worthwhile use involving acoustics.

The total grant of $15,000 may be split between several projects. Thus applicants should not assume that they would necessarily receive all the available funding if their application was successful and should not necessarily request the maximum amount of the award.

The closing date for submissions is the 31 July 2008 and the winning entry will be announced at the Annual Conference of the Australian Acoustical Society in November 2008.

http://www.acoustics.asn.au/general/education-grant.php

Telscreen II

Telscreen is Australian Hearing's new national telephone service that allows you to take a free hearing screening over the phone, any time, anywhere in Australia. Based on extensive research, Telscreen has been developed by the National Acoustic Laboratories (NAL) in conjunction with Australian Hearing and is one of the most sophisticated telephone hearing services in the world. Telscreen is a self check of hearing disability via the telephone and is not a replacement for a face-to-face hearing screening carried out by a qualified clinician. It should not be taken as medical advice. The recorded speech, a series of three numbers, is obscured by a pulsing noise. The listener must punch into the telephone the three numbers they thought they heard. A different set of numbers is then presented with a different signal to noise ratio.

Australian Hearing is introducing Telscreen to improve identification of hearing loss within the community and raise awareness of hearing loss as a significant health issue. National Acoustic Laboratories (NAL) will conduct research into the effectiveness of Telscreen and will continue to monitor its quality and use data for further research into hearing loss. Australian Hearing is keen to provide accessible services that help screen for hearing disability. Telscreen makes hearing screenings easier and more accessible, especially to those in rural and remote parts of the country.

If you wish to check your hearing then call 1800 826 500 from a phone (not a mobile!) Extracted from www.hearing.com.au

Sonification and stock trading

Professor Roger Dean and colleagues from the MARCS Auditory Laboratory at the University of Western Sydney are looking at ways to assist those working on the trading floors in financial institutions. Their idea is to use streams of sound so traders can take in information and warnings. The aim is to develop appropriate sound signals to assist the trader with decision making and, importantly, to attend to indicators of warnings. (extracted from Weekend Australian Nov 2007).

New environmental noise consultancy. Acoustics RB Pty Ltd

After 14 years with Ron Rumble Pty Ltd, Russell Brown has established a new consultancy company Acoustics RB Pty Ltd. This company is based in Wilston Queensland. The contact details for Russell are:

PO Box 150, Wilston QLD 4051

phone: 07 3356 5555 or 0407 555 850

email: russell@acousticsrb.com.au

Nations divided by a common language

The variation in the pronunciation of vowels is one of the strongest markers of the different accents in English. A new Australian project is quantifying the various accents of English in a novel way.

The traditional method for quantifying vowel sounds is to record and to analyse examples from a significant sample of speakers of each variant. This is a logical and direct, but very difficult and expensive to apply on a large scale.

The new project uses vowel perception rather than production. Volunteers are invited to map their own accent by plugging headphones into their computer, then listening to and identifying a series of sample words that span several variables in vowel space. The web site then plots the subject's own accent, with increasing detail as more samples are identified. The (anonymous) data are then added automatically to national and regional data bases, so volunteers can compare their accent with the existing data.

The project is the thesis work of computer science student Ahmed Ghonim, working in the Acoustics Laboratory of the University of New South Wales. The site curently has reasonably large sets for New South Wales, California and New York, but is short of volunteers in many regions of the anglophone world.

If you'd like to see where your accent fits on the map, the experiment and some of its current results are at http://project.phys.unsw.edu.au/swe

Hypertension and exposure to noise near airports

A recent study* reports the effects of night-time noise exposure on blood pressure monitored for 140 subjects living around four major airports at Heathrow (UK), Athens (Greece), Arlanda (Sweden) and Malpensa (Italy). The blood pressure of each subject was measured every 15 minutes throughout the night along with a continuous record of the noise level. A significant increase in blood pressure occurred in response to "noise events" greater than 35 dBA, even if the subjects did not wake up in response to the noise. The increases occurred if the noise source were due to aircraft, road traffic, or even loud snoring. Such increases in blood pressure due to noise are thought to play a role in the development of hypertension and cardiovascular disease.

* European Heart Journal doi:10.1093/eurheartj/ehn013

New Products

Brüel & Kjær introduce new Hand-held Analyzer

Brüel & Kjær have introduced a new hand-held analyser, the model 2270, to replace the model 2250. The objective was to provide a complete toolbox for sound and vibration professionals. Its features include dynamic range exceeding 120 dB, an integrated digital camera for documentation, integrated LAN and USB interfaces for data transfer, data storage via CF and SD memory cards and two channel measurement capability. The application modules can be licensed separately and currently include sound level metering, real-time frequency analysis, noise logging and sound and vibration recording. On-line video demonstrations are available at www.bksv.com.

Bruel & Kjaer Australia, Suite 2, 6- 10 Talavera Rd, North Ryde 2113.

Future Meetings

AAS2008 – Geelong

The AAS Annual Acoustics Conference will be hosted by the Victorian Division from 24th - 26th November, 2008. It promises to be 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. The theme of the Conference will be "Acoustics and Sustainability" and papers are invited in all aspects of acoustics with particular emphasis in 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 either to return home or to 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

Acoustics 08 – Paris

This conference is shaping up to be a spectacular event for acoustics. It is to be held in Paris at the Palais des Congrès from 29 June to 4 July 2008. The meeting is organised by the Acoustical Society of America, ASA, the European Acoustics Association, EAA, and the Société Française d'Acoustique, SFA. More than 3500 presentations distributed in 265 sessions are scheduled (up to 25 sessions will run in parallel) and more than 4000 participants are expected. Two major European conferences have been integrated into this event: ECUA, the European Conference on Underwater Acoustics and Euronoise the European Conference on Noise Control. Acoustics'08 Paris will also celebrate the 60th anniversary of the SFA. The plenary lectures will be: "How sound from human activities affects marine mammals," by Peter Tyack of the Woods Hole Oceanographic Institution, MA, USA , "New trends in aeroacoustics: From acoustic analogies to direct numerical simulations," by Daniel Juvé of the "Ecole Centrale de Lyon", France, "Binaural hearing and systems for sound reproduction," by Philip Nelson of the University of Southampton, UK. And "Light and sound: Ultrasonic imaging in molecular medicine," by Matthew O'Donnell of the University of Washington, WA, USA.

For more information:
www.acoustics08-paris.org/

ICSV 15 – Korea

The Fifteenth International Congress on Sound and Vibration (ICSV15) is to be held in Daejeon, Korea in 6 -10 July 2008. ICSV15 will be organised in the best tradition of these congresses, combining an excellent scientific program that includes the presentation of technical and experimental results with an attractive venue where friendship among the participants is renewed and refreshed.

The Congress Program will include distinguished Keynote Lectures by: Keith Attenborough, The University of Hull, UK on "Measuring, predicting and controlling outdoor ground effect"; Jin Chen, Shanghai Jiao Tong University, China on "Developments of new feature extraction methods in machinery fault diagnosis"; Patricia Davies, Purdue University, U.S.A on"Sound evaluation in the design of products and acoustic environments"; Yang-Hann Kim and Youngjin Park, KAIST, Korea on "3D Sound manipulation: its theory and applications"; Trevor Nightingale, National Research Council, Canada on "Controlling airborne and impact noise in wood-frame multi-unit buildings" and Nobuo Tanaka, Tokyo Metropolitan University, Japan on "Cluster control of distributed-parameter structures"

For more information: www.icsv15.org

ICBEN (Noise Effects) 2008 – Mashantucket, Conneticut

The 9th International Congress on the Biological Effects of Noise (ICBEN) will be held in Mashantucket, Conneticut 21 to 25 July 2008. These conferences are only held every 5 years and many will remember the Noise Effects 98 in Sydney, which was followed by the conference in the Hague in 2003. The 2008 conference will be of interest to researchers, policy makers and anyone with an interest in the impact of noise on public and industrial health. The scientific program includes papers on all aspects of noise effects on humans (and animals). ICBEN comprises 8 teams to deal with the topics: Team 1: Noise-Induced Hearing Loss Team 2: Noise and Communication Team 3: Non-auditory Physiological Effects Induced by Noise Team 4: Influence of Noise on Performance and Behavior Team 5: Effects of Noise on Sleep Team 6: Community Responses to Noise Team 7: Noise and Animals Team 9: Regulations and Standards. Each of these teams will have plenary sessions during the congress.

This congress is dedicated to the memory of the third Chair of the International Commission on Biological Effects of Noise, Henning E. von Gierke whose combined interest in human responses and their governing mechanical processes formed the basis of his four-decade professional career in studying the interaction between acoustic, mechanical energy and the human organism.

More information: www.icben2008.org

INTER-NOISE 2008 – Shanghai

The 37th International Congress and Exposition on Noise Control Engineering, will be held in Shanghai, China on 26-29 October 2008. The Congress is sponsored by the International Institute of Noise Control Engineering (I-INCE), co-organised by the Acoustical Society of China (ASC) and the Institute of Acoustics, Chinese Academy of Sciences (IACAS). The theme of the Congress is "From Silence to Harmony".. The Congress will feature a broad range of high-level technical papers from around the world. There will be distinguished lecturers, technical sessions as well as extensive exhibitions of noise and vibration control technology, measurement instrumentation and equipment and a varied social program. There will be a special session on Wind Turbine Noise.

More information: www.internoise2008.org

Book Review

Principles of Environmental Engineering and Science

Chapter 15 Noise Pollution

Mackenzie L Davis and Susan J Masten

McGraw Hill, 2004, 704pp
(Hardcover edition)

ISBN-0-07-119449-5

Approximately $AUD135
(www.mcgraw-hill.com.au)

Texts on environmental engineering generally focus on air, waste and water issues, with only a passing mention of environmental noise. In this book by Davis and Masten it is refreshing to see one complete chapter on noise pollution, with the other 15 chapters dealing with other aspects of environmental science. In over 40 pages this chapter introduces the key factors associated with environmental noise. Commencing as usual with the sound and then the hearing mechanism, it moves onto annoyance and interference. It describes both percentile and equivalent energy levels before small sections on transportation and construction noise. One criticism is the omission of any mention of noise from industry or mining. The section on propagation includes a simple calculation of the attenuation with wind effects highlighting the difference upwind and down wind. The summary of approaches to noise control could be improved with a simple calculation of the effect of a barrier. The final paragraphs list measures to protect hearing with the final sentences describing personal stereos as an 'ear-destructive device". In the style of McGraw Hill text books, there are problems and discussion questions. The additional reading and references while valid are somewhat dated with the majority from the 1970s.

While this chapter on noise pollution would not provide sufficient detail for the acoustic consultant, it is a reasonably good overview of noise pollution and would serve as an introduction to noise for a generalist environmental engineer. It is easy to read with diagrams and tables and only a few equations and calculations. Having read through this chapter, an environmental engineer should have a better understanding of the issues associated with managing noise and be able to communicate more effectively with the acoustic consultants on the project team.

Marion Burgess

Marion Burgess is involved with teaching an environmental noise portion of an environmental engineering course and is familiar with the difficulties of finding suitable reference books for such courses.

ROBERT HOOKER (1930 – 2007)

The acoustical profession in Australia has sadly lost another of its pioneers with the recent passing of Professor Robert J Hooker; Bob to his friends and colleagues.

After attending and enjoying the Cairns conference in July, Bob become ill and was diagnosed with cancer. After a short but dignified struggle, he sadly succumbed on 1st December; mourned by his loving wife Joyce, their four children and seven grandchildren.

Bob was born in Adelaide in November 1930 and studied for his degree in Mechanical Engineering at Adelaide University. After graduation, he worked with Metropolitan Vickers in the UK and later at the Weapons Research Establishment in South Australia. In 1959, he commenced his academic career as a lecturer at the University of Queensland where he worked until his "retirement" in 1995; progressively advancing through the roles of Senior Lecturer, Reader, Associate Professor and Departmental Head. Retirement was in name only as he maintained an office at the University and continued to encourage students of acoustics by supervising their various undergraduate and post-graduate thesis projects. His tireless energy with this work persisted even on his hospital bed.

Widely regarded as the father figure of acoustics, within Queensland, Bob has played a major role in the establishment and development of the acoustical profession in Queensland. His interest and dedication as a teacher have inspired an interest in acoustics for his students; many of whom have gone on to develop careers in the field. His continued involvement as a mentor and friend for many of his former students has lasted for periods of up to forty years.

Bob was instrumental in the establishment of the Queensland Division of the Acoustical Society and served as its inaugural chair. He has worked tirelessly for the Society, serving in various capacities on the Federal Council including a stint as Federal President. He was elected as a Fellow of the Society in 2004.

Members of the Society will remember Bob as a teacher, mentor, colleague and friend. He will be missed by many. *Ron Rumble*

TIBOR VASS (1925 – 2007)

It is with sincere regret that the AAS records the passing of former Federal President and Fellow of the Society, Dr Tibor Vass, in December 2007.

Tibor was born in Hungary, the younger of two sons. His interest in acoustics emerged during World War II while in high school, when he wrote a paper explaining why a US bombing raid some 100km away sounded as if it were only 1km distant!

After graduation he entered the Budapest University of Technology to study Architecture. At this time an incident with his brother – a soldier who had served at the Eastern Front – inspired in Tibor a lifelong interest in hospital design and the acoustic environment. In his own words:

"When I was shown where my brother lay with multiple wounds, I thought that I was not in a hospital but at a railway station. Nurses, doctors and orderlies were rushing, shouting and slamming doors as if the patients did not exist. When I cornered a sister and pointed out the extremely noisy conditions, she said 'Look, most of these patients on this floor were wounded by Russian tanks and their wounds were caused by shrapnel exploding practically in their faces, which left them with temporary or permanent hearing loss. Believe me, the noise does not disturb them; furthermore we have to make room for the next train load arriving from the Russian front tomorrow.' My brother's wounds were serious, but he did not suffer hearing loss and the terrible noise in the ward distressed him."

In January 1945, Tibor and his University friends were conscripted into the German Army and transhipped to Germany. By the end of the war (1945 to 1946), Tibor was a prisoner of the American forces. After his release he found work in an architect's office in Paris, where he was involved in the reconstruction of a bombed-out hospital. At a time of material shortages, he was able to implement well-sealed double-glazing on all exterior windows for thermal and acoustic benefits, as the hospital was on a busy street.

In 1951 he emigrated to Australia, choosing to live in Tasmania, where he met his wife Denise. Married in 1954 they settled in Hobart where their three children were born. Tibor finished his architecture studies and worked in his brother-in-law's practice. He proudly became an Australian citizen in 1957.

In 1966 the family moved to Perth, where Tibor pursued an academic career, initially as a lecturer at Perth Technical College, then as a founding staff member of the Department of Architecture at the Western Australian Institute of Technology (now Curtin University of Technology). Here he was instrumental in establishing an open plan laboratory for Lighting and Acoustic Studies, including a reverberation room and anechoic chamber. Specialising in acoustics, Tibor developed courses in a range of related areas.

Tibor took a strong interest in his profession, and was a founding member of the WA Acoustical Society, being present at its inaugural meeting on May 7, 1970. A very early Bulletin of the AAS (Vol 1, No.2, Winter 1972) delighted in the presence of the recently-formed WA Division in the newly-incorporated Australian Acoustical Society, and records the name of one Tibor Vass in the Members register for WA.

Tibor went on to occupy all AAS Divisional Committee positions and was a Federal Councillor for many years. He served as President of the AAS in 1984-85 and was made a Fellow of the Society in 1999.

Tibor Vass was a great believer in the value of education. He completed his Masters Degree in Building Science at UWA, and then his PhD at about the time of his retirement! After retirement he continued his involvement with the AAS, took up painting, and undertook refresher courses in the various languages that he spoke.

His care and attention to students and associates is reflected in their many fond memories of him. His dry sense of humour, his many fascinating stories and his considerable contribution to acoustics through his work in WA and his AAS involvement are his legacy to us.

Deepest sympathies are extended to Denise and the family at their loss. *JohnMacpherson*

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

Following the formal launch of the International Congress on Acoustics for 2010 (ICA2010) during the closing ceremony of ICA2007 in Madrid, the organizing committee have proceeded with the task of planning for this event. The small web presence has now been replaced by the real website although many of the pages at the site www.ica2010sydney.org will not be filled with information till closer to the event. While we have already had compliments about the clear layout, suggestions for improvement will be welcomed. Even Google searches now list the site, although we do have competition from a conference of actuaries in South Africa which is also named ICA2010!

We are very grateful for the advice and assistance from Nicole Kessissoglou and her committee following the organization of the ICSV14 in Cairns in 2007. We are using the same web designer as he showed his ability to develop a clear and concise web site layout. We are also working with the developer of the paper management system to achieve a smoother transfer of information between it and the registration system held by the professional conference organisers.

Neville Fletcher has taken on the task of seeking funding opportunities from Federal and State government agencies for support of scientific exchanges and showcasing Australian technological achievements. Fergus Fricke is assisting with the framework for the scientific program and in particular the selection of keynote and distinguished speakers. We intend to have 4-5 plenary speakers and 8-10 distinguished speakers. Even with this number of specially invited speakers it will be a challenge to cover the entire range of topics in acoustics that form part of the ICA. The organising committee would appreciate any suggestions for plenary and distinguished speakers. These speakers should be well known internationally and able to present the latest trends in their fields in an interesting and informative manner.

Promotion is essential to achieve a good attendance at the ICA. This year is a particularly busy year for acoustics conferences which will all provide opportunities for promotion. We are fortunate to have been offered a booth at the Acoustics08 congress in Paris in July which has an anticipated attendance of over 3,500. We anticipate a similar offer of a booth at Internoise 08 in China. We hope that Australians and New Zealanders attending the other international conferences, including ICBEN and ICSV15, will take opportunities to encourage colleagues to atttend ICA 2010

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

Marion Burgess, Chair ICA 2010

Standards Australia

Standards Australia New Business Model

Standards Australia currently has 1,400 active Standards projects and more projects are being commissioned each month. Standards Australia states that this workload is not sustainable and the problem is a lack of rigour in the way projects are selected, inflexible processes and too much work for the resources available. As Standards Australia prepares to implement the New Business Model, the current 1,400 Standards projects will be reviewed to establish which of these are priority projects, and which may need to be earmarked for alternative pathways under the New Business Model.

A comprehensive proposal will be required for each new Standards project, articulating the need and demonstrating wide support and commitment. Each proposal will be subject to a Net Benefit Assessment against a common set of criteria. A positive balance between imposts and benefits must be declared and demonstrated in order to ensure the delivery of net benefit to the nation. Standards Australia will provide guidance to stakeholders on preparing their proposal for assessment. Relationship Managers will assist stakeholders choose the right pathway for the development of a Standard and determine the appropriate resource mix.

The National Standards Office (NSO) has been established to ensure all Standards developers work together, avoid duplication and work in harmony with International Standards. The Accreditation Board of Standards Development Organisations (ABSDO) will accredit other Standards Development Organisations to develop Standards and have these recognised as internationally aligned Australian Standards.

Standards Australia will apply its limited resources to projects using objective criteria, including demonstrated net benefit. Standards Australia's Project Management Group (PMG), under delegated authority from and oversight by the Standards Australia Board's Standards Development Committee, will determine which standards and solutions Standards Australia will support, service and approve.

The latest updates about the New Business Model will be posted on the Standards Australia website www.standards.org.au

A1055-1997: "Acoustics - Description and measurement of environmental noise".

Technical Committee EV10 has commenced a review of this standard. Consideration has been given to using ISO 1996-1:2003 "Acoustics - Description, measurement and assessment of environmental noise - part 1: Basic quantities and assessment procedure" to replace part 1 of the Australian Standard. However, it was considered too complex and technical. With the Australian Standard being widely used across Australia, a more user friendly Standard than the ISO was thought preferable. There are sections of ISO 1996 that are being considered for use, such as those dealing with facade correction, measurement intervals, percentile descriptors, tonality measurements, background noise, extraneous noise influences on the background noise measurement, measurement and prediction of uncertainty. New Zealand Standards has also been reviewing their environmental noise Standards, NZS 6801 and 6802 (see below), and these will also be reviewed as a part of the process. The Committee has also been considering a review of "AS2436:1981 Guide to noise control on construction, maintenance and demolition sites". This has included a review of the BS5228 series. Hopefully, the review of both of these Standards will progress during 2008. If any members have comments about how AS1055 could be improved, they would be welcome to send their comments to the Committee Secretary, Suzanne Wellham, email Suzanne.Wellham@standards.org.au.

Colin Tickel

Diary

2008

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

9 – 11 June, Bremen
11th International Conference on New Actuatorswww.actuator.de

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

2 – 4 July, Southampton.
Workshop on Transportation Noise Sources in Europe
www.quiet.org/
or contact Louis Challis
at challis@unwired.com.au.

6 – 10 July, Daejeon, Korea.
ICSV15: 15th International Congress on Sound and Vibration.
www.icsv15.org

7 – 10 July, Stockholm
18th International Symposium on Non-linear Acoustics (ISNA18)
www.congrex.com/18th_isna/

9 – 12 July, Athens
3rd IC-SCCE, From Scientific Computing to Computational Engineering
www.scce.gr/

21 July – 25 July, Mashantucket
ICBEN9 Int Cong Noise as a Public Health Problem.
www.icben.org

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

15 - 17 September, Leuven
Int Conf on Noise&Vibration Engineering.
http://www.isma-isaac.be

3 - 5 October, Oslo
7th Int Conf on Auditorium Acoustics.
http://ioa.org.uk

21 – 23 October, Tokyo
The 13th International Conference on Low Frequency Noise and Vibration
www.lowfrequency2008.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

4 – 6 January, Cairo
Advanced Materials for Application in Acoustics and Vibration (AMAAV)
www.amaav.org

19 - 24 March, Dallas
Int Conf on Acoustics, Speech, and Signal Processing.
icassp2010.org

5 – 8 April, Oxford
NOVEM 2009, Noise and Vibration: Emerging Methods
http://www.isvr.soton.ac.uk/novem2009/index.htm

13 - 17 April, Shanghai
2nd Int Conf on Shallow Water Acoustics.
www.apl.washington.edu

5 – 9 July, Krakow
ICSV16: 16th International Congress on Sound and Vibration.
http://www.icsv16.org

6 – 10 September, Brighton
Interspeech 2009
www.interspeech2009.org

26 - 30 September, Makuhari
Interspeech 2010.
www.interspeech2010.org

26 - 28 October, Edinburgh
Euronoise 2009
www.euronoise2009.org.uk

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/calendar.html

New Members

Member

Cornelis Petersen (SA)

Sarabjeet Singh (SA)

Glenn Wheatley (SA)

Luke Zoontjens (SA)

AUSTRALIAN ACOUSTICAL SOCIETY ENQUIRIES

NATIONAL MATTERS

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

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For 2005/2006 Financial Year:

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Retired	$35.00
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Including GST

DIVISIONAL MATTERS

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

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ACOUSTICS AUSTRALIA ADVERTISER INDEX - VOL 36 No 1

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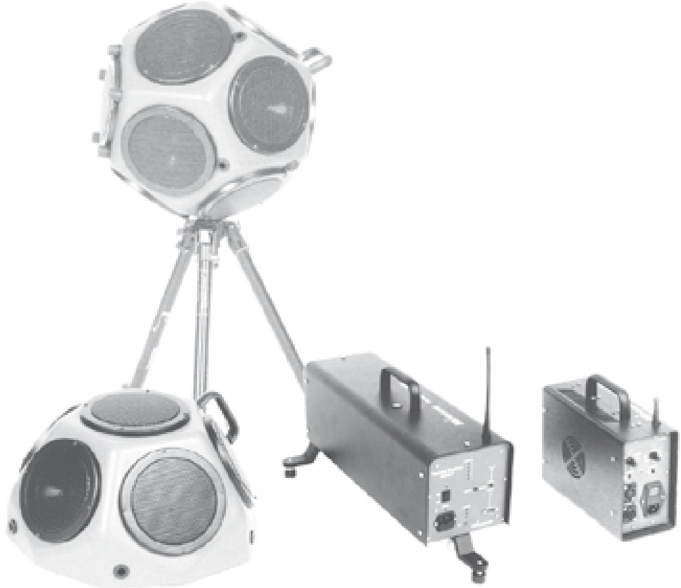

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