Acoustics Australia

Analysing and reducing blade noise in the Mupod

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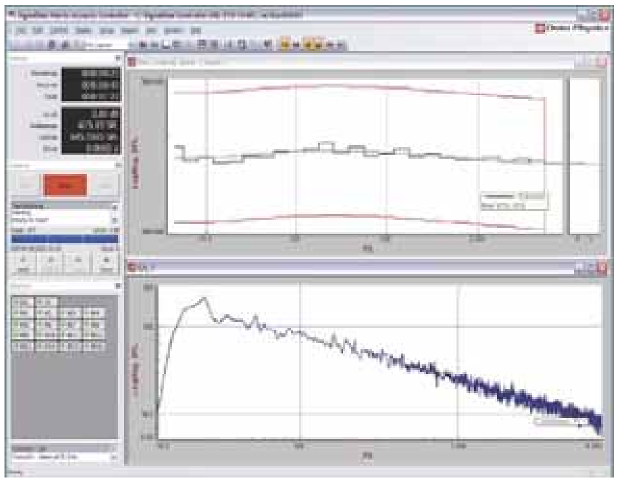

Closed Loop Random Acoustic Control. Top graph:
Octave response control trace over profile reference from 16 inputs with abort lines above and below . RMS = 639.609
Bottom Graph:
G1,1 Auto Power spectrum of response.
Ordinate units in both graphs = LogMag SPL.

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PAPERS

Cover design: Heidi Hereth *Mupod photo courtesy of Entecho Pty Ltd*

Message from the President

Well, it's a busy time of year! Preparations are well underway for the upcoming 20th ICA in Sydney and the program is promising to be a very exciting one. As of this message, there have been over 950 abstracts lodged, mainly from overseas, of course. So, if you haven't registered yet for this great opportunity to meet your local and overseas colleagues, please do so now by visiting the registration page at http://www.ica2010sydney.org/registration.htm. Once again, there will be a very good technical exhibition where you can catch up on the latest products and innovative ideas.

And don't forget the Associated Satellite meetings held in the region after the main congress and the links to these are available from the ICA website under Associated Meetings. These are:

ISMA 2010 (International Symposium on Music Acoustics) begins as sessions of ICA on the 26-27 August and continues on the 30-31 in Katoomba. ISMA brings together the world's researchers on music acoustics to discuss strings plucked and bowed, wind, brass and percussion, organs, keyboards, the voice, the nature of music, physical phenomena, techniques and modelling in music, perception and recognition of music.

ISSA 2010 (International Symposium on Sustainability in Acoustics), 29-31 August in Auckland, New Zealand. This symposium intends to explore sustainability in acoustics, including areas such energy generation, the use of sustainable materials in building and product manufacture, as well as mechanisms to achieve reduction of noise emission and the preservation of hearing.

ISRA 2010 (International Symposium on Room Acoustics), 29-31 August in Melbourne. ISRA 2010 provides a forum for an in-depth exchange of scientific and design research on room acoustics. A special treat for this Conference is that Leo L. Beranek will give the banquet speech.

And now to some very exciting news! After 5 years of very hard and consistent work, Joe Wolfe and his team hang up their red pen and delete key as editors of Acoustics Australia. Joe, as manuscript editor, has been very ably supported over the years by John Smith, who has been news editor and has laid out the journal, and by Marion Burgess, who assisted John in sniffing out and writing the news from around the traps. Emery Schubert was the treasurer. All the editorial staff were well supported by Leigh Wallbank who continues as the business manager and does a great job. And finally, we could not do without our printers, Cliff Lewis Printing. So this edition will be the last by this wonderful team and, on your behalf, I wish to thank the team for all their hard work over the last 5 years, and believe me, it can be hard work!

The challenge is now handed over to Nicole Kessissoglou, who is putting together her new team and we look forward to seeing their first edition in August!

Best regards for now,

Norm Broner

From the Editors

As Norm has announced above in his presidential messge, this will be our last editorial. In the months leading up to the August number, we shall be involved in the organisation and editing for two of the international conferences that Norm has mentioned.

Symbolically, the editorial office is moving 100 m East, remaining on the UNSW campus but moving from Physics to Mechanical and Manufacturing Engineering. For the last five years, however, the journal has been edited and laid out electronically, with copyright forms and financial records being the only remaining paper copies to be handed over. The journal 'office' lives on a couple of (geographically separated) servers and we'll simply hand over some disks and URLs to the new team.

Every issue, it's been satisfying to polish the papers and news items, to lay out the journal and to see the final product evolve. Something that worries most journal editors is the 4n problem: what to do when the contents available do not fill 4n pages, n an integer. If you have noticed compression or omission of your lovingly crafted news item, the appearance of an extra advertisement or even some unexpected text written somewhat hurriedly by the editors, you and we can blame 4n.

We've been pleased with the quality and the breadth of topics of the formal papers, but there have been fewer technical notes than we should have liked. There were three in April 09, one in December 09 but none in the current issue, which we expect will disappoint many readers. Of course, many readers also have their own interesting reports and cases to share. So we urge you to help the new committee by turning some of these into technical notes.

We thank all those of you who have submitted papers, technical notes, forum pieces and news stories for the journal. Hearty thanks to Leigh Wallbank, who continues as business manager and to Sandra Lewis and Louise Gault, who have been our contacts at Cliff Lewis Printing. Louise will continue in that role. Special thanks to Heidi Hereth, whose artwork has been a feature of the covers of the journal over the last five years. And we thank the advertisers who have allowed us to provide the journal to AAS members at very low cost, thereby allowing the Society to support other worthy projects.

Above all, we thank our reviewers: those anonymous experts who have carefully read manuscripts. They have been reliable, thorough, polite and often, thankfully, rapid – but never publicly acknowledged. On behalf of all the authors whose papers have been improved by their insights, suggestions and comments, a warm thank you.

And to the new team, thank-you and good luck!

Marion Burgess, Emery Schubert,
John Smith and Joe Wolfe

ANALYSIS AND REDUCTION OF BLADE PASSING NOISE OF THE ENTECHO MUPOD

Pan[1] , J., Sun. H. M., Walsh, B. S., Do, K. D., O'Neill, P. and Ranasinghe, J.
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Rotor-stator interaction has been identified as the dominant noise source of the Mupod, a vertical take-off and landing aircraft, developed by Entecho, a WA company. This paper reports field measurement results of blade passing noise of the Mupod together with its analysis and control. The blade passing event was simulated in a wind tunnel experiment. The flow speed, rotor blade position and rotor stator blade spacing were varied while the chord-wise pressure distribution of the leading surface of the rotor blade was measured by an array of 6 flush mounted microphones. The features of the pressure distribution on the rotor blade influenced by an upstream stator blade were used for the qualitative analysis of the sound radiation from the Mupod. The result of the reduction of the blade passing noise using angled stator blades is also presented.

INTRODUCTION

Recent development of an unmanned aircraft (the Mupod), capable of vertical takeoff and landing (VTOL) and flight through air, by a WA company, Entecho, has stimulated this research on analysis and control of noise radiated from the aircraft. Figure 1 is a photo of the Mupod in a wind tunnel. This device includes a centrifugal fan that draws air in vertically and dispels it radially and vertically, providing a means to lift and propel this craft through the air. The air from the fan is contained and directed by a skirt, which expels most of the air vertically giving rise to the predominant lift force. This is illustrated in Figure 2 where a cross section of the Mupod is shown to describe the inlet air flow (the arrow towards the stator) towards the stator blade. The skirt in the down stream of the rotor blade (Figure 2) also has the ability (through changing its shape) to direct the air flow from the the rotor (the blue arrow from the rotor) with a horizontal component giving rise to the thrust force. Unlike other craft that are capable of VTOL, the Mupod accelerates the air radially as opposed to axially, resulting in a more compact vehicle footprint, higher lift/power ratio and safer VTOL.

Figure 1. Image of the Entecho Mupod.

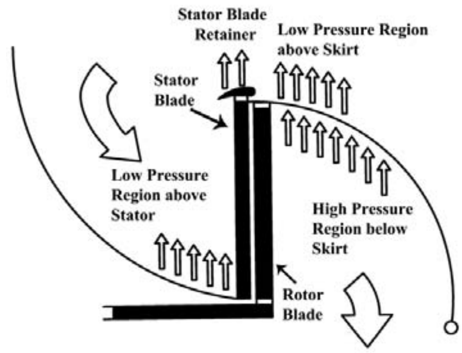

Figure 2. Illustration diagram of lift generation in the cross section of the Mupod.

The centrifugal fan in the Mupod consists of a rotor with equally spaced rotor blades and upstream stator blades that provide axial moment balance and structural support. Figure 3 shows the configuration of the rotor and stator blades and the flow in the radial direction.

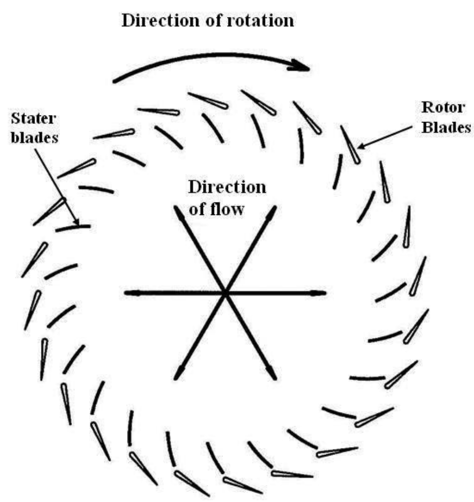

Figure 3. Rotor and stator blades in the Mupod.

Measured noise demonstrates that the Mupod noise is dominated by its blade passing components attributed to the aerodynamic interaction between the rotor and stator blades. Figure 4 shows the noise spectra (measured at 1 m away from the source and at the same height with the source) of the Mupod for two different configurations, tested at a constant angular velocity of the rotor.

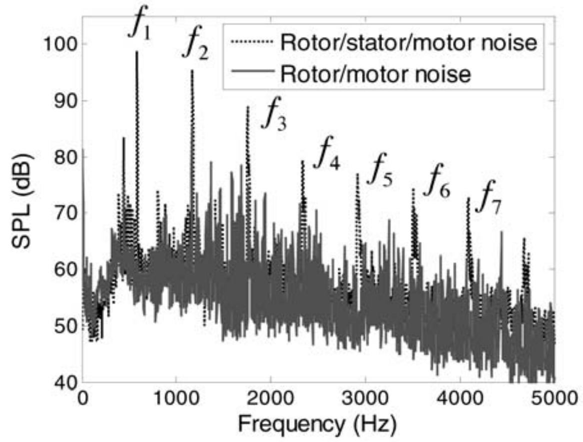

Figure 4. Sound pressure spectra (dB re 20 μPa) of Entecho VTOL aircraft (Mupod) at 1500 rpm rotor speed.

The dashed curve in Figure 4 is from the completely assembled Mupod. Noise for this case is contributed by the electric motors, turbulent noise generated by the blades passing the flow and blade passing noise due to rotor and stator blade interaction. The peak sound levels corresponding to the blade passing event are readily identified at the blade passing frequencies and its harmonic frequencies, which can be determined from

$$f_k = kN_R\Omega \qquad (1)$$

where k = 1, 2, 3..., Ω and N_R are respectively the rotational frequency and number of rotor blades.

The solid curve in Figure 4 is from the Mupod with the stator blades removed. This is the case where rotor and stator interaction is removed. A significant difference in overall noise level is evident for this configuration. The large sound level for the first case is clearly attributed to the blade passing event.

The blade passing event of the Mupod is analysed in this paper. The understanding of the blade passing mechanism has led to a recommended change to the Mupod configuration that has significantly reduced the blade passing noise.

NOISE ANALYSIS

Blade passing noise

When a rotor blade rotates at a constant angular velocity in an air flow, it experiences aerodynamic forces. In turn the reaction force from the rotating blade accelerates the surrounding air and generates pressure disturbances in the air as sound waves. A specific case of interest of the force transfer from blade to the air is when the blade passes a closely placed obstruction such as a stator blade. During this blade passing, the force on the air in the vicinity of the rotor blade often increases significantly. The work by Curle [1], Ffowcs Williams and Hawkings [2] have provided the mathematical and physical foundation for understanding aerodynamic sound when boundary surfaces are present. For the radiated sound pressure at observation location $\vec{r}_o$ by the reaction force $\vec{F}$ at the blade passing location $\vec{r}_3$, Morse and Ingard [3] provided the following expression:

$$p(\vec{r}_o, t) = -\frac{1}{4\pi}\nabla_{\vec{r}_o}\cdot\left[\frac{\vec{F}}{R}\right]_\tau \qquad (2)$$

where $R = |\vec{r}_o - \vec{r}_s|$. $[\ \ ]_r$ indicates that quantities inside are evaluated at the retarded time $\tau = t - R / c$, where c is the speed of sound. It is noted that the retarded time in Equation (2) does not include Mach number. This is because the sources of the the blade passing sound are located near the stators and have no relative motion with the observer.

A single rotor blade passing a stator (Figure 5) is considered first to demonstrate that the blade passing noise is dependent on the reaction force (blade passing force). The reaction force $\vec{F}$ at $\vec{r}_s$ and instant τ generates pressure $p(\vec{r}_o, t)$ at $\vec{r}_o$ and time t:

$$p(\vec{r}_o, t) = \frac{1}{4\pi}\frac{\vec{R}\cdot\vec{F}(\tau)}{R^3} + \frac{1}{4\pi}\frac{\vec{R}}{cR^2}\cdot\frac{\partial\vec{F}(\tau)}{\partial\tau}. \qquad (3)$$

For simplicity, we assume the sound generated by a single force component $\vec{F}(\tau) = [0F_y(\tau)\ 0]^T$ only at position $\vec{r}_s = (a, 0, 0)$, where $F_y(\tau)$ is modelled by impulses:

$$F_y(\tau) = \begin{cases} F_{yo}, & \frac{n}{\Omega}\le\tau\le\frac{n}{\Omega}+\Delta T, \quad n=0,\pm1,\pm2,... \\ 0 & otherwise \end{cases} \qquad (4)$$

In Equation (4), ΔT is the duration of blade passing wherein the force is assumed to be a constant. Equations (4) and (3) lead to the time history of sound pressure in the form of a series of pressure impulses with period of $1/\Omega$. This gives rise to the fundamental frequency (blade passing frequency) Ω of the sound pressure.

The above analysis is extended to the blade passing noise and frequencies of the Entecho Mupod, which has N_R equally spaced rotor blades. If we only consider the aerodynamic interaction when the rotor and stator blades are very close, the time history of the force on air near a stator blade can still be expressed by Equation (4), but with Ω replaced by $N_R\Omega$. This is because the stator is passed by N_R rotor blades in one cycle of rotation. As a result, radiated sound by the N_R rotor blades passing the stator blade is described by the discrete frequency components of sound pressure at zero frequency k = 0, blade passing (k = 1) and its harmonic (k > 1) frequencies:

$$p(\vec{r}_o, t) = \frac{1}{4\pi}\frac{y_o F_{yo}}{R^2}\sum_{k=-\infty}^{\infty} C_k\left(\frac{1}{R}+\frac{j2\pi k}{cT}\right)e^{j2\pi k\frac{\tau}{T}} \qquad (5)$$

where y_o is the location of the observer on the Y axis and

$$C_k = (\frac{\Delta T}{T})\frac{\sin(\pi k\Delta T/T)}{\pi k\Delta T/T}e^{-j\pi k\Delta T/T}. \quad (6)$$

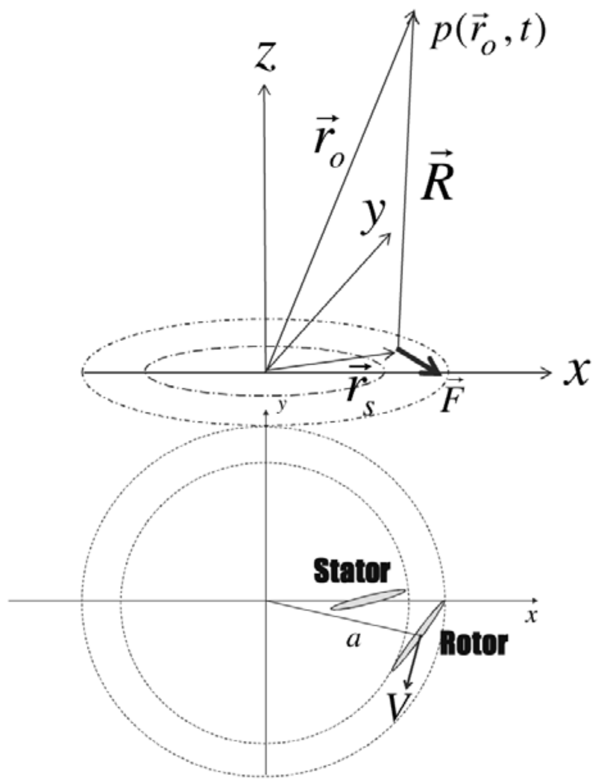

Figure 5. Coordinates for blade passing sound generated by a blade passing force.

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

Overall blade passing noise

The Mupod actually has N_S (different from N_R) equally spaced stator blades as shown in Figure 3. Consequently, the total sound pressure is due to the superposition of all sound produced by the N_R blade passing events in the vicinity of the N_S stator blades.

If the the first stator blade is located at $\vec{r}_{s1}$ = (a, 0, 0), then the position of the n_s^{th} (increasing in the anti-clockwise direction) stator blade is described as

$$\vec{r}_{sn_s} = (a\cos\frac{2\pi(n_s-1)}{N_S}, a\sin\frac{2\pi(n_s-1)}{N_S}, 0) \quad (7)$$

and the time delay between the adjacent blade passing events (for clockwise rotation of the rotor) is

$$\Delta\tau = \tau_{n_s} - \tau_{n_s-1} = -\frac{1}{N_S\Omega}. \quad (8)$$

Thus when Equation (3) and the superposition principle are used to calculate the total sound pressure produced by the blade passing at all stator blades, the phase of each blade passing event is also contributed by the distance $|\vec{r}_o - \vec{r}_{sn_s}|$ in the retarded time τ_{n_s}. The magnitude of blade passing noise from each stator also depends on the orientation between the force vector $\vec{F}_{n_s}(\tau_{n_s})$ and the position vectors $\vec{r}_o - \vec{r}_{sn_s}$.

Huang's work on the noise control of axial fans [3] indicated that blade passing noise from axial fans can be reduced by optimising the relative phases of the noise from each stator blade. An optimal combination of the numbers of rotor and stator blades may be found that results in minimum noise radiation to certain directions (based on spatial interference of radiated sound) and at certain blade passing frequencies. A similar approach can be undertaken for the Mupod after the properties of the blade passing forces are understood.

Properties of the blade passing force

It is clear that any further analysis of the blade passing noise must require the property of the blade passing forces. However, neither experimental nor numerical data of such forces are available for the specific blade passing of the Entecho Mupod.

A preliminary experiment has been conducted in a wind tunnel to measure the surface pressure on a stationary rotor blade located downstream from a stator blade. Six mini-microphones were flush mounted along the chord of the rotor blade (see Figure 6). The pressure was measured on the rotor blade at 13 different angular positions (see Figure 7) on an arc simulating the rotor path. The minimum clearance between the blades was 2mm. The angle of the trailing edge of the stator blade was -30° to the direction of the incoming air flow. The orientation angle of the rotor blade was -70° from the radial direction. Both rotor and stator blades were made of wood. The dimensions of the blades are listed in Table 1. The area testing cross sectional area of the wind tunnel working section was 380 x 250 mm^2.

Table 1. Dimensions of rotor and stator blades

	Chord length	Thickness	Blade length
Rotor	49mm	10mm	260mm
Stator	47mm	10mm	260mm

Figure 8 shows the sound pressure level measured by the microphone (mic 1) near the leading edge of the rotor blade at three flow speeds. The result shows an increase of pressure up to 15dB when the rotor blade is placed in the vicinity of the stator blade. The location of the rotor blade is given by its angular location on a circle of radius 210 *mm*, where the angular position of zero corresponds to rotor blade position 7 in Figure 7.

The distributed sound pressures (with and without the effect of the upstream stator) along the chord of the rotor blade for rotor position 7 are shown in Figure 9. The increase in the sound pressure level near the trailing edge of the blade when the stator is absent is due to the unsteady flow generated near the trailing edge of the rotor. With the influence of the stator, the pressure distribution on the rotor blade finds its maximum level near the leading edge of the blade. An increase in peak pressure by 15dB near the leading edge of the blade is evident for the case where the stator blade is located upstream. Since this surface pressure contributes to the blade reaction force to the air, the reduction in the radiated sound pressure (see Equation (3)) is expected if the stator blades in the craft are removed. This explains the noise reduction observed in Figure 4.

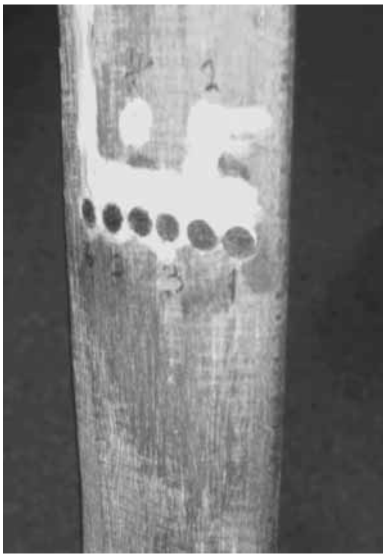

Figure 6. Rotor blade with flush mounted microphones and their locations.

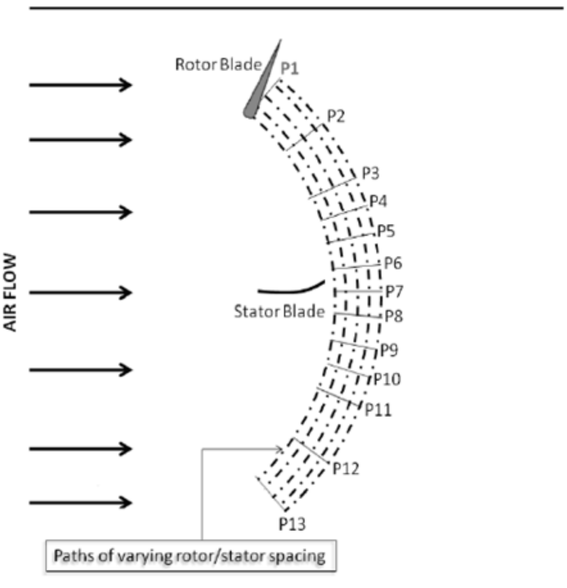

Figure 7. Rotor blade positions for surface pressure measurement.

Observations from Figures 8 and 9 also provide data for further analysis of the blade passing noise using quasi-steady approximation for the force estimation [4], where the effect of the rotor blade speed on the characteristics of the force is ignored. Figure 8 shows that the effective angular range where the blade passing force is significant is about 0.1 (rad).

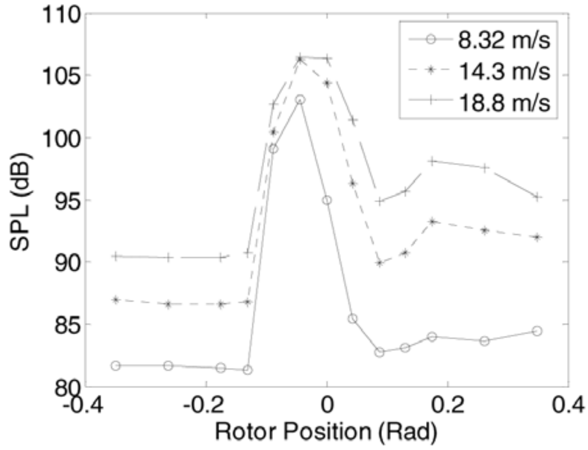

Figure 8. Pressure near the leading edge of the rotor blade at different angular positions.

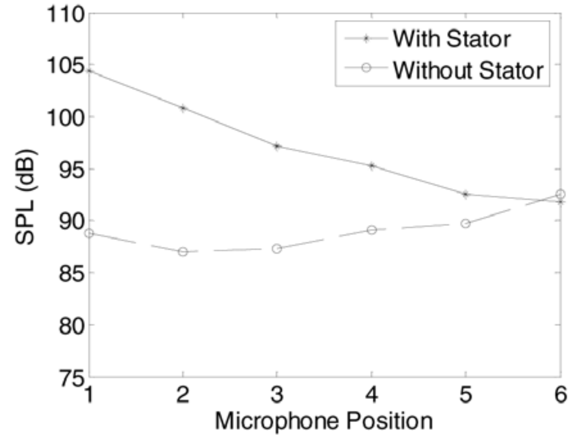

Figure 9. Pressure distribution along the chord of the rotor blade when rotor blade is located at position P7.

This range yields the blade passing duration $\Delta T = \frac{0.1}{2\pi\Omega}$. Figure 9 indicates that most blade forces are contributed by the pressure over approximately the first quarter of the chord length from the leading edge.

The pressure on the stationary rotor blade in the wind tunnel may not be a correct representation of that on the rotating blades. Because of the increased relative velocity, the change in the momentum in the air near the area of blade passing may be increased, so too the relative magnitude of the force components. Nevertheless, the confined angular region of pressure rise in the vicinity of the stator blade and pressure concentration near the leading edge of the rotor blade during blade passing are features useful for the following qualitative analysis of control of blade passing noise.

REDUCTION OF BLADE PASSING NOISE

Noise control mechanism

The above analytical and experimental results are used to assist the control of the blade passing noise of the Mupod. Equation (4) shows that the main characteristics of the force during blade passing are the duration of blade passing ΔT and force magnitude F_{yo}. The F_{yo} can be approximately estimated by the pressure (during blade passing) integrated over a surface area near the leading edge of the rotor blade. This area equals 1/4 of the rotor's chord multiplied by the length of stator section (in the stator length) which is involved in the blade passing event. The location of the force can be approximated as a point at the leading edge of the rotor blade and close to the trailing edge of the stator blade. If the rotor blade is parallel to the stator blade, then the effective blade passing area is 1/4 chord times the height of the stator L. For this case the force magnitude is the largest as the blade passing area per unit time is the largest. When the blade passing area is the largest, the blade passing duration ΔT is also the shortest ($\frac{0.1}{2\pi\Omega}$). The magnitude and duration of the force for this parallel blade configuration is illustrated in Figure 10 by rectangular pulses with solid boundaries.

If angled stator blades are used for the Mupod (see Figure 11), the force magnitude $F_{yo,A}$ during blade passing can be significantly reduced due to the reduced blade passing area per unit time. The blade passing duration is increased to

$$\Delta T_A = \frac{0.1}{2\pi\Omega} + \frac{L\sin\vartheta}{2\pi a\Omega} \tag{9}$$

where ϑ is the angle between the stator blade and the vertical direction (see Figure 11) and L is the length of the stator blade. Those features of the blade passing force due to the use of an angled stator are illustrated in Figure 10 by the dashed pulses.

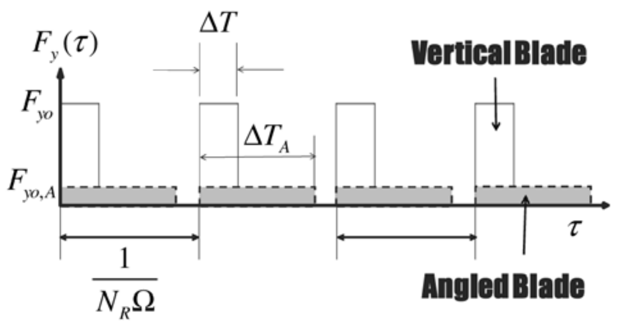

Figure 10. Illustration of time history of forces during blade passing. Solid impulses: stator blade is parallel to the rotor blade; dashed impulses: stator blade is at an angle to the vertical rotor blade.

Figure 11. Angled stator for the VTOL aircraft with a blade angle of 15 degrees.

Intuitively, the reduction of blade passing noise by using the angled stator blades can be understood from Figure 10. Increasing the stator's blade angle not only reduces the blade passing force, but also increases the blade passing duration. As ΔT_A is increased towards the blade passing period T = $\frac{1}{N_R\Omega}$, the force approaches a constant. This results in a significant reduction in noise components at the blade passing frequency and its harmonic frequencies. Mathematically this mechanism of reducing blade passing noise is also observed from Equation (6), i.e.

$$\lim_{\Delta T_A/T\to 1} C_k = \begin{cases} 0, & k\neq 0 \\ 1, & k=0 \end{cases}. \tag{10}$$

Figure 12 provides more details of $|C_k|$ as a function of $\Delta T_A / T$. There are only two regions of $\Delta T_A / T$ ($\Delta T_A / T \to 0,1$) where all the blade passing components can be reduced. Design for $\Delta T_A / T \to 0$ is not feasible because of the inherent non-zero blade passing duration ($\frac{0.1}{2\pi\Omega}$) as observed experimentally. However, design of $\Delta T_A / T \to 1$ can be achieved by using an angled stator as illustrated by Equation (9).

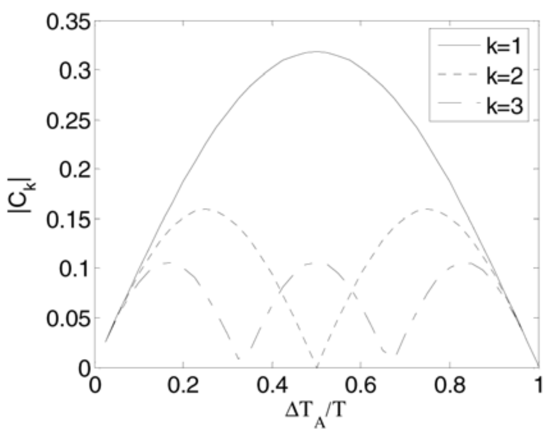

Figure 12. $|C_k|$ as a function of $\Delta T_A / T$.

Experimental confirmation

A comparison experiment was conducted to measure radiated sound pressure at 1.5m away from the Mupod. The first measurement was for the noise from the Mupod with the normal configuration where the stator blades are parallel to the rotor blades. In the second measurement, the normal stator was replaced by a stator with a blade angle of 15 degrees from the rotor blades. The test was conducted at three different motor speeds.

The sound pressure spectra for these two measurements and for motor speed at 1500rpm are shown in Figure 13. The results demonstrate that the angled stator significantly reduces the blade passing noise.

The Mupod used for the test has a rotor diameter of a = 206mm, $N_R = 24$, $L = 84.8mm$ and $\Omega = 25Hz$. This configuration results in T of 0.0017s. The inherent blade passing duration when rotor and stator blades are parallel is $\Delta T = 0.0006s$. The estimated increase of blade passing duration due to the 15 degree angled stator blade is $\frac{L \sin \vartheta}{2\pi a\Omega} = 0.0007s$. As a result, $\Delta T_A / T = 0.76$, This indicates that the angled stator blades almost doubled the blade passing duration and made $\Delta T_A / T$ approach the unity limit.

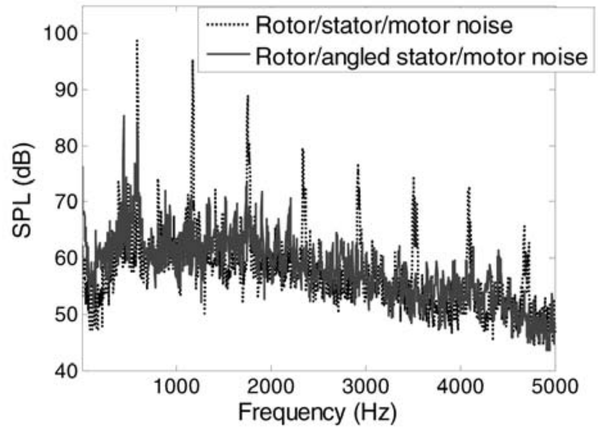

Figure 13. Comparison of sound pressure spectra of Entecho VTOL aircraft (Mupod) at 1500 rpm. Dotted curve: Mupod with normal stator blades; solid curve: Mupod with angled stator blades.

Further confirmation of the effectiveness of the blade passing noise reduction using angled stator blades is shown in Figure 14, where overall dBA level of the Mupod with normal stator and with angled stator are compared at three different rotor speeds. Up to 10 dB noise reduction is observed for all the rotor speeds.

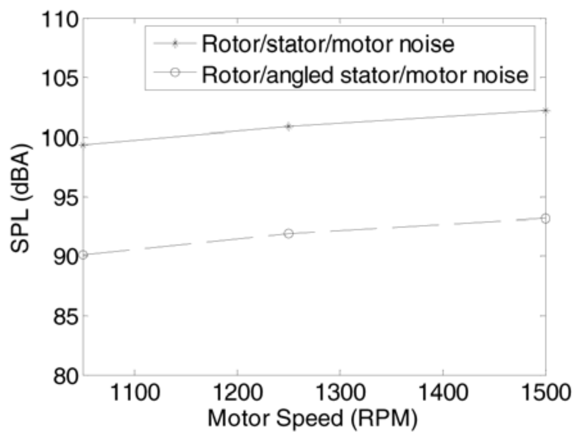

Figure 14. Comparison of overall A-weighted noise levels of the Mupod sound radiation at three different speeds. Solid curve: Mupod with normal stator; dashed curve: Mupod with angled stator.

CONCLUSIONS

Analytical and experimental approaches have been used to study the blade passing noise radiated from an Entecho Mupod. The blade passing sources are spatially stationary sources and the blade passing force plays a dominant role in producing the radiated noise. The features of the force are characterised by the duration of the peak blade passing force and the distribution of the blade passing pressure. Wind tunnel measurement of the blade passing pressure on a stationary rotor blade has provided some preliminary information of the blade passing duration and pressure distribution. This understanding of the blade passing force (which is the surface intergration of the blade passing pressure) and the analytical model of the blade passing noise provides a useful explanation for the effective reduction of blade passing noise using angled stator blades.

Future work includes the measurement of the blade passing force when relative motion between rotor and stator blades is involved. In this paper, the blade passing duration through a stator blade is obtained from a quasisteady approximation and based on measured pressure on a stationary rotor blade downstream from a stator blade. Further experiment is required to study the effect of blade size, angle of attack and relative velocity between rotor and stator blades on the blade passing force.

ACKNOWLEDGEMENTS

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REFERENCES

[1] N. Curle, "The influence of solid boundaries upon aerodynamic sound", Proceedings of the Royal Society of London. Series A, Vol.231(1187), 505-514, (1955).

[2] J. E. Ffowcs Williams and D. L. Hawkings. "Sound generation by turbulence and surfaces in arbitrary motion', Proceedings of the Royal Society of Lon-don. Series A, Vol.264(1151), 321-342, (1969).

[3] P. M. Morse and K. U. Ingard, "Theoretical Acoustics", McGraw-Hill Book Co. New York, (1968).

[4] L. Huang, "Characterizing computer cooling fan noise", J. Acoust. Soc. Am 114(6), 3189-3199, (2003).

[5] D. Olivari, "An investigation of vortex shedding and galloping induced oscillation on prismatic bodies", Journal of Wind Engineering and Industrial Aerodynamics, 11, 307-319, (1983).

A NUMERICAL AND EXPERIMENTAL STUDY OF THE TRANSMISSION LOSS OF MUFFLERS USED IN RESPIRATORY MEDICAL DEVICES

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Mufflers are incorporated into continuous positive airway pressure (CPAP) devices to reduce noise in the air paths to and from the flow generating fan. The mufflers are very small, irregularly shaped, and are required to attenuate noise up to 10kHz. It is important that the acoustic performance of these mufflers is reliably predicted and optimised, in order to improve the user experience and maximise compliance with the CPAP therapy. In this study, the acoustic performance of three reactive muffler designs similar to those used in CPAP devices is presented. Transmission loss predictions obtained using analytical and finite element methods are compared with experimental data measured using a test rig based on the two-microphone acoustic pulse method. The analytical methods were found to be unsuitable for predicting the transmission loss of CPAP muffler designs due to the complexity of the muffler geometries. Good agreement between the finite element and experimental results were obtained.

INTRODUCTION AND BACKGROUND

Obstructive sleep apnoea (OSA) is a medical condition whereby the smooth muscles of the upper airway lose sufficient condition during sleep that the airway becomes constricted, resulting in partial or complete obstruction of the airway. OSA can be successfully managed through the application of a positive pressure to the airway during sleep. This elevated airway pressure is produced by a flow generating fan within a continuous positive airway pressure (CPAP) device. Noise from the flow generator is controlled using mufflers situated in the flow path at the fan inlet and the flow generator outlet. The mufflers are very small and are required to attenuate noise up to 10 kHz. Often these mufflers are irregularly shaped and consist of a number of interconnected volumes. They are predominantly reactive, although absorptive materials have been utilised. An assessment of the air path noise characteristics of three ResMed® CPAP device designs has identified that the most significant noise levels are present at frequencies below 4 kHz. This frequency range encompasses the dominant noise sources associated with the rotation of the blower shaft, blade pass frequency and shaft bearing harmonics. With the exception of the narrow peak sound levels associated with these discrete sources, the region of greatest benefit for targeting noise attenuation lies below 1.5 kHz.

The most common type of linear acoustic model used to predict the performance of mufflers applies classical electrical filter theory and is most widely known as the transfer matrix method, although it is also referred to as the two port approach or 4-pole parameter method [1-3]. The acoustic parameters of many of the individual design elements that are frequently used in mufflers have been well characterised [4-6]. Kim and Soedel [7] and Wu *et al*. [8] present a transfer matrix method which rearranges the variables used in the original method such that only the velocity boundary condition is used in the calculation of the matrix parameters. This improved method offers several advantages over the original method when applied with the finite element method to evaluate transmission loss [9]. Other methods include the 3-point method [10] and impedance method (also known as transmission line theory) [11]. Numerical approaches used to predict the performance of mufflers include the finite element method (FEM) [12-14], the boundary element method (BEM) [15, 16], and computational fluid dynamics (CFD) [17]. Barbieri *et al*. [13, 14] implemented the transfer matrix method to predict the acoustic performance of expansion chambers using both the original parameter formulation and the improved method. A comparison of the various numerical methods has been given by Bilawchuk and Fyfe [18].

This study considers the application of the analytical impedance method and transfer matrix method to three reactive muffler designs having dimensions and geometric complexity similar to those found in CPAP devices. These designs correspond to a single expansion chamber design, an integrated multi-chamber design and a multi-chamber design consisting of three interconnected volumes. Acoustic finite element modelling of the designs was conducted by the authors [19] and the results obtained using the ANSYS package are reproduced in this paper. The transmission loss of each of the mufflers was measured using a two-microphone acoustic pulse method which was based on the procedure developed by Seybert and Ross [20]. Experimental results for the three muffler designs are compared with the analytical and computational results.

MUFFLER DESIGNS

The first design shown in Figure 1 consists of a single unbaffled chamber having coaxial inlet and outlet ports. The close proximity between the inlet and outlet ports creates a narrow, short opening between the ports and the muffler chamber. While the level of geometric detail in the design is high, the underlying configuration is that of a Helmholtz resonator. The acoustic characteristics of this design are well known and, as such, it serves as a suitable design for initial comparison between the analytical, computational and experimental results. The characteristic dimension of the muffler chamber will result in the propagation of some higher order modes within the chamber at frequencies within the desired attenuation range.

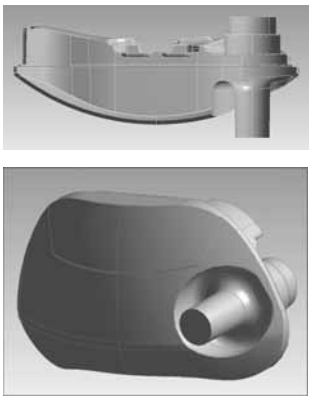

Figure 1: Cross-section and air volume of single chamber design

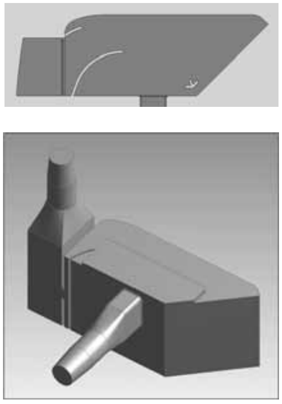

Figure 2: Cross-section and air volume of integrated multi-chamber design

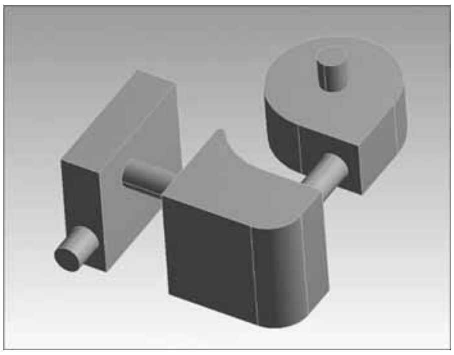

Figure 3: Air volume of three chamber muffler design

The second design shown in Figure 2 consists of two integrated chambers and presents a complex path between the inlet and outlet ports. If air is flowing through the device it would be deflected around a vertical internal baffle before passing through a narrow slot into the final chamber. The third design shown in Figure 3 consists of three interconnected expansion chambers each having orthogonal inlet and outlet ports. The chambers are geometrically simple and contain no internal baffles. A cylindrical pipe of 43mm length and 18mm internal diameter connects each chamber to isolate through-wall noise transmission between adjacent chambers. The dimensions of each muffler design are given in Table 1. The lengths are measured in the direction normal to the chamber inlet and the cross-sectional areas are calculated by dividing the total chamber volume by its length.

Table 1: Dimensions of muffler designs

Design	Chamber 1		Chamber 2		Chamber 3	
	L_1 (mm)	S_1 (mm^2)	L_2 (mm)	S_2 (mm^2)	L_3 (mm)	S_3 (mm^2)
1	35	5,728	-	-	-	-
2	50	6,489	28	2,770	-	-
3	40	5,920	47	7,077	27	7,247

ANALYTICAL APPROACHES

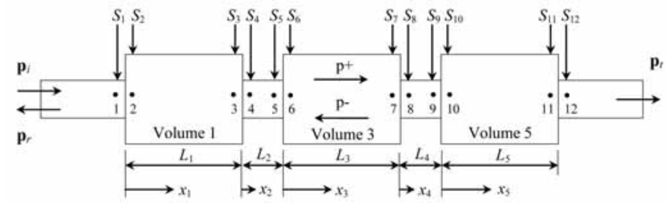

Figure 4: Five volume expansion chamber reactive muffler

Two analytical approaches have been considered: the impedance method and the transfer matrix method. The development of each method is introduced by considering a design comprising three coaxial expansion chambers connected in series by small lengths of pipe, as shown in Figure 4. Unique cross-sectional areas are provided at each point (1 to 12) to allow for the representation of the non-uniform cross-sectional areas that are present in the designs being considered in this study.

Impedance Method

The impedance method is based on the premise that equality of acoustic impedance, Z, is maintained at a change of section. Development of the final equation for transmission loss requires calculating the impedance and pressure at each of the locations 1 to 12 as identified in Figure 4.

Impedance calculations proceed from the outlet (point 12) to the inlet (point 1). The acoustic impedance at point 12 is given by $Z_{12} = z_{12} / S_{12}$, where z_{12} is the specific acoustic impedance and S_{12} is the cross-sectional area of the outlet pipe. By applying an anechoic termination to the outlet duct at point 12, the specific acoustic impedance at that location is equal to the characteristic impedance (ρc), where ρ is the fluid density and c is the speed of sound. Thus the acoustic impedance is given by

$$Z_{12} = \frac{z_{12}}{S_{12}} = \frac{\rho c}{S_{12}} \qquad (1)$$

The acoustic pressures and volume velocities at points 11 and 12 are equal, hence the acoustic impedances at these points are equal. The specific acoustic impedance at point 11 is then given by $z_{11} = Z_{11} S_{11}$ or

$$z_{11} = z_{12} \frac{S_{11}}{S_{12}} \qquad (2)$$

The specific acoustic impedance at point 10 can be found from that at point 11 by considering the impedance formula for undamped plane acoustic waves in a gas column. The complex representation of the actual acoustic pressure is obtained by the superposition of the acoustic pressures associated with the positive and negative travelling one dimensional plane acoustic waves. Thus [2]

$$\mathbf{p}(x,t) = \mathbf{P}_+ e^{j(\omega t - kx)} + \mathbf{P}_- e^{j(\omega t + kx)} \qquad (3)$$

where the amplitude and phase information has been grouped as $\mathbf{P}=Pe^{j\Phi}$. $\mathbf{P}_+$ and $\mathbf{P}_-$ are the complex amplitudes associated with the positive and negative travelling waves, respectively, k is the wave number and ω is the radian frequency. The corresponding particle velocities associated with the positive and negative travelling waves are related to the acoustic pressures by the characteristic impedance. Thus the complex representation of the particle velocity is given by [2]

$$\mathbf{u}(x,t) = \frac{\mathbf{P}_+}{\rho c} e^{j(\omega t - kx)} - \frac{\mathbf{P}_-}{\rho c} e^{j(\omega t + kx)} \qquad (4)$$

At $x_5 = L_5$ (corresponding to point 11 in Figure 4), the specific acoustic impedance becomes

$$z_{11} = \frac{\mathbf{p}_{11}}{\mathbf{u}_{11}} = \rho c \left(\frac{\mathbf{P}_+ e^{j(\omega t - kL_5)} + \mathbf{P}_- e^{j(\omega t + kL_5)}}{\mathbf{P}_+ e^{j(\omega t - kL_5)} - \mathbf{P}_- e^{j(\omega t + kL_5)}} \right) \qquad (5)$$

which can be rearranged to give

$$\mathbf{P}_- = \mathbf{P}_+ \frac{(1 - \rho c / z_{11})}{(1 + \rho c / z_{11})} e^{-j2kL_5} \qquad (6)$$

Similarly, at $x_5 = 0$ (corresponding to point 10 in Figure 4), the specific acoustic impedance can be found as

$$z_{10} = \frac{\mathbf{p}_{10}}{\mathbf{u}_{10}} = \rho c \left(\frac{\mathbf{P}_+ + \mathbf{P}_-}{\mathbf{P}_+ - \mathbf{P}_-} \right) \qquad (7)$$

Equations (2), (6) and (7) can be used to show that

$$z_{10} = \rho c \left(\frac{\left(1 + \frac{\rho c S_{12}}{z_{12} S_{11}}\right) e^{jkL_5} + \left(1 - \frac{\rho c S_{12}}{z_{12} S_{11}}\right) e^{-jkL_5}}{\left(1 + \frac{\rho c S_{12}}{z_{12} S_{11}}\right) e^{jkL_5} - \left(1 - \frac{\rho c S_{12}}{z_{12} S_{11}}\right) e^{-jkL_5}} \right) \qquad (8)$$

The specific acoustic impedance at point 10 is given by $z_{10} = Z_{10} S_{10}$. The acoustic pressure and volume velocity at points 9 and 10 are equal, hence the acoustic impedances at these points are equal $Z_9 = Z_{10}$. The acoustic impedance at point 9 can then be obtained as

$$Z_9 = \frac{\rho c}{S_{10}} \left(\frac{\left(1 + \frac{\rho c}{Z_{12} S_{11}}\right) e^{jkL_5} + \left(1 - \frac{\rho c}{Z_{12} S_{11}}\right) e^{-jkL_5}}{\left(1 + \frac{\rho c}{Z_{12} S_{11}}\right) e^{jkL_5} - \left(1 - \frac{\rho c}{Z_{12} S_{11}}\right) e^{-jkL_5}} \right) \qquad (9)$$

Equation (9) may also be re-stated in non-complex form as

$$Z_9 = \frac{\rho c}{S_{10}} \left(\frac{Z_{12} S_{11} \cos(kL_5) + j\rho c \sin(kL_5)}{\rho c \cos(kL_5) + jZ_{12} S_{11} \sin(kL_5)} \right) \qquad (10)$$

The acoustic impedance at points 1 to 8 may be obtained by following the same methodology used in the development of equations (2) to (10), and considering the muffler to comprise five volumes (three expansion chambers plus two interconnecting pipes). The resulting five sets of equations can be combined using the overlap which occurs at the interface between each volume.

Pressure calculations proceed from inlet (point 1) to outlet (point 12). Considering the first volume shown in Figure 4, the pressure and specific acoustic impedance at point 1 are given by

$$\mathbf{P}_1 = \mathbf{P}_i + \mathbf{P}_r \quad , \quad z_1 = \rho c \left(\frac{\mathbf{P}_i + \mathbf{P}_r}{\mathbf{P}_i - \mathbf{P}_r} \right) \qquad (11, 12)$$

which can be rearranged to give the acoustic pressure at point 1

$$\mathbf{P}_1 = \frac{2\mathbf{P}_i}{(1 + \rho c / z_1)} \qquad (13)$$

Unity pressure amplitude of the incident wave ($\mathbf{P}_i = 1$) has been considered. Equality of the acoustic pressures at points 1 and 2 leads to $\mathbf{P}_2 = \mathbf{P}_1$. At $x_1 = 0$ (corresponding to point 2 in Figure 4), the acoustic pressure given by equation (3) and the specific acoustic impedance can be arranged to give

$$\mathbf{P}_+ = \frac{\mathbf{P}_2 (z_2 + \rho c)}{2 z_2} \quad , \quad \mathbf{P}_- = \frac{\mathbf{P}_2 (z_2 - \rho c)}{2 z_2} \qquad (14, 15)$$

The acoustic pressure at point 3 can be obtained from that at point 2 by substituting these relationships into equation (3) defined at point 3 ($x_1 = L_1$).

$$\mathbf{P}_3 = \frac{\mathbf{P}_2}{2} \left((1 + \rho c / z_2) e^{-jkL_1} + (1 - \rho c / z_2) e^{jkL_1} \right) \qquad (16)$$

Equality of the acoustic pressures at points 3 and 4 leads to $\mathbf{P}_4 = \mathbf{P}_3$ and, given $z_2 = Z_1 S_2$, the pressure at point 4 can be written in terms of that at point 1 by

$$\mathbf{P}_4 = \frac{\mathbf{P}_1}{2} \left((1 + \rho c / Z_1 S_2) e^{-jkL_1} + (1 - \rho c / Z_1 S_2) e^{jkL_1} \right) \qquad (17)$$

Equation (17) may also be re-stated in non-complex form as

$$\mathbf{P}_4 = \frac{\mathbf{P}_1}{Z_1 S_2} (Z_1 S_2 \cos(kL_1) - j\rho c \sin(kL_1)) \qquad (18)$$

The acoustic pressure at points 5 to 12 in response to an incident wave of unity pressure amplitude may be obtained by following the same methodology used in the development of equations (16) to (18). As the outlet duct is anechoically terminated, the magnitude of $\mathbf{P}_{12}$ is simply the transmitted pressure, $\mathbf{P}_r$

Transmission loss is calculated using the ratio of the incident acoustic power at the inlet of a system and the transmitted acoustic power at the outlet. The sound power of a travelling harmonic plane acoustic wave is defined as [21]

$$W_{rms} = \int_S \frac{p_{rms}^2}{\rho c} dS \qquad (19)$$

p_{rms} is the root mean square pressure and S is the area of the surface through which the sound power is passing. The corresponding form of the transmission loss equation is

$$TL = 10 \log_{10} \left(\frac{W_i}{W_t} \right) \qquad (20)$$

W_i and W_t are the incident and transmitted sound power, respectively. For unity pressure amplitude of the incident wave ($\mathbf{P}_i = 1$) the sound transmission loss (TL) for the reactive muffler shown in Figure 4 can now be found from

$$TL = -10 \log_{10} \left(\left| \mathbf{P}_{12}^2 \right| \frac{S_{12}}{S_1} \right) \qquad (21)$$

Note that the above derivation accommodates differing cross sectional areas at each of the points 1 to 12. For the simplified case of a single expansion chamber of constant cross section and having inlet and outlet ducts of equal cross section, the above process can be shown to produce a well known form of the transmission loss [20]

$$TL = 10 \log_{10} \left[\cos^2(kL) + \frac{1}{4} \left(\frac{1}{m} + m \right)^2 \sin^2(kL) \right] \qquad (22)$$

where m is the expansion ratio (chamber cross section divided by duct cross section).

Transfer Matrix Method

The transfer matrix method (also known as the two port method) uses 2 x 2 matrices to relate two variables at planes on either side of an acoustic component [2]. Matrices for individual components can be readily combined to form a single, overall matrix that describes the behaviour for a multi-component muffler system.

Adopting acoustic pressure (p) and volume velocity (U) as the two state variables, the following general transfer matrix may be written to relate the state variables on either side of an expansion chamber reactive muffler.

$$\begin{bmatrix} p_1 \\ U_1 \end{bmatrix} = \begin{bmatrix} A & B \\ C & D \end{bmatrix} \begin{bmatrix} p_2 \\ U_2 \end{bmatrix} \quad (23)$$

For the case of the simple cylindrical expansion chamber reactive muffler, only the transfer matrices for (i) a uniform tube, (ii) a sudden expansion and (iii) a sudden contraction need to be considered. For a uniform tube, the transfer matrix is given by [2, 4]

$$\begin{bmatrix} A & B \\ C & D \end{bmatrix} = \begin{bmatrix} \cos(kL_c) & j\dfrac{\rho c}{S_c}\sin(kL_c) \\ j\dfrac{S_c}{\rho c}\sin(kL_c) & \cos(kL_c) \end{bmatrix} \quad (24)$$

where L_c is the length of the expansion chamber. When the muffler cross section, S_c, is small compared with the wavelength, and in the absence of air flow, the sudden expansion and contraction at the discontinuities (ends) of the expansion chamber may be represented by simple unit matrices [2]. When this assumption cannot be made, additional elements referred to as "Karal's correction" [5] should be introduced at the discontinuities. The correction matrix is given by

$$\begin{bmatrix} A & B \\ C & D \end{bmatrix} = \begin{bmatrix} 1 & j\omega L_K \\ 0 & 1 \end{bmatrix}, \qquad L_K = \frac{8\rho}{3\pi^2 r_p} H\left(\frac{r_p}{r_c}\right) \quad (25)$$

where L_K is the analogous acoustical inductance, r_p is the radius of the pipe, r_c is the radius of expansion chamber and $H(r_p/r_c)$ is given by Figure 2b in Miwa and Igarashi [5] or may be approximated by

$$H(r_p/r_c) = 0.6857(r_p/r_c)^3 - 0.4312(r_p/r_c)^2 - 1.2501(r_p/r_c) + 1.0006 \quad (26)$$

With reference to the series connected expansion chambers shown in Figure 4, the final transfer matrix, T, can be derived using simple matrix multiplication of the appropriate combination of the above matrices

$$\begin{bmatrix} A_T & B_T \\ C_T & D_T \end{bmatrix} = \begin{bmatrix} 1 & B_{K1,2} \\ 0 & 1 \end{bmatrix} \begin{bmatrix} A_{2,3} & B_{2,3} \\ C_{2,3} & D_{2,3} \end{bmatrix} \begin{bmatrix} 1 & B_{K3,4} \\ 0 & 1 \end{bmatrix} \cdots \begin{bmatrix} A_{10,11} & B_{10,11} \\ C_{10,11} & D_{10,11} \end{bmatrix} \begin{bmatrix} 1 & B_{K11,12} \\ 0 & 1 \end{bmatrix} \quad (27)$$

For the case of a non-reflecting termination of the system, the corresponding form of the transmission loss equation incorporating the transfer matrix constants can be shown to be [2]

$$TL = 20\log_{10}\left[\frac{1}{2}\left(\frac{S_1}{S_{12}}\right)^{1/2}\left(\left|A_T + B_T\left(\frac{S_{12}}{\rho c}\right) + C_T\left(\frac{\rho c}{S_1}\right) + D_T\left(\frac{S_{12}}{S_1}\right)\right|\right)\right] \quad (28)$$

For the simplified case of a single expansion chamber of constant cross section having inlet and outlet ducts of equal cross section and only a very small expansion ratio ($m \sim 1$), the correction matrix given by equation (25) simplifies to become a unit matrix and the above process can be shown to produce the same result as given by equation (22).

As the single chamber design presented in Figure 1 is expected to perform as a Helmholtz resonator rather than a simple expansion chamber, it is appropriate to also consider the transfer matrix for a resonator. Miwa and Igarashi [5] define the transfer matrix parameters for a resonator comprised of a side branch with a closed cavity as:

$$A = D = \cos^2(kL_m) - \sin^2(kL_m) - \frac{\sin(kL_m)\cos(kL_m)}{S_m}R \quad (29)$$

$$\left(B / j\frac{\rho c}{S_m}\right) = 2\sin(kL_m)\cos(kL_m) - \frac{\sin^2(kL_m)}{S_m}R \quad (30)$$

$$\left(C / j\frac{\rho c}{S_m}\right) = 2\sin(kL_m)\cos(kL_m) + \frac{\cos^2(kL_m)}{S_m}R \quad (31)$$

$$R = \frac{S_b\sin(kL_b)\cos(kL_c) + S_c\cos(kL_b)\sin(kL_c)}{\cos(kL_b)\cos(kL_c) - (S_c/S_b)\sin(kL_b)\sin(kL_c)} \quad (32)$$

where L_m is the length of the main pipe extending equally either side of the resonator, S_m is the cross-section of the main pipe, L_b is the length of the side branch (measured from main pipe centreline to start of cavity), S_b is the cross-section of the connecting branch, L_c is the length of the resonator cavity and S_c is the cross-section of the cavity. For the case of a non-reflecting termination of the system, parameters A, B, C and D given by equations (29) to (31) may be substituted into equation (28) to obtain the transmission loss for the resonator.

FINITE ELEMENT APPROACH

Acoustic finite element models of the three muffler designs have been developed using the finite element analysis (FEA) software package ANSYS (Version 11). Transmission loss is calculated by implementing the transfer matrix methodology using ANSYS Parametric Design Language (APDL), a scripting language that may be used to customise the FEA workflow. Kim and Soedel [7] and Wu *et al*. [8] present a method for the calculation of the four pole parameters which presents the general transfer matrix equation in the form

$$\begin{bmatrix} p_1 \\ p_2 \end{bmatrix} = \begin{bmatrix} A^* & B^* \\ C^* & D^* \end{bmatrix} \begin{bmatrix} U_1/S_1 \\ U_2/S_2 \end{bmatrix} \quad (33)$$

S_1, S_2 are the cross-sectional areas of the inlet and outlet pipes, respectively. Utilising this method, the transfer matrix parameters in equation (33) may be calculated by applying the following two load cases

$$\text{Case 1:} \quad A^* = p_1\big|_{u_1=1,\,u_2=0}, \qquad C^* = p_2\big|_{u_1=1,\,u_2=0} \quad (34,35)$$

$$\text{Case 2:} \quad B^* = p_1\big|_{u_1=0,\,u_2=1}, \qquad D^* = p_2\big|_{u_1=0,\,u_2=1} \quad (36,37)$$

where u_1 and u_2 are particle velocities on either side of the acoustic component. It is then possible to calculate the original four pole parameters by combining equations (23) and (33) to give

$$A = \frac{A^*}{C^*} \quad B = \left(B^* - \frac{A^* D^*}{C^*}\right)\left(\frac{1}{S_2}\right) \quad C = \left(\frac{1}{C^*}\right)S_1 \quad D = \left(\frac{-D^*}{C^*}\right)\left(\frac{S_1}{S_2}\right) \tag{38-41}$$

While each load case requires an acoustic particle velocity to be specified, ANSYS does not accept velocity as an applied boundary condition. Instead, the velocity must be converted to a displacement [22] using the relationship $X = -ju / \omega$. The inlet boundary condition can thus be specified with a velocity magnitude equal to unity (u = 1). Acoustic load cases are applied to the finite element model at single frequencies, and analyses are conducted across the frequency range of interest at regular frequency intervals, with the number of computational runs being dictated by the frequency resolution required. The output from the finite element analysis includes the acoustic pressure and particle velocity at each node in the finite element model. The acoustic pressure data from nodes located at the muffler inlet and outlet can be used to calculate the complex transfer matrix parameters using equations (34) to (41). The matrix parameters calculated at each frequency are substituted into equation (28) to obtain the transmission loss spectrum over the desired frequency range.

The finite element model for each muffler design was meshed using tetrahedral FLUID30 elements with mesh controls applied to adequately resolve the fine details and tight radii in the muffler geometries. The resulting mesh size produced 15-25 elements per acoustic wavelength at the upper bound of the frequency range being analysed (limiting case). This is very high compared with a widely accepted minimum acceptable mesh density of 6 elements per wavelength. The fluid (air) was assumed to be non-flowing and inviscid and acoustic damping was not utilised at the fluid-structure interface. That is, the walls were treated as acoustically hard boundaries. The lack of air flow and the absence of both air and structural damping from the models represent a simplification of the actual conditions present in a CPAP device muffler during operation. Further work is being conducted to incorporate damping into the finite element models and to assess the impact of typical CPAP device air flow rates on the acoustic performance of the muffler designs.

EXPERIMENTAL APPROACH

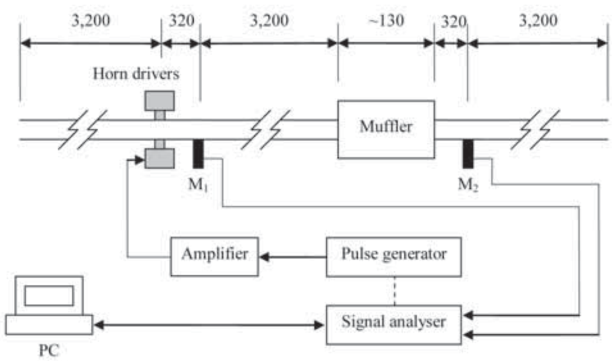

Figure 5: Schematic diagram of the two-microphone acoustic pulse experimental set-up (dimensions are mm)

Experimental data was obtained using a two-microphone technique based on a short duration acoustic pulse [19]. Figure 5 shows a schematic diagram of the experimental set-up used in the current study.

A transient acoustic pulse was generated from the Brüel & Kjær LAN-XI Pulse front end and fed to two TU-650 horn drivers via a PA-25E power amplifier. The pulse propagated down the 18mm diameter PVC conduit where it was measured by the upstream microphone, M_1, before continuing to the muffler inlet. The pressure of the corresponding pulse transmitted from the outlet of the muffler was measured by the downstream microphone, M_2. Utilising long lengths of pipe in the system provided a time separation of approximately 15ms between the arrival of the initial pulse and the arrival of the subsequent reflections of the pulse at the muffler and pipe ends. This time delay was sufficient to facilitate extraction of the time intervals that captured only the initial positive travelling wave from the total time histories recorded by the two microphones. Rectangular windowing with leading and trailing cosine tapers was applied to the time history measured by M_1 and exponential windowing with a leading cosine taper and 5ms decay constant (τ) was applied to the time history measured by M_2. These extracted time histories were captured for 100 individual pulses, Fourier Transformed, and the results averaged in the frequency domain. The transmission loss for the muffler was then obtained using

$$TL = 10\log_{10}\left(\frac{FFT_1}{FFT_2}\right) \tag{42}$$

where FFT_1 and FFT_2 are the Fourier Transforms of the time histories of the incident and transmitted waves, respectively.

RESULTS AND DISCUSSION

Single chamber design

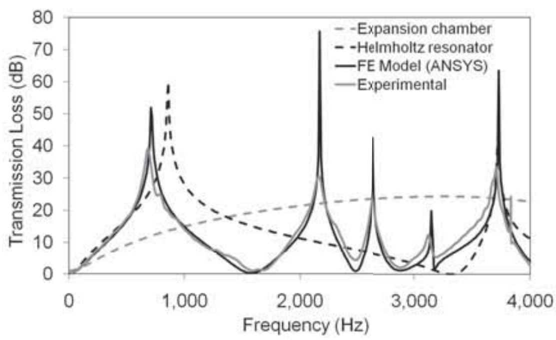

Figure 6: Single chamber muffler comparing analytical (transfer matrix method), FE and experimental results

Figure 6 contains the transmission loss obtained experimentally for the single chamber muffler and the transmission loss predicted by the analytical transfer matrix method (both as an expansion chamber and side branch Helmholtz resonator) and the finite element model. While both analytical models exhibit a poor correlation with the experimental results, the resonator model aligns more closely with the measured results than the expansion chamber model. This is attributed to the close proximity between the coaxial inlet and outlet ports. The FE results show excellent agreement with the experimental results over the frequency range assessed, with the exception that the magnitude at resonant frequencies is over-predicted by the FE model. This is attributed to the FE model assuming an inviscid fluid and rigid walls. The inclusion of damping in the FE model would result in a reduction in the peaks at the resonances.

Integrated chamber design

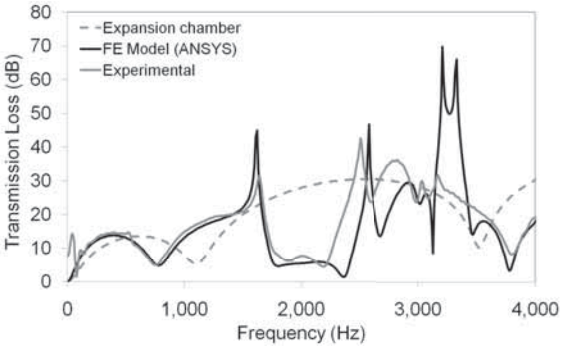

Figure 7: Integrated chamber muffler comparing analytical (transfer matrix method), FE and experimental results

Figure 7 contains the transmission loss obtained experimentally for the integrated chamber muffler and the transmission loss predicted by the analytical transfer matrix method and the finite element model. The analytical approach modelled the muffler chambers as two expansion chambers connected in series. While similarities may be observed, the analytical results show poor agreement with the experimental results. This is attributed to the simplified geometric representation used in the analytical model and the influence of the higher order modes as the frequency increases. The FE results show good agreement up to approximately 3 kHz. The underlying trend followed by both sets of results is similar over the remaining frequency range with the exception of the large double peak predicted by the FE model.

Interconnected chamber design

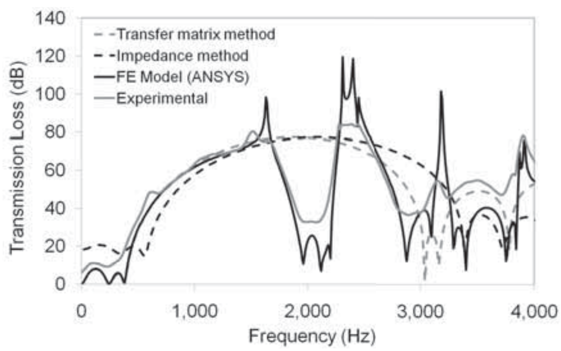

Figure 8: Interconnected chamber muffler comparing analytical (impedance and transfer matrix methods), FE and experimental results

Figure 8 contains the transmission loss obtained experimentally for the interconnected chamber muffler and the transmission loss predicted by the analytical impedance method, the analytical transfer matrix method, and the finite element model. Both analytical methods model the muffler as series-connected expansion chambers. The analytical results show reasonable agreement with the experimental results up to 1.6 kHz, with the transfer matrix method exhibiting closer agreement than the impedance method. Based on a limiting chamber diameter of 95mm, the plane wave cut-on frequency is approximately 2.1 kHz. Application of the analytical approach to this muffler design is complicated by the orthogonal alignment of the inlet and outlet connections on each chamber as the volumes are no longer simply coaxial. While the impedance method provides for differing chamber cross sectional areas at inlet and outlet, both of the analytical methods presented assume one primary path for forward and reverse travelling waves. The FE model results show good agreement with the experimental results for the majority of the frequency range. Departure between the FE results and experimental data at higher frequencies is attributed to assumptions of totally rigid walls and no acoustic damping in the FE model. Altering wall compliance has been shown to significantly affect resonant frequencies in the transmission loss results [23].

During experimental testing, it was noted that pressures in the FFT spectrum for the downstream microphone (M_2) above 500Hz were less than 20 x 10^{-6} Pa, resulting in poor coherence. These observations highlight the importance of producing an acoustic pulse of short duration which still has sufficient energy at high frequencies to provide an acceptable signal-to-noise ratio. A further challenge presented by the two-microphone method is the need to weight the time domain results to capture the initial incident (M_1) and transmitted (M_2) pulses while excluding any subsequent reflections. While use of long lengths of duct goes some way towards providing adequate time spacing, it must be balanced against the higher system losses attributable to the longer ducts. Internal reflection within each of the muffler chambers complicates the separation of the initial transmitted and subsequent reflected pulses due to the length of time decay and lack of clarity in the form of the pressure signal. Care was also required to avoid leakage errors associated with the FFT of the microphone results.

CONCLUSIONS

In this study, the acoustic performance of three reactive muffler designs similar to those used in CPAP devices have been numerically and experimentally compared. Analytical expressions for the transmission loss based on both impedance formulae and the transfer matrix method were developed. Finite element models of the three muffler designs were also generated based on the transfer matrix method. Experimental validation of the computational results was conducted using a test rig based on the two-microphone acoustic pulse method.

At lower frequencies, the analytical results showed reasonable agreement with the finite element and experimental results. However, they departed significantly before reaching the first cut-on frequency, beyond which the plane wave assumption is not valid. The analytical methods are not suitable for CPAP designs due to the complexity of the muffler geometries and the non plane wave behaviour that must be considered when the design incorporates orthogonal inlet and outlet ports.

In general, good agreement between the finite element and experimental results were obtained. The FE models over-predict the transmission loss at resonant frequencies and this is attributed to simplifying assumptions corresponding to the use of inviscid fluid and rigid walls. The experimental results predict that the interconnected chamber design has the most desirable transmission loss characteristics, which is attributed to the combined effect of three discrete chambers, the isolation provided by the interconnecting pipes and a 250% increase in total chamber volume compared to the single chamber design. The average transmission loss for the integrated chamber design is similar to that of the single chamber design despite a 60% increase in total chamber volume. However, the frequency response is more uniform and the design outperforms the single chamber muffler across much of the frequency range. The proximity between the inlet and outlet ports of the single chamber muffler design results in this design behaving as a Helmholtz resonator, producing narrow transmission loss peaks centred at specific frequencies but having significantly lower performance at other frequencies. Variants of this design may be useful where discrete frequencies are to be targeted but it has limited application over a broad frequency range.

Further refinement of the work covered by this study will include an assessment of the acoustic interaction of adjacent chambers and integration of wall compliance into the FE models. The acoustic performance of the predominantly reactive mufflers used in CPAP devices can be enhanced with the inclusion of dissipative materials. Current work is investigating appropriate experimental techniques for the acoustic characterisation of polyurethane foams with the aim of incorporating significant foam volumes into the muffler FE models.

ACKNOWLEDGMENT

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REFERENCES

[1] Jones, A.D. (1984) “Modelling the exhaust noise radiated from reciprocating internal combustion engines – A literature review”, *Noise Control Engineering*, **23**, 12-31.

[2] Munjal, M.L. (1987) *Acoustics of Ducts and Mufflers*, Wiley, New York.

[3] Davies, P.O.A.L. (1988) “Practical flow duct acoustics”, *J. Sound and Vibration*, **124**, 91-115.

[4] Igarashi, J. and Toyama, M. (1958) “Fundamentals of Acoustic Silencers: (I) Theory and experiment of acoustic low-pass filters”, Report 339, *Aeronautical Research Institute*, University of Tokyo, 223-241.

[5] Miwa, T. and Igarashi, J. (1959) “Fundamentals of Acoustic Silencers: (II) Determination of four terminal constants of acoustical elements”, Report 344, *Aeronautical Research Institute*, University of Tokyo, 67-85.

[6] Sullivan, J.W. (1979) “A method for modelling perforated tube muffler components: (I) Theory”, *J. Acoust. Soc. America*, **66**, 772-778.

[7] Kim, J. and Soedel, W. (1989) “Analysis of gas pulsations in multiply connected three-dimensional acoustic cavities with special attention to natural mode or wave cancellation effects”, *J. Sound and Vibration*, **131**, 103-114.

[8] Wu, T.W., Zhang, P. and Cheng, C.Y.R. (1998) “Boundary element analysis of mufflers with an improved method for deriving the four-pole parameters”, *J. Sound and Vibration*, **217**, 767-779.

[9] Barbieri, N., Barbieri, R. and de Lima K.F. (2004) “Errors in transmission loss prediction – the bispectrum and kurtosis approaches”, *Mechanical Systems and Signal Processing,* **18**, 223-233.

[10] Wu, T.W. and Wan, G.C. (1996) “Muffler performance studies using direct mixed-body boundary element method and a three-point method for evaluating transmission loss”, *J. Vibration and Acoustics*, **118**, 479-484.

[11] Byrne, K.P. (1980) “Calculation of the specific normal impedance of perforated facing-porous backing constructions”, *Appl. Acoustics*, **13**, 43-55.

[12] Young C-I.J. and Crocker, M.J. (1975) “Prediction of transmission loss in mufflers by the finite-element method”, *J. Acoust. Soc. America*, **57**, 144-148.

[13] Barbieri, R., Barbieri, N. and de Lima K.F. (2004) “Application of the Galerkin-FEM and the improved four-pole parameter method to predict acoustic performance of expansion chambers.” *J. Sound and Vibration*, **276**, 1101-1107.

[14] Barbieri, R. and Barbieri, N. (2006) “Finite element acoustic simulation based shape optimization of a muffler”, *Appl. Acoustics*, **67**, 346-357.

[15] Cheng, C.Y.R. and Seybert, A.F. (1991) “A multidomain boundary element solution for silencer and muffler performance prediction”, *J. Sound and Vibration*, **151**, 119-129.

[16] Selamet, A. and Ji, Z.L. (1999) “Acoustic attenuation performance of circular expansion chambers with extended inlet/outlet”, *J. Sound and Vibration*, **223**, 197-212.

[17] Middelberg, J.M, Barber, T.J., Leong, S.S., Byrne, K.P. and Leonardi, E. (2004) “CFD analysis of the acoustic and mean flow performance of simple expansion chamber mufflers”, *Proc. ASME Intl. Mechanical Engineering Congress and Exposition*, Anaheim, California.

[18] Bilawchuk, S. and Fyfe, K.R. (2003) “Comparison and implementation of the various numerical methods used for calculating transmission loss in silencer systems”, *Appl. Acoustics*, **64**, 903-916.

[19] Jones, P. and Kessissoglou, N. (2009) “An evaluation of current commercial acoustic FEA software for modelling small complex muffler geometries: Prediction vs Experiment”, *Proc. ACOUSTICS 2009*, Adelaide.

[20] Seybert, A.F. and Ross, D.F. (1977) “Experimental determination of acoustic properties using a two microphone random-excitation technique”, *J. Acoust. Soc. America*, **61**, 1362-1370.

[21] Reynolds, D.R. (1981) *Engineering Principles of Acoustics*, Allyn and Bacon, Massachusetts.

[22] Imaoka, S. (2009) “ANSYS Tips and Tricks: Acoustic Elements and Boundary Conditions”, Memo Number *STI:05/01B* [http://ansys.net/?mycat=search&mytype=Tips&mycategory=Sheldon]

[23] Munjal, M.L., Venkatesham, B., Tiwari, M. (2009) “Four-Pole Parameters of a Rectangular Expansion Chamber with Yielding Walls”, *Proc. 16th Intl. Congress on Sound and Vibration*, Krakow.

USE OF PiP TO INVESTIGATE THE EFFECT OF A FREE SURFACE ON GROUND VIBRATION DUE TO UNDERGROUND RAILWAYS

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The Pipe-in-Pipe model (PiP) is a quick and accurate tool for calculating vibration from underground railways and for assessing the performance of vibration countermeasures. The original model formulation simulates a tunnel buried in a fullspace but has recently been extended to account for a free-surface (i.e. halfspace). Results from the two versions are compared to quantify the effect of the free-surface on soil power spectral density (PSD) values. The study suggests that it is reasonable to assume the PSD surface results predicted from the free-surface model will be approximately 6dB more than those predicted by the fullspace model when the tunnel is at a depth of two tunnel-diameters or more. For tunnel depths less than two tunnel-diameters it seems beneficial to account for the free surface in the simulation as there is significant variation in the results invalidating the 6dB assumption.

1. INTRODUCTION

Underground railways are an effective means of transport in urban areas: locating the railway infrastructure below ground aids in reducing surface vehicle congestion and per-capita pollution emissions are lower than from the equivalent number of personal vehicles. A main concern with underground railways is vibration which propagates to nearby buildings causing annoyance to people [1]. The vibration may be perceived either directly through motion of floors and walls or indirectly as re-radiated noise. The vibration frequencies of interest typically range between 15-150 Hz [2]; higher frequencies are generally attenuated rapidly with distance along the transmission path through the soil [3]. The concern over vibration from underground railways has spurred the development of BS ISO 14837-2 to quantify acceptable vibration levels from underground railways.

Designers of underground railways and surrounding buildings rely on vibration predictions from numerical simulations to ensure they do not exceed these specified vibration levels; failure to do so can result in costly retrofitting of vibration countermeasures. Numerical methods commonly used for simulating ground vibration due to underground railways include two-dimensional [4, 5] and three-dimensional [6, 7] finite-element (FE) and boundary-element (BE) models; however, each method suffers from its own set of difficulties. Andersen and Jones [8] show that while 2D models require less computation effort they prove to be only qualitatively useful to indicate whether reductions in vibration can be achieved through structural changes. Three-dimensional models provide more quantitative results but require significantly more computational effort.

Semi-analytical modeling is also an accurate and efficient method for simulating ground vibration. Forrest and Hunt [9, 10] present a computationally efficient, three-dimensional semi-analytical model for calculating soil vibration in a fullspace from underground railways, known as the Pipe-in-Pipe model (PiP). As the name implies, the PiP model represents the tunnel and soil as concentric, coupled "pipes" as shown in Figure 1. The tunnel pipe is modeled using thin-shell theory while the soil pipe is modeled using elastic continuum theory. The outer radius of the tunnel pipe is equal to the inner radius of the soil pipe, and the outer radius of the soil pipe is infinite to simulate a fullspace.

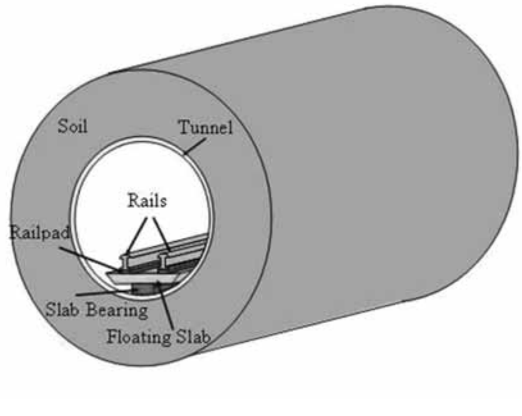


Figure 1. A floating-slab track coupled to the tunnel-soil model

Forrest [9] describes the method of transforming the governing equations of motion for the tunnel and the surrounding soil into the frequency, wavenumber, and circumferential ring-mode domains using Discrete Fourier Transforms (DFT); this allows the transfer functions for the tunnel and soil to be written in the simplified manner as follows

$$\tilde{\mathbf{U}} = [\tilde{\mathbf{A}}]_{tunnel} \left(\tilde{\mathbf{F}} - \tilde{\mathbf{P}}\right) \qquad \tilde{\mathbf{U}} = [\tilde{\mathbf{A}}]_{soil}\tilde{\mathbf{P}} \qquad (1)$$

where $\tilde{\mathbf{U}}$ represents the cylindrical displacements at the tunnel-soil interface, $\tilde{\mathbf{F}}$ describes the traction applied to the inner surface of the tunnel, and $\tilde{\mathbf{P}}$ is the reactionary traction developed at the tunnel/soil interface. The transfer function matrices $[\tilde{\mathbf{A}}]_{tunnel}$ and $[\tilde{\mathbf{A}}]_{soil}$ are both 3×3 matrices where the components are functions of material properties, wavenumber, frequency and ring-mode only. The tunnel and soil are coupled at the interface by enforcing continuity of displacements and equilibrium of reaction forces which results in the coupled equation of motion

$$\tilde{\mathbf{U}} = \left([\mathbf{I}] + [\tilde{\mathbf{A}}]_{tunnel}[\tilde{\mathbf{A}}]_{soil}^{-1}\right)^{-1}[\tilde{\mathbf{A}}]_{tunnel}\tilde{\mathbf{F}} \tag{2}$$

The actual displacements are then calculated by transforming the solution back into the spatial domain using inverse DFT. For full details on this method, please refer to references [9, 10].

A slab and rails are mounted to the bottom of the tunnel using Euler-Bernoulli beam theory; rail pads and slab bearings are represented by two separate, continuous layers of hysteretically damped springs shown schematically in Figure 1. Hussien and Hunt [11] show that the transfer function in the frequency and wavenumber domain for the coupled rail-slab section subjected to harmonic moving loads can also be expressed in a simplifed matrix form as

$$\tilde{\mathbf{U}}_{rail} = [\tilde{\mathbf{A}}]_{track}\tilde{\mathbf{F}}_{rail} \tag{3}$$

where $\tilde{\mathbf{U}}_{rail}$ describes the vertical motion of the rail and the slab, $\tilde{\mathbf{F}}_{rail}$ describes the moving harmonic force, and $[\tilde{\mathbf{A}}]_{track}$ is the 2×2 transfer matrix built from the coupled Euler-Bernoulli equations. The slab displacements are used to calculate the forces at the wheel/rail interface which are used as inputs into the fully coupled PiP model.

The theory of random vibration [12] is used to calculate vibration levels in the soil due to an infinite train passing along a rough track. The model used for this purpose is shown in Figure 2 where an infinite number of axles (only 3 axles, i.e. 6 wheels are shown in the figure) are used to determine the resultant moving loads due to an uncorrelated rail roughness spectrum; it is assumed that the roughness on the two rails is identical. It is reasonable to ignore the sprung masses of the train (i.e. main body of the carriages) due to the low stiffness of the primary suspension isolating the carriage from the axle assembly (i.e. the unsprung mass). Standard random vibration theory states that the power spectral density (PSD) for displacement response at any point in the pipe-in-pipe soil model can be calculated using

$$PSD(\omega) = \sum_{j=1}^{N} \left|H_j(\omega)\right|^2 S_\delta(\omega) \tag{4}$$

where $H_j(\omega)$ is the transfer function describing the displacement at the observation point in the soil due to a unit harmonic roughness applied under the jth axle of the train, and $S_\delta(\omega)$ is the single-sided rail-roughness spectrum experienced by an axle at an angular frequency ω. A realistic value of the rail roughness spectrum can be calculated from the following empirical formula [13]

$$S_\delta(\omega) = \frac{a}{v\left(b + \frac{\omega}{2\pi v}\right)^3} \tag{5}$$

where v is the load velocity (m/s), ω is the forcing frequency (rad/s), and a and b are constants describing the rail unevenness (eg. average rail roughness values are given as 1.31×10^{-2} mm^2/m^2 and 2.94×10^{-2} /m, respectively). This allows the calculation of power spectral density and insertion gain (IG) of the vertical displacement for any point in the soil; IG is defined as the ratio between the PSD displacement before and after changing parameters of the track, tunnel or soil.

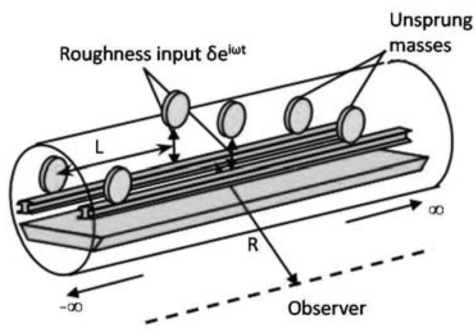


Figure 2. Unsprung masses moving over rough rails causing random force input

The PiP model has been validated against a coupled BE-FE model and shown to have good agreement over the frequency range of interest [14] but with a computational cost which is orders of magnitude less than the BE-FE model. The combination of model accuracy and computational efficiency makes PiP a powerful computational tool for calculating vibration from underground railways and for assessing the performance of vibration countermeasures. The reader is invited to evaluate a free version of the software at www.pipmodel.com.

Hussein and Hunt [15] have recently extended the PiP model to account for a tunnel embedded in a halfspace rather than a fullspace. The standard pipe-in-pipe arrangement is not suitable for including a free-surface since the soil is modeled as a cylinder with infinite radius. To account for the free surface an extension to the model is incorporated which calculates the Green's functions for a homogeneous halfspace [16]; the standard PiP method predicts the loading at the tunnel-soil interface and is used as input into the halfspace model to calculate surface response.

The purpose of the current work is to investigate the effect of this free-surface on ground vibration levels. Equivalent models are run using the fullspace model (PiP version 3) and halfspace model (PiP version 4) and compared to determine how the inclusion of the free-surface affects the vibration levels along the surface. The paper is broken into three sections: a description of the parameters used for the comparison between fullspace and halfspace models, a discussion of the results, and concluding remarks.

2. MODEL DESCRIPTION

Figure 3 shows the software user-interface for PiP v4 (note the halfspace schematic of the buried tunnel at the top-center of the screen). Default parameters for the soil, tunnel, floating-slab track and train can be altered by the user as necessary for specific simulations. The observation point can be set to any location in the fullspace (v3) or halfspace (v4) at which the PSD of the vertical displacements will be calculated. As a demonstration of the computational efficiency of the software the default parameters are used as inputs for a simulation with frequency range of 2-150 Hz with 2Hz intervals. The running time to produce the response curve shown in the plot window of Figure 3 is 42 seconds using PiP v4; to run the equivalent simulation in PiP v3 takes only 6 seconds. The difference in computational time is due to the homogeneous halfspace Green's function formulation required for simulating the free-surface in v4. As PiP v3 is less computationally expensive some users may prefer to trade the added accuracy of accounting for the free-surface for maintaining quick run-times. It would be useful to understand the effect that the free-surface has on PSD values so the uncertainty associated with making this compromise can be quantified.

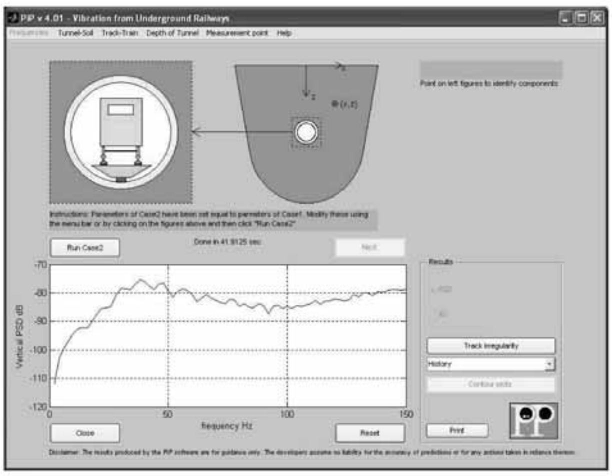


Figure 3. The graphical user interface of the PiP software

Table 1. Model Properties

Soil	
Elastic modulus	0.55 GPa
Density	2000 kg/m^3
Poisson's ratio	0.44
Shear loss factor	0.06
Dilation loss factor	0
Tunnel	
Elastic modulus	50 GPa
Density	2500 kg/m^3
Poisson's ratio	0.3
Shear loss factor	0
Dilation loss factor	0
Outer radius	3 m
Wall thickness	0.25 m
Train	
Unsprung axle mass	500 kg
Spacing between axles	20 m
Rails	
Mass of one rail	50 kg/m
Bending stiffness of one rail	5 MNm^2
Bending stiffness loss factor	0.02
Railpads	
Railpad stiffness per rail	200 MN/m/m
Railpad loss factor	0.3
Slab	
Bending stiffness	1430 MNm^2
Bending stiffness loss factor	0.05
Slab mass	3500 kg/m
Slab bearing	
Bearing stiffness	221 MN/m/m
Bearing loss factor	0.5

To investigate the effect of a free-surface on vertical soil PSD values, a set of representative properties were chosen to simulate an underground railway in the London, UK area. The parameters tabulated in Table 1 are used in both PiP v3 and v4 over a frequency range of 2-150Hz using 2Hz spacing. Figure 4 details the location of five observation points (x =0, 2.5, 5, 10, 20m) used to compare the results from the two models. The tunnel depth is defined as the distance between the free surface and the center of the tunnel; as there is no free surface in PiP v3 equivalent observation points are placed in the fullspace. Five tunnel depths are investigated: 3.5m, 5m, 10m, 20m, 40m. The results at the observation points for the two models are compared in the following section to determine the effect of the free surface.

3. RESULTS AND DISCUSSION

The vertical power spectral density at each observation point was calculated using PiP v3 and v4 for the parameters specified above. The results comparing the two models at x=10m for a tunnel depth of 5m are presented in Fig 5. This typical response shows the model containing the free-surface (PiP v4) predicts PSD values which are offset by a relatively constant margin from those of the fullspace model (PiP v3).

The differences in PSD response for the two models at all observation points are presented in Fig 6. The PSD difference is calculated in dB (ref 1 mm^2/Hz) by subtracting the free-surface response by the fullspace response: a positive value indicates PiP v4 predicts higher PSD values than does PiP v3 at equivalent points.

The results suggest the offset between the two models is relatively insensitive to the observation point, increasingly so at greater tunnel depths. At the deepest tunnel depth used in this study (Fig 6(e) at TD=40m) the results for the five observation points are closely banded around the 6dB level. As tunnel depth decreases the spread between the observation points' responses increases but stays relatively centered around the 6dB offset level.

This 6dB offset is not entirely unexpected. Consider axial vibrations in the infinite and semi-infinite bars shown in Fig 7. If the bars are nominally identical and subjected to the same incoming pressure wave-field, it is known that the displacement at the free end of the semi-infinite bar (u_2) will be twice that of the infinite bar at the same location (u_1) [17]; in the decibel scale this doubling of the response is equivalent

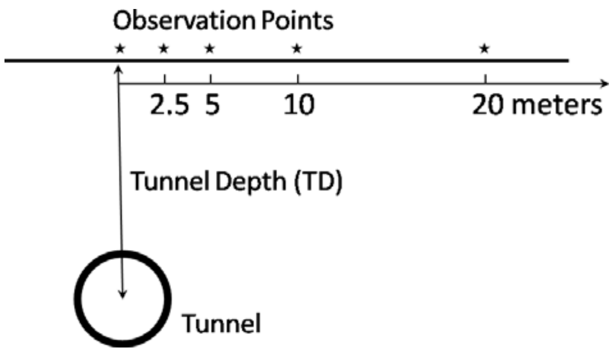


Figure 4. Schematic showing location of observation points

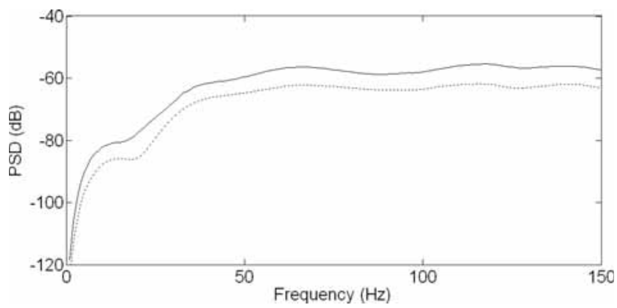
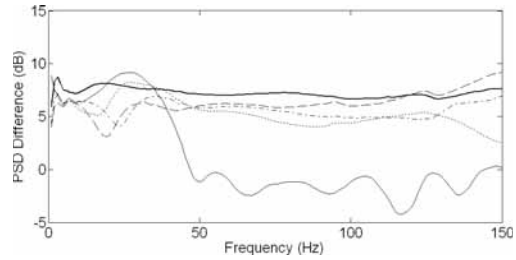


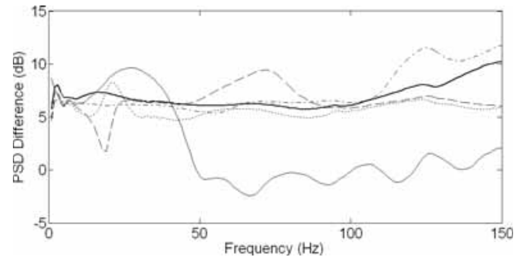
Figure 5. Response at x=10m for a tunnel depth of 5m: (solid-line)-PiP v4 with free-surface; (dotted-line)-PiP v3 using fullspace solution; results presented in dB (ref 1 mm^2/Hz)

to an increase of 6dB. The "doubling effect" is attributed to the superposition of the incoming and reflected waves. This effect can be extrapolated to the fullspace vs. halfspace argument if the incoming wave field is traveling perpendicular to the surface (i.e. wavefronts parallel to the surface). In the case of a tunnel buried in a halfspace, waves are emitted from the tunnel exterior with cylindrical wavefronts; if the tunnel is at great depth the radius of curvature of the wavefronts will be quite large thus the waves will be virtually parallel to the surface when they are reflected.

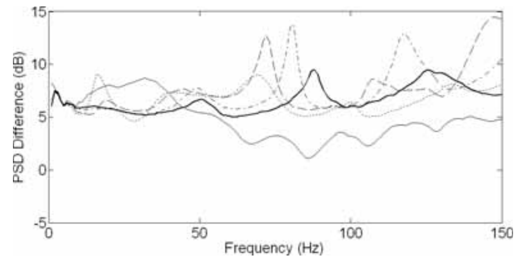
The increase in spread around this 6dB offset for shallow tunnel depths is attributed to the complexities of wave-surface interaction above the tunnel. When the tunnel is near the surface the wavefronts still have relatively small radii of curvature when they are reflected by the free-boundary. The amount of energy that is reflected and the superposition of that energy with the incoming field depends on the incident wave angle. Therfore changing the location of the observation point on the surface (i.e. the angle between the source and the surface point) significantly affects the amount of energy transformed into vertical particle motion on the surface. This general trend in angle dependance can best be seen in Fig 6(a) where the PSD difference drops as the observation point moves away on the surface. At $x = 20$m the incident angle is steep compared to the other observation points thus little superposition of reflected energy occurs, resulting in a PSD difference which approaches 0dB.



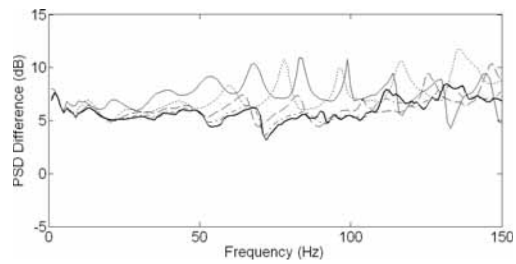
(a) Observation points for 3.5m tunnel depth



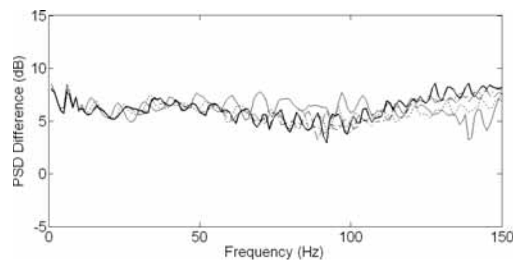
(b) Observation points for 5m tunnel depth



(c) Observation points for 10m tunnel depth



(d) Observation points for 20m tunnel depth



(e) Observation points for 40m tunnel depth

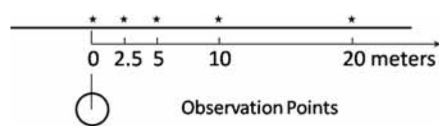


Figure 6. PSD difference [dB (ref 1 mm^2/Hz)] between free-surface model (PiP v4) and fullspace model (PiP v3) at various observation heights; positive value corresponds to an increase in PSD when free-surface is included compared to fullspace model. Legend: (thick-solid) x=0m; (dash-dot) x=2.5m; (dashed) x=5m; (dotted) x=10m; (thin-solid) x=20m

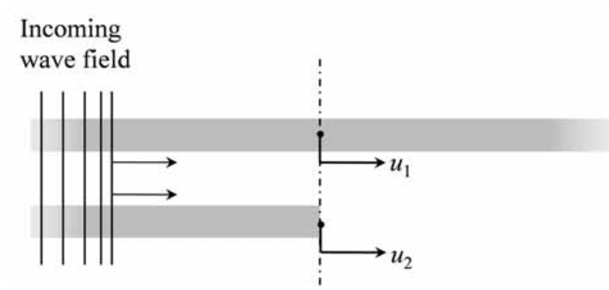


Figure 7. Schematic showing an infinite bar (upper) and semi-infinite bar (lower) subjected to an equivalent pressure wave field and the respective displacements at the boundary of the semi-infinite bar

4. CONCLUSIONS

The Pipe-in-Pipe model (PiP) is a powerful computational tool for calculating vibration from underground railways and for assessing the performance of vibration countermeasures. PiP v3 simulates a tunnel buried in a fullspace using analytical models for the tunnel and soil which results in rapid computational times (6 seconds for 150 Hz frequency sweep). PiP v4 accounts for a free surface by extending the model to include homogeneous halfspace Green's functions; however this added calculation complexity increases the computational time (42 seconds for a 150 Hz frequency sweep). Results from the two models are compared to quantify the effect of the free-surface on soil power spectral density (PSD) values. The study suggests that it is reasonable to assume the PSD surface results predicted from the free-surface model will be approximately 6dB more than those predicted at an equivalent "surface" location by the fullspace model when the tunnel is at a depth of two tunnel-diameters or more, regardless of the surface location of the observation point. This is a useful finding as the computationally efficient fullspace model can be confidently used early in the design process when numerous sensitivity studies are necessary without concern for how the free-surface alters the results. Once the design parameters are narrowed down to specific cases a free surface can be incorporated into the model to determine the soil response more accurately. If the tunnel depth is less than two tunnel-diameters there is significant variation in the results between the fullspace and halfspace models; this is attributed to the increased incident angle of the incoming vibration wavefield. For these instances the results suggest it would be beneficial to include a free surface in the simulation to obtain more accurate predictions of the surface PSD levels.

REFERENCES

[1] British Standards Institution, "Guide to measurement and evaluation of human exposure to whole body mechanical vibration and repeated shock" *BS 6841:1987*

[2] British Standards Institution, "Mechanical vibration - ground-borne noise and vibration arising from rail systems - part 1: general guidance" *BS ISO 14837-1:2005*

[3] C. Jones and J. Block, "Prediction of ground vibration from freight trains" *Journal of Sound and Vibration* **193**(1), 205-213 (1996)

[4] K. Chua K, K. Lo and T. Balendra, "Building response due to subway train traffic" *Journal of Geotechnical Engineering, ASCE* **121**(11), 747-754 (1995)

[5] C. Jones, D. Thompson and M. Petyt, "A model for ground vibration from railway tunnels" *Transport* **153**(2), 121-129 (2002)

[6] M. Mohammadi and D. Karabalis, "Dynamic 3D soil-railway track interaction by BEM-FEM" *Earthquake Engineering and Structural Dynamics* **24**, 1177-1193 (1995)

[7] A. Stamos and D. Beskos, "3D seismic response analysis of long lined tunnels in half-space" *Soil Dynamics and Earthquake Engineering* **15**, 111-118 (1996)

[8] L. Andersen and C. Jones, "Coupled boundary and finite element analysis of vibration from railway tunnels - a comparison of two- and three-dimensional models" *Journal of Sound and Vibration* **293**(3-5), 611-625 (2006)

[9] J. Forrest and H. Hunt, "A three-dimensional model for calculating of train-induced ground vibrations" *Journal of Sound and Vibration* **294**(4-5), 678-705 (2006)

[10] J. Forrest and H. Hunt, "Ground vibration generated by trains in underground tunnels" *Journal of Sound and Vibration* **294**(4-5), 706-736 (2006)

[11] M.F.M. Hussein and H.E.M. Hunt, "Modelling of floating-slab tracks with continuous slabs under oscillating-moving loads" *Journal of Sound and Vibration* **297**(1-2), 37-54 (2006)

[12] Newland D. An introduction to random vibrations, spectral and wavelet analysis, 3rd ed., Longman, Harlow, Essex, England, 1993.

[13] F. Frederich, "Effect of track geometry on vehicle performance" *Zeitschrift für eisenbahnwesen und vekehrstechnik - Glasers auualen* **108**(12), 355-362 (1984)

[14] S. Gupta, M. Hussein, G. Degrande, H. Hunt and Clouteau, "A comparison of two numerical models for the prediction of vibrations from underground railway traffic" *Soil Dynamics and Earthquake Engineering* **27**(7), 608-624 (2007)

[15] M.F.M. Hussein, S. Gupta, H.E.M. Hunt, G. Degrande and J. Talbot, "An efficient model for calculating vibration from a railway tunnel embedded in a half space" *Proceedings from the 13th International Congress of Sound and Vibration* Vienna, Austria (2006)

[16] A. Tadeu and E. Kausel, "Green's functions for two-and-a-half-dimensional elastodynamics problems" *Journal of engineering mechanics-ASCE* **126**(10), 1093-1097 (2000)

[17] K. Graff, "Wave motion in elastic solids" Clarendon Press, Oxford (1975)

NOTEVIEW: A COMPUTER PROGRAM FOR THE ANALYSIS OF SINGLE-LINE MUSICAL PERFORMANCES

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Newly developed software, NoteView, is used to analyse the fundamental frequency (F0) and categorical as well as microtonal pitch from audio recordings of music performances from a single line (monophonic) instrument. The code is based on the SWIPE algorithm developed by Camacho and Harris. The features of the interface are described and a comparison of two performances of a familiar piece played by a professional French horn player is used as an example for the purpose of investigating the pitch stability of the two renditions. Results produced by NoteView indicate that pitch pairs across the two performances differed by a mean of 7 cents, and that within-note standard deviation was typically 6 cents. These results are examined using the various customisable views and statistics returned by the software. Some of the features and limitations of NoteView are discussed. The software is currently implemented in Matlab and is freely available from the UNSW Empirical Musicology web site http://empa.arts.unsw.edu.au/research-and-creative-practice/research-projects/empirical-musicology/.

INTRODUCTION AND RATIONALE

Research on acoustics of speech and music frequently involves analysis of F0 (the fundamental frequency of a periodic signal), and a range of software exists for analysing this property [e.g. 1, for a review of recent such software see 2]. However, the acoustic analysis of music, speech and noise at times require significantly different approaches. An important example is pitch, which is a property of subjective music perception that can be expressed as a logarithmic transform of F0, and is reported here in units of semitones or cents[1]. Pitch tends to be produced and perceived categorically in Western music but not in most Western speech [3, 4]. While efficient software exists for analysis of many musical features [5, 6], the present paper reports our attempt to provide a graphic representation of musical pitch that facilitates comparison of different performances of the same piece.

We sought to develop a freely available tool that could take as its input a sound recording of a single line instrument (in the present case a French horn), parse the notes of the performance into a list of events (that could be inspected in both tabular and graphic forms), and to provide a comparison of this event list with an event list of another performance (also reported via tables and graphs). We wanted to be able to answer questions like ‘how close in pitch is player A playing piece X to that of player B playing the same piece’, or ‘how close in pitch is player A playing piece X to that of player A playing the same piece under a different circumstance’? While Dixon and Widmer’s [6] MATCH software can perform such functions, we wanted to have a strong focus on graphic and tabular representation of pitch and pitch comparison, as well as a range of statistics on pitch related information. In addition, the algorithmic foundations of our coding is different to that of Dixon and Widmer, who apply positive spectral difference vectors, whereas we focus on pitch strength, according to the SWIPE [Sawtooth Waveform Inspired Pitch Estimator] algorithm. The former has advantages in identifying note onset times with efficiency and speed. And while we too were interested in identifying temporal position of note events such that they could be matched across two renditions of the same performance, detailed information about F0 or pitch was the more important consideration here. As a preliminary exercise we examined a recording of a horn player playing the same familiar piece twice.

NoteView overview

NoteView is a music signal analysis toolbox we developed to analyse and visualise music performances in the Matlab computing environment. Reading in a monophonic sound recording, NoteView’s *nAnalyse* function begins by analysing the signal to determine the time-localised fundamental frequency and RMS power information. Onsets and offsets are then derived from the pitch and power information to form a series of audio *events*. For each event, various parameters are calculated, including various within-event fundamental frequencies, timing and RMS power parameters, in addition to

[1]A cent is a ratio of $2^{1/1200}$ between two F0s. One semitone in equal temperament corresponds to 100 cents. Other tuning systems can be represented with non-integer values of equal tempered semitones between intervals. An example is given in Table 1.

several statistical descriptions. These parameters are collated into a *signal* structure, which forms the basis for more complex analysis and visualisation.

NoteView also provides an automated comparative tool, which allows two *signal* structures to be compared to each other. This can be used to compare a particular performance against a template (e.g. one derived directly from a music score to exported audio, such as is available on many music notion software packages), or to compare two performances. This functionality is provided by the *nCompare* function, and is capable of automatically determining the most appropriate events from each signal structure to be matched and compared with each other. Having matched the appropriate event pairs, the *nCompare* function then provides statistical information regarding each pair of events and these are stored in a *compare* structure.

In addition to its automated analysis and comparative tools, NoteView provides a set of tools to facilitate the manual editing of the computed structures. Events can be added, removed, split, swapped, and the onset and offset information can also be manually modified. Any manual edits also initiate the automatic recalculation of the statistical parameters, thereby ensuring that the information stored in the structures remains up to date.

While NoteView is capable of displaying the information contained in its structures in tab-delimited lists, it also provides visualisation tools, which can expedite the analysis of large sets of data. The information contained in *signal* structures can be visualised using the nView function, in either a frequency-time plot or a signal amplitude-time plot. Information contained in the *compare* structure is accessed using the *ncView* function. The *ncView* function displays the information of two signal structures concurrently in the frequency-time domain, with an option to time-align the onsets of matched events to facilitate visual comparison. An additional visualisation option allows up to 3 parameters to be plotted simultaneously in 3 dimensions, allowing many salient factors in musical performances to be identified.

NoteView Specifications

The *signal* structure acquires the F0 information in the current implementation using the SWIPE' algorithm [7], which is a sawtooth waveform inspired monophonic pitch estimator. It uses the spectrum of a sawtooth wave which is adjusted to best match the signal spectrum under investigation, and was the technique selected because of its computational efficiency and good performance compared to a range of other approaches [see 7 for details], and its compatibility with public domain audio analysis frameworks such as PsySound3 [5]. The F0 estimates are calculated for non-overlapping windows sampled at 100 Hz, potentially providing accuracy to less than a musical cent. The F0 estimates are formed into tracks (a time series containing an array of F0 estimates over the time for the audio file being analysed) based on frequency deviations over time, track length and pitch strength. The short-time RMS power is calculated for non-overlapping windows sampled at 50 Hz.

The events are then identified according to the following rules. The attack (event onset) portion of the event is defined as the time taken to reach 80% of the maximum pitch strength, and the end of the note defined as the point in time when the pitch strength drops below 20% of the median pitch strength for the note or F0 deviates by more than 40 cents from the median F0, whichever is the smaller. These thresholds are customisable, but our experiments have produced good results with these values. Pitch strength here refers to the salience of a pitch (as distinct from the more commonly understood property of height, which is measured by F0 and is commonly called 'pitch'[2]) [8, 9]. For example, a complex tone is likely to be perceived as having more pitch strength than the same tone with added narrow-band noise at the centre frequency of the tone, despite having identical pitch height.

The parameter that reports pitch height in NoteView is based on the MIDI note numbering system. F0 is converted to equal tempered semitone count according to the MIDI (Musical Instrument Digital Interface) protocol where A4 (defined as F0 with 440Hz) is assigned the value 69, C4 is assigned the value 60, C#4 61 and so on. By addition of two decimal places we are also able to represent pitch in units of cents (see footnote 1). For example 60.03 can be a quantification of an equal tempered middle C played three cents sharp, and so other tuning systems that require non-integer representation of cents could, in principle, be analysed with an accuracy of ±0.5 cents, provided that the note was sustained for a sufficiently long time. In NoteView, the summary value of pitch height reported for an event is the median of F0 estimates across windows within the event, reported in these MIDI units.

Pitch deviation of an event indicates the amount of variability in F0 during the event. It is measured as the standard deviation (SD) of the F0 estimates produced by the array of windows of the event and is reported in units of cents.

Finally, pitch stability is reported. This indicates how stable a played pitch remains for the duration of the event. In the current implementation of NoteView this stability is reported by comparing the temporally split (into two even halves) event. The pitch variance (in cents) is calculated for each half, and an F-test is conducted that compares the two variances. If the F test is not statistically significant at $p = 0.05$, a value of 0 is assigned to the stability change. If the F-test is significant, then the log of the F statistic is assigned to the stability change. A positive value indicates that the second half of the note (event) has statistically less pitch variability than the first half. A negative value indicates that the first half has statistically less pitch variability than the second half. The value is not indicative of the actual variance/standard deviations, just the amount by which one half changed relative to the other,

[2]We use the term pitch and pitch height interchangeably here.

as reflected by the F-test. Therefore it is possible to have, for example, an increased stability as the note unfolds (less stable to more stable, reflected by a positive value) while the overall variability of the event (reported by NoteView as SD) is very small. In such a case, the stability rate value may have limited utility. Because the F statistic is only reported when the difference is significant, the absolute value will generally be greater than 1 (and therefore its log greater than 0) but zero when not significant. It should also be noted that this value has some dependence on the length of the event (long events are more likely to be reported as having non-zero stability rates). This is an artefact of the statistic.

In summary, the parameters that the NoteView *signal* structure reports for the purpose of the current investigation are onset time, offset time, pitch strength, pitch height, pitch deviation and pitch stability change. Additional parameters that are variants of the above are also accessible for table reporting and visualisation, and plans for further expansion and flexibility may be considered in future versions of the software.

The *compare* structure calculates the frequency distance between each event pair across the 2 signal structures using a dynamic programming algorithm [10] to determine the optimum matching of events. Parameters returned by the compare structure include several temporal relations concerning the relative locations and amount of temporal overlap of the two notes (consider the case when a note played in one performance is slightly longer than the corresponding note played in another). In addition, and of particular relevance to the analysis we present in the following section, the *compare* structure also returns the difference in F0 medians between event pairs and interval difference relative to the previous event, each in MIDI units.

WORKED EXAMPLE

To observe some of NoteView's capabilities, the recordings of a professional horn player were evaluated, with the objective of determining his pitch production accuracy with respect to just intonation (JI) and between renditions. The player was a professional horn player and composer[3]. The player reported the intention of using a just intonation system, which formed the basis of some of our analyses.

The accuracy with which flautists and violinists can reproduce pitches depends on the playing condition, their expertise and several other factors, and is typically greater than 10 cents [11, 12]. Furthermore, Sundberg and colleagues [13] reported that professional singers had a mean difference of 7 cents across renditions. We did not find published literature on the pitch 'accuracy' of a horn player reproducing the same piece and decided to test the hypothesis of 7 to 10 cent accuracy for the task.

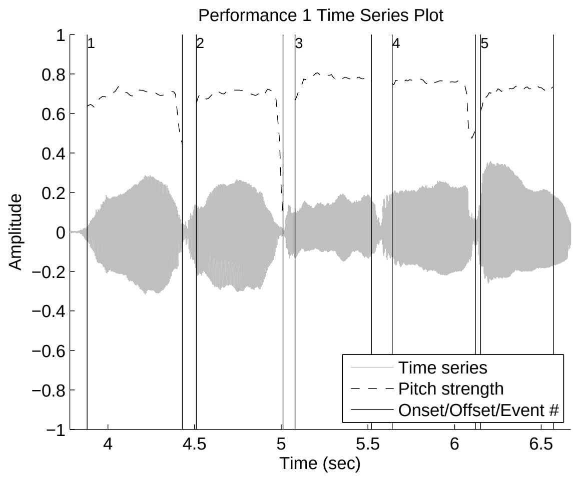

Figure 1. Time series plot of events 1-5 of performance 1 (first five notes of 'Twinkle Twinkle Little Star'), parameters relating to the event segmentation of the audio data.

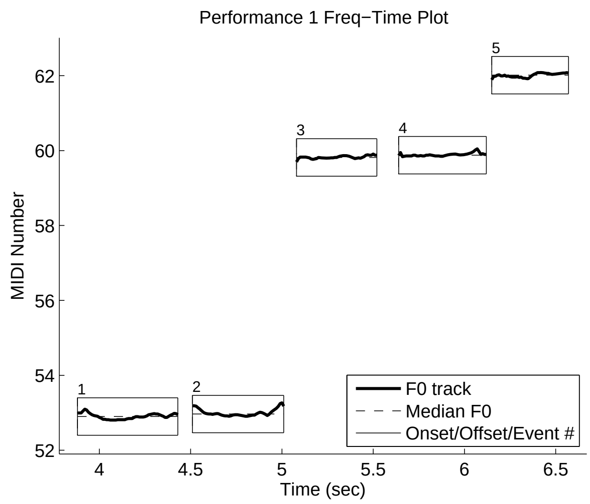

Figure 2. Frequency-time plot of events 1-5 for performance 1. The F0 track denotes the fundamental frequency (in MIDI units) as a function of time. The boxes outline the event number, encapsulating the event onset (left edge), offset (right edge), and ±50 cents either side of the median F0 (top/bottom edge).

To determine the pitch production accuracy of the horn player, the musician was instructed to play 'Twinkle Twinkle Little Star'. The player was not told that he would be playing the piece twice, and was not told that the intention was to examine pitch accuracy. After the piece was recorded, the player was asked to play the piece again, resulting in two recorded performances. The recordings were made at the recording studios of the Australian Institute of Music, 1-51 Foveaux Street, Surry Hills, NSW, Australia using ProTools audio editing software, with recordings saved as wav files at 16 bit depth, 44.1kHz sampling rate, suitable for NoteView input. The two performances were then analysed using NoteView's *nAnalyse* function to generate *signal* structures

[3]We are grateful to Michael Dixon, who agreed to be named as the performer in this study.

for each performance separately. Fig 1 and Fig 2 illustrate the visualisations generated by the *nView* function given a signal structure, as a time series, frequency-time plot as well as an event list (Table 1).

Table 1. List view of events 1-5 of performance 1. F0 is the fundamental frequency in Hz to two decimal places (two decimal places are returned by the software, though here and in typical performance conditions the error is about ±1Hz) . On and Off are the note onset and offset times with respect to the time elapsed in the sound file. MIDI# is the pitch in MIDI units. The player reported the intent to use just intonation (JI). His starting tone (and musical tonic) was F3, with an empirical F0 of 174 Hz.. With this F0 for F3, the ideal JI tunings are 196, 217, 231, 260 and 289 Hz for G3, A3, Bb3, C4 and D4 respectively (204, 386, 498, 702 and 884 cents above F3 respectively). Error is the difference in cents between the played F0 and the ideal JI F0.

Event#	F0(Hz)	On(sec)	Off(sec)	MIDI#	Note	Error(cts)
1	173.62	3.88	4.43	52.90	F3	0
2	174.17	4.58	5.01	52.96	F3	6
3	259.16	5.08	5.52	59.84	C4	-8
4	260.06	5.64	6.12	59.90	C4	-2
5	291.24	6.15	6.57	61.86	D4	11

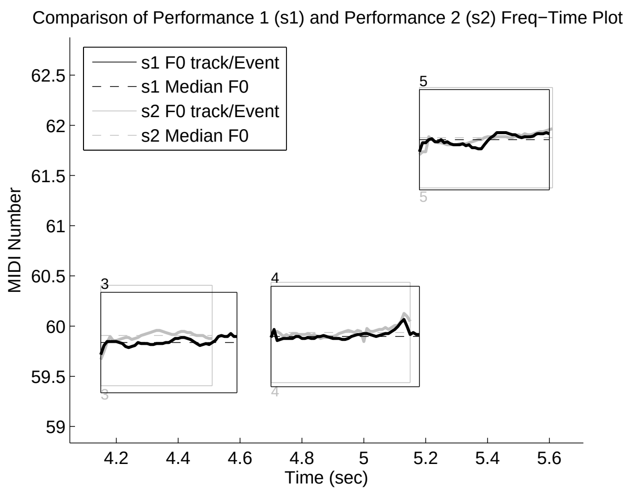

Figure 3. Frequency-time plot comparing the onset-aligned events 3-5 (third to fifth note of Twinkle Twinkle Little Star) of 2 performances.

The two performances were then compared using the *nCompare* function, generating a comparison structure matching events from the first and second performances. Fig 3 illustrates a comparative frequency-time plot generated using *ncView*, whose onsets are time-aligned to facilitate visual comparisons of the matched events. Fig 4 shows the deviation from ideal just intonation for each of the 42 events of the two performances (see caption for Table 1). The distribution of the events towards the top right side of the centre suggests that more pitches were played slightly sharp (above the median F0). Further, the occurrence of these points in the top right quadrant indicates a consistency of slightly sharp notes across the two performances. Visual inspection shows that these intonation variations fall within a boundary of ±20 cents with the calculated mean of the absolute value of the deviations being 7.2 cents, minimum of 0 and maximum of 25 cents.

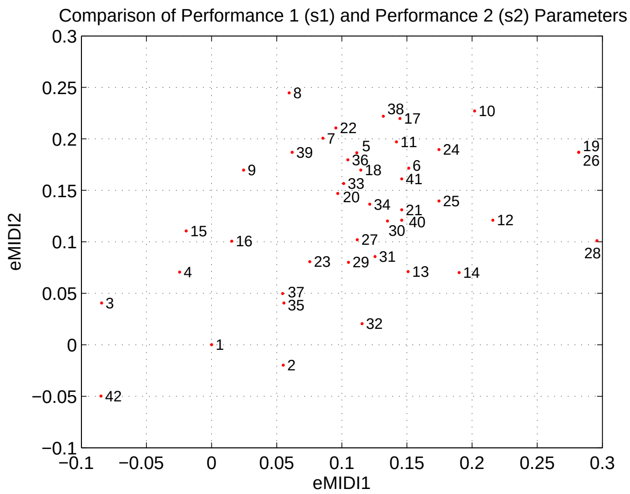

Figure 4. Deviation from just intonation pitch (in semitones) for performance 2 (y axis) plotted against deviation for performance 2, for the 42 notes in each performance.

In Fig 5 and Fig 6, the within-event variation of F0 is plotted. Fig 5 comparatively illustrates the standard deviations of the fundamental frequency for each event of the performances. We can see that the SDs of nearly all events fall within a square bounded by ±16 cents. The median of the within-note SD across the two performances was 6 cents with a maximum of 18 cents and a minimum of 2 cents. This means that, under the assumption of normal distribution, 68% (2 SDs) of notes vary by 12 (2 x 6) cents or less while the note is being played. This statistic is sensitive to the note onset and offset criteria, because including transients will inflate the standard deviation. The criterion for an event onset is the time at which the note has reached 80% of the pitch strength, as described above in NoteView Specifications.

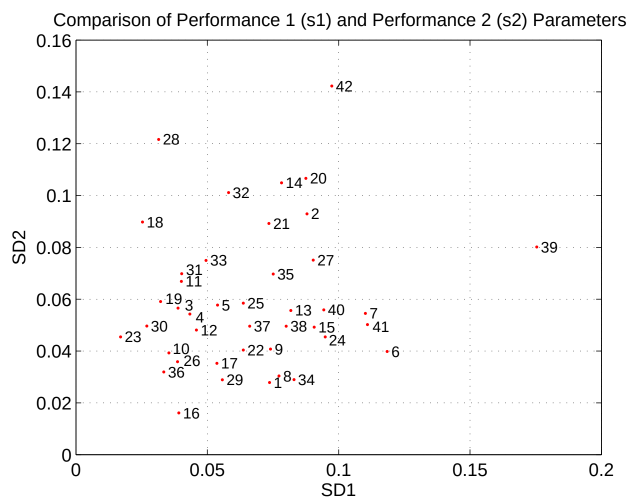

Figure 5. Standard deviation of F0 within each note in performance 2 plotted against that for each note in performance 1, for each of the 42 notes.

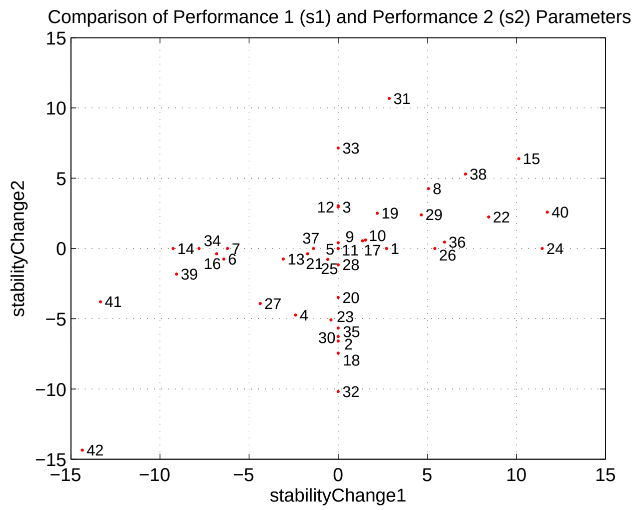

Figure 6. Stability change in performance 2 plotted as a function of that for performance 1. Units are log(F) when p =0.05, otherwise 0. Positive value denote variance decrease significantly in time from the first half to the second half of the event.

Fig 7 shows a different visualisation of the difference in pitch between events, listing them in the order in which they were played. The difference in pitch between notes in the second and first performances are plotted in semitones (MIDI units). Of the 42 events, 95% were within ±15 cents.

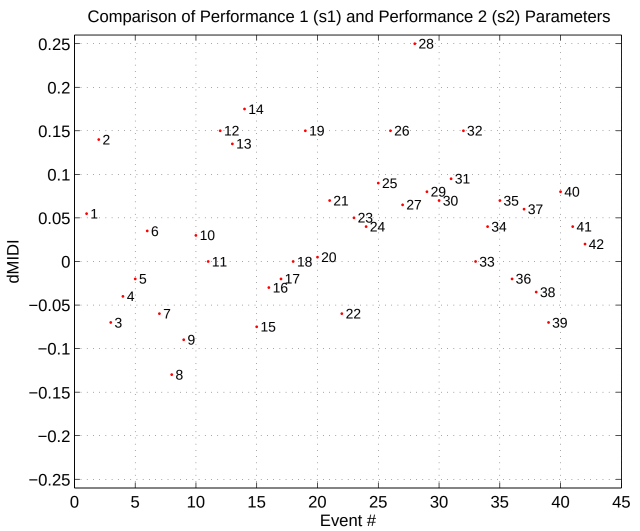

Figure 7. F0 in performance 1 minus F0 in performance 2 for each of the 42 notes, plotted in semitones.

A comparison of the within-event stability change is shown in Fig 6. 26% of events had no significant change in stability (each reported with a value of 0). 35% become significantly less stable (negative value) and 39% became more stable (positive value), with data pooled across performances. It needs to be kept in mind that some of these 'unstable' notes occur within the context of small overall variability within the note, and with the artefact of note duration affecting the calculation to some degree, as discussed above.

LIMITATIONS OF THE SOFTWARE

The current implementation of NoteView (Beta version 0.5) is limited to monophonic F0 detection between 30 and 5000 Hz. The event detection is generally quite robust, however there are issues when trying to track automatically events that have vibrato greater than 40 cents. In these circumstances, events can be manually edited using the *nEdit* function. There are also limits to the automated matching algorithm used in *nCompare*, which can skip a maximum of 2 consecutive events at a time. The system is expected to work for performances with legato and slurs (joining one event to the next event with little or no transient noise across the transition) when compared against a score, but otherwise is limited by the ability of the algorithm to identify such subtle transitions.

Additionally it should be noted that different instruments have different nuances that affect the pitch particularly around the note onsets and offsets. In the case of the horn, the inherent response time of the lip muscles can affect the pitch between notes and thus could warrant the manual editing of event onset and offset times to compensate for these physical limitations.

SUMMARY AND CONCLUSION

By applying a small set of the analytic tools available in NoteView we were able to examine the accuracy of the pitching of a professional horn player who was asked to play a simple piece twice without notice. The player was able to perform the two versions to ±7 cent accuracy of each other (pair by pair analysis). Both mean difference in pitch across versions and within-pitch variation were typically under 10 cents, with a mean of around 7 cents for both (paired comparison of median, and within event variation). We note that these data are in response to analysis of a simple piece played by a professional player, but are consistent with the literature we cited that investigated performance accuracy of other instruments including the human voice.

NoteView provides several statistics and allows visualisation of data in various, user-controlled ways [with examples shown in Figs 3-6]. Further, it calculates within-note pitch accuracy using standard deviation of pitch, and stability change statistics. It is able to deal with microtonally differentiated tunings such as equal temperament and just intonation.

The software described is a music performance analysis tool that provides information about individual signals as well as comparisons between signals using existing algorithms, with a strong focus on visual display of features concerning pitch. The system also provides timing information which can be used to investigate temporal aspects of a performance, such as tempo and articulation (ratio of note duration to inter-onset interval). Further, future versions of NoteView may apply different pitch extraction algorithms as required. The present algorithm does not do inherently well in identifying offsets and onsets when notes are played legato or slurred. Our approach mitigates the problem in two ways: (1) a score can be used as one of the signal inputs to help the algorithm identify the likely location of a new event in the second signal input that was otherwise played in a connected manner with its preceding event, and (2) post analysis editing allows the user to manually adjust any notes that were incorrectly identified due to slurring or for any other reason. Coupled with a variety of visualisation options, NoteView provides flexible ways of accurately analysing performances to facilitate the investigation of music performance. The description in this paper refers to the Beta version 0.5, which is available for download from UNSW Empirical Musicology web site http://empa.arts.unsw.edu.au/research-and-creative-practice/research-projects/empirical-musicology/.

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REFERENCES

[1] P. Loizou,, *COLEA: A MATLAB software tool for speech analysis* http://liceu.uab.es/~ joaquim/phonetics/fon_anal_acus/herram_anal_acus.html, Dept. Electrical Engineering, U. Texas at Dallas: Richardson, TX (1998-1999)

[2] A. Marsden, A. Mackenzie, A. Lindsay, H. Nock, J. Coleman, G. Kochanski, “Tools for Searching, Annotation and Analysis of Speech, Music, Film and Video A Survey”. *Literary and Linguistic Computing*, **22**, 469-488 (2007)

[3] J. Wolfe, “Speech and Music: Acoustics, Signals and the Relation between them’”, Proc. Inaugural International Conference on Music Communication Science (ICoMCS), 176-179 (2007)

[4] J. Wolfe, “Speech and music, acoustics and coding, and what music might be ‘for’”, Proc. 7th International Conference on Music Perception and Cognition (ICMPC) http://www.phys.unsw.edu.au/%7Ejw/ICMPC.pdf (2002)

[5] D. Cabrera, S. Ferguson, F. Rizwi, E. Schubert, “PsySound3: a program for the analysis of sound recordings”, Proc. Acoustics 2008 – Joint conference of Acoustical Society of America and the European Acoustics Association (2008)

[6] S. Dixon, G. Widmer, “Match: A music alignment tool chest”, Proc. 6th International Conference on Music Information Retrieval (ISMIR 2005), 492–497 (2005)

[7] A. Camacho,, J.G. Harris, “A sawtooth waveform inspired pitch estimator for speech and music’” *J. Acoust. Soc. America*, **124**, 1638-1652 (2008)

[8] H. Fastl, G. Stoll, “Scaling of pitch strength”, *Hearing Research*, **1**, 293-301 (1979)

[9] J. W. Hall III, D.R. Soderquist,, “Encoding and pitch strength of complex tones”, *J. Acoust. Soc. America*, **58**, 1257-1261 (1975)

[10] Ellis, D. *Dynamic Time Warp (DTW) in Matlab Web resource.* http://labrosa.ee.columbia.edu/matlab/dtw/ (2003)

[11] A. Botros, J. Smith, J. Wolfe, “The Virtual Boehm Flute-A Web Service that Predicts Multiphonics, Microtones and Alternative Fingerings”, *Acoustics Australia*, **30**, 61-66 (2002).

[12] Pierce, J.R. *The science of musical sound*. Freeman, New York (1992)

[13] J. E. Sundberg, E. Prame,. J. Iwarsson, “Replicability and accuracy of pitch patterns in professional singers”, in *Vocal fold psychology, controlling complexity and chaos* ed. P.J. Davis and N.H. Fletcher, Singular Publishing, San Diego (1996) pp. 291-306

WHAT IS OFFENSIVE NOISE? A CASE STUDY IN NSW

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INTRODUCTION

A recent decision in the NSW Land & Environment Court relating to noise emitted from an outside play area of a private school is used as a case study to demonstrate the distinction between “offensive noise” in enforcement actions and “environmental impact” in planning matters. On the surface, the two terms appear to be a manifestation of the same effect, however, there is a difference.

In NSW, the three principal legislation documents pertaining to the planning and enforcement of noise pollution are:

1. Environmental Planning & Assessment Act 1979 (EP&A Act) [1],
2. Protection of the Environment Operations Act 1997 (POEO Act) [2] and
3. Protection of the Environment Operations (Noise Control) Regulation 2008 (POEO Regulation) [3].

The EP&A Act is relevantly about planning matters and ensuring that “environmental impact” associated with new developments or the intensification of existing developments are properly considered and are reasonable before granting development consent to development.

On the other hand, the POEO Act and its Regulation are mostly concerned about enforcement: that is, preventing or putting a stop to the emission of “offensive noise”.

In planning, the assessment of “environmental impact” invariably relies upon the use of acceptability criteria which either may be explicitly defined in a Development Control Plan (DCP) or, derived from first principles using the guidelines promulgated in the Industrial Noise Policy (INP) [4] or the Noise Guide for Local Government (NGLG) [5] both of which are produced by the NSW Department of Environment Climate Change and Water (DECCW, formerly the EPA). In general terms, compliance with the acceptability criteria would infer that “environmental impact” or relevantly “noise impact” is acceptable.

In enforcement, “offensive noise” must be proved for there to be a prosecution. The term “offensive noise” is defined in the POEO Act (to be further discussed in detail below).

It has been common practice in the acoustics profession to use the term “offensive noise” when dealing with planning matters and in deriving noise criteria from first principles as if the term were defined in the EP&A Act. In fact, the distinction between “environmental impact” and “offensive noise” has been blurred to such an extent that they are thought of in practice as one and the same thing. That is, having established an acceptability criterion then exceedence of that criterion is an unacceptable “environmental impact” and therefore an “offensive noise”.

However, a recent judgement in the NSW Land and Environment Court concerning a resident and a Sydney private school [6] crystallised the difference between planning and enforcement regimes and the use of noise criteria in each.

This paper explores the issues involved in noise impact assessment in planning and enforcement and highlights the different approaches which should be taken (according to the judgment). Note that the judgment relevantly pertains only to the NSW jurisdiction and does not translate to other Australian States where the legislation differs and so does case law.

PLANNING LEGISLATION AND MATTERS FOR CONSIDERATION

When considering whether or not to approve a development, the consent authority (usually the local council) must take into account the requirements of Section 79C “Evaluation” of the EP&A Act, commonly referred to as the “Heads of Consideration”.

The Heads of Consideration states that a consent authority is to take into consideration *“the likely impacts of that development, including environmental impacts on the natural and built environments, and social and economic impacts in the locality, the suitability of the site for the development, any submissions made in accordance with this Act or the regulations, the public interest.”*

There are no prescriptive acoustic standards to be met in the EP&A Act and there is no requirement in respect of “offensive noise”.

In NSW, there is no consistency in codes and standards between regulatory authorities. Whilst it would appear desirable to have common policies across the State (as they do in Victoria) it is up to each regulatory authority to determine what standards it wants to impose on development if any at all.

When deciding whether to approve a development and what conditions to impose, the consent authority, must take into account all the relevant matters it is required to in the Heads of Consideration, the need for the development, the social consequences, noise impacts, its own policies, the views of those persons who object to or support the development and anything else that it reasonably considers relevant.

Upon granting consent, the resulting planning decision is one that should have certainty for everyone, both developers and objectors and result in finality for all parties. Otherwise developers will not be able to rely on the decision and would have no certainty in investing their capital and objectors will not be able to rely on the decision to put a stop to their plight so that they can get on with their life.

In order for this to occur it is not appropriate, when contemplating enforcement action, to commence the process again from the start and to ignore the regulatory authority’s contemplation described above.

In other words, when contemplating an enforcement action, one should not treat the matter as if it were a development application *de novo*. The starting point should be to consider objectively what the consent authority had in mind when approving the development and to interpret the conditions of consent pertaining to that development in that light. This then leads to contemplation of how "offensive noise" should be interpreted and what noise criteria are appropriate

ENFORCEMENT LEGISLATION

Once a planning consent is given, the consent conditions imposed must be complied with (Section 121B of the EP&A Act prosecuted in Class 4 Proceedings in the Land & Environment Court). Therefore, if specific noise conditions are imposed, it is a matter for the development to comply with those noise conditions.

However, not all noise conditions are written in an unambiguous way, such as the following noise condition pertaining to the development application for the school, the subject of this paper:

8 Any noise emanating from the use at any time shall not have any detrimental effect on the adjoining residential amenity.

How should this condition be interpreted in circumstances where development consent was given for up to 640 students in a school which incorporates outside play areas in locations approved by Council? Does the condition mean that the noise should not be detrimental in an absolute sense (that is the absolute level of noise or exceedance of background) or does it mean any noise in excess of what would normally be expected from the school operating in accordance with its consent conditions (that is, noise above what one would expect from 640 students in this location)?

Neighbouring residents who claim they are adversely affected by noise have a number of choices for advocating their grievances, including a Class 4 prosecution in the Land & Environment Court alleging a breach of a condition of consent. In normal circumstances, this may be the simplest action an aggrieved person could take against a development.

Another action which could be taken is a prosecution under the POEO Act. Under this Act, authorised personnel (e.g. the police and local council officers) and the public (by application to a local court) may direct persons causing "offensive noise" to be emitted from premises to abate that noise. There are various forms of directions which may be given to such persons including a Noise Control Notice, a Noise Abatement Order or a Noise Abatement Direction. A Noise Control Notice and a Noise Abatement Direction may only be issued by authorised personnel. A Noise Abatement Order may be brought by a member of the public and it is an offence to act contrary to an order issued by the Local Court.

What difference is there in seeking a Noise Abatement Order instead of prosecuting a breach of a condition of consent? Put simply, in the former, the applicant must prove that "offensive noise" is emitted from the premises. In the latter, the applicant must prove there is a breach of a condition of consent (such as condition 8 above).

The reader may think that the two are synonymous, however, there is a difference. In the case study described in this paper, the resident decided upon a path of proving "offensive noise".

In the definitions of the POEO Act,
"offensive noise" means noise:

(a) *that, by reason of its level, nature, character or quality, or the time at which it is made, or any other circumstances:*
(i) *is harmful to (or is likely to be harmful to) a person who is outside the premises from which it is emitted, or*
(ii) *interferes unreasonably with (or is likely to interfere unreasonably with) the comfort or repose of a person who is outside the premises from which it is emitted, or*
(b) *that is of a level, nature, character or quality prescribed by the regulations or that is made at a time, or in other circumstances, prescribed by the regulations.*

In respect of the part of the definition comprising (a)(ii), offensive noise does not mean noise that is unpleasant, irritating, annoying, abhorrent, abusive, detestable, disagreeable or any other like words. It does not even mean noise that is "offensive" in the way that word is used in every day life. That meaning was expunged from the definition of "offensive noise" when the POEO Act replaced the former Noise Control Act 1975.

The Noise Control Act 1975 then defined "offensive noise" as:
"noise that by reason of its level, nature, character or quality, or the time which it is made, or any other circumstances, is likely to be harmful ***or offensive****, or to interfere unreasonably with comfort or repose".* [emphasis added]

The modification of that definition probably occurred because it was seen to include noise which is offensive solely by virtue of its content, such as a noise which is indecent or racist [7].

The salient requirement in the POEO Act is that the noise "interferes unreasonably". In that sense, the term "offensive noise" could equally well be thought of as "unreasonable noise" or even "noise which is an offence under this Act". In fact, it may be better to think of "offensive noise" in that way to avoid confusing it with noise that is (by its ordinary meaning) offensive.

When interpreted in that sense, noise that "interferes unreasonably" must take into account whether or not the activity producing the noise was consented to by council under the Heads of Consideration for not to do so would subvert Council's decision. As stated in the NGLG (Section 2.2.1 Intrusive Noise), *"in the absence of a council policy, intrusive noise would not automatically be considered offensive noise"*. Therefore, when considering the "level" of noise, one should not only take into account the intrusiveness of the noise in an absolute sense (i.e. its level above background) but also the level of the noise in comparison with the level that council has assessed as being reasonable in the circumstances when approving the activity or development.

Put simply, if council approved the development and the associated emission of noise subject to certain conditions and the user of the property complied with those conditions, then it should follow that the use would not "interfere unreasonably" and therefore the noise could not be categorised as "offensive noise" notwithstanding that it may be at a level above acceptability standards used to assess planning developments.

THE COURT DECISION

We now turn to the application of the principles discussed above to a recent decision of the Land & Environment Court referred to in the introduction. That case concerned a dispute between a resident and an adjacent private primary school in the Sydney suburb of Strathfield. Private schools (unlike public schools) are subject to the POEO Act (see Figure 1).

The resident complained of noise from the school, particularly the sound of children playing outdoors on the school's lawn in the morning (before school), recess, lunch time and after school and children's outside activities at other times of the day. In addition, noise from the use of yard maintenance equipment such as leaf blowers, gurneys and edge trimmers.

The resident applied to the Burwood local court for a Noise Abatement Order (pursuant to section 268 of the POEO Act 1997) seeking an order for the school to abate the "offensive noise".

The competing arguments are this: The resident says noise emitted from the school is loud, is intrusive, causes him distress and therefore is offensive. The school says that notwithstanding the loudness of the noise and its exceeding of planning criteria which may apply today, the school is nevertheless compliant with its consent conditions issued by Strathfield Council and therefore it should be permitted to operate in accordance with those conditions. Therefore, the central question is whether or not in these circumstances, noise from the school is "offensive noise".

In the local court hearing, the magistrate took a fairly simple view that where there is a conflict, the parties should negotiate and settle the matter otherwise, he said, he would issue directions that may not be palatable to either party. He concluded based on the evidence that noise from the school should comply with a planning criterion of "background plus 5" and issued orders requiring the construction of a noise barrier 5m in height located close to the common boundary of the properties and the double glazing and mechanical ventilation of the affected residence.

The school appealed that decision in the Land and Environment Court which re-heard the matter in its entirety (a *de novo* hearing). That judgement set aside the decision of the local court and concluded that the school does not emit "offensive noise". The judgement concluded as follows:

"My conclusion on the appropriate assessment is that the advice of [the School's Consultant] *is to be preferred. It needs to be borne in mind that I cannot assess the school activities in a planning sense. The assessment criteria are one aid to determining what the level of acceptable noise should be, as part of the information relevant to determining whether the noise emitted is offensive. The criteria are not to be applied as a standard the school has to meet where there is no evidence that it is not otherwise complying with its conditions of development consent."*

The school argues that it operates within its consent conditions and as anticipated by Council when it made its decision. The Council planner's report stated as follows:

"It is considered that from observations the noise from the play time activities, are not considered excessive and are considered reasonable and accepted by the general community."

In other words, Council's position is that it approved the development in its current form, the development appears to be operating as anticipated by Council and therefore it complies with its consent conditions.

The judgement states as follows:

"All noise that emanates from the normal activities at a school is not offensive. The focus of the case should be that element of the noise above normal school operations which is identified as offensive but no such category of noise has been clearly identified by the Respondent despite attempts to define the offensive noise by him. There is no particular sound above the usual ambient noise expected of a school environment which is particularly identified as giving rise to offensive noise apart from noise resulting from the children's use of the Jobling lawn, and the use of blowers and gurneys. In the absence of such specificity in the Respondent's case, I do not consider there is evidence to enable me to consider any other aspect of the activity at the school which may give rise to noise beyond these two areas."

Therefore, in respect of condition 8 referred to earlier and in paraphrasing the judgement cited above, that condition should be interpreted as *"any noise emanating from the use at any time above normal school operations shall not have any detrimental effect on the adjoining residential amenity"*.

CONCLUSION

The decision in this case study makes a distinction between the assessment of noise impacts in planning and enforcement. "Offensive noise" is an enforcement term defined in the POEO Act and an assessment of unreasonable intrusion under that Act should take into consideration what was permitted by the responsible authority and not in isolation of it. Environmental impact is a planning term referred to but not defined in the EP&A Act and involves the assessment of noise and the consequences of noise emission. Noise criteria are pertinent to both enforcement and planning but, in deriving enforcement criteria where specific criteria are not stipulated, those criteria should not be derived *de novo* from a planning perspective.

In conclusion, it is important to distinguish between the two processes of planning and enforcement. It is common practice for acoustic engineers to refer to the definition of "offensive noise" in the POEO Act in environmental noise assessments and to treat enforcement from a planning perspective. One must differentiate the two processes, firstly by not referring to the "offensive noise" definition in the POEO Act is if it were a planning term and secondly, by not treating an enforcement prosecution as if it were a planning process.

REFERENCES

[1] NSW Environmental Planning & Assessment Act 1979

[2] NSW Protection of the Environment Operations Act 1997

[3] NSW Protection of the Environment Operations (Noise Control) Regulation 2008

[4] NSW EPA Industrial Noise Policy January 2000

[5] NSW EPA Noise Guide for Local Government August 2009

[6] Meriden School v Pedavoli [2009] NSWLEC 183 (22 October 2009) http://www.austlii.edu.au/au/cases/nsw/NSWLEC/2009/183.html

[7] The New South Wales Noise Control Act 1975. Julian Disney. The University of New South Wales Occasional Papers 1-1976

News

Heggies Merge with SLR

Heggies Pty Limited has announced its merger with UK-based international environmental consultancy SLR Management Limited.

Heggies Pty Limited is an employee-owned environmental consulting company, which was established in 1978 by Richard Heggie, who remains the Managing Director. From its roots in Sydney, this acoustics, vibration and air quality consultancy has expanded to over 130 staff members with nine offices across the Australia and Singapore. SLR presently has over 750 employees working from more than 45 offices in the UK, Canada, Ireland and USA, with a strong presence in waste management services, the oil and gas industry and in renewable energy from wind, waste, hydropower and biomass.

New Products

Bruel & Kjaer announce Noise Sentinel

The Noise Sentinel is a new internet-based subscription service for urban and industrial noise monitoring. Noise Sentinel has applications in any industry that makes significant noise in the community and is required to report compliance with noise regulations. Examples include mines, ports, industrial premises, power stations, wind farms and large construction projects.

Noise Sentinel is web-based and shows the current real time noise situation, alerts the operator if any threshold is exceeded and delivers regular noise compliance reports that are tailored to individual legislative requirements. At any time, any particular noise situation can be investigated in detail though ad-hoc analysis. “Noise Monitoring is complex when it is done properly. It requires investment in quality instrumentation, non-core skills and accredited personnel. When your operating licence is under threat you really need to ensure you do it properly otherwise you can increase business risk within your organisation. Noise Sentinel provides a compelling solution”, said Doug Manvell, Noise Sentinel Product Manager.

Further information from www.bksv.com/NoiseSentinel or via www.bksv.com.au Tel.02 9889 8888

3M launches noise monitor

The new 3M™ Noise Indicator N1-100 is designed to alert users to potentially dangerous noise levels and to help identify areas where hearing protection may need to be worn. Users simply clip the Noise Indicator to a shirt or jacket and its LED can deliver a clear indication when noise levels exceed a potentially hazardous threshold.

A red flashing LED indicates noise levels are equal to or above 85 dBA – hearing protection may be required.

A green flashing LED indicates noise levels are below 85 dBA – hearing protection may not be necessary.

The 3M Noise Indicator’s small size and lightweight design make it ideal for workers in a variety of industries. It has a rechargeable battery that operates for up to 200 hours between charges. The NI-100 can be used as an effective training tool within a Noise Management Program (consult AS/NZS1259 series) and help to ensure workers know when and where to wear hearing protection. It can also be used as a mapping tool to determine where noise studies are necessary.

For more information about this product and other 3M solutions for worker safety, visit www.3m.com/au/ohs or phone 136 136.

Peace introduces the Sound Barrier

The most effective sound absorbent walls are typically built of materials that are thick and heavy. Peace has developed a lightweight solution to noise control that they call ‘Sound Barriers’ and that can be flat packed and installed anywhere, quickly and easily in a few hours.

Concrete road barriers form the base and provide not only strength, but also contribute to sound absorption performance. Peace Sound Barrier acoustic panels then slide on top of the concrete barriers to create an acoustic wall up to 3 metres high acting as an absorption and transmission barrier. They provide effective noise reduction for an entire site in applications where large acoustic enclosures are not feasible.

The panels can be installed anywhere in any configuration, so they’re ideal for when construction or public works are occurring close to houses. Peace Sound Barriers can be painted and even act as a security wall to help keep out intruders.

More information can be found at www.peaceengineering.com.au or you can sound out the team at Peace Engineering at (02) 4647 4733 for the full story.

Divisional News

Queensland Division Acoustics Awards 2009

The Queensland Division awards program was established to support education and research in acoustics in Queensland schools and universities. Awards are granted on an annual basis in two divisions, a schools division (Division I) and a tertiary division (Division II).

The Division 1 Bursary is awarded as part of the Queensland Science Contest. The 2009 Bursary of $450 was split between two year 11 students from Trinity Anglican School, White Rock, Cairns. First prize ($300.00) went to Cary Wang: for his project entitled “Groper Triangulation”; second prize ($150) went to Chris Reymond for his project “Underwater Ears”. Both projects involved triangulation of underwater sound sources using a simple two channel hydrophone rig (put together from its basic components – duct tape, peanut oil and silastic in liberal quantities). External supervisor was Geoff Macpherson - well known to AAS members for his work with DPI & Fisheries.

Division II comprises two categories. The Category I bursaries are directed towards final year undergraduate or first year postgraduate research projects in acoustics or vibration. Two $1500 bursaries are available, the Acoustic Bursary and the RJ Hooker Bursary, with the latter intended to encourage projects conducted in the context of a professional placement. The Category II bursaries (of $150) are awarded to the most outstanding student in an undergraduate acoustics course at a Queensland university. Four courses are supported: MECH3250 Engineering Acoustics and AUDL 7800 Acoustics and Psychoacoustics in Audiology at University of Queensland, MMB413 Industrial Noise and Vibrations at Queensland University of Technology and ME3511 Dynamics and Acoustics at James Cook University.

The 2009 Acoustic Bursary was split between Dong Yang and Nicholas Powers (James Cook University). An amount of $800.00 was voted for Dong Yang’s project, Study on Fluid/Acoustic-Structure Interaction for Noise, Vibration and Failure Analysis in Oil Pipelines, while $700.00 was voted to Nicholas Power’s project, Optimal Design of Noise Barriers with Smart Material Structures against Environmental Noise Pollution and Ecosystems. The awards were presented at JCU on 26 August through the good offices of Associate Professor Chengwang Lei of the School of Engineering and Physical Sciences.

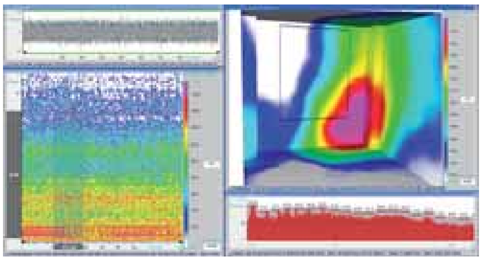
The 2009 Category II bursaries were awarded to Nikilesh Kumar (ME3511 Dynamics and Acoustics), Calem Walsh (MECH3250 Engineering Acoustics) and Annika Batros

AC easy

acoustic camera
simply perfect – perfectly simple

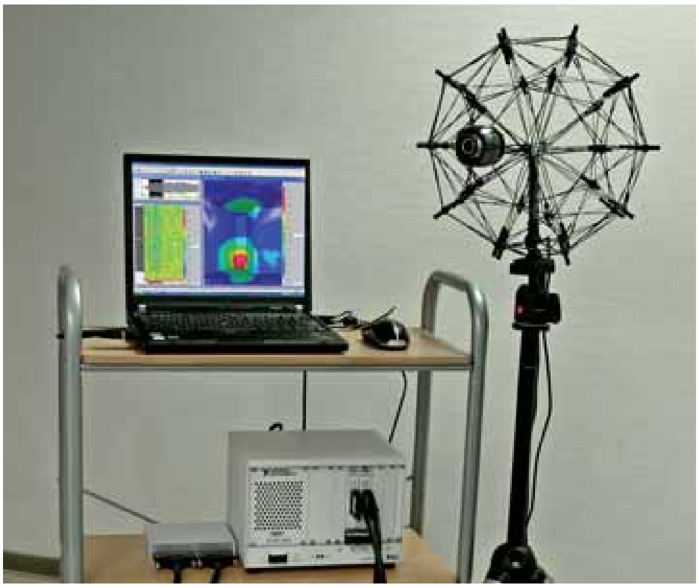
acoustic camera

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



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

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



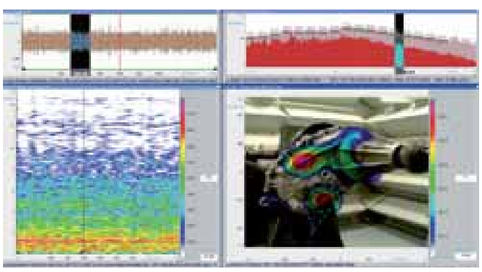
AC easy key features:

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

Specifications – system with notebook

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

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



Specifications – system with desktop PC

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

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(AUDL 7800 Acoustics and Psychoacoustics in Audiology). Nikilesh Kumar's award was presented at JCU in August. Calem Walsh and Annika Batros were presented with their bursaries at the Queensland AGM in September. At the March EPP[Noise] technical meeting, the 2008 AUDL7800 Bursary was presented to Michelle Nicolls. The RJ Hooker Bursary and the Category II bursary for MMB413 Industrial Noise and Vibrations were not awarded in 2009.

A travel bursary of $750 was presented to Ross Batten towards his attendance at the 2009 Australian Acoustical Society Conference for purposes of presentation of a paper entitled "Investigation into the effects of asymmetric train speed distribution on rail corrugation growth in cornering". Ross has been invited to make a presentation of his results at a Division technical meeting during 2010. Additional student travel bursaries are under consideration for 2010 and on an ongoing basis.

Details of submissions for the 2010 Acoustic Bursary and the RJ Hooker Bursary are given at http://www.acoustics.asn.au/general/awards-index.shtml

Ian Hillock

Museum takes H.V. Taylor equipment

H. Vivian Taylor was the founding President of the Australian Acoustical Society in 1971. He had been described as "A careful, obedient and industrious lad" in a reference he was given on 27 June, 1916, after about 4 years as a carpentry and joinery apprentice with Clements Langford, a builder in Richmond, Melbourne. He later developed into one of Australia's leading Architects, with a special interest in acoustics.

He was commemorated in the 1999 annual conference in Victoria, "Acoustics Today", and some of his original acoustic measurement equipment, framed awards and other memorabilia were put on display at the venue. Afterwards the Society was presented with this material, which has since been stored in the Society archives. When it was decided by Council in late 2008 to close the archives, a new home for the equipment was sought. The former General Secretary of the Society, David Watkins, initiated discussions with several places, which has resulted in the Melbourne Museum agreeing to permanently accept both the equipment and the associated items. As shown in the photograph, they were formally handed over to David Dermant, Senior Curator, Information & Communication, at the Museum on 11 November, 2009. Acting on behalf of the Society were Charles Don, most recent Society Archivist, and Louis Fouvy.

The equipment presented to the museum included the following three General Radio Co. instruments;

(a) Vibration Analyzer type 762,

(b) Sound Analyzer type 760,

(c) Octave Band Analyzer type 1550.

In addition there was also a Byer Model 66 Tape Recorder, as well as four Victor 78 rpm test records, containing tracks of single frequency sounds, probably used by HVT for reverberation testing. Amongst the small collection of books and papers was a well used copy of the third edition [1956] of the General Radio "Handbook of Noise Measurement and Measurement of Vibration", which our more senior members will recognise as a most practical guide to making acoustical measurements.

In their day, the above were all high-precision instruments. The Vibration and Sound Analyzers were tunable filters having a constant percentage bandwidth of 2% [in musical terms, a fine bandwidth of about 1/5 tone]. Their frequency ranges were, respectively, 2.5 to 750 Hz and 25 to 7500 Hz, each in five ranges, beginning with 2.5-7.5 or 25-75 Hz. In addition, the Type 762 Vibration Analyzer had a broad selectivity bandwidth of constant percentage of 30% for measuring vibration signals of slightly varying frequency. The Type 1550 Octave Band Analyzer covered the audible frequency spectrum of 20 Hz to 10 kHz in 8 bands. These comprised a low pass filter to 75 Hz, six octave filters using the earlier series of octave bands from 75-150 Hz to 2400-4800 Hz, instead of the current standard "preferred" series, and a high pass filter from 4800 Hz. With more intense sounds, this analyzer could be used directly with a microphone. The Byer tape recorder [of Melbourne manufacture] was a high-quality precision piece of equipment, with a tape speed up to 38 cm/s [15 in/s], and a high frequency response extending probably to at least 10 kHz.

Unfortunately, this equipment lacked the microphone that HVT must have used as the input. At this stage a recent Fellow of the Society, Louis Fouvy, an electrical engineer formerly employed by the Melbourne & Metropolitan Tramways Board in their Testing Laboratories became involved with the discussions about the equipment. By chance, he had a GR Type 759 Sound Level Meter of the same vintage as the HVT equipment and of the type almost certainly used by him. There is an interesting story associated with this instrument. Back in 1960, a Society member, Gerald Riley, spotted this instrument in a disposal store, after it had been made redundant by the Tramways Board in favour of a more modern instrument. After purchasing it, Gerald eventually handed it over to his friend Louis, realising that Louis would have been involved with the instrument at the Tramways Laboratory. They originally intended to place the instrument in the AAS archive, however, they have now graciously donated it to the Melbourne Museum, to effectively complete the set of instruments.

Before it was superseded by the Type 1551 in the late 1940s, the GR Type 759 Sound Level Meter had been the top instrument of its type for about a decade, with a sound level range of from 20 to at least 120 dB, using the then standard A, B or C frequency weightings. Using valve amplifiers, the first stage was somewhat sensitive to being microphonic. It could also be used with the Types 759-P35 and 759-P36 Vibration pickup and Control box [included among the HVT instruments], and was equipped with an AC signal output jack for connexion to an analyzer or graphic level recorder for more detailed analysis of a sound or vibration signal.

One aspect which helped Melbourne Museum decide to become caretaker of the HVT collection was that the Acoustical Society could provide information about his activities, which complemented the framed certificates, a diary, equipment handbooks and other memorabilia which accompanied the equipment. For example, after becoming registered as an Architect in 1923, he became involved professionally in acoustics from 1928, designing both the architecture and acoustics of around 55 churches and halls, before starting to specialise in the 1930's in picture theatres. By the 1940's, he was involved with over 430 theatres and halls: sometimes advising on acoustic treatment and on others handling the complete design. Later he acted as consultant to the ABC in the design of their sound studios in all states. Although dominantly a Melbourne based Architect, he also worked in many parts of Australia and on 31st July, 1962, he even set up an office at 42 Bridge St., Sydney, under the name "Acoustic Advisory & Consulting Services of H.Vivian Taylor". He was elected the first Fellow of the AAS in September, 1972.

As part of the background information of HVT, the Melbourne Museum has asked if the Society could provide as many stories about his exploits as possible. One anecdote – which indicates how even a top consultant, by falling victim to a technology failure, can supposedly "get it wrong" – relates to the design of the new sound reinforcement system for the 1955 replacement Wilson Hall at the University of Melbourne. As part of the cutting edge technology at the time, it was decided to install delayed signals to speakers along the length of the hall. HVT was not an electrical engineer and had nothing to do with installation of the electrical equipment, however, as architect he copped the flack when the initial results were a disaster.

Have you any other interesting stories about the man? Please send them to the Editor, so they can be passed on to the Museum.

Louis Fouvy and Charles Don

Meeting Reports

VICTORIA DIVISION

Melbourne Recital Centre

The meeting on 8 September 2009 was the technical meeting that accompanied the Victoria Division AGM. It involved 30 participants visiting the recently completed Melbourne Recital Centre at 31 Sturt St, Southbank. This centre comprises;

[a] the Elisabeth Murdoch Hall, a 1000 seat auditorium with stage to accommodate chamber music groups of up to 45 musicians,

[b] a 500-seat theatre for the Melbourne Theatre Company,

[c] a small 150-seat auditorium for contemporary-type performances, together with

[d] broadcasting, dressing, practice and recording rooms.

All these auditoriums and rooms were visited as part of the site tour.

This meeting also provided the opportunity for the team led by Andrew Nicol, and supported by Kym Burgemeister, Sylvia Jones and Peter Bickle [of Arup and the associated building and architectural firms] to describe acoustical and technical features of the Murdoch auditorium and of the building in general.

The need for a music recital centre in Melbourne to complement the Hamer Hall and State Theatre of its adjacent Arts Centre was recognized as early as 1982, when lobbying began. Design work was not begun until 2001, building work started in 2006 and was completed in late 2008. The opening concert in the Murdoch Hall was held on the 8th of February 2009. Arup Acoustics was responsible for the acoustical design, Ashton, Raggatt, McDougall [ARM] for the architectural design, and Bovis Lend Lease for the construction work. This design and construction work was carried under a prime requirement that the acoustical quality of its auditoriums would be of uncompromised excellence for musical recitals, chamber music and choral music.

Kym Burgemeister described how, because the site was surrounded on three sides by streets and two of these streets carried trams, the recital centre building interior had to be isolated from both exterior noise and vibration. Vibration from passing trams was found to occur mostly between 60 and100 Hz [as distinct from the lower frequency vibrations from rubber-tired vehicles with reciprocating engines]. To achieve a sufficient degree of vibration isolation, the inner shell of the building, was designed as part of a box-in-box arrangement with air spaces between inner and outer shells and was mounted on 38 bearings made up of units of steel helical springs. All springs were of known measured stiffness to maximize uniformity of load distribution. Subsequent measurement showed the vibration isolation of these bearings between 25 and 100 Hz to be similar to that as designed.

Sylvia Jones, in describing details of the building construction, emphasized that constant co-operation between architectural, acoustical and building teams was maximized so that all workers on the site understood the importance of all the anti-noise and anti-vibration treatments used both in the structure itself, as well as in the ventilation and other installed services. The ventilation system selected operated at low air velocities and met an overall building services noise criterion of PNC 15. For general insulation from exterior noise, the building shell walls were of 256 mm thick concrete. The 75 mm thick timber panels [of ply rather than solid timber] were tested for their acoustical absorption at low frequencies.

The Centre was designed as a completely above-ground building because of the “Coode Island silt” below ground. Peter Bickle described various aspects of the building and auditorium architectural designs, and the close co-operation that was maintained between acousticians and architects from their earliest stages. These aspects included everything from the building’s exterior shape to the shape and decoration of the auditoriums. Because all these aspects needed to be distinctive, many shapes of existing buildings were studied and compared in the process of achieving the final design. This was accomplished with the aids of both computer 3-D design and a 1:25 scale model. This scale model was particularly useful in simulating the auditorium’s sound diffusion patterns with considerable accuracy. These patterns are determined primarily by the angles of the ceiling and wall panels with respect to the auditorium’s principal axes.

The building’s design distinctiveness began with its exterior shape, windows and wall decoration. Inside, the entrance foyer was located below the main auditorium whose preferred shape was a “shoe-box”. Its chosen length and width of 37 and 20 m allowed for 1000 seats, with 700 in the stalls and 300 in rear and side balconies. A volume per seat of 9 m3 was chosen to allow a reverberation time of 1.6 to 1.8 s. The seats were designed to minimize the difference in reverberation times with the hall empty or occupied.

The shape and height of the walls and ceiling around the stage were designed to provide the musicians with sufficient early sound to enable them to hear themselves and each other without resort to an orchestral reflector [not desired by the client]. Their design was checked by computer simulations to achieve an ST1 early sound support parameter of between –10 and –13 dB.

In answer to a question about designing auditoriums in the light of previous developments, Andrew Nicol replied that good auditoriums are replicated, but bad ones don’t get continued use. After a tour of the whole building, the chairman thanked Andrew Nicol and his colleagues for their most interesting talk and tour, a vote confirmed by applause.

Architectural Acoustics

At a meeting held in the SKM theatrette, Armadale, on 10 November, Dr Ken Roy, a former Acoustics Consultant and now the Senior Principal Research Scientist in US firm Armstrong, described the LEED system of rating the architectural acoustics of “green” buildings. Charles Don acted as chairman with 22 present.

Ken Roy’s daily work included persuading architects of the need for the acoustical design of buildings – for offices, homes, schools and hospitals – of suitable layouts for partitioned offices and other rooms, and advising them of materials and equipment suitable as acoustic absorbers, barriers, and cover to mask background sound and ensure speech privacy. As examples he described several typical situations.

Of fundamental importance was that architects understood the acoustical ratings of the various architectural components, of partitions and ceilings.

In large open office areas, architects tended to arrange the partitions as though “teaming” work was primary, whereas around 60% of time was taken up on individual thinking, and 15% on conferencing, for which 1.5 m high partitions were quite inadequate. It was found that partitions needed to be higher [even to the ceiling], ceiling [and upper walls] needed absorptive treatment and, if also necessary, broad band masking sound at 43 dB[A] was recommended. Doors and doorways might also need remedial treatment. Recessed ceiling light fittings were found to reduce ceiling area by around 15%, whereas pendant fittings caused no reduction. There was need of a measurable privacy factor.

Acoustical treatment of school classrooms through reducing their reverberation time was found to be especially beneficial. Before treatment, noise from transient sounds such as paper shuffling would reduce speech intelligibility. After treatment, it was found that classroom acoustics were significantly improved, and class behaviour was much better.

As a result of US experience with many buildings not being acoustically satisfactory, the US building council approached ASHRAE to develop Performance Measurement Protocols, and acoustics was included as a building IEQ factor. This resulted in;

• sound quality criteria based on NC [or similar] curves,

• acceptable room reverberation times,

• room designs that provided satisfactory reduction of exterior noise, and

• the use of surveys for measuring personal

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satisfaction in workplaces.

Interior acoustical and thermal comfort are difficult to measure. As yet, there are no officially adopted performance standards in acoustics.

At the end of the meeting, Charles Don thanked Ken Roy for his most interesting talk, a vote carried with applause.

Animal Behaviour and Noise

The Division End-of-year dinner was held at the Malvern Valley Golf and Reception Centre on the 1st of December. The guest speaker was Dr Robert Holmes, a veterinarian with a PhD and Fellowship in animal behaviour, who recounted some of his experiences with animal behaviour and noise. The meeting was chaired by Geoff Barnes, and attended by around 25 members and friends.

At the beginning, and during his talk, Dr Holmes used recorded sounds to illustrate his points. His first two recorded sounds were of arias from opera singers to illustrate the exciting effect of music on us humans. From a loud cat purr, largely low frequency sound, it is assumed the cat is contented. But cats also purr through to near the end of a terminal illness. When dogs obey the commands to 'sit' and 'walk', it is the manner, not the content, of the spoken words that control them.

Distress in animals can be recognized by the sounds they make, such as from a puppy in distress, sheep in a pen, or an excited and upset dog. Kennel doors with the upper part transparent and the lower part opaque are less likely to cause dogs to become upset and more likely to bark. Dogs scratching at and damaging doors to get inside are exhibiting separation distress. Dogs are phobic to electric saw and nail gun noise. Around 40% of dogs with noise phobias also exhibit separation distress. Around 8% of dogs show distress at thunder. During thunderstorms, when dogs suffering separation distress get inside and hide under a bed, it is the owner's bed with the owner's smell.

Elephants have been assumed from evidence in other environments to respond to ground-transmitted seismic activity, in that a tsunami and its preliminary vibratory wave causes them to head for higher ground. An audience comment that magpies, usually timid except during the nesting season, can be attracted by a quiet whistle, evoked the comment that their small brain size is not correlated with intelligence.

In conclusion Dr Holmes said that animals perceive the world differently from humans. Some communicate through ultra-sound or infra-sound. Concerning the occurrence of dogs going to a window just prior to their master's return, it was observed that dogs might do so often, as frequently as every 10 minutes ! Bird song is a signal to others to "get out of my territory". Is a tail wagging dog friendly, social or aggressive? Animals become accustomed to sound more readily than sight. Constant noise is accepted; sudden noises may be perceived as a threat.

After a brief discussion, Geoff Barnes thanked Dr Holmes for a most interesting and informative talk, and asked those present to confirm the thanks with applause.

Music rehearsal booths / Acoustic perception of space

The meeting on 28 January was held in conjunction with the deferred national AGM at Swinburne University of Technology, Glenferrie, Melbourne. There were 20 present. The speakers were Geoff Barnes of Acoustical Design, and Victoria Division chairman, and Jim Barbour, lecturer in radio, audio and multi-media in the Swinburne University Media and Communications Department.

Geoff Barnes described his Musolabs design, which are music rehearsal booths, built to a design which was developed from his audiometric test booths. Some Musolab booths were shipped to Singapore 15 years ago for use in radio broadcasting. Four years ago, to a modernized design, a suite of seven booths was installed at Swinburne University Media and Communications Department for use in student projects. Two years later, they were dismantled and re-located in a new building.

These booths, with fail-safe door handles, and a vibration-isolated floor, provide, with a single wall, an interior reverberation time reduced to 0,25 seconds, and an acoustical isolation of 35 to 40 dB[A], such that, in use, an original background noise level of 40 dB[A] rises to 62 dB[A], mostly low frequency, with a level of 85 dB[A] of music in an adjacent booth. Their ventilation is provided by silenced fans and lined ducts. However, the Swinburne booths, being without the special floor, are not completely proof against the impact noise of footsteps, etc.

Following Geoff Barnes' talk, Jim Barbour then, by means of a power-point display with sound, conducted those present through an exploration of the height dimension in acoustic space and audio reproduction, as a demonstration of how clearly and accurately we as humans perceive the height dimension in free and bounded sound fields.

His thesis was built around the auditory experience that in any vertically bounded acoustic space, the height of the ceiling above the floor indicates the size of the room we are in, this being aided by our ability to combine early reflections with the direct sound. This results from our experience of having learned something of how we associate what we hear with the spaces we occupy. However, our aural discernment of our position in a space is not as refined as our sense of musical pitch, particularly when, through a limited number of loudspeakers, we are listening to sound recorded in an acoustic space with its own reverberation, etc, characteristics.

The first sound examples were of the general sounds characteristic of an enclosed atrium, an open-air cafe, under a railway viaduct, in the Melbourne Recital Centre and in a cathedral. Attention was drawn also to the effect of the Sydney Opera House overhead 'flying saucers' that provide early reflections from above.

From tests inside a loudspeaker test rig with several stereo arrangements to measure perceived elevation for Inter-channel Amplitude Differences [IADs] from –15 to +15 dB for front and side elevation [with speakers at elevations of 0º and 60º], the localization in the frontal [vertical right-left] plane was closer and with somewhat smaller standard deviations than in the median [vertical front-back] plane. For overhead localization [with speakers at elevations of 60º and 120º], that in the median plane was noticeably less well defined and with significantly greater standard deviations than in the frontal plane.

That is, median plane localization was poor, there was significant localization blur, pinna [head turning] effects worked but not well, and one or more speakers in the median plane are of limited benefit, with one directly overhead of no use. By contrast, frontal plane localization was generally good, with better accuracy and decreased localization blur at greater heights, with the optimum being two speakers in the frontal plane at left and right elevations of 60º.

The factors influencing spatial localization include

- inter-aural differences in time, level and spectrum;
- distance effects;
- the ratio of direct to reverberant sound;
- Doppler shift ;
- pinna effects.

Typical elevated sounds were of aeroplanes and helicopters and their different characteristics, birds high-flying, low-flying or in trees, and overhead service pipes, air-conditioning outlets and wind.

For recording overhead sounds, soundfield and optimized cardioid array microphones are available, with the soundfield microphone having limitations such as good localization within its critical distance and good envelopment outside it. For their reproduction, several speaker arrays are available to create localization, envelopment and spaciousness effects. With elevated and surround speakers, post-production processing includes adding reverberation and early reflections [de-correlated left-right] from above, with later reflections and de-correlated reverberation from the rear. Current release formats of recorded sound include DVD-audio, SACD and Blu-Ray, with Blu-Ray having some advantages over SACD.

Jim Barbour summed up by saying that the auditory perception of elevated sounds relies on Inter-channel Amplitude Differences, and that overhead speakers enhance spaciousness and envelopment. Geoff Barnes, as chairman, then thanked Jim for his interesting and stimulating presentation, confirmed by the applause of all present.

ICA 2010

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

The first months of 2010 have been the beginning of an increased work load for those involved with ICA 2010 and associated meetings.

The number of submitted abstracts is currently over 970 and these are being processed for the development of the program and the production of the book of abstracts. The next rush will be at the time of the deadlines for the paper submission which is 21 May for those seeking peer review and 28 May for all the other submissions.

The response indicates that the congress will have a full and exciting technical program over the 5 days. All those planning to attend are encouraged to complete their registration early and in particular to note the deadline for Early Bird registration of 28 May.

There will also be a large exposition and only a few booths are still available. Companies and organizations can also bring their products and services to the attention of the delegates via the range of options for sponsorship. The prospectus is available from the webpage for ICA 2010 under the link "Exhibition and Sponsorship".

There will be 3 associated symposia held in the region after the main congress:

ISMA 2010 International Symposium on Music Acoustics 25-26 and 30-31 August in Sydney and Katoomba

ISSA 2010 International Symposium on Sustainability in Acoustics 29-31 August in Auckland, New Zealand

ISRA 2010 International Symposium on Room Acoustics 29-31 August in Melbourne

The web page is the main source of information on the ICA and the associated meetings **www.ica2010sydney.org.** As well as planning to attend this outstanding congress we ask that all members of the AAS assist with the promotion of the ICA and associated meetings among your national and international colleagues.

Marion Burgess, Chair ICA 2010

Noise and Industry

The second technical meeting for 2010 was held on 15 March at the EPA offices in Carlton. Its purpose was to promote discussion on the 'Noise from Industry in Regional Victoria' [NIRV] draft regulations and 'Industry Noise and Statutory Approvals' guidelines, focussed on aspects where NIRV differs from the N3/89 regulations, how acoustical advice is called up under the statutory approvals guideline, how the documents might be improved, and technical or policy questions. This discussion was chaired by Eliot Palmer [EPA].

Most of those attending this meeting were acoustical consultants who stated that they were there to learn more about the practical application of the new regulations, which were considered generally good. Several typical questions arose as to;

- the responsibilities of the EPA and municipal councils in administering these regulations;
- whether the new regulations recommend noise reducing measures in buildings to cope with, e.g., single noise events;
- whether the regulations give sufficient detail for adequately quantifying the acoustical effects of weather conditions such as temperature inversions;
- how are perennial questions settled such as "who got there first, the complainant or the noise ?"
- using the best practical noise measures, octave band or overall levels;
- whether distinctions are applied in dealing with noise from profit-making and social-value industries;
- whether specific requirements result from interpreting these regulations ;
- are there specific provisions for assessing low-frequency and background noise?

Answers to some of these questions arose from the ensuing discussions, in which considerable stress was laid on the justice and strictness with which the EPA, municipal councils and VCAT administer the noise and associated planning regulations.

The following answers arose from these discussions:

- With the new NIRV regulations there is probably no change in the responsibilities of the EPA and municipal councils in administering them and granting approvals. EPA representatives don't normally attend VCAT hearings when noise is an important issue.
- The EPA regards the applying of these regulations to be an agent for change towards environmental improvement. Their application also depends on the local planning scheme provisions. Their basis involves polluters as those who pay for the noise reduction treatments.
- Authorities allowing the later encroachment of new residences closer to a noisy industry is a recurring enforcement problem. Municipal councils can be sued if, after having given approval to such an industry, they then allow closer residences. The establishment of suitable noise buffer zones can be required. Under these circumstances, the NIRV regulations and noise nuisance law have to be examined and compared and consistency sought. As far as possible, the practicability concept is to be adopted in applying the regulations.
- Whereas the NIRV regulations apply in regional areas outside towns, N-1 is to be used in urban Geelong, Ballarat and Bendigo.
- It was recognized that different conditions may apply to industries in an outer suburban area and to those in a rural area.
- For areas where industry is developing and increasing, the regulations provide, inter alia, that noise limits are 3 dB less for the first comers and 10 dB less for later comers. This raised the question whether the regulations are discouraging continuing industrial development in an area. It was considered here that cost constraints usually apply in such cases. Regional towns where industry is declining also raise their particular noise-zoning problems.
- The regulations were considered to embody a best practice concept, involving routine noise controls and providing environmental gains. Even standard noise control measures involve a cost, and can determine, for example, whether a whole factory is protected by a solid building, or each noisy machine is individually isolated.
- It was suggested by some present that it would be helpful if the regulations included explanatory examples.
- On the question of assessing low frequency noise, it is considered necessary that such noise is distinctly audible above the general background. In some such cases, planning scheme provisions require octave or one-third octave spectrum analysis.
- When derived noise levels are calculated from measured levels by adding or subtracting allowances for eg, tonal sounds, etc [as in AS 1055 on environmental noise], on the question whether these additions and subtractions are referred to as "adjustments" or "corrections", the EPA regularly uses the term "adjustment" as being appropriate [because "correction" applies strictly to instrument readings known to be in error].
- Considerable discussion occurred as to whether the procedures for including the effects of wind and other weather noises in the EPA regulations are more accurate or reliable than the internationally recognized and simpler procedures described in ISO 9613.2. The local procedures require, for example, that noise under down-wind conditions [is it frequent or rare ?] during both day and night, and during different seasons is to be considered to obtain the worst conditions and then come back to the 20% criterion. Such procedures may involve using a nearer location as a "derived point" [as in N-1] and converting the measured levels to suit the noise-sensitive area.

After the discussions, Eliot Palmer thanked all those present for their questions, comments and helpful suggestions, and closed the meeting.

Louis Fouvy

Conference Reports

Acoustics 2009: Research to Consulting

Three students received AAS travel awards to attend Acoustics 2009 in Adelaide last year. These three students recount their experiences.

Made it! I had spent the previous two days riding my motorcycle from Melbourne to Sydney in 42 degree heat and had arrived home, parked the bike, packed my bags, and flown to Adelaide. Having checked into my hotel I now found myself at the Acoustics 2009 reception at the National Wine Centre. As the Young Adelaide Voices provided a soothing backdrop, the contrast with the previous 48 hours was stark. The reception provided a great opportunity to catch up with some familiar faces and to make some new acquaintances while the enthusiastic staff enticed us with the range of fine wines and tasty morsels.

Locating the conference rooms at the University of Adelaide the following morning was straightforward with maps, ample signage and an encouraging trail of delegates (despite the best efforts of the staff at the reception the previous night). Coming to my PhD studies following several years in industry I found the first keynote by Dr David Rennison, titled "Industry and University partnerships in Acoustic Research - Factors for success", to be particularly interesting as it highlighted the need for both partners to be aware of the differences between each other's environments and the challenges that these can present. The program of presentations that followed was full, but time limits were well managed (by organisers and delegates alike) and adequate time was provided to move between sessions.

While some conferences might boast upwards of eight parallel sessions, there was a level of intimacy present at this conference that I found to be really enjoyable. Also, as a PhD candidate, it was valuable to be able to present to a wider audience than might have otherwise been possible had there been multiple sessions to choose from. I fielded some questions following my presentation that were encouraging and received suggestions from a learned audience that have assisted me focus my research.

Of course, conferences are not all about hard work and the organisers delivered with the heritage tram ride to Glenelg Beach and a fantastic banquet dinner at The Stamford Grand. Sipping bubbly while crammed together with the other delegates on the tram as it rolled through the afternoon traffic and the sunset over the Glenelg Jetty were highlights.

The conference closed with drinks and a BBQ on the university lawns and offered a generous range of wines, beers, meats and breads. The trailer decked out with BBQ hotplate and windmill was something that had to be seen. The opportunity to tour some of the university laboratories after the BBQ was very interesting and served to confirm why so much good acoustic work (and people!) can be found at the University of Adelaide.

I'd like to thank the Australian Acoustical Society (NSW Division) for the Student Travel Award that provided financial assistance for me to attend Acoustics 2009.

Peter Jones

Sitting in Sydney airport, I can't say I particularly looked forward to travelling to Adelaide. Sydney was in the middle of a heatwave, and the reports were that it was even worse in Adelaide. I had never travelled to Adelaide before, and of the people I knew who had, the one thing they agreed on is "Adelaide is hot". On arriving at Adelaide airport, I was pleasantly surprised. The record heatwave had broken, and temperatures had halved. Conditions remained comfortable during my entire stay, which was fortuitous considering the previous week.

The University of Adelaide was a wonderful venue for the Acoustics Conference, with an abundance of accommodation and facilities within walking distance (and normal walking distance too, as opposed to the Olympic variety). The venue itself had everything within a convenient distance.

I would like to thank the organisers of the conference, for organising everything simply and seamlessly, making my own presentation a painless experience. I would also like to thank all the presenters for their informative and interesting presentations over the three days of the conference. I particularly enjoyed seeing how the same techniques I use myself have been applied to quite dissimilar fields.

I would also like to thank all the exhibitors at the conference, for their excellent displays and information, and for supporting the conference. I enjoyed looking at many of the technical displays, as I had never had an opportunity to see a lot of the technology first-hand before.

The tour of the Facilities at the University was also very interesting, seeing the current projects of the students at the University, many of which were in markedly different topics than the work being undertaken at UNSW.

The conference banquet at Glenelg was an excellent night out, with superb water-views, music and food. I particularly enjoyed Prof. Brook's talk on Nuclear Power with respect to climate change. Regardless of opinions for or against Nuclear Power, it was good to see some issues presented and discussed which are regularly dismissed out-of-hand in Australia.

I would like to thank the Australian Acoustical Society for organising an excellent conference, and to thank the NSW division of AAS for providing the travel scholarship that gave me the opportunity to attend and present my work.

Michael Coats

I'd very much like to thank the AAS Queensland division for supporting me to attend the AAS conference in Adelaide last year. Without the support given I would not have been able to attend. As an attendant I was greatly appreciative of the chance to speak more with industry representatives relevant to my field of post grad research study in railway vibrations and noise as well as present some of my own most recent research for outside review. I also learnt of other projects being run throughout Australia and witness work being done at the University of Adelaide and talk to those involved to find out alternate approaches to my own research work. I felt very privileged to be given the opportunity to be there.

Ross Batten

Standards Australia

Submissions open for Standards Australia resourced projects

Standards Australia is implementing a new and transparent process to prioritise Standards development projects that will be resourced by Standards Australia. Proposals for these projects will need to be submitted at set dates throughout the year so they can be considered together and prioritised if necessary. Following the conclusion of the consultation period on the Discussion Paper outlining the proposed framework for the prioritisation and selection process, Standards Australia is now accepting submissions from proponents for Standards Australia resourcing for their Standards development projects. Completed proposals for the first round are to be received by 15 June 2010. Standards Australia will release in mid-May the Finalised Prioritisation Framework addressing feedback gathered during the consultation period. Any stakeholders wishing to amend their proposals in light of the Framework may do so until 15 June. Proponents interested in applying for Standards Australia resourcing are encouraged to consider their program of work and consult with relevant stakeholders about the priorities within their industry sectors. Standards Australia's Relationship Managers are available to discuss options and assist with completing proposal forms. Standards Australia will also hold workshops in Melbourne and Sydney—and other states as required—to provide guidance on the process for proposing a Standards project. There will be another opportunity to submit proposals for Standards Australia resourced projects to start in the first half of 2011. Stakeholders wishing to undertake an externally funded project with Standards Australia may submit proposals at anytime throughout the year. Such projects are resourced separately from the Standards Australia resourced pathway. To provide assistance to stakeholders in completing submissions, a revised Proposal Form and revised Guide to Net Benefit are now available on Standards Australia's website at http://www.standards.org.au/cat.asp?catid=149 Relationship Managers can be contacted through Standards Australia's Customer Information Service on mail@standards.org.au or 1800 035 822.

Stakeholder engagement

A Council meeting will be held mid-year to provide Standards Australia Councillors with an opportunity for an informal briefing and exchange of issues. The meeting will be held on 4 June 2010 in Sydney. Also, later this year Standards Australia will hold sector specific forums to obtain feedback and provide details on standardisation activities across the specific industry sectors. These forums will be in addition to the continuing role of Standards Australia's Relationship Managers in identifying the interests, activities and concerns of individual stakeholders. Details on dates and locations of the sector forums will be communicated.

From Standards Bulletin #7

FASTS

The AAS is a member of the Federation of Australian Scientific and Technical Societies, FASTS. The following is a summary of the contributions from FASTS during 2009 and the plans for 2010.

FASTS ongoing contribution to Australian science includes:

- Science meets Parliament – FASTS' annual flagship event, where more than 200 scientists have face-to-face meetings with Federal Parliamentarians on science issues;

- Highlighting science with the Prime Minister and the Cabinet through FASTS' ex-officio membership of PMSEIC;
- Organising forums and workshops on significant science issues;
- Developing science policy at a high level and providing input to Parliamentary Committees, Government Departments and Government reviews and inquiries;
- Assisting Member Societies to raise and develop issues;
- Distributing information to Member Societies regularly and responding to feedback.

2009 highlights include:

- Roll out of FASTS Heads Up Program including presentations on Quantum Cryptography, and Emissions Reduction Targets and the Great Barrier Reef;
- National roadshow to gather responses to the Government's: Powering Ideas: An Innovation Agenda for the 21st Century;
- Presentation to the PMSEIC on Epidemics in a Changing World and contribution to the PMSEIC foresighting committees;
- Provision of examples of science success stories from FASTS' members to the Prime Minister;
- Launch of major document in Parliament on Women in Science in Australia: Maximising Productivity, Diversity and Innovation;
- Release of major reference document When is Science Valid? – a Short Guide on How Science Works and When to believe it;
- Formation of the Great Barrier Reef Climate Change Alliance and briefing to politicians, the media and the bureaucracy on the impact of climate change on the GBR;
- Release of Policy Discussion Paper: Giving Preparedness a Central Role in Science and Innovation Policy ;
- Commissioned a study to investigate the changing nature of scientific and technological work;
- Submissions to reviews including ARC Centres of Excellence and NHMRC Fellowship Consultation Paper.

Book Reviews

Acoustic Absorbers and Diffusers
Theory Design and Application
2nd Edition
Trevor Cox and Peter D'Antonio

Taylor Francis 2009, 476 pp

Hard cover ISBN10:0-415-47174-5 and ebook ISBN10:0-203-89305-0

This book has one of the more interesting Prefaces that I have seen for technical books. In four pages the history of the interest in absorbers and diffusers of the authors is explained. As well the narrative highlights the benefits of cross discipline interactions – one of the authors was originally a diffraction physicist with an interest in music. Various coincidental events led to his greater involvement with diffuser technology and almost a decade later a meeting with the second author at a Standards Committee Meeting in 1994 which led to an initial informal collaboration. After working on various research and consulting projects, they published the first edition of this book a decade later in 2004.

Trevor Cox is Professor of Acoustics at Salford University and Peter D'Antonio is CEO of RPG Diffusor Systems Inc. Their combined skills have led to a comprehensive and authoritative book on acoustic absorbers and diffusers. These are the main design tools for altering the acoustic conditions of rooms and spaces. The early chapters clearly set out the basic principles with excellent diagrams. A set of 3 simple sketches encapsulates how the relative importance for three acoustic treatments – absorption, specular reflection and diffuse reflection – varies for sound production, sound reproduction and noise control. After these introductory chapters there are chapters on the different types of absorbers including porous, resonant and other types like seating and audience. Then follows chapters on scattering, Schroeder diffusers, geometric reflectors and diffusers, hybrid surfaces and concluding with active absorbers and diffusers.

The second edition brings the content up to date, including new materials like microperforated designs. It also includes updated sections on measurement techniques and prediction methods including the newer time domain methods.

This book will be a valuable resource for those involved with room design and concerned with ensuring ideal conditions for sound production and sound reproduction. The book would also be a useful reference for those involved with noise control, especially where sound absorption is an important component of the overall design to achieve the required outcome.

Engineering Noise Control
Theory and Practice
4th Edition
David Bies and Colin Hansen

Spon Press, Taylor Francis 2009, 746 pp

Hard cover ISBN10:0-415-8706-4, soft cover ISBN10:0-415-8707-2 and ebook ISBN10:0-203-87240-1

The fourth edition of this well known and well respected title follows a similar layout to the earlier editions, but the content has been considerably expanded and enlarged although only has about 30 more pages than my (well-used) version of the 3rd edition (2003).

I would guess that most people working in engineering noise control would either have a copy or have referred to Bies and Hansen at some time in their work. The strength of the book has been that the extensive coverage means the reader can find some introduction and detail on just about all aspects of engineering noise control.

Each chapter has been updated and expanded with the new understandings of the mechanisms of noise control over the last 6 years. One important change is the removal of the chapter on active noise control. The preface alerts the reader to a more comprehensive treatment in "Understanding Active Noise Cancellation" by Hansen. An important addition in the light of the developments in recent years is a chapter on practical numerical acoustics. This chapter has been written by Carl Howard and includes examples of how free and open source software can be used to solve complex problems.

Without wishing to criticise the overall value and quality of the book it is a little disappointing to note that some content has not been updated. For example reference in the text to the 1987 version of AS 2107 has not been updated to the 2000 version, but this later version is given in the reference listing at the end of the book. Similarly it would be of assistance to the reader to have something on the background of a particular measure or a reference to further reading. Just one example of this is the 'Noise Impact" method using total weighted population gives no guidance to the reader of the source of this relationship.

The previous editions have been the standard reference for all those working in noise control in Australia and procurement of the updated and expanded fourth edition is highly recommended.

Marion Burgess is a research officer with the Acoustics and Vibration Unit of UNSW at the Australian Defence Force Academy.

Diary

2010

06 - 07 May, Paris
2nd International Symposium on Ambisonics and Spherical Acoustics.
ambisonics10.ircam.fr

24 – 27 May, Sydney
OCEANS'10 IEEE
www.oceans10ieeesydney.org/

09 – 11 June, Aalborg
14th Conference on Low Frequency Noise and Vibration.
lf2010.org

13 – 16 June, Lisbon
INTER–NOISE 2010
www.internoise2010.org

18 – 22 June, Cairo
ICSV17
www.icsv17.org

23 – 27 August, Sydney
ICA2010
www.ica2010sydney.org

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

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

2010

29 – 31 August, Auckland
Sustainability in Acoustics
issa.acoustics.ac.nz

30 – 31 August, Singapore
Non Linear Acoustics and Vibration
www.ica2010sydney.org

23 – 27 August, Seattle
11th International Conference on Music Perception and Cognition
TBA

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

11 – 14 October, San Diego
IEEE 2010 Ultrasonics Symposium.
bpotter@vectron.com

19 -– 20 November, Brighton
Reproduced Sound 25
www.ica.org.uk/viewupcoming.asp

2011

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

24 – 28 July, Tokyo
19th International Symposium on Nonlinear Acoustics (ISNA 19)
Web: TBA

27 June – 1 July, Aalborg
Forum Acusticum 2011
www.fa2011.org

27 – 31 August, Florence
Interspeech 2011
www.interspeech2011.org

05 - 08 September, Gdansk
2011 ICU International Congress on Ultrasonics.
Web: TBA

4 –7 September, Osaka
INTER-NOISE 2011
office@ince-j.or.jp

2012

20 – 25 March, Kyoto
IEEE International Conference on Acoustics, Speech, and Signal Processing.
www.icssp2012.com

2013

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

02 - 07 June, Montréal
21st International Congress on Acoustics (ICA 2013)
http://www.ica2013montreal.org

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