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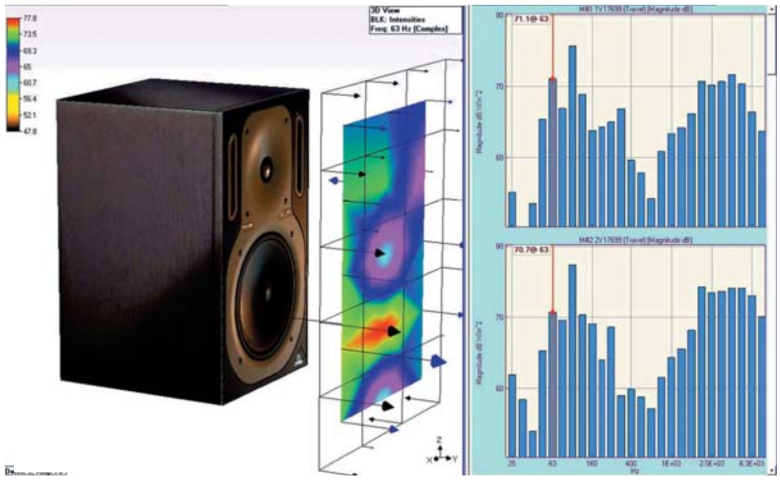

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Acoustics Australia
General Business
(subscriptions, extra copies, back issues, advertising, etc.)
Mrs Leigh Wallbank
P O Box 70
OYSTER BAY NSW 2225
Tel (02) 9528 4362
Fax (02) 9589 0547
wallbank@zipworld.com.au

Acoustics Australia
All Editorial Matters
(articles, reports, news, book reviews, new products, etc)
The editors, Acoustics Australia
School of Physics
University of New South Wales
Sydney 2052 Australia
61-2-93854954 (tel)
aaeds@phys.unsw.edu.au
AcousticsAustralia@acoustics.asn.au
www.acoustics.asn.au

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Enquiries see page 38

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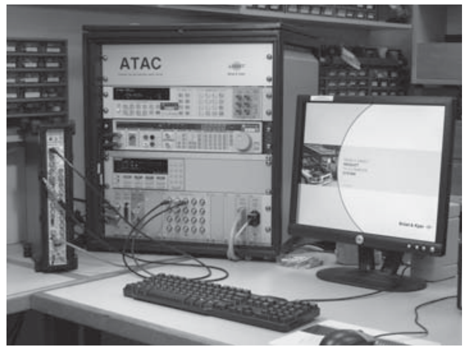

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Message from the President

It is hard to believe that a quarter year has already gone by but here we are in April already!

The "financial crisis" has been on everyone's mind and certainly is affecting our acoustics fraternity. I have spoken to a number of people and consultants about their current workloads and while most are reasonably busy, it seems that everyone is expecting a period of reduced activity as a result of reduced spending by corporations and developers. Certainly, there is an expectation that government will step in and provide some infrastructure projects for us to participate in. But in terms of hiring new graduates, the current outlook is not too promising.

However, we do have some good things to look forward to. While attendances at overseas conferences may drop due to financial considerations and the weakened Aussie dollar, the good news is that we have our own high standard **Annual Conference** coming up in **Adelaide** this **November 23 – 25, 2009**. So now is the time to get your papers ready and put the date into your diary.

Another date to put into your diary is the **ICA 2010** which will be held in **Sydney August 23 -27, 2010**. This promises to be an exciting international conference and expectations are for over 1000 attendees. Satellite conferences will be held in **Melbourne** (the theme is Room Acoustics) and in **New Zealand** (the theme Acoustics and Sustainability). So again, now is the time to plan for papers and attendance.

Our website has a new look and I want to thank Terry McMinn, our webmaster, for organising this and keeping it up to date. We are continuously updating this site as well as adding more and more information and soon you will be able to access your Acoustics Australia magazine electronically (see the details elsewhere in this journal). Check out http://www.acoustics.asn.au/joomla/

This year, our general secretary, Byron Martin, has sent our membership renewal invoices by email except to a few members who do not have email access. This is part of our move to making things simpler and more efficient. Please pay ASAP so that our collections are complete before the end of this financial year.

Recently, I have tried to contact some members using the members directory via our site. It is vital that members update this information and make sure that contact details are valid, particularly email addresses. If you move locations or change addresses, please remember to update your records. Best wishes for now,

Norm Broner

From the Editors

In December's editorial, we lamented the absence of technical notes in recent issues of Acoustics Australia, and urged readers to send some. Perhaps it is just a coincidence, but this issue has three. While the formal papers are sent for formal peer review, technical notes are reviewed only informally: they are read by some of the editors, who may then suggest revisions. The process is usually rapid. If you have a project that would interest readers, please consider submitting a brief report.

In January, I expect that all but one of the editors around the country thought ''I'm glad that wasn't me''. The political journal *Quadrant* published a hoax article, containing a number of unlikely claims [1]. It began with a reference to another hoax: Alan Sokal's amusing collection of postmodern foolishness published by *Social Text* [2]. Articles about the Quadrant hoax usually mentioned Ern Malley, a fictitious poet whose nonsense verse, created by James McAuley and Harold Stewart, beguiled Max Harris, editor of the modernist magazine *Angry Penguins* [3].

McAuley and Stewart had aimed to embarrass Harris and to disparage modernism. Sokal had wondered whether a leading postmodern journal would ''publish an article liberally salted with nonsense if (a) it sounded good and (b) it flattered the editors' ideological preconceptions.'' The author of the Quadrant hoax had aimed to embarrass Windschuttle, the editor of Quadrant, who is known for disputing the extent of the violence with which Europeans displaced the original Australians.

Not that anyone would ever publish a hoax in acoustics… Well, there was our press release about the acoustics of the air guitar. ''The body of a traditional guitar acts to match the high specific mechanical impedance of the strings to the low acoustic impedance of the radiation field'' we said. ''In an air guitar, the strings and body both have the same specific impedance as the radiation field, so the transmission should be maximised. So we wondered why the air guitar was so quiet.'' Student Avril Poisson, we continued, had solved the conundrum: ''The perfect impedance match between strings and bridge it is good for transmission, but the absence of the reflection at the bridge means that the standing waves are not produced and so the sound is not radiated." We then speculated that the absence of standing waves might be related to the observation that, on the air guitar, even a beginner can play extremely rapid passages with ease.

We sent the story to Column 8, a whimsical section of the Sydney Morning Herald, who ran it, drawing attention to the date (1 April) and observing that poisson d'avril translates as April fool. Now we're going back to check the latest submissions.

1. Ferrari, J. and Maiden, S. (7/2/2009) ''Keith Windschuttle caught in Quadrant hoax'', *The Australian* on line.
2. Sokal, A. (1996) ''Transgressing the Boundaries: Towards a Transformative Hermeneutics of Quantum Gravity'' *Social Text*, **46/47**, 217-252.
3. Wikipedia (4/4/2009) ''Ern Malley'' en.wikipedia.org/wiki/Ern_Malley
4. Column 8 (5/4/2006) *The Sydney Morning Herald*, p24.

Joe Wolfe

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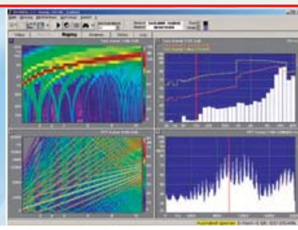

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A COMPARISON OF TWO TECHNIQUES THAT MEASURE VOCAL TRACT SHAPE

Catherine I. Watson[1], C. William Thorpe[2], Xiao Bo Lu[2]
[1]**Department of Electrical and Computer Engineering**
[2]**Bioengineering Institute, The University of Auckland**

This study compares vocal tract shapes of four vowels for a single speaker estimated via analysis of magnetic resonance images and acoustic reflectometer measurements. The first and second resonances of the vocal tract shapes from the two different methods are compared to the first and second formants obtained from an acoustic analysis of the speech signal. It is demonstrated that speech production is compromised by the mouthpiece used for the acoustic reflectometer, and as such this tool is not useful for studying articulatory phonetics.

INTRODUCTION

The sounds of speech are produced by movements of the speech organs and their effect on the air flow through the vocal tract. By changing the position of the articulators (i.e. tongue, jaw, lips), and the nature of the acoustic air flow through the vocal tract we produce speech. However, the speech production mechanism is hidden, and it is non-trivial to ascertain the precise configuration corresponding to different speech sounds. Acoustic phoneticians are interested in determining differences in production for various environmental effects such as accent, aging, and pathology, and utilise acoustic analysis techniques that relate specific spectral features of the speech signal to features of the vocal tract shape. However, such analysis relies on acoustic models of speech production to solve the inverse problem, and does not always result in reliable estimates of vocal tract shape.

The fundamental acoustic model of speech production is the source filter model [1], in which the acoustic energy source is separated from the time-varying filter which imparts a specific spectral shape to the speech sound. In vowel production the acoustic source consists of a quasi-periodic train of pulses of air emitted by the vibrating vocal folds, and the filter can be well modelled by small number of resonances corresponding to the resonating cavities in the vocal tract. The spectrum of a (sustained) vowel sound is therefore a line spectrum (with spacing equal to the vibration frequency of the vocal folds) with several distinct peaks. These peaks are termed formants and it has been shown that the first two or three formant frequencies collectively determine the identity of the vowel sound [1, 2]. Since the resonances (and consequently the formants) depend on the vocal tract shape, they will be affected by factors such as size, effects of aging, and manner of articulation (how the vocal organs are moved during speech). Unfortunately – especially for high pitched voices – the relatively wide spacing between the spectral lines means that there can be insufficient information to determine uniquely the centre frequency and bandwidths of the resonances and therefore a unique vocal tract shape. Thus, other methods of determining the vocal tract configuration are of interest if differences in its shape are to be investigated.

Researchers have obtained measurements of the vocal tract shape by various imaging techniques including X-rays (e.g. [1]), computer–tomography (C-T) (e.g. [3]), and magnetic resonance (MR) (e.g. [4]). The latter two approaches enable 3-D shapes of the vocal tract to be constructed through post-processing of the images. All these methods involve expensive equipment and well trained operators. With X-rays and C-T scans there is also some risk to the subjects if they are exposed to repeated measurements. For a large scale study on speech production, all the above factors make it difficult to obtain comprehensive data from a large number of subjects using these techniques.

It is also possible to deduce the vocal tract shape using a technique called acoustic reflectometry (AR) [5]. This measurement technique is used for determining the cross-sectional area of ducts. It has been adapted for diagnostic measurements of upper respiratory airways, and has been previously used in studies of the vocal tract shape (e.g. [6, 7]). The technique involves transmitting pulse-like signals through a wave tube and into the vocal tract. The pulses are partly reflected when they encounter physical obstructions or changes in the cross-section of the tract. Analysis of the reflected waves gives the impulse response of the tract, from which the cross sectional area of the tract can be calculated (see [5] and [8] for a more in depth discussion of the technique). Acoustic reflectometry is easy to perform, the equipment is cheap in comparison to the former approaches, and it has no known side effects on the subjects. This makes it a potentially ideal instrument for a large scale study relating vocal tract shapes to specific speech features.

Once we have obtained the vocal tract shape (by MR images, AR, or any other measurement technique), it is a routine process to calculate the vocal tract resonances corresponding to that shape [9]. We can therefore compare the shapes obtained by different measurement techniques with the spectral patterns expected for different speech sounds (as determined by direct measurement of the acoustic output).

Because of their geometrical accuracy, MR image studies are the "gold standard" for determining physiological structure, but as mentioned the cost is prohibitive when considering a large scale study. AR is appealing to use in a large scale study due to the low costs associated with collecting the data. However, questions arise as to how it compares to the MR image approach; does it give accurate enough information about the vocal tract shape; and can we assess the effects of the speech produced by such a shape? To date there has been no acoustic phonetic study done comparing vocal tract shapes calculated via MR images and AR. The purpose of this study is to do that comparison, and also to contrast the vocal tract resonances calculated from these shapes to formants from recorded speech.

METHOD

The study involves three different data sets collected from a single male speaker of New Zealand English (NZE). With this speaker we did an analysis of 3-D MR images of the vocal tract, an analysis of the cross sectional area of the vocal tract obtained from AR and an acoustic analysis of the speech. Four vowels were studied /iː, aː, ɔː, ɜː/, the NZE vowels in the words “heed”, “hard”, “hoard” and “heard” respectively.

2.1 Magnetic Resonance Imaging Analysis

The MR images were acquired with a 1.5T Siemens Magnetom Avanto MRI scanner. The scanning parameters were: T1-weighted image; parallel sagittal planes; 7 mm slice thickness; no gaps between slices; 200x250 mm field of view; 1660 ms repetition time; 9.4 ms echo time; 1 mm resolution; 20 slices and a total scanning time of 21 sec. MR images were collected in a supine position with the head supported to prevent movement. Images were obtained for all four vowels. Since the MR images produce a three dimensional image of a single vocal tract shape, the subject had to maintain a sustained production of each vowel for the entire scan time of 21 seconds. For this reason these vowels were pronounced in isolation, rather within a word context. The subject’s background in voice science meant he was able to ensure the articulator positions were appropriate for each vowel. The MR images were stored on the computer and can be viewed as DICOM images.

To determine the vocal tract area, cross-sections of the vocal tract were obtained at 15 points along the mid-line of the vocal tract. All image processing was performed using the CMGUI image processing and analysis software (http://www.cmiss.org/cmgui). A centre-line was constructed through the visible vocal tract on the mid-sagittal plane. Next, a smooth line was fitted through these points with a cubic Hermite spline having 15 equally spaced nodes. At each node a plane was constructed perpendicular to the centre-line, as illustrated in Figure 1 (left), and the 3-D image stack resampled onto each plane, thereby producing a sequence of images that cut the vocal tract perpendicularly throughout its length. The boundary of the vocal tract was then manually marked on each of the planes (see Figure 1(right)). Finally, a smoothing spline was fit to these data and the internal area computed at each of the planes. This sequence of measurements forms a 1-D vocal tract area function. This results in a discrete model of the vocal tract, approximating the varying area of the tract by a series of concatenated tubes of uniform thickness and varying cross-sectional area. From this the resonant frequencies were calculated from custom functions based on the standard linear prediction model of speech (e.g. [9]). All functions were implemented in R (http://www.r-project.org/).

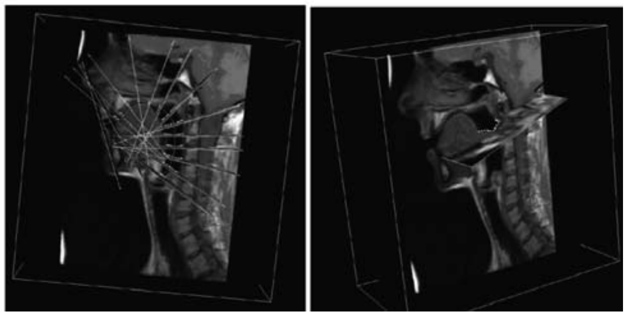

Figure 1: Illustration of how the cross section areas of the vocal tract were determined from the MR images: Slices computed perpendicular to the vocal-tract midline (left), and the cross-sectional area is obtained by manually locating points on the edge of the vocal tract cavity on each slice (right).

2.2 Acoustic Reflectometery

The AR vocal tract profiles were acquired by the ECCOVISION Acoustic reflectometer. It provides a non-invasive assessment of the cross-sectional area profile of the oral and pharyngeal spaces down to the larynx. Subjects place the wavetube in their mouth, position their articulators for the target vowel, and hold that position for two to three seconds whilst a series of sonic pulses are sent down the vocal tract and measurement takes place. The subjects are required to seal their lips tightly around the mouthpiece to prevent acoustic leaks of the sonic pulses. In addition the vocal folds need to be closed during the measurement (achieved by gently blocking expiratory airflow). Vowels can not be voiced during the measurement since the glottal excitation interferes with the measurement pulses, and therefore subjects do not receive any aural feedback on the production of their vowels.

To aid the subject to get the correct tongue placement for the vowels (the jaw placement was compromised due to the wavetube) we collected the data in a specific way. Firstly, we collected speech recordings immediately before the AR data (see section 2.3 for more details about this process). Secondly, although only four vowels were investigated in this study, we collected AR data for nine of the eleven monophthongs in New Zealand English. Two of the vowels, /ʌ/ (as in “hud”) and /ʊ/ (as in “hood”), were excluded on the grounds that they have been shown to differ primarily in duration, but not in quality, with /a:/ and /ɔ:/ respectively [10]. The vowel order the data were collected was also important - each consecutive vowel was both an articulatory and acoustic neighbour, e.g. /e/ (as in “head”) was recorded after /i:/,and /æ/ (as in “had”) was recorded after /e/. Thirdly, a series of four separate vocal tract measurements were obtained for each vowel, and after each measurement, the vocal tract profiles were checked to ensure consistency and a clear glottal closure. Any flawed data were rejected, and the measurements were retaken. All the measurements were done by a trained research assistant.

A collection of custom functions have been developed in R which allow the data from the reflectometer to be visualised and processed. Using this software, the start and end of the vocal tract (i.e. lips and glottis) were manually identified, and resonances calculated from the resulting vocal tract shape using the same algorithms as for the MRI data (See section 2.1). For the AR data the vocal tract was subdivided into 11 tube segments of uniform length.

2.3 Formant analysis of Speech

We recorded the subject's speech in an acoustically isolated sound booth (Whisper Room MLD8484E) directly on to a Marantz PMD670 Solid State Recorder at a sampling rate of 20 kHz, using a Shure SM58 Microphone. We collected citation form speech of nine words "heed", "head", "had", "hard", "hod", "hoard", "who'd, "herd" and "hid", four of which were used in this study. Five tokens of each vowel were recorded, with the order of the vowels randomised within each repetition. The speech data were transferred to the computer and the vowel portions of each word phonetically labelled using the EMU speech database system (http://emu.sourceforge.net/). The first three formant centre frequencies and their bandwidths were calculated (the settings were 12^{th} order linear predictive coding analysis, cosine window, 49-ms frame size, and 5-ms frame shift). All formant tracks were visually checked, and tracking errors were corrected. For each vowel, the target was manually identified. The acoustic vowel target is presumed to be the section of the vowel that is least influenced by phonetic context effects. The criterion for identifying the vowel targets varies between the different vowels (see [10] for more details). The formant values were extracted at the vowel targets and analysed in R/EMU.

RESULTS

The mid-sagittal images of the vocal tract from the subject when producing sustained productions of the three NZE point vowels /i:/, /a:/, and /ɔ:/, and the central vowel /ɜ:/ can be seen in Figure 2. The vocal tract is the black region which is bounded by the lips and the vocal folds. Note the markedly different dimensions of the vocal tract for each vowel configuration. For each of the point vowels, the tongue tip, jaw opening, and tongue body respectively are essentially at their articulation extremities. For /i:/ the greatest point of narrowing in the vocal tract is at the hard palate; the tongue surface is close to the roof of the mouth, as far forward as the alveolar ridge; and the jaw opening is very small. For /a:/ the jaw is at its most open and the tongue body is further back in the vocal tract than /i:/. For /ɔ:/ the tongue body is even further back than for /a:/ with the constriction location (greatest point of narrowing due to the tongue) at the pharynx; and the jaw opening is similar to that for the production of /i:/. For the central vowel /ɜ:/ there is an almost uniform cross-sectional area along the length of the vocal tract, unlike for the other three vowels where there are distinct wide and narrow sections of the tract.

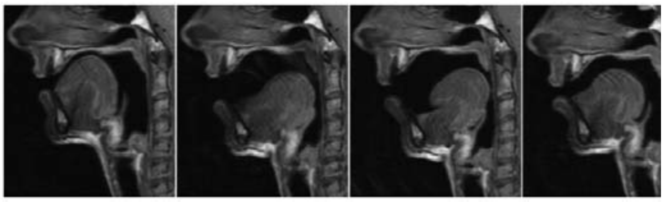

Figure 2. Mid sagittal MRI images of a male speaker doing a sustained production of: (left to right) /i:/, /a:/, /ɔ:/ and /ɜ:/ vowels.

Figure 3 shows the cross-sectional areas of the vocal tract for the four vowels /i:, a:, ɔ:, ɜ:/ obtained from AR (top four plots), and the MR images (bottom four plots). The cross-sectional areas obtained from the MR data for the four vowels are consistent with the mid-sagittal MR images in Figure 3. As expected, where there is a small constriction in the vocal tract in Figure 3, there is a corresponding small cross-sectional area, and where the vocal tract is wide there is a large cross-sectional area.

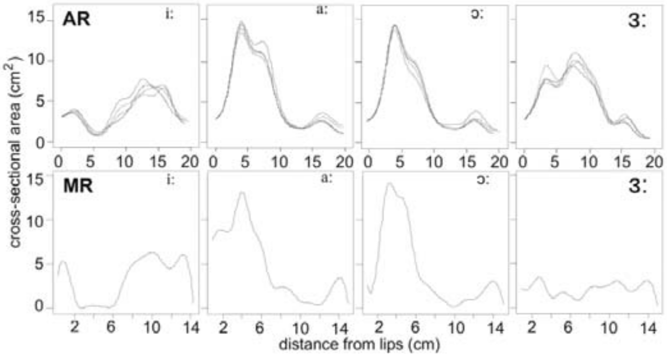

Figure 3: The cross-sectional area function of the vocal tracts for the four vowels indicated at the top, obtained from the AR (upper) and the MR (lower) data.

We obtained four readings for each vowel from AR, and all four readings have been plotted. The consistency in the vocal tract shape across repetitions is noteworthy, considering that the vowels could not be voiced during the measurement of the vocal tract. The cross-sectional area shapes obtained from the both the MR images and AR are similar for /i:, a:, ɔ:/ except around the lips. This is because the wavetube used in AR fixes the jaw position to the width of the wavetube, whereas the subject is free to move their jaw to any position for the MR measurements. There was a difference in the area functions for /ɜ:/, with the shape determined from the AR analysis having an unexpected large cavity in the front of the mouth. It may be that the fixed placement of the lips for the AR measurements interfered with the ability to correctly place the articulators, although more data from other subjects using AR will need to be analysed before that can be determined.

The two methods differed substantially in the measurement of the vocal tract length. For the MRI data the vocal tract length varied between 16.2 cm (for i: and ɜ:) and 17.8 cm (for a:). For the AR data the lengths varied between 19 cm for /i:/ and 20.6 cm for /ɔ:/.

Figure 4(a) plots the mean values of the first and second formants (F1 and F2) of the four vowels on a traditional F1 vs. F2 plot. The formant values for our speaker are typical for an NZE speaker (c.f. [10]). F1 was lowest for /iː/, and highest for /aː/. F2 is lowest for /ɔː/ and highest for /iː/. The /ɜː/ vowel (and the remaining NZE monophthongs) falls within the space between the vowels /iː, aː, ɔː/.

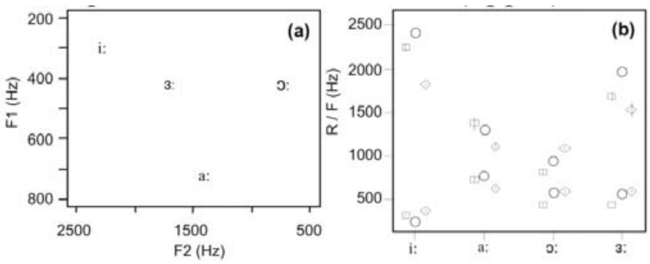

Figure 4: (a) The mean formant frequencies of the four vowels in the study on an F1 vs. F2 plot (note how the scales of the F1 and F2 axis have been reversed, thus enabling a direct comparison between the acoustic and articulatory spaces). (b) For the four vowels in the study, the mean first and second formants (□), the derived first and second resonances from the MRI data (○) and AR data (◊). The vertical lines indicate the standard deviation of the multiple measurements.

Figure 4(b) contrasts the mean F1 and F2 of the recorded speech tokens with the first and second vocal tract resonances (R1 and R2) calculated from the MR data and AR data for each of the four vowels. The AR resonance values are also means of the four measurements, but the MR resonances were calculated from the single image for each vowel. For /iː, aː, ɔː/ there is a good match between R1 and R2 from the MR method and the first two formants. However the values are more extreme. For /ɜː/ however the R2 value is much higher than expected. The AR data follow similar patterns to the formants for /iː/ and /aː/, i.e. /iː/ has the lowest R1, and highest R2, and /aː/ has the highest R1, however there is an issue with R1 and R2 values for /ɜː/ and /ɔː/. For both vowels, the R1 and R2 values are higher than expected. Also the R1 and R2 values for /aː/ and /ɔː/ are very similar

DISCUSSION

The necessity to form a seal with the lips on the wavetube in AR means that the vocal tract shape is necessarily compromised for almost all speech sounds. For example /aː/ and /ɔː/ differ mainly in jaw opening (see the images in Figure 2), which has the effect of changing both F1 and F2 for the two vowels substantially (see Figure 4(a)). However, the use of the wavetube fixes the jaw position and thereby removes this point of difference between the vowels – consequently in the AR data the derived vocal tract shapes for these two vowels are similar (see Figure 3), as are the R1 and R2 values (see Figure 4(b)). The fixed jaw opening imposed by the wavetube is also the reason why the R1 and R2 ranges for the AR measurements are much more constrained than for the MR data.

There is some suggestion in the data that the pharyngeal portion of the vocal tract is reasonably comparable between the AR and MR derived vocal tract shapes, at least for three of the four vowels. However the difference in the vocal tract length measurement between the two techniques is of some concern. There are a number of possible reasons for this difference. Firstly it may have been due to the position the subject was in whilst the measurements were taken. The AR data were collected whilst the subject was sitting holding the wave tube whereas the MR data were collected whilst the subject was lying supine. We subsequently repeated the AR measurements on these four vowels in the supine position. The vocal tract lengths for /iː/ and /aː/ remained about the same but for /ɜː/and /ɔː/ the mean lengths were reduced 1cm and 2 cm respectively. This is possibly due to a raised larynx in the supine position, however the change did not account for all the differences between the AR and MR derived vocal tract lengths.

In a previous comparison between X-ray data and MRI data [4], it was found that the MR data tended to underestimate the vocal tract length. The underestimation was attributed to the post processing method used to obtain the vocal tract shape. However we used a different method to get the shape, so it is unlikely this is the reason. Another consideration is that with the MR imaging, the determination of the precise vocal tract end-point at the lips is difficult because the opening at the lips is not a plane but is curved with some parts of the boundary being effectively longer than others, depending on the vowel (i.e. for /iː/: the corners of the mouth are retracted relative to the front). Whilst this may be a factor, it is important to note that the MRI analysis yielded vocal tract lengths in the expected region of 17 cm [1], whereas the AR derived lengths were longer than expected. In another study of vocal tract lengths measured using the AR technique [7], the vocal tract lengths were also around 17 cm. In that study however the subjects had the vocal tract in a rest position, not a speech like shape.

For all vowels, R1 and R2 from the MR data matched the F1 and F2 from the speech data much better than R1 and R2 from the AR data. For AR, the necessity to form a seal around the wavetube compromised the subject's ability to put his articulators in the appropriate position to say the vowel. But it is also notable that for both the AR and MR data the R1 and R2 values for /ɜː/ and /ɔː/ were higher than expected when looking at the overall vowel space (e.g. Figure 4(a)). Both vowels are produced with rounded lips (the lips are protruded and puckered). Epps and colleagues [11] measured R1 and R2 values for Australian English monophthong vowels using a different technique, and also found that the R1 and R2 values for the lip rounded vowels (such as /ɔː/, /ʊ/)[1] were higher than the would be expected from acoustic formants of Australian English monophthongs (e.g. see [10]). A possible explanation is that the lip rounding is affecting the (acoustic) formants by some mechanism (such as a radiation effect) that is not part of the actual vocal tract resonance.

[1] but note /ɜː/ is not produced with lip rounding in Australian English

CONCLUSIONS

We have compared two measurement methods which enable us to study the vocal tract shape, and compared the resonances obtained from these shapes with formants obtained from acoustic speech recordings. As expected there is reasonable agreement with the first and second formants and the first and second resonances from the vocal tracts measured from the MR data. There was also considerable agreement between the MRI and AR data for the vocal tract shapes of the three of the four vowel studies in the pharyngeal region. However the calculated vocal tract resonances from the AR data are not able to be compared meaningfully to the formant data from recorded speech. The wavetube used in the AR technique appears to compromise the speaker's ability to produce a meaningful vocal tract shape for the vowels where the mouth opening does not closely match the wavetube size. Further, the inability to vocalise whilst the measurement is being taken is also a methodological difficulty. Whilst the data were collected in a very specific manner, which created an optimal environment to get the correct articulator placement, the above two factors mean that speech production data across all vowels can not be collected using the AR technique. Consequently, it seems that acoustic reflectometry has limited used as an articulatory phonetic tool.

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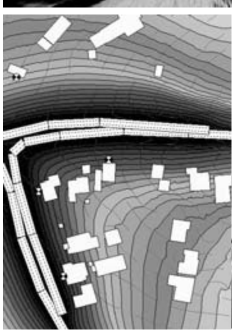

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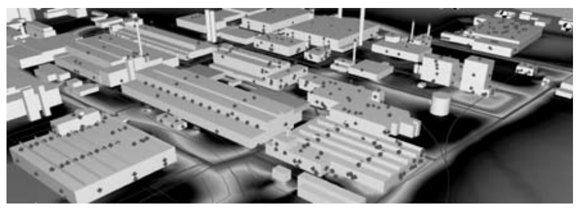

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INFLUENCE OF RESONANCE CHANGER PARAMETERS ON THE RADIATED SOUND POWER OF A SUBMARINE

Sascha Merz, Nicole Kessissoglou and Roger Kinns

School of Mechanical and Manufacturing Engineering, The University of New South Wales, Sydney NSW 2052, Australia

sascha.merz@gmx.net

Sound radiation from a submarine in the low frequency range is mainly due to fluctuating forces at the propeller. The forces arise due to the operation of the propeller in a non-uniform wake and are transmitted to the hull via the shaft and the fluid. The overall sound radiation from the submarine is the combination of sound radiated from the hull and sound radiated from the propeller. A hydraulic vibration attenuation device known as a resonance changer can be implemented in the propeller/shafting system in order to reduce the overall radiated sound power. In this paper, the influence of the virtual stiffness and damping of the resonance changer on the radiated sound power is investigated, where the importance of sound radiation from the propeller and the resulting re-excitation of the hull is of particular interest. Finite and boundary element methods are employed to model the structure and the fluid, respectively.

1. INTRODUCTION

The minimisation of sound radiated by a submarine is a significant research field as the importance of submarine stealth increases with more advanced detection techniques. A significant part of noise radiated from a submarine in the low frequency range can be correlated to the propeller. Broadband noise arises due to fluid flow over a wide frequency spectrum, however, tonal noise is prevalent in the low frequency range as shown in Fig. 1.

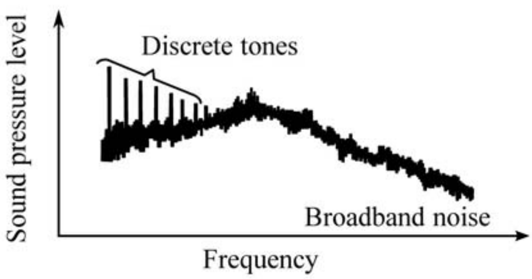

Figure 1. Non-cavitating noise of a submarine propeller [1]

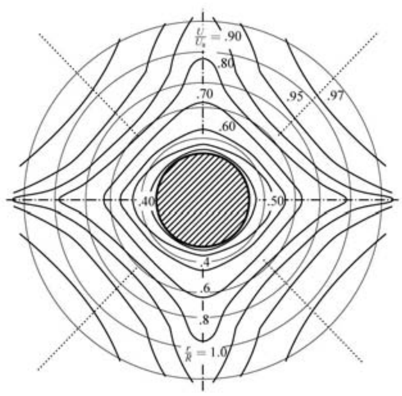

Figure 2. Wake of a torpedo [3]

Figure 3. First axial mode for a cylinder with rigid end plates and two internal bulkheads

The tonal noise can be correlated to the operation of the propeller in a non-uniform wake, as shown in Fig. 2. As the propeller blades pass through sections of different volume flow rate, they experience a temporal variation in drag. This results in a harmonically varying force on the propeller shaft as well as a harmonically varying pressure field originating from the propeller, at the blade-passing frequency (*bpf*) and its multiples [1, 2].

The pressure field as well as the structural force excite axial hull resonances correlated to accordion modes which are efficient sound radiators [4]. The first axial mode of a simplified submarine hull is depicted in Fig. 3. Furthermore, the structural force excites axial vibration of the propeller/shafting system, leading to additional sound radiation from the propeller. The overall radiated sound power is due to the combination of the sound fields radiated from the propeller and the hull.

In order to minimise sound radiation correlated to propeller forces, a hydraulic vibration attenuation device known as a resonance changer (RC) can be implemented in the propeller/shafting system between the thrust bearing and the foundation, as shown in Fig. 4. It detunes the natural axial resonant frequency of the propeller/shafting system and dissipates vibratory energy by hydraulic means. This results in a reduction of axial propeller vibration as well as a reduction of the vibratory energy transmitted from the propeller to the hull. In addition, excessive vibration at hull or propeller/shafting system resonances may be avoided.

Dylejko used analytical models to find optimum parameters for the RC [5]. However, the complex interaction between the propeller and the hull via the fluid has been

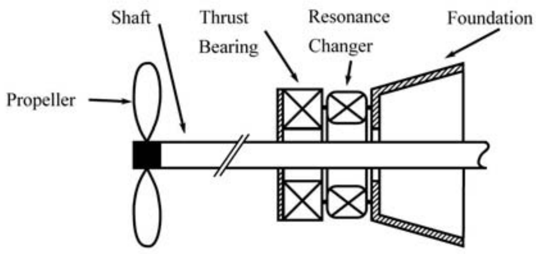

Figure 4. Propeller/shafting system

ignored as the use of analytical models requires numerous simplifications. For example, strong coupling between the structure and fluid cannot be considered for complex geometries. Numerical methods allow for more detailed models. A common approach is to use the finite element (FE) method to represent the structure and the boundary element (BE) method to represent the fluid [6, 7]. Strong coupling between non-matching meshes can be achieved by means of Lagrange multipliers at the fluid/structure interface in order to establish a coupled system [8].

In this work, a cost function is developed to assess the stealth of a submarine. For structural-acoustic optimisation, the integral of the overall radiated sound power over a predefined frequency range is often used [9, 10]. A simplified axisymmetric FE/BE model of a submarine is presented. The cost function is defined in terms of the overall radiated sound power due to sound radiation from the hull as well as sound radiation from the propeller. Scattering and re-excitation effects of the hull due to propeller noise are considered. The propeller is modelled as a rigid disc. The propeller/shafting system is represented by discrete finite elements, whereas the hull is modelled using shell elements based on Reissner-Mindlin theory. The fluid domain is represented using direct boundary elements. Results are presented for the cost function as a function of the stiffness and damping parameters for the RC.

2. PHYSICAL MODEL OF THE SUBMARINE

The simplified physical model of the submarine used in this paper is an extension of the model developed in [11]. In addition to the submarine hull, the propeller/shafting system and the tailcone have been included. The hull is considered as a thin-walled cylindrical shell with two evenly spaced internal bulkheads, rigid end plates at each end of the cylindrical hull and ring stiffeners. In addition, lumped masses are attached at the ends to represent the water in the ballast tanks and free flooded structures. The tail cone of the submarine is modelled as a rigid structure. The on-board machinery is considered as an added mass at the cylindrical shell surface. The model for the hull is depicted in Fig. 5.

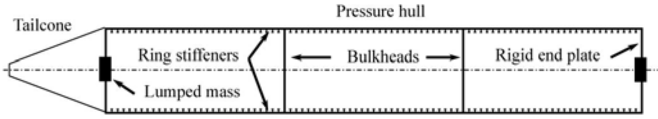

Figure 5. Submarine hull

A modular approach for the propeller/shafting system has been presented by Dylejko *et al.* [12]. The propeller/shafting system consists of the propeller, propeller shaft, thrust bearing, resonance changer and the foundation. The foundation is simplified to an axisymmetric, thin-walled, truncated cone attached to the stern side end plate of the pressure hull. Both the thrust bearing and resonance changer are represented by individual spring-mass-damper systems. The shaft and propeller can be envisaged as a solid rod with a lumped mass attached at the end. The effect of the entrained water at the propeller blades is taken into account as an additional lumped mass. The model for the propeller/shafting system is shown in Fig. 6, where f and v are the axial force and velocity components, respecively. m denotes a lumped mass. c and k are damping and stiffness coefficients, respectively. E and ρ denote Young's modulus and density, respectively. l_s is the propeller shaft length, l_se is the effective propeller shaft length and A_s is the cross-sectional area of the propeller shaft. ν_f is the Poisson's ratio for the foundation, h_f is the foundation shell thickness, a is the foundation minor radius and b is the foundation major radius. 'p', 's', 'b', 'r', 'f' and 'h' denote parameters for the propeller, shaft, thrust bearing, resonance changer, foundation and hull, respectively.

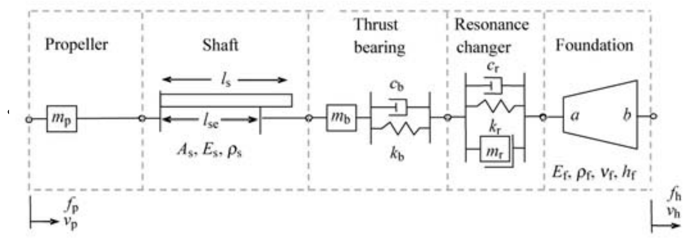

Figure 6. Modular approach for the propeller/shafting system [12]

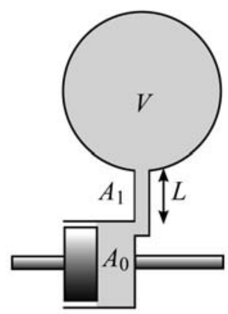

Figure 7. Resonance changer

The RC consists of a hydraulic cylinder that is connected to a reservoir by a pipe as shown in Fig. 7. The geometric properties of the assembly as well as the fluid properties determine the dynamic behaviour of the RC. It can be described as a spring-mass-damper system with the following virtual mass m_r, damping c_r and stiffness k_r parameters [13]:

$$m_\mathrm{r} = \frac{\rho_\mathrm{r} A_0^2 L}{A_1}; \qquad c_\mathrm{r} = 8\pi\mu L \frac{A_0^2}{A_1^2}; \qquad k_\mathrm{r} = \frac{A_0^2 B}{V}. \tag{1}$$

ρ_r is the density of the hydraulic medium, μ is the dynamic viscosity and B is the bulk modulus of the oil in the RC. V is the volume of the reservoir, A_1 is the cross-sectional area of the pipe, L is the pipe length and A_0 is the cross-sectional area of the cylinder.

3. SOUND FIELD RADIATED BY THE PROPELLER

There are two mechanisms involved in the low frequency range that cause sound radiation from the propeller: (i) the operation of the propeller in a non-uniform wake and (ii) axial fluctuation of the propeller blades due to vibration of the propeller/shafting system. The overall sound radiation from the propeller can be simplified to a superposition of dipoles resulting from (i) and (ii), where a dipole is described as

$$p(r,\theta) = \mathrm{j}k\mathrm{g}(r)f\left(1-\frac{\mathrm{j}}{kr}\right)\cos\theta \qquad (2)$$

where k is the wave number, θ is the angle between the field point direction and the force direction, f is the amplitude of the exciting force, r is the distance between the source and the field point and $\mathrm{g}(r)$ is the free space Green's function. The directivity pattern of a dipole is shown in Fig. 8.

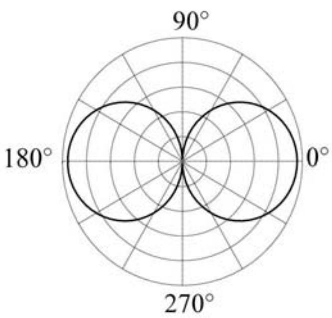

Figure 8. Dipole directivity pattern

A rigid disc approximation can be used to calculate the contribution from (ii). The force corresponding to the propeller axial velocity and the propeller added mass of water are expressd in terms of the radiation impedance as a function of wave number times disc radius [14].

4. NUMERICAL MODELLING

4.1 Sound Power Far Field Approximation

The radiated sound power through a surface Λ in the far field is given by [3]

$$\Pi \approx \frac{1}{2\rho c}\int_{\Lambda} pp^{*}\mathrm{d}\Lambda \qquad (3)$$

where p is the sound pressure, ρ is the density of the fluid and c is the speed of sound.

When Λ is subdivided into polygons and the pressure is expressed as a piecewise quadratic approximation, equation (3) can be represented in a discretised form similar to the procedure described in ref. [15]

$$\Pi = \mathbf{p}_{\Lambda}^{\mathrm{H}}\Theta\mathbf{p}_{\Lambda}, \qquad (4)$$

where $\mathbf{p}_{\Lambda}$ is the vector of pressures in the integration points and the diagonal matrix Θ describes the geometry of Λ and the fluid properties.

4.2 Representation of the Acoustic Domain using BEM

The direct BEM is based on the Kirchhoff-Helmholtz integral equation [16]. An integral equation for a scattering problem can be obtained by using a combination of the Kirchhoff-Helmholtz integral equations for the interior and exterior problems [17]. Let Ω be the exterior acoustic domain and Γ is its boundary, then

$$c(P)p(P) = -\int_{\Gamma}\left(\mathrm{j}\rho\omega v(Q)\mathrm{g}(|P-Q|) + p(Q)\frac{\partial \mathrm{g}(|P-Q|)}{\partial n}\right)\mathrm{d}\Gamma(Q) + p_{\mathrm{inc}}(P) \qquad (5)$$

where P is the field point and Q is the source point, p is the pressure, v is the normal fluid particle velocity and p_{inc} denotes the pressure contribution from a discrete source such as a dipole. For a smooth boundary, $c(P)=\frac{1}{2}$ if $P\in\Omega$ and $c(P)=1$ if $P\in\Gamma$.

For discretisation, the continouous integral equation is tested at a set of points P^{*} on Γ called collocation points, by employing $\delta(P-P^{*})$ as a test funtion [7]. Subsequently Γ is subdivided into elements, where the field variables are elementwise interpolated through the collocation points. This allows for establishing a system of equations by numerical integration over the elements:

$$\mathbf{G}_{\Gamma}\mathbf{v}_{\Gamma} + \mathbf{H}_{\Gamma}\mathbf{p}_{\Gamma} = \mathbf{p}_{\mathrm{inc},\Gamma} \qquad (6)$$

where $\mathbf{v}_{\Gamma}$ and $\mathbf{p}_{\Gamma}$ denote the normal fluid particle velocity and the surface pressure in the collocation points, respectively. The matrices $\mathbf{G}_{\Gamma}$ and $\mathbf{H}_{\Gamma}$ are called 'BEM influence' matrices. $\mathbf{p}_{\mathrm{inc},\Gamma}$ represents the pressure contribution from discrete sources in the collocation points of the surface Γ. For the presented models, $\mathbf{p}_{\mathrm{inc},\Gamma}$ has been evaluated using equation (2) to consider the dipole that is due directly to operation of the propeller in a non-uniform wake.

As the dipole pressure $\mathbf{p}_{\mathrm{inc},\Gamma,\mathrm{prop}}$ due to axial propeller fluctuation depends on the axial surface normal velocity of the propeller, it can be expressed in terms of $\mathbf{v}_{\Gamma}$:

$$\mathbf{p}_{\mathrm{inc},\Gamma,\mathrm{prop}} = \mathbf{G}_{\Gamma,\mathrm{prop}}\mathbf{v}_{\Gamma} \qquad (7)$$

The sparse matrix $\mathbf{G}_{\Gamma,\mathrm{prop}}$ is computed using equation (2) together with the radiation impedance of the propeller [14] and subtracted from matrix $\mathbf{G}_{\Gamma}$ to consider the additional dipole in the coupled system of equations.

The vector $\mathbf{p}_{\Lambda}$ can also be obtained using the Kirchhoff-Helmholtz integral equation by numerical integration, once the pressure and normal velocity on the surface Γ are known:

$$\mathbf{p}_{\Lambda} = \mathbf{G}_{\Gamma\Lambda}\mathbf{v}_{\Gamma} + \mathbf{H}_{\Gamma\Lambda}\mathbf{p}_{\Gamma} + \mathbf{p}_{\mathrm{inc},\Lambda} \qquad (8)$$

where $\mathbf{p}_{\mathrm{inc},\Lambda}$ is the pressure at the integration points of the surface Λ due to discrete sound sources. For the presented models, $\mathbf{p}_{\mathrm{inc},\Gamma}$ has been evaluated using equation (2) to consider the dipole that is due directly to operation of the propeller in a non-uniform wake. The contributions from the dipole that is due to axial propeller fluctuations has been considered by subtracting $\mathbf{G}_{\Lambda,\mathrm{prop}}$ from $\mathbf{G}_{\Gamma\Lambda}$, where $\mathbf{G}_{\Lambda,\mathrm{prop}}$ is obtained according to $\mathbf{G}_{\Gamma,\mathrm{prop}}$ but for the surface Λ.

4.3 Representation of the Structural Domain using FEM

The structure that interacts with the fluid as well as the foundation of the propeller/shafting system is represented by a thin-walled axisymmetric shell of finite elements based on Reissner-Mindlin theory, where transverse shear stiffness is finite [18]. The stress component normal to the shell is assumed to be zero throughout the shell thickness. A simple, one-dimensional rod element has been used to model the section of the propeller shaft between propeller and thrust bearing. Lumped masses in the nodes were utilised to represent the propeller, the mass of the remaining section of the propeller shaft, the RC virtual mass, the mass of the thrust bearing and the lumped masses at the end plates. One-dimensional spring-damper elements were employed to represent the virtual stiffness and damping of the RC as well as the stiffness and damping of the thrust bearing. A detailed description of the aforementioned element types can be found in [19].

Applying the principle of D'Alembert, a finite element formulation for the structural part of the dynamic problem can be obtained. The finite element formulation can be expressed in matrix form

$$\mathbf{A}\mathbf{u} + \mathbf{L}_{\mathrm{sf}}\mathbf{p}_{\Gamma} = \mathbf{f}_{\mathrm{s}}, \tag{9}$$

where $\mathbf{A}$ incorporates the structural stiffness, damping and mass matrices and $\mathbf{L}_{\mathrm{sf}}$ is a geometrical coupling matrix. The vector $\mathbf{u}$ represents the nodal displacement for the FE mesh and $\mathbf{f}_{\mathrm{s}}$ is the load vector of concentrated forces.

4.4 Combined FE/BE Problem

Strong coupling of the acoustic BE and the structural FE models is achieved by imposing the following conditions at the structure/fluid interface; (i) the normal velocity of the structure equals the normal velocity of the fluid and (ii) the normal distributed surface load of the structure equals the acoustic surface pressure. Condition (ii) has already been implicitly considered in equation (9). For non-conforming meshes at the coupling interface, condition (i) cannot be considered in a strong sense. Therefore, an approach similar to that used in ref. [8] is employed, where the pressure can be interpreted as a Lagrange multiplier and continuity of the surface normal velocity is only established in a weak sense [20]. The resulting system of equations can the be written as

$$\begin{bmatrix} \mathbf{A} & \mathbf{L}_{\mathrm{sf}} \\ \mathbf{G}_{\Gamma}\mathbf{L}_{\mathrm{fs}} & \mathbf{H}_{\Gamma} \end{bmatrix} \begin{Bmatrix} \mathbf{u} \\ \mathbf{p}_{\Gamma} \end{Bmatrix} = \begin{Bmatrix} \mathbf{f}_{\mathrm{s}} \\ \mathbf{p}_{\mathrm{inc},\Gamma} \end{Bmatrix} \tag{10}$$

where $\mathbf{L}_{\mathrm{fs}}$ and $\mathbf{L}_{\mathrm{sf}}$ are geometrical coupling matrices.

When the solution vector of equation (10) is known, the pressure vector $\mathbf{p}_{\Lambda}$ can be found using equation (8) and the identity $\mathbf{v}_{\Gamma} = \mathbf{L}_{\mathrm{fs}}\mathbf{u}$, resulting in

$$\mathbf{p}_{\Lambda} = \begin{bmatrix} \mathbf{G}_{\Gamma\Lambda}\mathbf{L}_{\mathrm{fs}} & \mathbf{H}_{\Gamma\Lambda} \end{bmatrix} \begin{Bmatrix} \mathbf{u} \\ \mathbf{p}_{\Gamma} \end{Bmatrix} + \mathbf{p}_{\mathrm{inc},\Lambda} \tag{11}$$

The radiated sound power can then be obtained using equation (4). A cost function representing the sound power over a given frequency range, can be defined as [9]

$$J = \frac{1}{\Delta\omega}\int_{\omega} \Pi \, \mathrm{d}\omega. \tag{12}$$

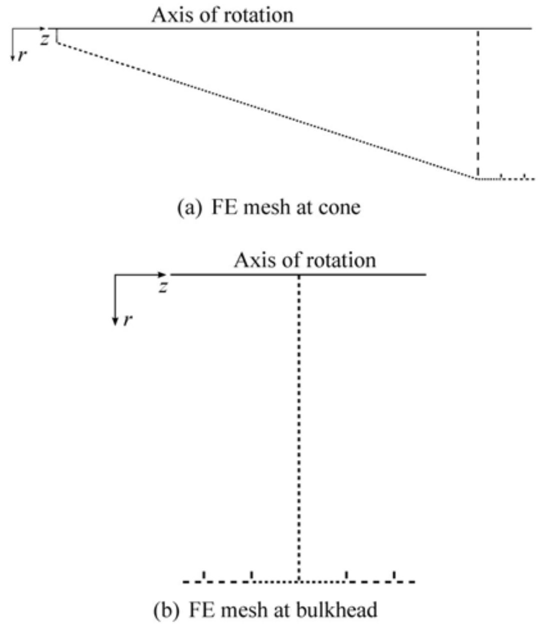

Figure 9. Details for the FE mesh of the submarine hull

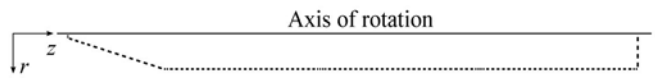

Figure 10. BE mesh of the submarine hull

5. RESULTS

Computations have been conducted for a fully coupled submarine model. ANSYS 11 was used to generate the FE and BE meshes and to compute the FE stiffness, mass and damping matrices. Details of the FE mesh for the submarine hull are shown in Fig. 9. The BE mesh is shown in Fig. 10

For both the FE and BE meshes, at least 10 elements per wave length were used. Computation of the BE and coupling matrices as well as equation solving was conducted by a software developed by the first author using SciPy and C++. Parameters for the propeller/shafting system and hull are given in Tables 1 and 2, respectively, as well as in ref. [14]. Results for the structural and acoustic responses are presented, where a fixed configuration of the RC parameters has been used (section 5.1) and for the cost function as defined in equation (12) (section 5.2). The fixed RC parameters were found by Dylejko *et al.* [12], where a simplified representation of the submarine hull was used and acoustic excitation was ignored. The results for the cost function were obtained by employing a frequency weighted, exciting force in order to take into account that the magnitude of the force is proportional to the square of the propeller rotational frequency. The force amplitude is defined as $(\omega/\Delta\omega)^2$, where a frequency range from 1 to 100 Hz was considered.

5.1 Structural and Acoustic Responses for fixed RC Parameters

The structural and acoustic responses of the submarine hull are presented, where the tailcone was modelled as a rigid structure. Both structural excitation through the

Table 1. Parameters for the propeller/shafting system

Parameter	Value	Unit
Propeller diameter	3.25	m
Propeller structural mass	10000	kg
Propeller added mass of water	11443	kg
Shaft cross-sect. area	0.071	m^2
Shaft length	10.5	m
Effective shaft length	9	m
Resonance changer mass	1000	kg

Table 2. Parameters for the hull

Parameter	Value	Unit
Cylinder length	45.0	m
Cylinder radius	3.25	m
Shell thickness	0.04	m
Stiffener cross-sectional area	0.012	m^2
Stern lumped mass	188×10^3	kg
Bow lumped mass	200×10^3	kg
Cone length	9.079	m

propeller/shafting system and acoustic excitation of the submarine hull have been considered. The acoustic excitation is due to dipole sound radiation caused by operation of the propeller in the non-uniform wake and axial propeller fluctuation due to vibration of the propeller/shafting system. The acoustic response in the far field is a combination of sound radiated from the submarine hull due to structural and acoustic excitation and sound radiated directly from the propeller. An exciting force of 1 N throughout the frequency range has been assumed.

Results for the structural response with and without the acoustic excitation are shown in Fig. 11. The three major peaks at about 20, 43 and 70 Hz represent the first three axial resonances of the hull. It can be seen that there occurs significant re-excitation of the hull due to the propeller sound field. The importance of the sound field radiated from the propeller becomes even more evident for the acoustic response of the submarine, as shown in Fig. 12. The radiated sound power is significantly increased at higher frequencies, when sound radiation from the propeller is taken into account. In addition, a peak can be identified at about 12 Hz that can

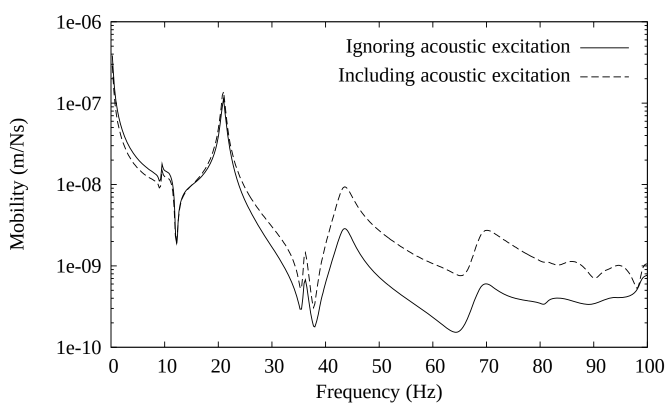

Figure 11. Mobility of the stern side end plate

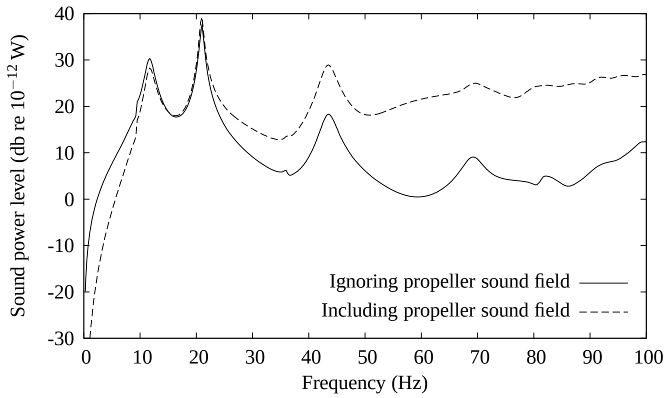

Figure 12. Sound power level for 1 N propeller force

be correlated to the fundamental resonant frequency of the propeller/shafting system.

5.2 Structural and Acoustic Responses for varying RC Parameters

By taking into account physical feasibility of the resonance changer, the RC virtual stiffness was varied between 15×10^6 and $1{,}500 \times 10^6\,\mathrm{N/m}$. A range from 5,000 to 1,100,000 kg/s was chosen for the RC virtual damping. A frequency range between 1 and 100 Hz was considered in order to cover sound radiation that is due to the first four harmonics of blade-passing frequency.

Results for the cost function are shown in Fig. 13. It can be concluded that an increase of the RC virtual damping generally leads to lower values for the cost function J. Two distinct local maxima of the cost function can be identified. The first local maximum occurs at the lower limits $c_\mathrm{r} = 5{,}000\,\mathrm{kg/s}$ and $k_\mathrm{r} = 15 \times 10^6\,\mathrm{N/m}$ for both the RC virtual damping and stiffness. In this case, the cost function is dominated by the sound power due to propeller vibration in the high frequency range, as shown in Fig. 14. This is due to the fact that a decrease of the values for c_r and k_r involves an increase of the propeller/shafting system axial flexibility. The second local maximum occurs at the upper limit $k_\mathrm{r} = 1{,}500 \times 10^6\,\mathrm{N/m}$ for the virtual stiffness and the lower limit $c_\mathrm{r} = 5{,}000\,\mathrm{kg/s}$ for the virtual damping. The cost function is dominated by sound radiation at the fundamental propeller/shafting system resonance. The global minimum occurs at the lower limit $c_\mathrm{r} = 1{,}100{,}000\,\mathrm{kg/s}$ for the virtual damping and $k_\mathrm{r} = 540 \times 10^6\,\mathrm{N/m}$ for the virtual stiffness. For the minimum cost function value, the radiated sound power at the fundamental hull resonance is negligible. Furthermore, the sound power due to propeller fluctuation in the high frequency range is low compared to the sound power for the RC configuration correlated to the first maximum.

6. CONCLUSIONS

A fully coupled vibro-acoustic model for a submarine has been developed in order to find optimum design parameters for a passive vibration attenuation device known as a resonance changer. The objective is to minimise the overall radiated sound power due to propeller forces in the low frequency range. The overall radiated sound power is due to

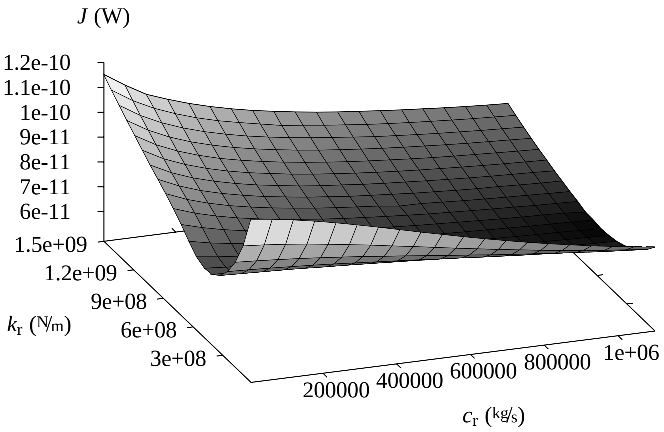

Figure 13. Cost function

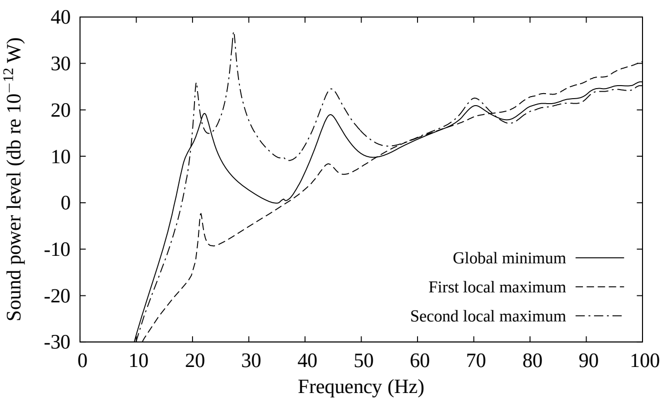

Figure 14. Sound power level for the minimum and maximum cost function values

sound radiated from the hull as well as sound radiated from the propeller, where the importance of sound radation from the propeller was of particular interest. A cost function has been obtained by integration of the radiated sound power over the investigated frequency range.

The structural and acoustic responses of a fully coupled submarine model for fixed and varying RC parameters have been presented. In both cases, there is a significant influence on overall sound radiation and re-excitation of the structure due to the propeller sound field. The variation of the RC stiffness was shown to have a significant effect on the fundamental resonant frequency of the propeller/shafting system. In contrast, an increase of the RC damping leads to a reduction of sound radiation due to axial propeller vibration.

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SAXOPHONE ACOUSTICS: INTRODUCING A COMPENDIUM OF IMPEDANCE AND SOUND SPECTRA

Jer-Ming Chen, John Smith and Joe Wolfe,
School of Physics, University of New South Wales, Sydney 2052 NSW
J.Wolfe@unsw.edu.au

We introduce a web-based database that gives details of the acoustics of soprano and tenor saxophones for all standard fingerings and some others. It has impedance spectra measured at the mouthpiece and sound files for each standard fingering. We use these experimental impedance spectra to explain some features of saxophone acoustics, including the linear effects of the bell, mouthpiece, reed, register keys and tone holes. We also contrast measurements of flute, clarinet and saxophone, to give practical examples of the different behaviour of waveguides with open-open cylindrical, closed-open cylindrical and closed-open conical geometries respectively.

INTRODUCTION

Saxophones are made in many sizes. All have bores that are largely conical, with a small, flaring bell at the large end and a single reed mouthpiece, fitted to the truncation, that replaces the apex of the cone. The soprano (length 710 mm, including the mouthpiece) and smaller saxophones are usually straight. Larger instruments (the tenor has a length of 1490 mm) are usually bent to bring the keys more comfortably in the reach of the hands (Fig 1). The half angles of the cones are 1.74° and 1.52° for the soprano and tenor respectively. These are much larger than the angles of the orchestral woodwinds: the oboe and bassoon have half angles of 0.71° and 0.41° respectively, while the flute and clarinet are largely cylindrical. The relatively large angle of the cone gives saxophones a large output diameter: even the soprano saxophone has a considerably larger end diameter than oboe, bassoon and clarinet. Because of the geometry and other reasons [1], the saxophone is noticeably louder than these instruments, which was one objective of its inventor, Adolphe Sax.

Figure 1. The soprano (bottom) and tenor (top) saxophones used here, shown with a metre rule.

The playing of reed instruments produces coupled oscillations involving the reed and standing waves in the bore of the instrument [e.g. 2–5], and sometimes also in the vocal tract of the player [6].

Many of the important acoustical properties of the instrument's bore can be described by its acoustic impedance spectrum, Z, measured at the embouchure or input of the instrument. For each note, there is at least one configuration of closed and open tone holes, called a fingering, and the impedance spectrum for each fingering is unique. Impedance spectra for a small number of fingerings on the saxophone have been reported previously [7, 8], but measurement technology has improved considerably since then [9, 10]. This paper reports an online database comprising, for each standard fingering on both a soprano and a tenor saxophone, an impedance spectrum, a sound file of the note produced and the spectrum of that sound. It also includes such data for a number of other fingerings. It thus extends our earlier online databases for the flute [11] and clarinet [12]. Beyond its acoustical interest, this saxophone database will be of interest to players and teachers. Another application is intended for the future: The analogous database for the flute was used in the development of 'The Virtual Flute', an automated web service that provides fingering advice to flutists for advanced techniques [13, 14]. A similar service for saxophone could use these data.

We use our experimental measurements to show separately the effects of the bell, mouthpiece, reed, tone holes and register keys. We also include a number of comparisons to illustrate the different behaviours of cylindrical and conical waveguides.

MATERIALS AND METHODS

The saxophones were a Yamaha Custom EX Soprano Saxophone and Yamaha Custom EX Tenor Saxophone: both high-grade models from a leading manufacturer.

The impedance spectra were measured using a technique described previously [10, 12]: see [10] for a review of measurement techniques. Briefly, it uses two non-resonant calibrations and three microphones spaced at 10, 50 and 250 mm from the reference plane (Fig 2). The smallest microphone separation is a half wavelength and therefore measurements are unresolved around 4.3 kHz, which is well above the cut-off frequency of both instruments. Measurements were made between 80 Hz and 4 kHz, which includes the range of fundamental frequencies of both instruments.

The impedance head has a diameter of 7.8 mm, whose cross sectional area is smaller than the internal bore of the mouthpiece, but larger than the average opening between the reed and mouthpiece, through which air flows into the instrument. Using a rigid seal, it was fitted to the tip of the mouthpiece, with the reed removed (Fig 2).

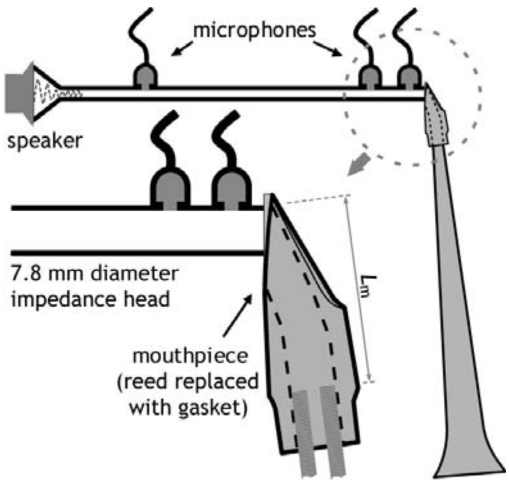

Figure 2. Schematic for measuring saxophone acoustic input impedance using three microphones. Not to scale.

Three different cones with the same half-angle as the soprano saxophone were made. The first was cast in epoxy resin with a length chosen to produce an impedance peak at the nominal frequency of C5 (523 Hz). The second cone was made (from Lexan) to replace the mouthpiece with an appropriate conical section. To allow input impedance measurements, both of these cones were truncated at 7.8 mm (the diameter of the impedance head) and extended with a cylindrical volume equivalent to that of the missing truncated section of cone. The third conical section was made (using a moulding compound) to fit inside the bell and to extend the conical bore over the full length of the instrument.

Sound files and the sound spectra files are not shown here but may be found at www.phys.unsw.edu.au/music/saxophone. They were recorded in the laboratory using two microphones, one placed about 30 cm directly in front of the bell, and the other one metre away. The player is a distinguished soloist who performs principally in jazz and contemporary concert styles. Sound spectra vary with distance from the instrument and direction, so these cannot be considered representative of all possible recordings.

RESULTS AND DISCUSSION

Fig 3 shows the impedance spectra measured on a soprano and a tenor saxophone for their lowest notes, sounding G#2 (tenor) and G#3 (soprano), both of which are written A#3 for these transposing instruments. In both cases, the first maximum determines the played pitch and the next several maxima closely match its harmonics. However, the first maximum is weaker than subsequent maxima, which is not the case for the clarinet [12]. This has the effect of making the lowest notes difficult to play softly, particularly on the tenor.

This weak first maximum of a cone is predicted by explicit models, but can be explained qualitatively if we use the geometric mean of the impedance of two adjacent extrema as an estimate of an effective characteristic impedance $Z_{0\mathrm{eff}}$ for a frequency between them. The characteristic impedance Z_0 associated with the bore cross-section decreases with distance down the bore. At sufficiently low frequencies, the air in the narrow section near the mouthpiece requires only small pressures to accelerate it, so $Z_{0\mathrm{eff}}$ is closer to the Z_0 of the larger bore downstream. The effect is stronger for the tenor saxophone, because its lowest resonances fall at lower frequencies. (Later we show that, for a first maximum at sufficiently high frequency, as in Fig 6 and 7, the first maximum is not much weakened.) The weak extrema at high frequencies are (in part) the results of increased viscothermal losses near the walls and increased radiation at the bell.

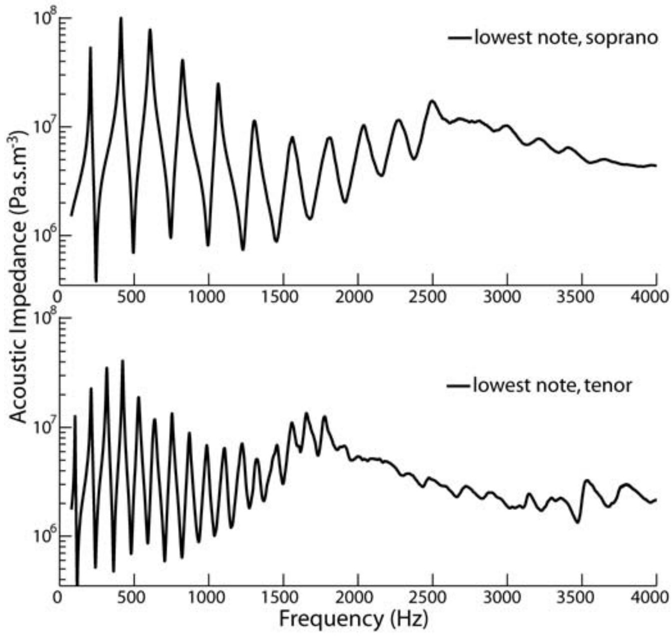

Figure 3. The measured impedance spectra of a soprano and a tenor saxophone for their lowest notes (G#3 and G#2 respectively, both written A#3 on these transposing instruments).

For the lowest note of the soprano, the maxima in Z occur at frequencies 207, 411, 606, 824, 1063, 1305, 1561, 1801, 2038 Hz, all ± 1 Hz. These are a good approximation to a (complete) harmonic series, f_1, $2f_1$, $3f_1$ etc. This contrasts with the case for the clarinet with cylindrical bore, whose impedance maxima occur at the odd members of the harmonic series, with the lowest having a wavelength about $4L$ and a frequency $c/4L$, where c is the speed of sound and L the length of the bore (~660 mm for the clarinet). Consequently, in spite of being 50 mm shorter than the soprano saxophone, the clarinet has a much lower lowest note (sounding D3 compared with G#3). The impedance spectra of the flute (also largely cylindrical, bore length of ~620 mm) are somewhat like that of the clarinet, but for the flute the minima rather than the maxima determine the playing regime, and so its lowest note is C4 – almost an octave higher than that of the clarinet [12]. We compare these three instruments below.

A completely conical bore of length L' (where L' includes the end correction of about 0.6 times the exit radius) would theoretically have maxima in Z at frequencies of about $c/2L'$ and all integral multiples of this. The frequency of the first maximum for the soprano saxophone, f_1 = 207 Hz, agrees well with this prediction. The cone of the saxophone, of course, is incomplete: if it were continued to a point at the mouthpiece, there would be no cross-section for air movement or place for a reed. The cone is truncated at a diameter of 9.2 mm, and the missing cone of length ~150 mm is replaced by a mouthpiece with a volume of 2.25 cm^3. When this is added to the effective volume due to the compliance of a reed (between 1.2 and 1.9 cm^3, discussed later), it is comparable with that of the missing cone (3.35 cm^3). This replacement has the affect of achieving resonances that fall approximately in the harmonic series expected for the complete cone [15, 16]. (The impedance peaks of a simple truncated cone are more widely spaced and not harmonically related.)

Effect of the bell

The loss of structure in Z above about 2.6 kHz for the soprano and above about 1.8 kHz for the tenor is due, in part, to the bell (Fig 3), which enhances radiation at high frequencies; the greater radiation means less reflection and therefore weaker standing waves. Fig 4 demonstrates this by plotting Z for the lowest note on the soprano with the bell replaced by a conical section of equal length and the same half-angle as the saxophone bore. The effective length with the cone is slightly greater, so the maxima appear at lower frequencies.

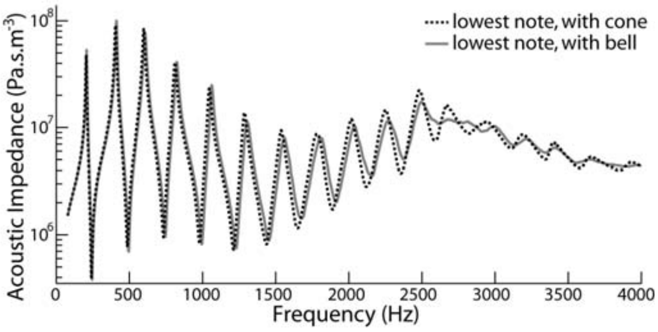

Figure 4. Measured impedance spectrum for G#3 (written A#3) on the soprano saxophone, its lowest note. The solid curve is for the normal saxophone (including mouthpiece and the compliance of the reed). In the dotted curve, the saxophone bell is replaced with a conical section of equal length and the same half-angle as the saxophone bore.

Mouthpiece and reed

The mouthpiece of the soprano has a volume of 2.25 cm^3 and an internal length, L_m, (see Fig 2) of 44 mm. At wavelengths very much longer than its length, it approximates a local compliance, in parallel with the rest of the bore. At low frequencies, the effective characteristic impedance of the bore is low, so the compliance will have only a modest effect, but will lower the frequencies of the maxima. At higher frequencies, it may lower the parallel impedance and, at still high frequencies, it is no longer appropriate to treat it as a simple compliance. The effect of the mouthpiece is shown in Fig 5. Neither the truncation nor the mouthpiece size scales exactly with the size of the instrument. Consequently this effect, which lowers the impedance in the range 1 – 2.5 kHz for soprano and 0.5 – 1.5 kHz for tenor, is not exactly scaled with the octave difference between the instruments (Fig 3).

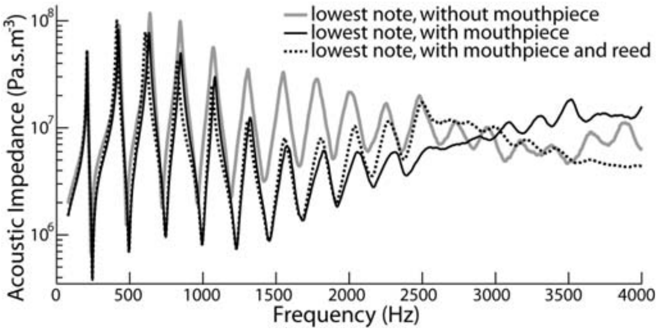

Figure 5. Effect of the saxophone mouthpiece and reed on the acoustic impedance of a soprano saxophone, shown for its lowest note. The dark curve is the measured impedance of a soprano saxophone with the mouthpiece attached, while the pale curve is measured with the mouthpiece replaced with a cone and cylinder. The dotted curve shows the effect of the reed on the acoustic impedance of a normal saxophone (with mouthpiece).

The reed has a mechanical compliance and may, to first order, be replaced by an acoustic compliance in parallel with the input. Its effect is also shown in Fig 5. Reed makers and saxophonists grade the reed according to hardness (a harder reed has a smaller compliance). The reed compliance here is for a number 3 reed. All else being equal, softer (more compliant) reeds lower the frequencies of the peaks in Z, and so play flatter. Of course, all else is not equal: the player may reduce the mouthpiece volume by sliding it further onto the instrument, or may increase the effective hardness of the reed by pushing it harder against the mouthpiece and reducing its effective length. S/he may also change the configuration of the vocal tract.

Bore comparisons

Musicians are generally puzzled by the fact that the (approximately) conical winds behave so differently from the (approximately) cylindrical clarinet. All have a reed at one end, which is therefore a region of large pressure variation, while the bell is open and approximates a pressure node. Yet the clarinet plays about an octave lower than a cone of the same length, its first two maxima in Z have a ratio of three instead of two, and its low notes have predominantly the odd harmonics, whereas the conical instruments have maxima in Z in the ratios 1:2:3 etc and all harmonics are present, even on low notes. Approximating the clarinet as a cylinder of length L with a pressure antinode in the mouthpiece (x = 0) and a pressure node at the bell ($x = L$), it is obvious that a pressure amplitude $p(x)$ of $\cos\{2(2n–1)\pi(x/4L)\}$ satisfies the boundary conditions for integers n, and that it gives a lowest note with a wavelength of approximately $4L$.

To acousticians, the explanation is simple: for the waveguide with constant cross section, solutions to the one dimensional wave equation describe propagating modes, and these are readily written in terms of sine and cosine functions. For a waveguide whose cross section goes as r^2, where r is the distance from the apex, the solution is a sum of spherical harmonics. These include a pressure term proportional to $(1/r)\sin\{2n\pi(r/2L)\}$, which has an antinode at the origin and a node at L for all integers n. Comparisons of the relevant functions are given elsewhere [17], but in this study we are able to give explicit experimental comparisons using the impedance curves.

Fig 6 presents the measured impedance spectra of several bores, each of which has an effective length of about 325 mm. One is a cylinder of that length, one a truncated cone, with the truncation replaced by a cylinder of equal volume. The others are a clarinet with the fingering for the note C4 (written D4) and a flute and soprano saxophone with the fingering for C5 (written D5 for saxophone), the latter using an alternative or

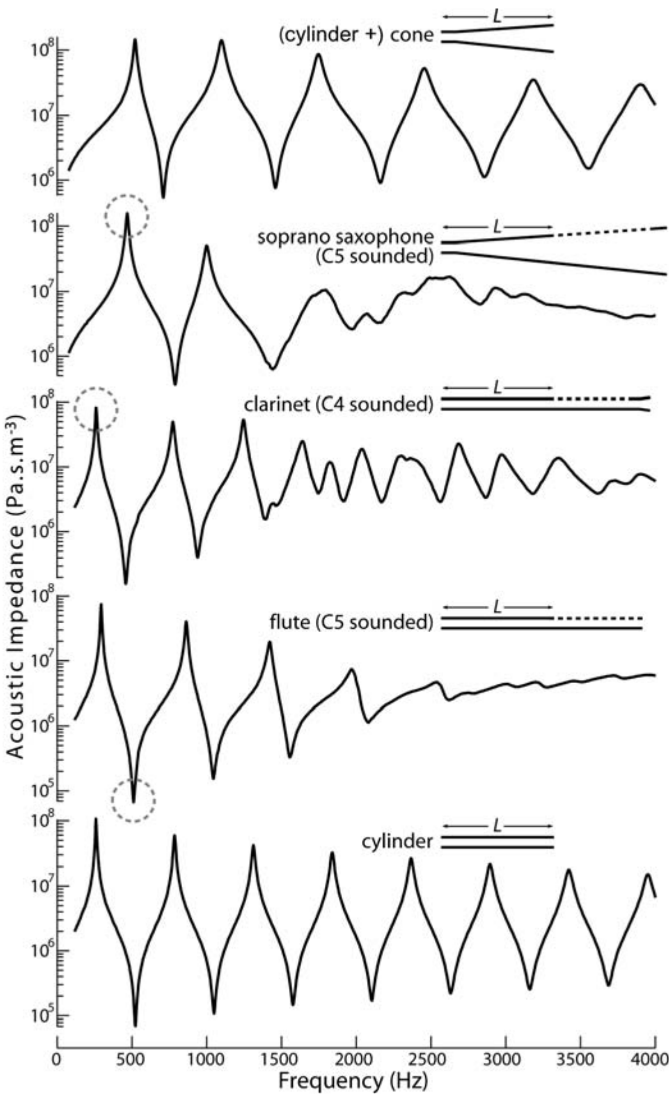

Figure 6. The acoustic impedance of (bottom to top) a simple cylinder, flute, clarinet, soprano saxophone and a truncated cone, all with an equivalent acoustic length. The circle indicates the maximum or minimum where each instrument operates. For the instruments, other notes are readily compared using the online databases reported here and in [11] and [12].

trill fingering that puts that note in the first register: i.e. the lowest frequency maximum in Z is used.

At low frequencies, the cylinder-cone combination has maxima at approximately $n(c/2L)$, which correspond to the minima for the cylinder. (At high frequencies, the replacement of the truncation of the cone becomes important for both the cylinder-cone and the saxophone.) So the saxophone and the flute can play C5 and C6 with these fingerings (although a saxophonist will usually use different fingerings for both notes). The maxima for the cylinder are at approximately $(2n–1)(c/4L)$, so the clarinet can play C4 and G5 with this fingering (although the player will usually use a register key for the latter, as explained below). The behaviour of the instruments at high frequencies illustrates several interesting effects, which we discuss below.

Cut-off frequency

At sufficiently high frequencies, the inertance of air in the tone holes also generates a non-negligible pressure difference between the bore and outside. An array of open tone holes and the short sections of bore that connect them thus resemble an acoustical transmission line comprising compliances and inertances. Above a cut-off frequency, tone holes can also effectively seal the bore from the air outside. Consequently, Z for many fingerings on a clarinet, flute and saxophone show, at sufficiently high frequency, a series of peaks spaced at frequencies corresponding roughly to standing waves in the whole length of the bore, irrespective of what tone holes are open, as shown, for example, in Figs 6 and 7.

The tone hole cut-off frequency, f_c, can be calculated for cylindrical waveguides using either the continuous transmission line approximation [2] or by assuming that the open tone holes approximate an infinite array [18]. Both give $f_c \sim 0.11\,(b/a)c(t_e s)^{-1/2}$, where a and b are bore and hole radii, s is half the separation, t_e is the effective tone hole length including end effects. No comparable expression currently exists for conical bores. Using this expression naïvely for the saxophone gives values of 1340 ± 240 Hz and 760 ± 250 Hz for soprano and tenor saxophone respectively, values that are similar to the frequencies at which there is a sudden change in the slope of the envelope of the sound spectrum [15 and the online database]. These values approximate the frequencies above which the broadly and regularly spaced maxima are replaced by the irregular and more narrowly spaced maxima (Figs 6 and 7, and others in the online database).

The periodic vibration of the air flowing past the reed (and also that of the reed itself) and that of the air jet exciting a flute, are both nonlinear processes, which therefore give rise to a spectrum with many harmonics [4, 5]. Only for notes in the low range of the instrument do several of these harmonics coincide with resonances in the bore. In that range, the spectrum of the clarinet contains predominantly odd harmonics while that of saxophone and flute have all harmonics. In the higher range, there is little systematic difference between even and odd harmonics [12].

Registers

For most standard fingerings in the first register, all tone holes are closed upstream of a point, which determines an effective length of the instrument for that fingering, and most of the tone holes downstream from that point are open. At low frequencies, these open holes approximate a short circuit between the bore and the air outside, effectively 'cutting-off' the bore at the first open tone hole, and producing the first two or more resonances seen in Fig 6.

A flutist can thus play C5 or C6 using the fingering whose impedance spectrum shown in Fig 6 by varying (chiefly) the speed of the jet of air. In clarinet and saxophone, the upper note corresponding to that fingering is selected using a register key.

To play in the second register of the saxophone, one of two register holes is opened. These are holes with small diameter (about 2 mm) and relatively long length (about 6 mm). At low frequencies, they allow air flow. At higher frequencies, the mass of the air in the hole can only oscillate substantially if there is a substantial pressure difference across it. Thus, at high frequencies, the air in a register key effectively seals it [2]. Consequently, when opened, a register hole weakens and changes the frequency of the first impedance peak, making it easier for the second peak to determine the playing frequency. Fig 7 shows the difference. This is how notes in the second register are produced in most reed instruments.

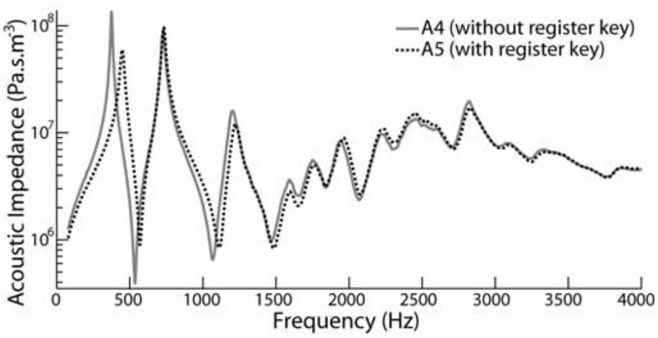

Figure 7. Effect of the register key, shown here for the fingerings for written A4 and A5 (sounding G4 and G5) on the soprano saxophone. The first impedance peak in the spectrum for the A4 fingering is weakened and detuned when the register key is engaged in the A5 fingering, so the reed now operates at the second peak. The high frequency impedance structure is less affected.

On the clarinet, with its largely cylindrical bore, notes in the first and second registers that use similar fingerings are separated by a frequency ratio of three (a musical twelfth) compared with a ratio of two (an octave) for the saxophone. The clarinet uses only one register hole to play the whole second register (which spans 13 semitones). Consequently, the position and dimensions of the register key appear to be less critical for the clarinet – so much so that it also uses this hole as a tone hole for the highest note in the first register.

In contrast, because its two registers are separated by only an octave, the saxophone requires two register holes to cover a second register (which spans 16 semitones). The register hole must be small enough so that it does not affect the second resonance too much. Therefore, to have a sufficiently large effect on the lower register, it must be located where the standing waves in that register have relatively high pressure. Consequently, the lowest seven notes are played using a register hole (2 mm diameter and 6 mm deep on the soprano), which is further from the mouthpiece than the register key used for higher notes. An automated octave key, operated by a system of mechanical logic, uses one key, for the left thumb, to open the appropriate register hole. Inspection of the *Z* plots for the highest notes using the lower register hole (G5 and G#5, sounding F5 and F#5: on the online database, not shown here) shows that it is rather less effective at reducing the first impedance peak than it is for lower notes.

The high range of the saxophone

For the saxophone, the combination of the cut-off frequency and the bore geometry strongly attenuate the magnitude and sharpness of peaks in *Z* above about 1.3 kHz for the soprano saxophone. The weak maxima in this range have important musical consequences. Traditionally, the range of the instrument comprises only two registers, playing notes corresponding to the first and second impedance peaks respectively. This range finishes (depending on the model) at written F6 or F#6 (for the soprano, this sounds D#6 or E6, about 1300 Hz; for the tenor, D#5 or E5, about 650 Hz). The higher resonances will not usually support notes on their own. However, with assistance of sufficiently large peaks in the acoustical impedance of the player's vocal tract, tuned to the appropriate frequency, expert players do achieve notes above the traditional range, in what is called the altissimo range [6].

Saxophone acoustics

Saxophone acoustics provides a range of physically interesting phenomena and musically interesting details [19], including subharmonics and multiphonics, which involve superpositions of standing waves, cross fingerings, and the relations between sound spectra, sound recordings and impedance spectra. These are best explored on-line. The compendium is located at www.phys.unsw.edu.au/music/saxophone.

ACKNOWLEDGEMENTS

We thank the Australian Research Council for support, Yamaha for the saxophones, and saxophonist Sandy Evans.

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

The ICA 2010 will incorporate the AAS 2010 annual conference and so will provide an opportunity to showcase the work that is being undertaken in Australia on a wide range of acoustics and vibration topics. All AAS members are strongly encouraged to put ICA 2010 in their calendar for 23-27 August as well as recommending participation from their national and international colleagues .

If anyone knows of an opportunity for promotion of ICA 2010 either using electronic distribution or for posters and flyers etc at a conference or meeting please contact Marion Burgess (m.burgess@adfa.edu.au) and it can be followed up.

Information on the conference can be found on the web page: **www.ica2010sydney.org**

Marion Burgess, Chair ICA 2010

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AMENDMENT FOR WIND FARMS ENVIRONMENTAL NOISE GUIDELINES

Valeri V. Lenchine

Science & Sustainability Division, Environment Protection Authority, GPO Box 2670 Adelaide SA 5001, Australia

At present, South Australia has a number of proposals to establish new wind farms. The State Strategic Plan encourages the use of renewable energy sources and highlights the need for the development of regulations and guidelines preventing the excessive exposure to noise sensitive areas.

It is important to promote wind farm development using contemporary assessment approaches to preserve the environmental and human well-being. The South Australian Environment Protection Authority (SA EPA) Environmental Noise Guidelines (the Guidelines) aim to assist developers, planning and governmental authorities and the general community to evaluate the noise impact from wind farms. The Guidelines were originally published in February 2003 with the intention to review them in the near future. During 2005 and 2006, preliminary compliance research and two rounds of public consultation were carried out. Responses to the consultation were received from the public, the wind farm industry and acoustic consultants. The preliminary research identified that further investigations regarding compliance checking methodology were required. As a result of the findings and consultation responses, the Guidelines were replaced in December 2007 by Interim Guidelines. The 2007 Interim Guidelines did not contain a compliance checking procedure. Other jurisdictions also intend to issue or to update regulation related to wind farm noise. Working groups have presented a draft Australian Standard and an update of the existing New Zealand Standard NZS 6808 for noise from wind turbine generators [1,2].

As there is no widely accepted regulation with respect to environmental impact imposed by wind farm operation, there are still controversial issues that should be addressed. Concerns expressed by the industry centred on whether the existing compliance methodologies could provide an adequate measure of the true contribution of wind turbine noise emissions against other sources (natural background, farming activities etc). Our recent investigations aim to address this issue.

Contemporary scientific methods and instrumentation can find a way to separate the contribution of a wind farm and other sources. However, a method or procedure included as a part of a statute should meet certain requirements. Preferably, the methods should not require employment of complex measurement techniques, special instruments or advanced post-processing of data.

Our research centred on:

- analysis of international practice for predicting and monitoring wind farm noise;
- noise criteria to estimate the noise exposure;
- peculiarities of the data post-processing for the background noise measurements and compliance checking procedures;
- comparison of the predicted noise levels with results of the case studies; and
- alternative methods for compliance checking procedure.

Our investigation suggests that the conventional correction for background technique [5,8] may be used in cases where the previously measured background data are still valid. Frequently, background monitoring is performed before the wind farm construction. Compliance checking measurements might follow a few years later. Validity of the background noise monitoring becomes very important under these circumstances. In this case, other methods can be considered as an alternative.

The correction for the reported sound power method can be used as an alternative procedure if acquisition of valid background noise levels cannot be arranged. Calculation of the wind farm noise is also possible by correction for the reference point sound pressure level (SPL) method. Implementation of the latest method requires a few conditions to be met, otherwise accuracy of the correction for the reference point SPL is doubtful. For example, Figure 1 shows an increase of the wind turbine generator (WTG) SPL over 20dBA if the wind speed varies from the cut-in speed to 12m/s. It is scarcely possible if the reported sound power varies less than 10dBA at the same time. Most likely that the reference point measurements are significantly affected by the operation of other WTGs, local topography and extraneous noise sources. Two other methods generally demonstrate good agreement with results of the case studies.

During our recent noise monitoring program, a BarnOwl directional monitor was used in addition to noise loggers and wind monitoring stations as a reference instrument since it allows the detection of noise contribution from particular directions with an angle resolution of 5º [3]. Comparison of data measured by the directional noise monitor and the calculated contribution (by different methods) from the wind farm demonstrates reasonable compliance with the correction for the reported sound power method (see Figure 2).

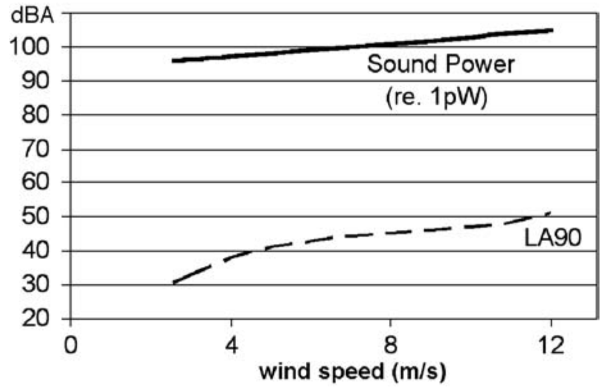

Figure 1 LA90 measurements and reported sound power level at 125m from WTG tower versus wind speed at 10 m height.

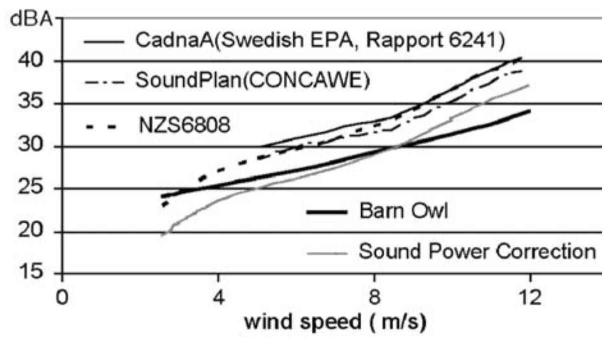

Figure 2 Comparison of wind farm noise measured by the directional monitor and calculated by different methods (versus wind speed at 10m height).

Details of investigations pertaining to the wind farm noise can be found elsewhere [6].

Noise modelling software can incorporate different algorithms to predict wind farm noise. Generally WTGs are represented as elevated point sources. Some researchers state that the ISO9613-2 procedure produces very accurate results for predicting noise impact from the wind farm. However, none of the algorithms can reproduce the pecularities of the noise emission in the source zone. For example, our recent investigations and other research show that the attenuation rate in close proximity to a wind farm can be 2~4dB/doubling of distance [4,7]. None of the commercially available noise prediction software, which utilises the point source representation of a WTG, can reproduce this effect. Generally, the software accuracy for the noise impact from wind farms is unsatisfactory. In practice, a descrepancy up to 10dBA between predicted results and in situ measurements, is not rare. The inability of conventional calculations to reproduce pecularities of wind farm noise immission is due to the fundamental problem of the generator being represented as a point source. This problem should be addressed by the introduction of a more sophisticated model, however, the complications of implementing and applying such a model may be an issue [4,7].

Results of the research are incorporated in the *Draft Wind Farms: Environmental Noise Guidelines*, which are available at www.epa.sa.gov.au/pdfs/. Publication of the final edition is expected soon.

ACKNOWLEDGMENTS

The South Australian Environment Protection Authority thanks the Australian Acoustical Society for contributing education grant funds towards this research.

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Technical note

USING ISO 8253 TO CALCULATE THE MAXIMUM PERMISSIBLE BACKGROUND SOUND PRESSURE LEVELS FOR AUDIOMETRIC TESTING

W. Williams

National Acoustic Laboratories, Chatswood NSW 2067

warwick.williams@nal.gov.au

For many years, various versions of *de facto* standards have been in used in Australia for specifying acoustic conditions in audiometric test areas. These specifications had their origin in combined *Australian/New Zealand Standard AS/NZS 1269.4: Occupational noise management, Part 4: Auditory assessment*, Appendix D, '*Maximum acceptable background noise levels for workplace audiometry programs*' [1]. Although the Standard states that the "*specifications in this Standard are not intended for clinical purposes*" (pg 6), over the last few years in Australia they have been specifically used for that purpose. *ISO 8253 Acoustics – Audiometric test methods* [2] presents internationally accepted maximum permissible ambient (i.e. background, in the ANZ standards) sound pressure levels for Hearing Level (HL) measurements.

If an individual's hearing threshold measurements (HTL) are to be measured, care must be taken to ensure that it is the HTL that is measured and not the background noise of the measurement space. If a test signal is presented to a subject then in order to respond to that test signal the subject must be able to distinguish the test signal clearly from the background noise in the test space. If measuring to 0 dB (HL) is the objective then the background sound pressure level (SPL) must be significantly below the SPL of the applied test signal over the whole frequency range measured and for bands that may mask the measured bands [3].

ISO 8253 – 1, Section 11 *Permissible ambient noise,* provides internationally agreed specifications on the maximum permissible background sound pressure levels desirable for both air and bone conduction audiometry. *Table 2* of ISO 8253 – 1 provides a summary of the "*maximum permissible ambient sound pressure levels*", L_{max}, in one-third octave bands for air-conduction audiometry to 0 dB HL using typically available supra-aural earphones. *Table 4* of ISO 8253 – 1 provides similar information for bone conduction audiometry to 0 dB HL. *Table 1* below summarises these data in octave bands.

Section 11 of ISO 8253 – 1 also indicates that by using the information supplied, the "*maximum permissible ambient sound pressure levels*" required for testing using noise attenuating headsets different to those described in the Standard and to hearing threshold levels other than 0 dB HL can be calculated in a reasonably straight forward manner.

AIR CONDUCTION

Audiometric testing using a noise excluding headset is

Octave band centre frequency (Hz)	*Max permissible sound pressure levels* L_{max} *(Reference 20 Pa)* (dB)				
	Test tone frequency range (Hz)				
	Air conduction audiometry			Bone conduction audiometry	
	125 to 8k	250 to 8k	500 to 8k	125 to 8k	500 to 8h Hz
31.5	52	62	73	47	56
63	38	48	59	30	39
125	23	39	47	17	21
250	18	18	33	11	11
500	18	18	18	8	8
1 kHz	20	20	20	7	7
2 kHz	27	27	27	6	6
4 kHz	34	34	34	2	2
8 kHz	33	33	33	9	9

Note: Using the above values provides an uncertainty of +2 dB due to ambient noise. If an uncertainty of +5 dB for the threshold value is acceptable the L_{max} values in the above table may be increased by 8 dB.

Table 1: Maximum permissible ambient sound pressure levels, L_{max}, for air and bone conduction audiometry for hearing thresholds down to 0 dB HL using typical supra-aural earphones such as the Telephonics TDH39 with MX 41/AR cushions or the Beyer DT48 (adapted from *Table 2* and *Table 4* of ISO 8253 – 1).

sometimes possible when background noise is greater than those levels specified in ISO 8253 – 1. The attenuation characteristics of the headset should come from a well recognised test procedure such as *AS/NZS 1270: 2005 Acoustics – hearing protectors* [4] where the attenuation values used are the mean attenuation minus one standard deviation. For consistency with Australian practice, the parameter L_{max} should be measured with the sound level meter on ‘S’ (slow) time weighting as recommended in *AS/NZS 1269.4: 2005*, Appendix B. If this is the case then the L_{max} parameter should more correctly be specified as $L_{S,max}$.

To calculate the required background levels when using a headset different from those specified by ISO 8253, the process is simply to add the extra attenuation provided by the headset intended for use to that provided by the typical specified headset. Further, if testing to a different minimum threshold is required, not to 0 dB HL, the difference between the new minimum threshold level and 0 dB HL is added to the given permissible maximum background values for 0 dB HL.

BONE CONDUCTION

HTL testing for bone conduction requires unoccluded ears, so noise excluding headsets cannot be used to reduce the background noise. However, testing to a different HL is carried out in a similar manner as for air conduction by simply adding the difference between 0 dB HL and the desired HL.

If both air and bone conduction are to be undertaken the lower background SPL requirements for bone conduction will predominate.

Using this template developed from *ISO 8253 – 1 Acoustics – Audiometric test methods – Part 1* the maximum permissible background sound pressure levels can be easily calculated to ensure satisfactory testing to the desired hearing threshold level to a specified accuracy of either +2 dB or +5 dB for both air and bone conduction.

ACKNOWLEDGEMENTS

The author would like to acknowledge the advice of Mr Dick Waugh in the revision of this work.

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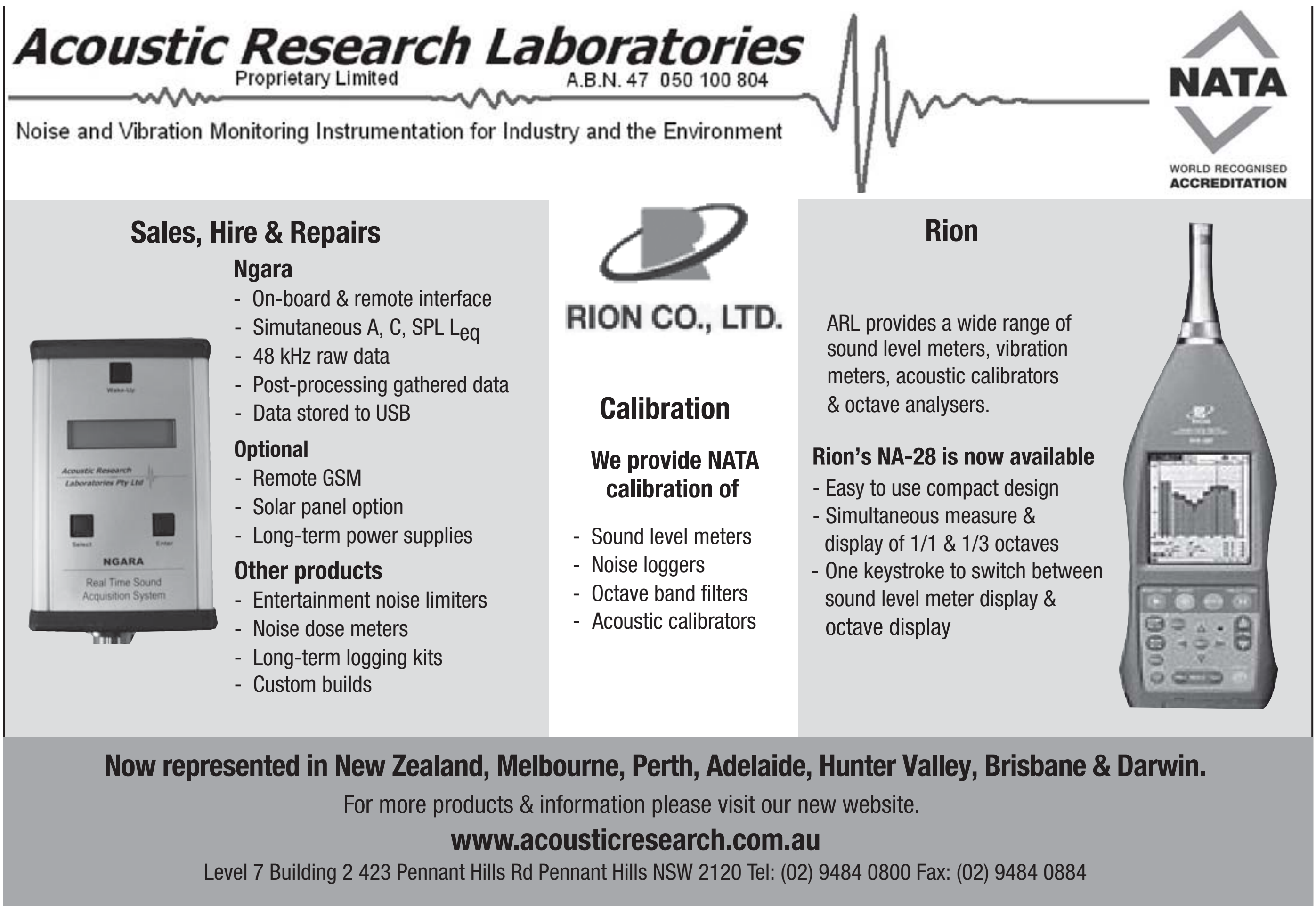

Technical note

A submission on this project won the AA-Bradford Excellence in Acoustics award for 2008

NOISE AND VIBRATION DESIGN OF THE MONASH CENTRE FOR ELECTRON MICROSCOPY BUILDING

Matthew Stead[1], Simon Moore[1], Jon Cooper[1] and Andrew Mitchell[2]

1 AECOM, Level 28, 91 King William Street, Adelaide SA 5000 Australia

2 AECOM, Level 6, 76 Cashel Street, Christchurch 8011, New Zealand

INTRODUCTION

The newly completed Monash Centre for Electron Microscopy (MCEM) building is a central research facility located at Monash University, Clayton Campus, which conducts world-leading research and undergraduate and postgraduate training. Modern electron microscopes can resolve detail at the atomic level, if the effects of building noise and vibration satisfy demanding specifications.

The MCEM building provides a world class low noise and vibration environment to optimise instrument performance and is one of the most stable buildings worldwide.

The key objectives of the noise and vibration design for the MCEM project were:

- Interpretation of the client brief and electron microscope noise and vibration specifications
- Create a world class low noise and vibration environment
- Coordinated design to encompass a range of disciplines
- Quality control inspections throughout construction

SITE

The MCEM building is located on the Monash University Clayton Campus. It is surrounded by other campus buildings, and a nearby university ring road. The ring road has a number of speed humps, which create a source of vibration when vehicles pass over them. Mechanical pumps, fans and other machinery in the surrounding buildings are also sources of vibration.

Further noise sources include vehicles using the university ring road, aircraft flying overhead, air-conditioning plant from adjacent buildings and other general activity on the Monash University Site. Noise levels on site were typically over 60 dB(A) or 70 dB(Lin).

Baseline vibration measurements showed that the ground vibration levels at the proposed site were close to the maximum that would be acceptable if the desired microscope performance were to be achieved. Hence, the building design needed to ensure that vibration levels were reduced, and not amplified in any way, particularly in the low frequency range (< 10 Hz).

All of the environmental noise and vibration sources were carefully identified and the building designed to minimise the effects due to these existing sources.

ELECTRON MICROSCOPE OPERATION

The electron microscope uses high energy electrons whose de Broglie wavelengths are much smaller than the wavelength of visible light. Because optical microscopes are usually diffraction limited, this allows a resolution thousands of times better than a light microscope – with images that can show molecular or even atomic detail.

As shown by Gordon and Dresner [1] electron microscopes are noise and vibration sensitive for a number of reasons:

- Relative motion of microscope components and the sample itself, even on the Angstrom scale, can cause blurring and reduced image resolution.
- Long exposure times are required to receive enough electrons to generate an image at the smallest scales, so noise and/or vibration must be minimised over such time scales.
- The high voltages required for the smallest wavelengths means that a long electron beam column is required. The consequent size of the microscopes lowers the frequency of their mechanical resonances and makes them more vulnerable to low frequency noise and/or vibration.

The maximum sensitivity of electron microscopes to vibration and noise disturbance occurs, typically, when components within the electron microscopes are excited at their resonance frequencies. At these resonances, significant relative movements can occur between components.

The lowest order resonances, those in the frequency range 10 to 50 Hz typically, may be excited by vibration of the floor on which the electron microscope is supported but are not readily excited by acoustic noise, probably because of the poor coupling at these frequencies between the sound field and the electron microscope. In the few measurements that have been made of microscope sensitivity to noise, maximum acoustic sensitivity has occurred in the frequency range 100 to 300 Hz. In this range the potential for efficient coupling between the sound field and the typical electron microscope is much improved. At these higher frequencies, resonances on, or within, the electron microscope are often thought to be the cause of relative movements and operational problems.

NOISE AND VIBRATION CRITERIA

Monash University provided detailed information about the acoustic standards achieved by existing electron microscope facilities and the acoustic requirements of the proposed new electron microscopes. This information was reviewed and the most stringent acoustic criterion in each class of laboratory identified. Figure 1 summarises the noise criteria for the most sensitive A Class of laboratories in one third octave bands. This demonstrates that significant noise attenuation was required, particularly for the low frequency region (< 50 Hz), which was difficult to achieve.

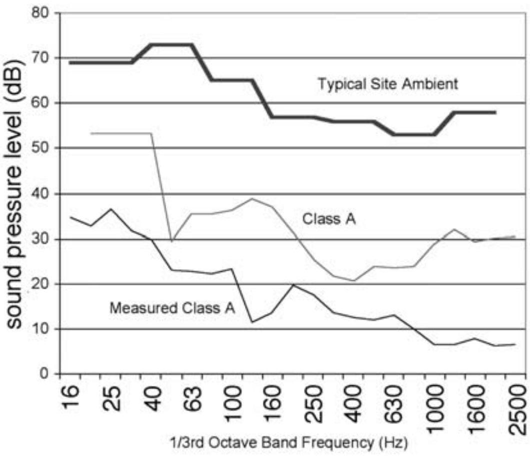

Figure 1: Measured noise levels in the most stringent (designated Class A) laboratory compared with the relevant criteria and ambient noise levels. This shows the significant noise attenuation of environmental and mechanical services noise that was achieved.

The vibration criteria were provided in a similar manner to the noise criteria. This required extensive consultation with the University who were not able to provide equipment details due to non-disclosure agreements with the microscope manufacturers. All data were provided in an anonymous format. Limited information was available on the measurement techniques used to obtain the data, which made it difficult to compare measurements with equipment criteria. As part of the project, a test method was developed to enable appropriate comparison of vibration measurement results with the equipment specifications. The vibration criterion is shown in Figure 2.

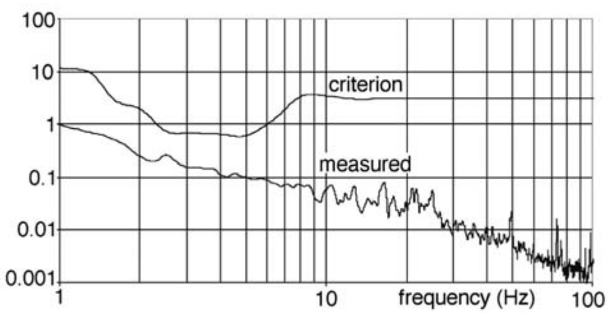

Figure 2. Measured vibration levels compared with vibration criterion for Class A laboratory.

NOISE AND VIBRATION DESIGN

The building had a number of stringent and particular building requirements. These required non-standard constructions to achieve the design intent. As the building contractors were not familiar with these construction requirements, the construction phase inspections were frequent and timed for key stages to ensure that defects did not compromise the noise and vibration design.

Mechanical plant associated with the facility was an additional source of noise and vibration. The mechanical plant was designed to achieve air flow velocities less than 1 to 2 m.s-1 for the most sensitive lab, which required large duct cross-sections. Attenuation of the plant noise down the ducts therefore necessitated long duct runs. Most of the plant was located in a separate building to reduce vibration transmitted to the laboratories. Flexible connections were utilised to minimise vibration transmitted across the isolated slabs and isolated walls. Acoustically lined ductwork formed with steel of increased thickness and acoustic attenuators were used to control noise break-in and break-out from the ductwork entering the laboratories.

Architecturally, the most critical laboratory building structure (a box in a box) incorporates a heavy masonry internal structure (200 mm thick blockwork walls and 300 mm concrete plank roof) and, externally, multiple layers of plywood fixed to the building structure including the walls and roof.

The floor slab in the Class A and B laboratories (the laboratories with the most stringent noise and vibration requirements) is isolated from the adjacent areas. The isolated floor slab was designed such that there resonances or vibration modes are above the critical frequency range. The thickness of the floor slab was the primary variable.

Research has shown [2] that it is difficult to achieve low frequency (< 10 Hz) vibration isolation from the site.

The goal of the isolated floor slabs was to maximise vibration isolation (at higher frequencies) and to ensure that the isolated slabs do not have any resonances or vibration modes, particularly near the critical 4 Hz frequency. The design is indicated in Figure 3.

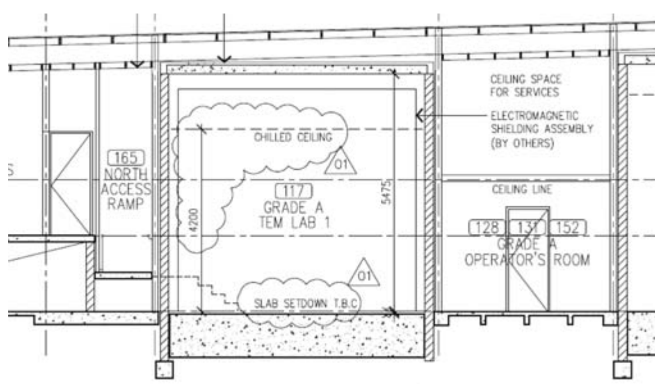

Figure 3: Cross section through an A Class Lab showing isolation between floor slabs, masonry walls and the inner laboratory floor slab.

Based on the floor slab design and measured soil properties, the design concrete slab thicknesses were up to 1000 mm thick.

Mechanical equipment has the potential to cause excessive vibration where the sensitive electron microscopes are to be installed. The major mechanical equipment types were each assessed according to power rating and normal minimum operating speed.

A schedule of vibration isolators was recommended for this project. The schedule included the spring isolator static deflection. The vibration isolator's selection depended on the distance to sensitive areas and the parameters outlined above.

CONCLUSION

An innovative design for noise and vibration was carried out and implemented in the building's construction. The building was commissioned with noise levels lower than those typically observed in a concert hall (< 20 dB(A)) and vibration levels approximately 1000 times less than can be felt. One of the Class A electron microscope laboratories is now fully operational and is operating within the environmental parameters specified as shown in Figure 1 and 2. This allows the very high resolution microscopes to achieve 0.1 nm resolution (approaching atomic scale).

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Figure 4: Site layout during construction showing laboratories (with isolated masonry walls) with isolated slabs yet to be poured.

Letter

SLINKIES AND STAR WARS SOUND EFFECTS

How to teach the dispersion relation in sound transmission? How to get a class interested in the important equation relating angular frequency ω and wave number *k* for all possible vibrational modes? And how to relate sound transmission to the atomic oscillator picture of a solid?

Seeking a demonstration, my mind returned to my childhood, and a fascinating documentary called '*SPFX*' that exposed the cinematic secrets behind George Lucas' "*The Empire Strikes Back*". The most memorable effect was the sound of the laser pistols, generated by hitting the guy-wire for a 60 m radio antenna with a spanner. In a stiff metal wire, transverse waves with higher frequencies travel faster, so the initial impulsive tap of the spanner travels up the guy-wire and back to return as the characteristic 'piow' noise of the laser pistol – a short whistling sound running from high to low pitch. A nice example of dispersion, but how to fit a 60 m radio antenna into a lecture theatre, and how to connect the resulting sound to a dispersion relation?

The solution to the first problem was to use a slinky. After all, a spring is just a huge length of wire conveniently coiled up into a much smaller package. Good results are obtained by suspending the slinky vertically with its free end just resting on the floor. You get the laser pistol sound by touching a microphone to the coil, and giving the bottom of the slinky a quick lift to make the free end tap the floor [1]. Alternatively, you can connect the slinky's top end to a soundboard, which can be as sophisticated as an acoustic guitar body or as simple as a styrofoam cup jammed between the coils.

Linking the sound to the dispersion relation is more complex, but an excellent account is given by Crawford [2]. The stiffness of the wire leads to an added energy cost for transverse waves that induce curvature in the wire. Although this bending energy is negligible in the very long wavelength limit (i.e., frequencies $\omega \to 0$), it isn't at acoustic frequencies, where the dispersion relation becomes parabolic (i.e., $\omega \sim k^2$) rather than linear. The result is an increased phase velocity for higher frequencies, and the characteristic high-to-low pitch laser pistol sound made by tapping the slinky against the floor. As to why a laser makes a noise, I'll leave that to George Lucas.

[1]. For a video of this demonstration, see http://www.youtube.com/watch?v=aqtqiuSMJqM

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Adam Micolich, School of Physics, UNSW

2009 Bradford CSR Insulation Excellence in Acoustics Award

Entries close 30 July 2009

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2009 AAS Education Grant

Nominations close 31 July 2009

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News

Acoustics Australia Archive soon on line

The Society is currently in the process of scanning all the back issues of Acoustics Australia with a view to making them available on line for individual researchers. The policy is that all current members will continue to receive a hard copy of the journal. Electronic versions of the last 3 issues will be available to members at no charge via their restricted area. However, there will be a charge of A$30 per issue for the last 3 issues for non members. Copies of all earlier issues will be available free of charge to all for individual download. The usual copyright requirements would apply for those seeking to make multiple copies of these past issues.

Tax Office ruling for retired members fees

A ruling from the Australian Tax Office of 3 September 2008 confirms that a retired person can claim a tax deduction (under section 25-55 of the Income Tax Act) for payment of membership of a trade, business or professional association up to a maximum of $42. More information on this ruling can be found from www.aip.org.au/news/197. The current AAS retired member fee of only $40 is consequently fully claimable under this ruling, so this is yet another incentive to maintain links with the AAS as a retired member.

Brüel & Kjær and Lochard join forces

Brüel & Kjær announces the acquisition of the Australian company Lochard Ltd. with the intention of providing customers with world-class Environment Management Solutions. Environment Management Solutions (EMS) involve monitoring and management of noise and climate parameters in cities and airports with the purpose of reducing environmental impacts and to ensure compliance of national and international regulations. The new global EMS business will have headquarters in Melbourne and be run by the existing Lochard executive team. The acquisition of Lochard will enable Brüel & Kjær to develop, market, and support an increased product portfolio, including Lochard's NoiseOffice suite of managed noise services and innovative, web-based services.

Lochard has a customer base of over 130 major airports and can deliver innovative solutions that include a complete range of quality services for airport noise management.

For more information about Brüel & Kjær and Lochard, www.bksv.com and www.lochard.com

Aviation Noise Information - Australia

WEBTRACK is an innovative system introduced by Air Services Australia to provide the community with information on where and how high aircraft fly, as well as noise levels of these operations. It allows members of the public access to detailed information on aircraft operations around major Australian airports.

WebTrak is part of the largest integrated noise and flight path monitoring system in the world. It uses information from air traffic control secondary surveillance radars to monitor aircraft within 55 km of the airport up to a height of 3000m above ground level. Aircraft noise data are downloaded daily from noise monitors strategically located about the communities close to the airport.

The information is then displayed on a detailed map (road or aerial) that enables the user to zoom down to their street level. In Current Flights mode you can view current operations (delayed by 40 minutes for aviation security reasons) around the airport. In Replay Mode you can access flight information and noise data for the previous two weeks.

The site uses Lochard software (see earlier news item).

For access to webtrak try www.airservices.gov.au/aviationenvironment/noise/webtrak/

Aviation Noise Information - USA

A new Web site developed through Penn State and sponsored by the Federal Aviation Administration (FAA) offers a new resource for residents, community leaders and airport officials grappling with aviation noise. The site includes educational content, a community forum that provides users an opportunity to participate in a moderated discussion on the Wyle Noise Bulletin and links to airport noise sites, environmental information and other educational resources. NoiseQuest visitors can also find how noise is measured, sources of aviation noise, effects of noise and other related topics. All of the site's information is based on government documents, news articles, research journals and current aviation practices. www.noisequest.psu.edu

National Acoustic Laboratories testing contract to Day Design Pty Ltd

National Acoustic Laboratories (a division of Australian Hearing) is pleased to announce that the contract for the conduct of all commercial acoustical testing in the NAL sound shell, has been awarded to Day Design Pty Ltd, a well-established firm of acoustical consultants in Sydney.

The building industry requires manufacturers and builders to provide sound ratings for internal building partitions (walls and floors) to comply with the Building Code of Australia. In addition, windows and external façade materials are often assessed to ensure adequate reduction of noise from aircraft, trains, road traffic and industry.

Accurate acoustical information is required by architects, engineers and government and local authorities to ensure that buildings meet the requirements of technical specifications and the Building Code of Australia.

The NAL sound shell is a superb facility that allows the accurate measurement of:

- the noise reduction properties of building elements such as walls between apartments, windows and doors.
- the noise emission of machinery such as fans, medical equipment, air conditioners and air compressors.
- the noise reduction performance of acoustically lined ducts and duct silencers.
- the sound-absorptive properties of insulating materials.

The NAL sound shell is comprised of a large reinforced concrete building located in a quiet park environment in Chatswood. Inside the shell a number of large rooms float on steel springs and rubber pads, which exclude all outside noise and vibration, creating a stable acoustic environment inside. It contains twin reverberation chambers, a number of anechoic chambers, a quiet air-flow tunnel and a wide range of precision acoustical instrumentation.

Day Design is a respected firm of professional acoustical engineers and a member of the Association of Australian Acoustical Consultants (AAAC). The company has been operating in Sydney for over 25 years, providing acoustical advice to the building industry. Day Design has a successful record of serving the industry and this contract expands their service to manufacturers and building suppliers. They will work with Australian manufacturers to acoustically test and rate their products.

Australian Hearing staff look forward to a positive relationship with Day Design as the operators of the NAL Sound Shell.

Please contact either Stephen Gauld or Athol Day on (02) 9584 2639 and they will be happy to take your enquiries. Alternatively, see www.daydesign.com.au.

BioLab Davidsons

Biolab Industrial Technologies - Incorporating Davidson Measurement have advised that their fax number has been incorrectly listed in the past.

Australia-Wide the Fax: is 1300 736 755 (International Fax: +613 9763-3832)

The other contact details remain the same: namely tel 1300 736 767 and www.davidson.com.au/

Safe Work Australia established

At the 80th meeting of the Workplace Relations Ministers' Council (WRMC) on 3 April 2009, Ministers noted the creation of the Safe Work Australia Council. Ministers also noted the creation of Safe Work Australia, the independent body that will support the Council.

Through a partnership of governments, employers and employees, the Safe Work Australia Council drives national policy development on OHS and workers' compensation matters and specifically: to achieve significant and continual reductions in the incidence of death, injury and disease in the workplace; to achieve national uniformity of the OHS legislative framework complemented by a nationally consistent approach to compliance policy and enforcement policy; and to improve national workers' compensation arrangements. The Safe Work Australia Council comprises an independent Chair and representatives from the Commonwealth, each State and Territory, employers and unions. More information: www.safeworkaustralia.gov.au/

Pam Gunn Elected as an AAS Fellow

At the November 2008 meeting of the AAS Federal Council, Pam Gunn was elected as a Fellow of the Australian Acoustical Society. The citation reads:

In recognition of her contribution to Occupational Noise, particularly in Western Australia, her dedication in promoting the field of acoustics and her involvement with the Australian Acoustical Society, Pam Gunn has been elevated to the grade of Fellow.

The fellowship certificate arrived in Western Australia just in time for it to be presented to Pam at the WA Division's Christmas function, only to be whisked away again for framing. Pam eventually received the framed version at the Division's technical meeting held at the NVMS training centre in February this year.

Pam's professional career in acoustics started in 1973 where she worked with a private consulting company in the UK and subsequently with the WA Main Roads Department in the field of traffic noise. In 1977 she became a member of the Noise Abatement Advisory Committee (set up under the Noise Abatement Act) acting as an expert in the effects of noise on people. Then in 1979 she became team leader of the WA Health Department's Hearing Conservation Team, providing an advisory service for government departments and small business in hearing conservation matters, establishing occupational noise surveys, educational programs and establishing a mobile hearing test service.

During the 1980's Pam was involved in setting up the Noise Officer Course, an approval scheme aimed at training people to assess noise in industry and produce noise reports. During this time she was also involved in the development of educational material and procedures to support hearing conservation initiatives and in providing technical input into policy bodies on the development and revision of standards and regulations on occupational noise, measurement and control.

During the 90's, Pam continued her work in occupational noise with membership of NOHC Occupational Noise Review Group, and Australian Standards working Group AV3-1-1 providing input into the noise measurement and assessment section of AS1269. She was also the Australian representative at international meetings of government occupational noise specialists in France (1988) and Denmark (1994).

As Senior Scientific Officer (Noise) in the WorkSafe Division, Department of Consumer and Employment Protection, Western Australia, Pam has in recent years maintained active involvement in occupational noise issues as a member of the AV003 Committee on revision of AS/NZS 1269 in 2003/5 and as a technical advisor to NOHSC for an overview document on noise as an occupational disease. She has also served as subject matter expert for noise at the NOHSC Occupational Disease Prevention and Indicators Workshop.

More recently, Pam has been involved in investigating emerging issues and liaising with international researchers on issues including ototoxins, impulse noise, low frequency noise, acoustic shock, and new types of hearing protectors. She is the WA representative on the Australian Safety and Compensation Council's Noise Technical Group that is conducting a major review of the National Code of Practice for occupational noise.

Pam has been an active member of the Society since 1976, and a committee member from 1979 until the early 90's. She served on the organising committee for the Society's 1984, 1990, and 2000 conferences, and was chair of the WA Division from 1984 to 1985. She was also a federal councillor 1984/85, and secretary of the WA division 1990 to 1992. Pam presented papers at the 1984 and 2001 AAS national conference, and the 2007 WA division state seminar.

Pam Gunn receives her framed fellowship certificate from Alec Duncan, chair of the WA Division.

New Products

New miniature accelerometers

Kingdom announce that Dytran Instruments in California have released 20 new sensors that include the following:

• D3092C, a charge mode self generating ultra high temperature accelerometer with a sensitivity of 3.5 pC/g, hermetically sealed stainless steel construction with a 10-32 mounting hole and a 10-32 electrical connector providing an increased MTBF. The 3092C sensors will be useful for general purpose ultra high temperature vibration measurements in Automotive - engine / exhaust analysis and turbine engine vibration monitoring

• D5334: an IEPE voltage mode <20mA ultra high temperature accelerometer with a sensitivity of 10 mV/g with other sensitivities available, 3-bolt pattern mount, an integrated hard line cable assembly, a case isolated hermetically sealed, stainless steel construction and integrated differential amplifier, The D5334 has a low profile and some benefits are the integrated cable and Industry standard tri-bolt mount which enables the ultra high temperature operation and the easily powered by IEPE data acquisition system. Potential applications are in aircraft turbine vibration measurements, industrial turbine vibration measurements, and ultra high temperature general-purpose vibration measurements.

• The Dytran 3224A1&2 Ultra Mini accelerometer weighs just 0.2 gram, has an integrated cable and sensitivities of 2 and 10 mV/g. The 3225Fx Ultra Mini accelerometer at 0.6 grams has a removable cable and a sensitivity of 10 mV/g. Both models are IEPE (voltage mode) sensors and are suitable for response measurement on objects which themselves are light, thin, fragile or delicate and on which a heavy mass in a sensor would be significant, disturbing the natural response of the object. They would also be suitable in medical application where a heaver or larger sensor would be uncomfortable, inconvenient, difficult to attach or disturb the patient unduly. Some models are available with TEDS or for charge mode use.

The catalogue of Dytran miniature accelerometers and pressure sensors can be

studied on the web graphical Product Selection Guide at

http://www.dytran.com/selector/Main.html

Please contact Kingdom Pty Ltd for more information. 02 9975 3272 or

visit www.kingdom.com.au

BSWA sound calibrator

CA111 is a small, battery-operated, sound source for calibrating measurement microphones, sound level meters and other sound measurement equipment and checking the linearity of acoustic equipment. The

Acoustic Indoor/Outdoor Monitoring

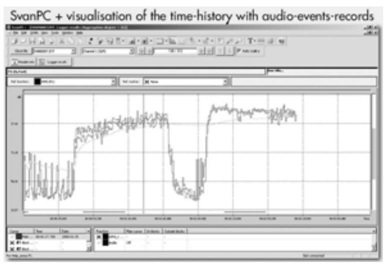
SV 215 Acoustic Indoor/Outdoor Monitoring

With Unique Remote Control Trigger

SV 215 is a complete system for identifying and measuring nuisance noise with the unique capability of performing Audio Events Recording (AER). AER is essential for identification of the related noise source & is accomplished via audio playback.

FEATURES:

- **Audio Event Recording** function with remote manual trigger
- Noise source identification
- Dynamic range 110 dB
- USB Memory Stick logging capability
- Type 1
- 3 parallel independent profiles
- 1/1 and 1/3 octave real-time analysis
- Time-domain signal recording
- Advanced automatic trigger function



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Pyrotek SOUNDGUARD
www.soundguard.com.au

calibrator can be used on 1/2-inch and on 1/4-inch microphones with an adapter. It conforms to IEC 60942:2003 Class 1, ANSI S1.40-1984 and GB/T 15173-1994. The calibration frequency is 1000Hz, which is the standardized international reference frequency, and enables the calibrator to be used for calibrating sound equipment with A, B, C, D weighting and linear weighting filters. The calibration pressure is 94 dB ±0.3 db (1Pa) and 114 dB ±0.3 dB (10Pa).

More information: Kingdom Pty Ltd. tel 02 9975 3272, or www.kingdom.com.au

Improved sound and vibration analyzers

Brüel & Kjær announces another milestone in the development of applications for Types 2250 and 2270 hand-held sound and vibration analyzers. Version 3 is a major advancement with new features and enhancements in almost every application module.

- CF Wireless LAN connectivity making it possible to synchronise measurement and setup data with a host PC
- Broadband Internet communication that can operate through routers and firewalls with security enhancements
- New measurement parameters – Loudness (ISO 532B), Noise Rating (NR), Noise Criteria (NC & NCB), Room Criteria (RC)
- Microphone signal integrity checks utilising Brüel & Kjær's patented Charge Injection Calibration (CIC) technique
- Advanced multilevel and timer-based event triggers and recordings.
- Support of SDHC (Secure Digital High Capacity) memory cards offering up to 32 GB of data storage.

More information from www.bksv.com
or tel +61 2 9889 8888 or email or
auinfo@bksv.com

Noise Monitoring Terminal

Brüel & Kjær announce a new Noise Monitoring Terminal based around Brüel & Kjær's award-winning Type 2250 Sound Level Meter. It includes many features and enhancements such as:

- Secured data 24 hours a day, 7 days a week with extensive battery lifespan
- Automated and live data download over the Internet
- Type approved NMT (IEC 61672-1 sound level meter standard Class 1)
- Removable award-winning Type 2250 which can be used as a stand-alone sound level meter
- Growth path to meet future requirements
- Complies with ISO 1996-2.2 regarding automatic windscreen correction

Using Environmental Noise Management Software Type 7843, the Noise Monitoring Terminal can be controlled by a remote PC, and the Noise Monitoring Software in Type 2250 automatically streams the data to the Environmental Noise Management Software, where the data can be viewed on maps.

More information from www.bksv.com
or tel +61 2 9889 8888 or email or
auinfo@bksv.com

Meeting Report

Environmental Protection Legislation

The Queensland Division recently held a well attended workshop to discuss revisions to the Environmental Protection Act 1994, the new Environmental Protection Regulation 2008 and the new Environmental Protection Policy (Noise) 2008. The event held on, Tuesday 24 March, at the Templeton Room, Tattersall's Club, Queen Street Brisbane and was attended by 46 members and guests.

Three guest speakers presented on the new legislation from a variety of viewpoints.

The first speaker, Frits Kamst, Director, ASK Consulting Engineers, walked the group through the new Legislation from a practitioner's point of view. He highlighted the differences between the old legislation and the new, such as the removal of noise characteristics from consideration and with economic benefits being no longer relevant.

The second speaker, Michael Labone, Barrister, discussed Clause 10(2)(a) of the Noise EPP from a legal perspective, including the Legislative context and specific wording of the clause. The audience found it very enlightening to listen to a legal interpretation of the policy.

The third speaker, Frank Henry, Principal Policy Officer Pollution Prevention, Brisbane City Council, discussed the new legislation from a local government's perspective. He was passionately critical of many of the facets of the new policies, from the lack of consultation to the problems with its application along with disappointment at what it could have been. Many of the issues facing local Councils such as reverse amenity situations, entertainment precincts and future industry, are not addressed in the new legislation.

The night was expertly moderated by Professor Lex Brown, Environmental Planning, Griffith University who not only managed to keep the meeting on time, but also tactfully contributed to the discussion.

The event, a great opportunity to discuss legislation and here the views of fellow members, was enjoyed by all.

Colin Speakman

Brisbane City Planning

Queensland Division held a technical meeting on Tuesday 21st April to hear about possible changes to Brisbane City Council Planning Policies that affect residential development adjacent to designated land transport corridors (major roads and railways). The speakers were Alex Marchuk and Frank Henry, from the Natural Environment & Sustainability Branch of the Brisbane City Council. The event was held at the Queensland Irish Club, Brisbane and was attended by around 40 members and guests.

Alex Marchuk spoke first and gave an overview of the status of the current review of Council Policy, noting that it was being conducted in cooperation with a number of state government departments and instrumentalities with the aim of ensuring that future policies are integrated as far as possible with those of other relevant parties, such as Queensland road and rail authorities. Cooperation with NSW DECC was also noted.

Frank Henry then outlined some of the details of the intended approach. It was made clear that the new approach will provide clear and upfront advice to developers and builders as to generic building design and construction requirements for new residential buildings within the designated corridors. The approach proposed includes generic "deemed to comply" solutions similar to those used by the Building Code of Australia in relation to inter-tenancy walls, with the aim being to improve acoustical outcomes, for residents in the vicinity of land transport corridors, relative to existing development assessment procedures. The system to be employed was compared with the Fortitude Valley Special Entertainment Precinct, another noise affected "zone", recently implemented in Brisbane.

The event, a great insight into the current thinking of the noise policy makers of Brisbane City Council and other relevant agencies, was enjoyed by all.

Matthew Terlich

Conference Report

Internoise2008

Internoise2008 was held in Shanghai, China from 26-29 October 2008 at the impressive Shanghai International Convention Center which is adjacent to the Huang Pu River in the Pudong area of Shanghai and immediately adjacent to the famous Oriental Pearl Tower. With Expo 2010 being held in Shanghai, this meant lots of construction (including a number of tunnels underneath the Huang Po River and 2 new metro lines), sometimes on a 24 hour per day basis. This makes environmental noise a challenging issue.

The theme of the Conference was "From Silence to Harmony". Four distinguished lectures were given in the fields of binaural hearing, real noise and virtual noise synthesis, sound absorption and noise reduction, and acoustical aspects of «Green» office buildings. In line with previous Internoise conferences there was a wide range of acoustical topics, evidenced by the number of parallel sessions (around 10). Basically there is something for everyone, whether you deal with architectural acoustics, environmental noise, noise control, road noise or sound propagation in ducts and pipes, just to name a few of the 45 different topics.

There were quite a number of exhibitors at the Conference, including developers of instrumentation and noise models, as well as a stand for ICA2010, promoting the International Congress on Acoustics 2010, to be held in Sydney in 2010. Approximately 16 Australians, some with their partners, attended the conference. New acquaintances were made, with an «Aussie» table at the congress dinner. The dinner was delightful and the shows put on by the Chinese artists were thoroughly enjoyable, particularly the changing masks. All in all the conference was worthwhile from a technical perspective, as well as providing an opportunity to make new and to renew acquaintances.

Frits Kamst

Future Meetings

AAS Annual Conference 2009

This will be held in Adelaide, 23-25 November 2009, with the theme "Research to Consulting". Keynote speakers include Dr David Rennison, Prof Jie Pan and Dr Brian Ferguson. Papers on all topics in acoustics are welcome and the submission details available from the website. A technical exhibition will be open for the duration of the conference. The social program will include a welcome function, dinner and farewell meal.

Deadlines are abstract submission 24 April 2009, paper submission 29 May and early bird registration fee closes 23 June. For more information on the conference follow the links from AAS webpage www.acoustics.asn.au/joomla/acoustics-2009.html

EURONOISE 2009

The international city of Edinburgh in Scotland is the setting, 26-28 October, for the 8th European Conference on Noise Control organised by the UK Institute of Acoustics on behalf of the European Acoustics Association. Edinburgh is an intriguing host city, rich in history and culture. The conference will be held at the Edinburgh International Conference Centre (EICC), which is in the heart of this dynamic city. With world class facilities, this is the perfect environment for a successful event.

The Conference programme will consist of key-note lectures, invited and contributed papers in structured parallel sessions, workshops, and poster presentations. The deadlines are: abstract submissionby 16 February 2009; paper submission by 30 June 2009. Further information on topics and registration from www.euronoise2009.org.uk

ICSV16

The Sixteenth International Congress on Sound and Vibration (ICSV16), will be held in Krakow, Poland, 5-9 July, 2009, in cooperation with the International Union of Theoretical and Applied Mechanics (IUTAM), the American Society of Mechanical Engineers International (ASME International), and the Institution of Mechanical Engineers (IMechE). The ICSV series of conferences is a major forum for presentation of papers in all branches of acoustics. For more information on keynote speakers and registration details see www.icsv16.org/

Internoise 2009 and Active 2009

The INTER-NOISE 2009 Congress, sponsored by the International Institute of Noise Control Engineering (I-INCE) and co-organized by the Canadian Acoustical Association (CAA) and the Institute of Noise Control Engineering-USA, will be held in Ottawa Canada, from 23–26 August 2009. The congress will feature a broad range of high-level research papers from around the world, as well as an extensive exhibition of noise and vibration control and measurement equipment and systems. Distinguished speakers will provide additional stimulation for our technical sessions and discussions with a focus on our theme of "Innovations in Practical Noise Control." Three key speakers will be Prof. Barbara Griefahn on "Noise induced sleep disturbances and after-effects on performance, well-being and health', Dr. David Quirt on "Controlling air-borne and structure-borne sound in buildings" and Prof. Rajendra Singh on "Gear Noise: Anatomy, Prediction, Solutions"

The 2009 International Symposium on Active Control of Sound and Vibration (ACTIVE2009) will be held 20-22 August, immediately before the INTER-NOISE 2009 congress. The ACTIVE symposia gather together international experts in active control of sound and vibration and are held every 2 or 3 years. ACTIVE 2009 will feature: Steve Elliott on "Active control of vibration in aircraft and inside the ear, " Ken Cunefare on "Distributed active control, Emmanuel Friot on "Estimation and global control of noise reflections" and Chris Fuller on topic to be confirmed. A workshop will also be held on the topic Is there a future for active control?

For information on both conferences

www.internoise2009.com.

Wespac X

The 10th Western Pacific Acoustics Conference (WESPAC X). will be held at the Friendship Hotel, Beijing, China on 21-23 September 2009.. WESPAC is one of the largest acoustical conferences in the western Pacific Area and is held once every three years with the purpose to promote communication and stimulate interactions in acoustics research. All areas pertinent to acoustics from around world are welcome.

As the capital of People's Republic of China, Beijing has a great culture and a long history of over 3000 years. The Conference will provide conference attendees not only an excellent platform for exchanging ideas, but also a unique opportunity to explore the rich culture and visit the attractions of Beijing and other places in China.

Deadlines are abstract submission April 30, 2009 and early registration by May 31, 2009. More information from:

http://www.wespacx.org/

FASTS

FASTS Update

The AAS is a member of FASTS, the Federation of Australian Scientific and Technological Societies which is Australia's peak science body, representing over 60 professional societies and 60,000 scientists and serves you, the AAS and the Australian scientific community in a number of ways. FASTS ongoing activities include 'Science meets Parliament'—FASTS' annual flagship event, where more than 200 scientists have face-to-face meetings with politicians on key science issues.

Other work includes:

- Highlighting science with the Prime Minister and the Cabinet through the Prime Minister's Science, Engineering and Innovation Council (PMSEIC)
- Organising forums and workshops on key 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, and
- Distributing information to member societies weekly, and receiving feedback.

Highlights of 2008 included:

- Forums on 'Rights and Obligations of Scientists and Researchers' and 'Supporting Risk-Aware Research'
- A national roadshow to gather inputs to FASTS' submission to the Cutler Review
- Submissions to reviews on Higher Education Research Training, Future Fellowships, Defence, Higher Education

Endowment Fund, ERA journal ranking, Questacon, CRC
- Continuation of FASTS' successful request for release of ARC grants in early October
- FASTS' statement on climate change – reported in 145 media outlets
- FASTS' Taxonomy paper highlighting this endangered species at SmP 2008.

In 2009 FASTS will:
- Hold 'Science meets Parliament' on 17/18 March
- Provide to Parliament examples of science success stories from FASTS' members
- Present 'On the Radar' briefings on upcoming issues in science that need to be addressed by government, industry and the media – contact FASTS with your ideas
- Contribute to the development of national curricula in science and mathematics
- Investigate whether science graduates have sufficient industry-ready practical skills.

In addition to its continuing and prospective activities FASTS will:
- Establish an expert list of FASTS members for media commentary – via AAS
- Hold a forum on governance of science – how can science self-organise better?

FASTS seeks your help to keep science at the forefront of the national agenda in these challenging times. For more information visit the FASTS' website www.fasts.org <http://www.fasts.org/> . If you have any suggestions on how AAS either can contribute to FASTS activities or can call on the assistance of FASTS please contact the AAS President to discuss president@acoustics.asn.au

Tom Gosling

Researchers in Business

Senator Kim Carr, Minister for Innovation, Industry, Science and Research has announced that great Australian ideas will have more chance of getting to market, with the launch of a $10 million Researchers in Business program. The program will bridge the gap between public sector research and business, giving great ideas more chance of being commercialised at home and boosting innovation in small and medium-sized firms.

For too long, Australia has ranked last in the OECD on collaboration between public sector researchers and industry. This costs us opportunities and leaves us falling further and further behind the rest of the world. Researchers in Business is one of a number of measures to boost the kind of collaboration that will make the most of great Australian ideas and make the most of the taxpayer's investment in higher education.

Researchers in Business will place researchers in small and medium-sized businesses for two to twelve months. The scheme will provide $50,000 per business to employ researchers from universities or public research agencies. This is a marvellous opportunity for innovative firms to develop new commercial ideas while giving Australian researchers a new hands-on career option.

Researchers in Business is one part of the Australian Government's $271 million investment in business growth through the Enterprise Connect network. For more information visit the Enterprise Connect website at www.enterpriseconnect.gov.au

Late News

Revised submission dates for AAS 2009

Abstracts are now due by 22nd May.

The Opening Reception will be held in the National Wine Centre, a showcase for SA Wines.

Sustaining Members

The following are Sustaining Members of the Australian Acoustical Society. Full contact details are available from http://www.acoustics.asn.au/sql/sustaining.php

3M AUSTRALIA
www.3m.com

ACOUSTIC RESEARCH LABORATORIES
www.acousticresearch.com.au

ACRAN
www.acran.com.au

ACU-VIB ELECTRONICS
www.acu-vib.com.au

ADAMSSON ENGINEERING
www.adamsson.com.au

ASSOCIATION OF AUSTRALIAN ACOUSTICAL CONSULTANTS
www.aaac.org.au

BORAL PLASTERBOARD
www.boral.com.au

BRUEL & KJAER AUSTRALIA
www.bksv.com.au

CSR BRADFORD INSULATION
www.csr.com.au/bradford

EMBELTON
www.embelton.com.au

ENERFLEX ENVIRONMENTAL
www.enerflexglobal.com

GEBERIT
www.geberit.com.au

NSW DEPT OF ENVIRONMENT & CLIMATE CHANGE
www.environment.nsw.gov.au

PEACE ENGINEERING
www.peaceengineering.com

PYROTEK SOUNDGARD
www.soundguard.com.au

SINCLAIR KNIGHT MERZ
www.skm.com.au

SOUND CONTROL
www.soundcontrol.com.au

SOUND SCIENCE
www.soundscience.com.au

VIPAC ENGINEERS AND SCIENTISTS
www.vipac.com.au

Diary

2009

18 – 22 May, Portland
157th Meeting of the Acoustical Society of America
http://asa.aip.org/meetings.html

18 – 22 May, Copenhagen
15th International Conference on Auditory Display (ICAD)
http://www.icad09.dk

24 – 27 May, Zakopane Poland
MARDiH 2009: 9th International Conference on Active Noise and Vibration Control Methods,
www.vibrationcontrol.pl

17 – 19 June, Aalborg
3rd International Conference on Wind Turbine Noise
http://www.windturbinenoise2009.org

21 – 25 June, St. Petersburg
13th International Conference "Speech and Computer."
www.specom.nw.ru

22 – 26 June, Nafplion
3rd International Conference on Underwater Acoustic Measurements: Technologies and Results.
www.uam2009.gr

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

20 – 24 July, Porto, Portugal
Symposium on Vibration and Structural Acoustics Analysis
http://paginas.fe.up.pt/clme/IRF2009

12 - 16 August, Jyväskylä, Finland
ESCOM 2009: 7th Triennual Conference of the European Society for Cognitive Science of Music
http://www.jyu.fi/hum/laitokset/musiikki/en/escom2009

20 – 22 August, Ottawa
ACTIVE2009: The 2009 International Symposium on Active Control of Sound and Vibration
http://www.internoise2009.com

23 – 28 August, Ottawa
INTER–NOISE 2009
http://www.internoise2009.com

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

23 – 25 September, Beijing
WESPAC X: 10th Western Pacific Acoustics Conference
www.wespacx.org

23 – 25 September, Xi'an
Pacific Rim Underwater Acoustics Conference (PRUAC)
lfh@mail.ioa.ac.cn

5 – 7 October, Tallinn
International Conference on Complexity of Nonlinear Waves.
http://www.ioc.ee/cnw09

26 – 28 October, Edinburgh
EURONOISE 2009 "Action on Noise in Europe"
www.euronoise2009.org.uk

23 – 25 November, Adelaide
AAS Annual Conference 'Research to Consulting'
www.acoustics.asn.au

2010

15 – 19 March, Dallas
International Conference on Acoustics, Speech, and Signal Processing.
http://icassp2010.org

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

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

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

26 – 30 September, Makuhari
Interspeech 2010 - ICSLP.
http://www.interspeech2010.org

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

2011

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

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

04 - 07 September, Gdansk
2011 ICU International Congress on Ultrasonics.

4 –7 September, Osaka
INTER-NOISE 2011

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

2009 AAS Education Grant

Nominations close 31 July 2009
www.acoustics.asn.au/joomla/notices.html

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