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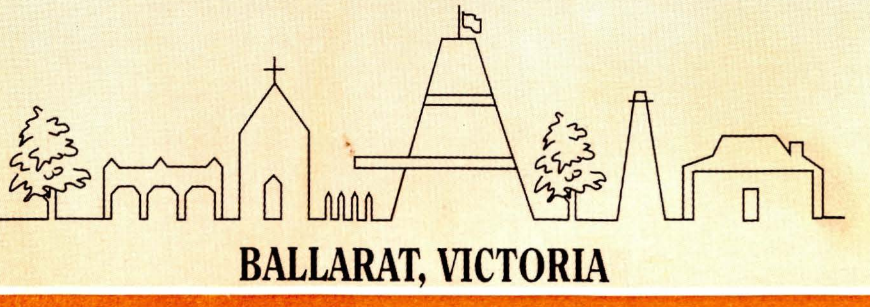
PRACTICAL ACOUSTIC SOLUTIONS

PROCEEDINGS



BALLARAT, VICTORIA

25–27th November, 1992

W. Williams

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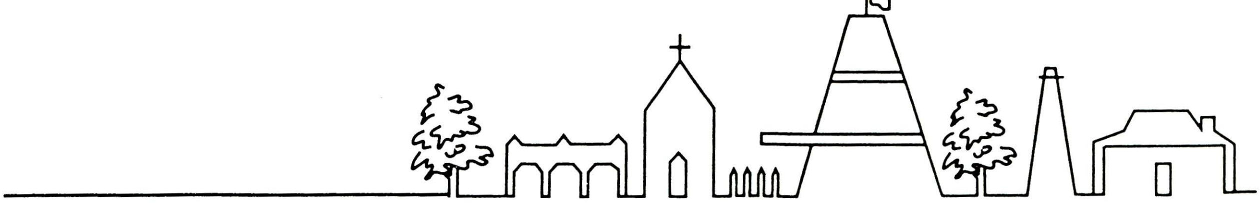

BALLARAT, VICTORIA

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The principal activities of the Society are the technical meetings held by each State Division, the Annual Conferences which are held by the State Divisions in rotation, and the publication of a journal.

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PRACTICAL NOISE CONTROL METHODS

FOR ALUMINIUM CUTTING SAWS

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Abstract

Despite the considerable body of research into engineering noise control methods for preventing noise-induced hearing loss in the workplace, there appears to be limited implementation of these techniques in industry.

An applied research project, funded by Worksafe Australia, was developed to address this "knowledge-action" gap in the particular case of aluminium extrusion saws. A second-hand saw was purchased and reconditioned and a range of engineering noise controls fitted to it, resulting in noise reductions of up to 12 dB(A).

As part of the promotional programme, the treated saw has been set up as a "hands on" demonstration at the "DOHSWA/TAFE Engineering Noise Control Centre" in Perth.

Introduction

The project[1] aims were as follows:–

(i) To identify and apply practical engineering noise control techniques for aluminium cutting saws, appropriate to W.A. industry and capable of implementation in a wide range of applications throughout Australia.

(ii) To develop a promotional programme based on a full sized working circular saw, fitted with the optimised engineered noise control treatment, with a view to achieving a high rate of implementation of the treatment in W.A. and providing the basis for effective implementation elsewhere.

Broadly the project was structured in three phases, as follows:–

(i) Literature search and industry survey.

(ii) Design of engineering noise controls and construction of a working demonstration.

(iii) Development of a promotional programme and evaluation.

Industry Survey

A total of 52 circular saws were surveyed in 16 workplaces in the Perth metropolitan area. Most of the workplaces were members of the Architectural Aluminium Fabricators' Association of Western Australia. The information gathered from the survey questionnaire was tabulated and analysed in terms of saw types and uses.

Noise levels were measured at the operator's position when cutting, and a summary of these results, in terms of $L_{Aeq,10s}$ levels measured over one cut, are presented in Table 1.

Table 1: Summary of Noise Levels from 52 Saws Surveyed

Noise Level Range LAeq,10s,dB(A)	Number of Saws (n)	Percentage (%)
80–84	2	4
85–89	4	8
90–94	14	27
95–99	18	35
100–104	12	23
105–109	2	4
Total	52	101

The results in Table 3.5 show that the vast majority (84%) of saws surveyed generated noise levels at the operator's ear in the range 90–104 dB(A). Two saws were above this range, while six (mostly treated for noise in some way) were below 90 dB(A). The median $L_{Aeq,10s}$ value was 96 dB(A).

In general, it was found that there was considerable potential for further reduction in noise levels, with most companies having done little effective work in this area.

Predominant Noise Sources in Aluminium Cutting

From the literature search it was generally recognised that the main sources of noise in aluminium extrusion sawing are:-

- blade vibration (both forced and resonant);
- workpiece vibration (both forced and resonant);
- aerodynamic noise from the blade.

The factors which influence saw noise may be summarised as follows:-

- blade rotational speed and feed rate;
- blade geometry, stiffness, damping and mounting;
- tooth number, geometry and condition;
- gullet geometry;
- workpiece geometry, clamping and damping.

Purchase and Reconditioning of Project Saw

A second-hand saw, MEP SL 350, approximately nine years old, was purchased for the project for $1300. This saw was typical of the most common in use, i.e., manual docking action, 350mm blade, belt driven from a two-speed 3-phase supply, with mitre cutting facility in two planes, manually operated adjustable workpiece clamps and a compact base. The blade was a typical aluminium cutting type with 84 tungsten-carbide-tipped teeth cut in a "triple chip" profile and with six plugged expansion slots.

The saw was in poor condition and required considerable reconditioning work, such as cleaning and repainting the many rusted parts, and replacing drive belts and other fixtures. The blade was cleaned and sharpened by a local saw doctor.

A series of baseline noise level tests were conducted on the saw in its free–running mode in order to minimise mechanical noise, and the results are presented in Table 2.

Table 2: Results of Baseline Noise Level Tests on Saw Machine (Idling)

Test Condition		Noise Level – dB(A) Low Speed	High Speed
i)	Motor only, original bearings	60.5	70.5
ii)	Motor and shaft, no blade, orig bearings	76	85
iii)	All bearings replaced, no blade	60.5	70
iv)	As (3), plain disc	60.5	72
v)	As (3), standard blade	70.5	87.5

Note: All tests at "operator" left ear position 1.5m above ground 0.4m out from saw base, with saw in "up" position and with the lower guard removed.

The noise level tests (without blades) showed a dramatic reduction of approximately 15 dB(A) when the motor and shaft bearings were replaced. The residual noise levels were dominated by motor fan noise (Test 3). There were no significant noise increases with the machine loaded by a balanced blank disc (Test 4), apart from a small increase at high speed attributable to disc resonance. Installing the "triple chip" blade (Test 5) increased the noise levels to 87.5 dB(A) at High speed and 70.5 dB(A) at Low speed, indicating that aerodynamic blade noise dominated over machine noise in the idle condition. Installation of the lower guard resulted in a further increase of about 3 to 5 dB(A) in noise level at High speed, possibly due to acoustic reflections.

Engineering Noise Control Measures

Various noise control measures were identified from the literature and from the project team's experience. These measures were implemented on the project saw and a series of repeatable tests carried out to determine any changes in noise levels.

For these tests, the saw was located in a typical workshop, at the same position, and a 100mm piece was cut from a 900mm long extrusion, Alcan E–66. Measurements of $L_{Aeq,10S}$ levels was taken at three positions as shown in Figure 1. The arithmetic average of the noise levels were taken over three cuts, all by the same operator using a "normal" stroke.

A subsequent series of tests was carried out in the new "DOHSWA/TAFE Engineering Noise Control" workshop at Leederville College of TAFE, in a similar type of location to the original workshop. Typical differences between the two workshops were determined in a series of tests under the same saw configuration to be approximately ± 1 dB(A) at High speed and results were normalised to the original workshop situation. Signals were also recorded on tape at two positions for subsequent analysis on an FFT analyser.

A summary of the results is presented at Appendix 2.

The results are discussed below in relation to the various noise control measures implemented:-

(i) **Blade Selection**

In all, five blades have been tested. Three of these were tungsten-carbide-tipped (TCT) blades while two were semi-high-speed steel (SHSS) ("cold sawing") blades. Details of the blade and tooth geometry are given in Appendix 1.

Blade TCT-2 caused severe noise and vibration on entry to the workpiece, primarily because of its much lower tooth passing frequency, and its use was discontinued.

Blade SHSS-1, a second-hand blade, exhibited problems in the form of a screeching noise due to inadequate taper at the sides (due to wear) and consequent scraping against the workpiece, so its use was also discontinued.

The use of a semi-high-speed-steel blade (SHSS-2), provided an overall reduction of about 5 dB(A) when compared with the tungsten-carbide-tipped, blade (TCT-1), originally supplied with the saw. This can be attributed to:-

- greater number of teeth (180 vs 84);
- reduced kerf width (2.7mm vs 3.5mm);
- lower aerodynamic noise (approx, 15 dB(A) lower) due to smaller gullets.

It should be noted that, while the SHSS blade produces slightly less noise than the TCT blade, it can only be used on non-anodized section, and is therefore limited in its application.

Recently, an improved TCT blade was investigated in the form of a DIMAR blade with 108 teeth (blade TCT-3). This was found to be significantly quieter than blade TCT-1 and only approximately 1-2 dB(A) noisier than blade SHSS-2, consistent with the factors listed above.

As a variation, this blade (TCT–3) was reground so that all teeth were trapezoidal, not just alternate teeth, the intention being to reduce noise by further decreasing chip size. Noise levels and chip sizes actually increased under this configuration, however, and the original tooth configuration, i.e., alternate square/trapezoidal teeth is preferred.

Blade speed was noted to be significant in that noise levels on Low speed (1700 rpm) were typically up to 5 dB(A) higher than those measured on High speed (3400 rpm), attributable to the greater tooth passing frequency speed and consequent smaller chip size on High speed.

(ii) **Blade Clamping**

Research by Stewart[2] indicated that significant reductions in blade vibration could be achieved through the use of appropriate damping collars, preferably covering about half the blade diameter, with a viscoelastic interlayer. Accordingly, collars were machined from sections of sawblade steel, of similar thickness and stiffness to the sawblade, with a slight concavity to provide a positive grip on the blade. A new drive shaft was manufactured to accommodate the additional thickness of metal and to facilitate changeovers for demonstration purposes.

The viscoelastic material was Dyad 606, 0.5mm thick, adhered to the collars over approximately the outer two thirds of the radius. The overall radius of the collars was 87.5mm, or half the blade radius. This resulted in a slight decrease in available depth of cut from about 110mm to 85mm. A schematic arrangement is shown in Figure 2.

The noise levels measured with blades TCT–1 and SHSS–2 were approximately 1–3.5 dB(A) lower in the damped condition, on High speed, than in the undamped position.

iii) **Workpiece Clamping/Damping**

Studies by Stewart[3] and others indicated that noise radiated from the workpiece is often a significant component of the overall noise. Consideration was given to a number of possible schemes for providing clamping (to reduce forced vibration at the tooth passing frequency and harmonies) or damping (to reduce resonant vibration of the workpiece).

A simple clamping system was developed, in which the existing metal clamping piece was replaced by a timber section, profiled to match the extrusion profile, with a matching timber section mounted onto the back fence of the saw, as shown in Figure 3. While this scheme initially limits the saw function, in that only one extrusion can be accommodated, it would clearly be possible to fabricate several timber sections to fit a range of extrusions. (The industry survey showed that about half the saws inspected were used for cutting ten extrusions or less).

With the SHSS–2 blade, changing from standard metal clamps to timber clamps resulted in a small reduction of 0–1 dB(A) at High speed, suggesting that the predominant residual noise may be from the blade. With TCT–1 blade, the timber clamps resulted in a 0.5–1.5 dB(A) increase in noise levels at High speed (and a reduction of 0–3 dB(A) at Low speed). One possible explanation is that the primary radiation from the workpiece may be from the horizontal surfaces which are clamped by the metal clamps, but are free to vibrate under the timber clamps which contact the vertical surfaces only. Further, the resonant and forced vibration components on these surfaces may be present in differing degrees at the two saw speeds.

iv) **Summary**

The change of blade from TCT–1 to TCT–3, combined with blade damping and workpiece clamping treatments resulted in an overall reduction of approximately 6 dB(A) at the three positions at High speed.

The use of a semi–high–speed–steel blade (SHSS–2) resulted in an overall reduction of approximately 7–8 dB(A) at High speed when compared with the original blade and original blade and workpiece clamps. If one compares these results for the High speed with the original results for Low speed, reductions of up to 12 dB(A) have been achieved.

Further reductions may be achieved through addition of resonant damping treatment to the workpiece, improving the present clamping system to engage both horizontal and vertical surfaces, and use of an acoustical enclosure.

Promotional Programme

The promotional programme is based at a new joint facility established in 1991, known as "DOHSWA/TAFE Engineering Noise Control". Located at Leederville TAFE in Perth, the facility comprises a workshop for construction of noise control demonstration items and a display area for conducting seminars on engineering noise control using the practical demonstration items, including the quietened aluminium saw.

The saw has been promoted through an initial launch to the industry and the media, followed by its use in seminars for Approved Noise Officers, persons from workplaces, DOHSWA Inspectors and others. While copies of the Research Report[1] have been circulated to interested industry members, a simpler pamphlet has been produced to present the most relevant findings in summary form.

It is proposed to follow up the promotional phase with a targeted inspection campaign during which workplaces with aluminium cutting saws will be visited to assess the effectiveness of implementation of this work.

Conclusion

Following a survey of saw usage in the aluminium fabrication industry, a typical saw was purchased, reconditioned and treated to reduce its noise levels. The noise reduction treatments applied to the saw included selection of appropriate blade parameters and blade speed, damping of blade vibrations and clamping of the workpiece to reduce its vibration. Reductions in noise levels of up to 12 dB(A) were achieved by these methods. Further reductions may be achieved through refinement of the workpiece clamping system, addition of damping to the workpiece and enclosure of the saw.

A promotional programme is currently in progress to demonstrate the saw to relevant persons and groups, with a view to the effective implementation of these and/or other control measures in workplaces in Western Australia and elsewhere.

Acknowledgement

The assistance of members in the project team is gratefully acknowledged, especially Jack Etherington, Ellen MacKenzie and Renata MacMillan.

The support of Worksafe Australia in funding the Project and the Department of Occupational Health, Safety and Welfare in Western Australia in providing resources for the project are also gratefully acknowledged.

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Table 6.1: Details of Blades Tested

Blade	TCT-1	TCT-2	SHSS-1	SHSS-2	TCT-3
Diameter	350 mm	250 mm	350 mm	350 mm	350 mm
Thickness	3.0 mm	2.0 mm	3.0 mm	2.7 mm	2.6 mm
No. of teeth	84	12	200	180	108
No. of expansion slots	6 (with copper rivets)	0*	0	0	6
Depth of slots	38 mm	–	–	–	36 mm
Gullet depth	10 mm	–	3 mm	3 mm	8 mm
Gullet width (at periphery)	7 mm	–	3 mm	3.2 mm	4 mm
Tooth const.	TCT	TCT	SHSS	SHSS	TCT
Tooth profile	Square/ trapezoidal (trap.)	Square	Alternate top bevel	Square/ trap.	Square/ trap.
Kerf	3.5 mm	3.15 mm	3.0 mm	2.7 mm	3.15 mm

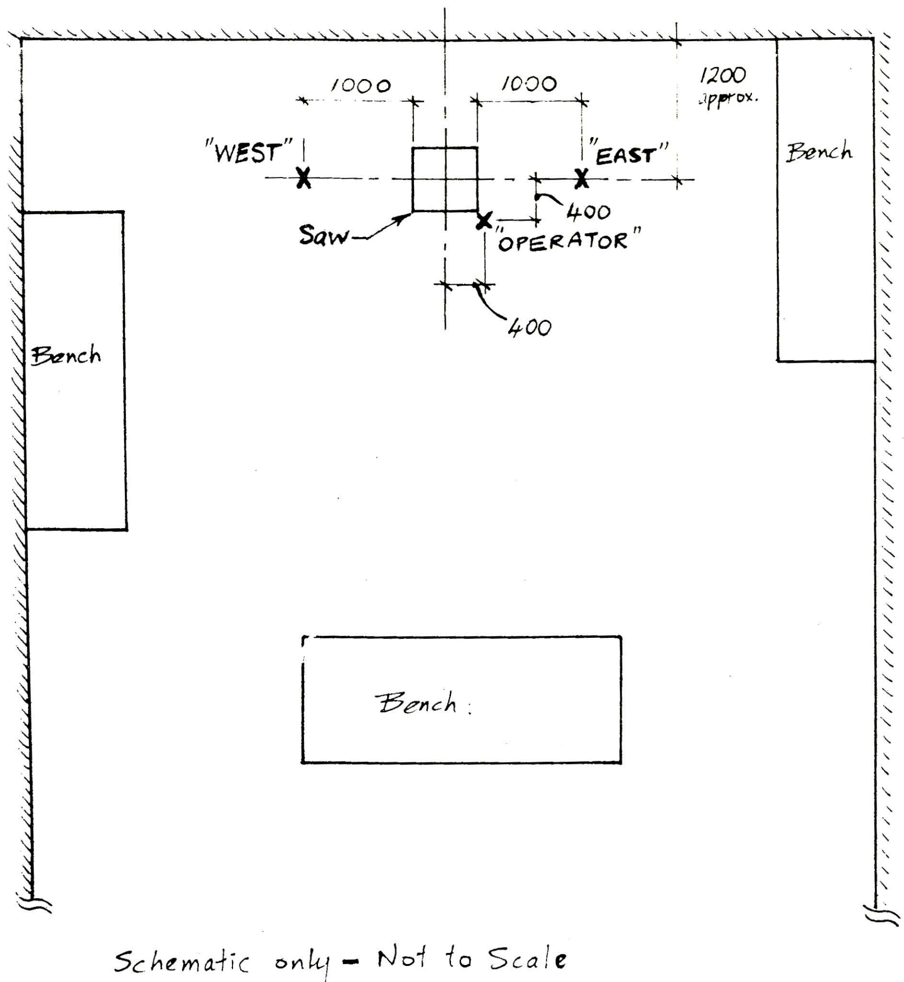
* This blade had no gullets as such; there was a small manufacturing slot in front of each tooth.

APPENDIX NO. 2

Test Results for MEP SL 350 Saw

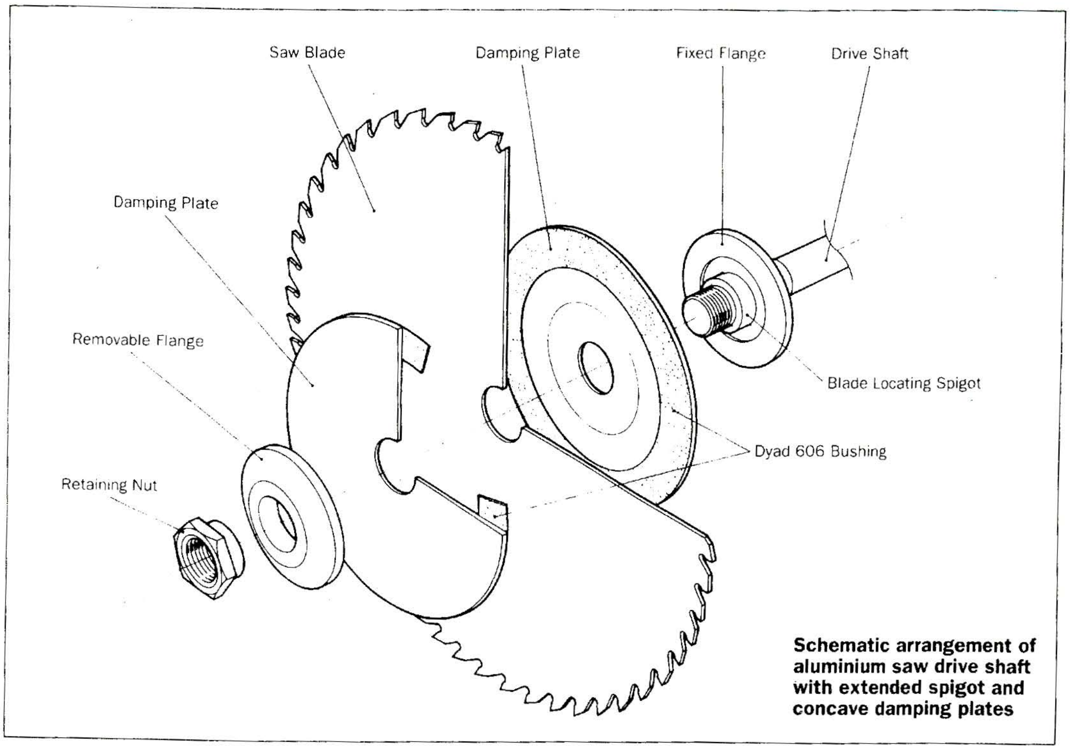
Cutting ALCAN E-66 Aluminium Section on High Speed and Aerodynamic Noise

Test Condition			Noise Level LAeq,10s-dB(A)		
Blade	Blade Damping	Workpiece Clamping	Microphone Position West	Operator	East
Cutting					
TCT-1	Undamped	Metal	96.7	98.1	95.4
TCT-2	"	"	101.9	102.2	101.9
SHSS-1	"	"	96.4	97.8	95.8
SHSS-2	"	"	91.5	92.3	90.6
TCT-1	Damped	Metal	94.3	94.7	92.9
SHSS-2	"	"	90.2	90.9	88.9
TCT-1	Damped	Timber	95.7	95.2	93.4
SHSS-2	"	"	89.0	90.9	88.6
TCT-3	"	"	91.2	91.9	89.0
Aerodynamic Noise - "Dummy cut"					
TCT-1	Damped	-	86.5	92.3	88.4
SHSS-2	"	-	72.6	77.0	73.0
TCT-3	"	-	79.6	85.4	80.5



LOCATION OF PROJECT SAW AT S.C.G.H.
METAL FABRICATION WORKSHOP
SHOWING MICROPHONE POSITIONS

FIGURE 1



Schematic arrangement of aluminium saw drive shaft with extended spigot and concave damping plates

FIGURE 2

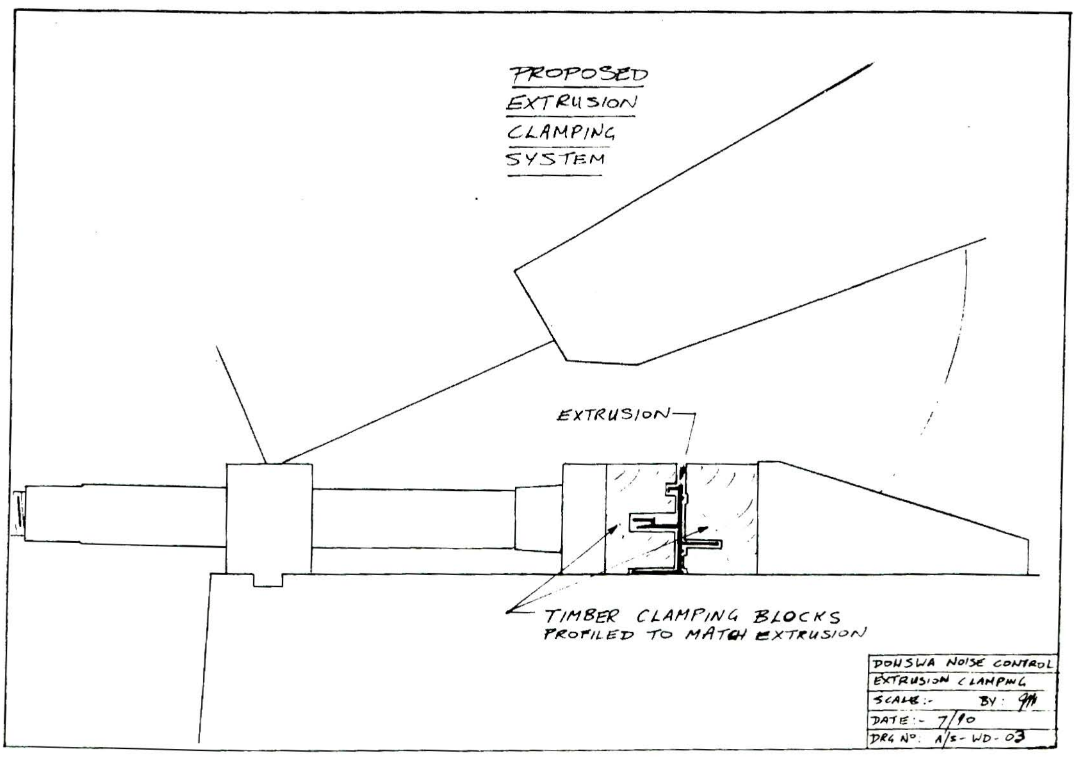


FIGURE 3

The Muting of Organ Pipes

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ABSTRACT

Many musical instruments (such as the stringed or brass sections) are capable of being muted in order to change their harmonic structure or acoustic output. The present investigation is aimed at determining how well wind instruments can be muted by studying ways of muting an organ pipe.

Three methods of muting were investigated:

1. The use of thin sheets of perforated plastic applied to the open end of the organ pipe.
2. Application of porous absorbers to the open end of the organ pipe.
3. The construction of a "double organ pipe".

The perforated sheets proved to be the most useful form of mute with reductions in the total SPL of up to 10dB and any required value from 0–10dB obtainable. The double organ pipe demonstrated larger reductions in the SPL but this value could not be altered easily. Factors that were thought to be of importance in determining a "good" mute were the reduction in the total SPL, the change in the fundamental frequency and the change in the relative intensities of the harmonics.

1 Introduction

The organ-pipe has it's precursors dating back to the time of the ancient Greeks. It then took the form of the *syrinx* (or pan-pipes) which was a collection of hollow reeds closed at one end and wind from the players mouth blown across the other to sound a note.

The form of the modern organ flue pipe soon followed from this simplistic design. It consists of a small slit cut into the closed end of the pipe and a hole cut above this slit into the side of the pipe. The edge of this hole farthest away from the slit is fashioned to give a sharp edge. The basic sounding mechanism of this organ pipe is that air is blown through the slit so that a thin air jet is produced; as this jet encounters the sharp edge of the hole in the pipe, it flicks alternately into and out of the pipe. The rate at which the jet flips is determined by acoustic feedback provided by the pipe esonance.

The organ, as an instrument, has many individual organ pipes grouped into *stops*. Each stop is used to produce a different sound giving variety to the organist. The purpose of designing mutes for organ pipes is that not only does the mute change the volume of the organ pipe, it also alters the harmonic structure of the organ pipe. This allows the organist the possibility of creating more stops for an even wider variety of musical sounds. Also, the methods employed here for muting organ pipes may be found to be applicable to other wind instruments. Mutes are generally used for violins (and similar stringed instruments) as well as brass instruments. A further application of this work is in combustion chambers where acoustic feedback from the resonating chamber can influence combustion.

2 Existing Mutes

2.1 Violin Mute

This consists of a three or five pronged device which attaches itself to the bridge of the violin with the prongs resting on the body. It mutes the violin because it fixes the bridge in place firmly and so does not transmit vibrations to the body as well. This mute also has the effect of changing the resonance frequencies of the violin, producing a different sounding violin. The mute reinforces the lower frequencies at the expense of the upper harmonics.

2.2 The Trumpet Mute

Investigations of various types of trumpet mute have been undertaken [1]. There are four main types of trumpet or cornet mute, these are the cup mute, harmon, straight shastock and the solotone. Each is inserted into the bell of the trumpet tightly. They consist of various resonant cavities and thin tubes that produce the

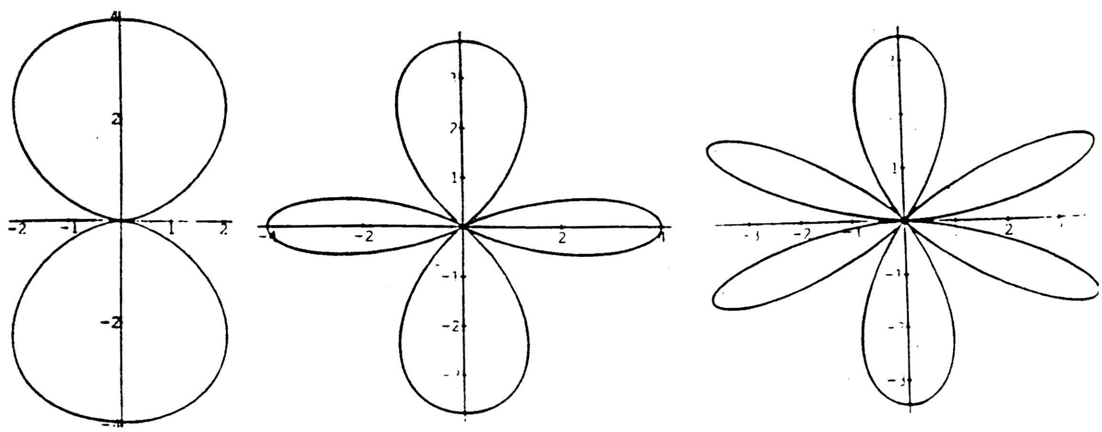

Figure 1: 1a. First mode: 1b. second mode; 1c. third mode of resonance showing the intensity lobes associated with interference from the mouth and open end of the organ pipe.

muting effect. These mutes drastically alter the harmonic structure of the sound so much that they almost make the instrument unrecognizable as a trumpet. The spectrum is usually altered to give more emphasis to the middle to high harmonics.

3 Background Experiments

The organ pipe used in the experiment was wooden, of rectangular cross-section (Gedeckt type). It's physical length was 83·5cm and had a fundamental frequency of around 189Hz, depending on the conditions. Measurements were made in an anechoic chamber at $20°C$ and 50% humidity.

In positioning a microphone to investigate the sound pressure levels from the pipe, it is noticed that there are two sound sources (the open end and the mouth) of approximately equal intensity. This will introduce interference effects for each harmonic depending on the relative position of the microphone. This is illustrated by Figure 1

$$I(\theta) = [1 - (-1)^n \cos(kL \sin\theta)] \tag{1}$$

and is derived from Equation 1 where θ is the angle between the direction the pipe is pointing in and the position of the microphone in the horizontal plane as the pipe lies horizontal. $k = 2\pi nf/c$ with f as the frequency of the nth harmonic and L is the physical length of the pipe.

It is noticed that the odd harmonics (1a and 1c) are extremely weak in the horizontal position. This corresponds to the microphone being placed along the axis of the pipe. At the vertical positions in 1, the harmonics have similar in-

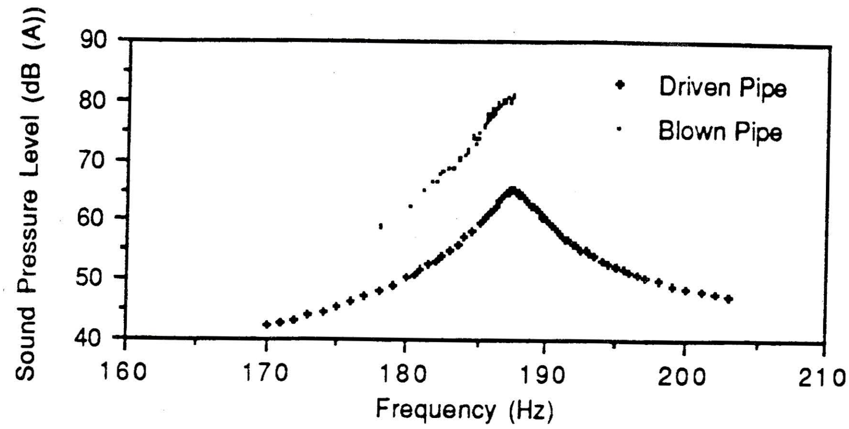

Figure 2: SPL's measured for exciting the organ pipe by a loudspeaker (crosses) and by blowing with air pressure (dots).

tensities. This position corresponds to the microphone placed at right angles to the pipe and equidistant from each sound source. thus producing no destructive interference effects. This was the position chosen to conduct further experiments so that the 'real' intensities of the harmonics would be measured.

3.1 Blowing Pressure

From theory [2, 3] it is noted that the sounding frequency increases as the blowing pressure is increased. A simplistic explanation of this is that as the blowing pressure increases so does the jet velocity. This means that the faster jet will flick across the upper lip more quickly and will raise the sounding frequency.

To investigate more closely the change in the sound pressure level (SPL) as the blowing pressure and hence the fundamental frequency is changed. one must look at the resonance curve of the pipe around the fundamental frequency. This is done by acoustically exciting the pipe with a loudspeaker positioned near the open end of the pipe with a thin tube delivering the sound to the end of the pipe from the loudspeaker. The thin tube is required because large objects placed over the open end of the pipe will change the fundamental frequency. The SPL from the mouth of the pipe was measured and from this the resonance curve could be established.

To compare the resonance curve to exciting the pipe in the normal way (ie. by the jet), one must examine how the SPL and the fundamental frequency change as the blowing pressure is altered. This was carried out by measuring the total SPL (in dB (A)) and the fundamental frequency with the microphone.

To compare the data. a difference of 3°C was taken into account since the measurements of the resonance curve were taken at a temperature of 17°C and measurements of the effect of blowing pressure were taken when the temperature was 20°C. Having applied this correction to the fundamental frequency, the two curves are plotted in Figure 2 showing the change in the total SPL as a function of frequency for when the pipe was acoustically driven with the loudspeaker and when the pipe was blown in the usual manner. The similarity of the two curves indicate that the output SPL from an organ pipe is due mainly to the resonance structure of the pipe. More acoustic output is expected closer to the natural resonance frequency of the pipe. One would expect that the offset in Figure 2 is unimportant as the loudspeaker could have been turned up to match the SPL's of the curve for the blown organ pipe. It is expected that there will be a smaller effect that will favour larger acoustic outputs at higher frequencies for the blown pipe since the input from the jet will be greater as the blowing pressure is increased.

3.2 Calculation of Experimental Error

To make an estimation of the error bars used in further experiments, it was necessary to locate the source of errors in the experimental procedure.

The equipment used to analyse the spectra from the organ pipe was a Brüel and Kjær type 2034 signal analyser. The procedure was to average 50 samples of the organ pipe's spectrum. There was an intrinsic error from the analyser due to it's gating process: the spectrum was displayed in a series of bins 2Hz wide. If a resonance peak was located in the center of a bin, a maximum obtainable SPL was read in this bin. If, on the other hand, the center of the peak was located at a frequency common to two adjacent bins (ie. the highest frequency of one bin and the lowest frequency of the other) then the spectrum would have two adjacent bins showing the same SPL but somewhat less than the maximum obtainable SPL. The difference of these two was treated as an error in the measurement of the SPL of a resonance peak (typically $\pm 1{\cdot}2$dB).

Errors associated with the instability of the organ pipe's output were taken into account by measuring similar spectra and comparing the SPL's of the same peaks in each run.

The errors used in further measurements were taken to be the largest of the two sources described above.

4 Muting by Perforated Sheets.

The first method of muting the organ pipe investigated was that of applying a thin sheet of perforated perspex to the open end of the pipe. The total SPL can be affected by applying a solid perspex sheet to the open end by partially covering it but this method also decreases the fundamental frequency by a relatively large amount. For example, covering the open end by 50% leads to a drop of 2dB in the SPL but also decreases the fundamental frequency by $4{\cdot}8$Hz.

There are practical limits associated with the sheets of perforated perspex. Firstly, the sheets must not be too thick. As a rule of thumb, the diameter of

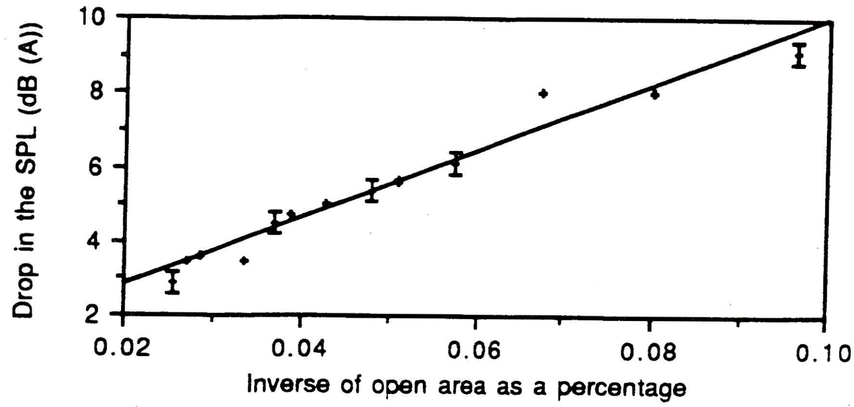

Figure 3: Graph showing the inverse relationship between the open area of the mutes and the drop in the total SPL.

the holes should be larger than the thickness of the sheets for sound to propogate through[4]. Also, the holes must not be too large or, in the worst case such as when a solid sheet partially covers the open end, the fundamental frequency will drop by a large amount. The sheets used were 1·5mm thick and the perforations were either 2 or 4mm in diameter. There was no observable difference between these two hole sizes in their performance as mutes.

To measure the performance of the perforated sheets, a number of sheets were made up with varying open areas and tested on the organ pipe. There were also practical limits associated with the open area used: the largest open area was around 40%. Above this value the perspex sheets were too fragile to perforate. Also, the effects of sheets with higher open areas become slight and subtle and possibly inaudible. The lower limit on the open area was found to be around 10%. Below this point the fundamental mode would not resonate.

The measurement of the performance of the sheets as mutes depends on three criteria:

1. The drop in the total SPL.
2. The drop in the fundamental frequency and
3. The change in the harmonic structure of the organ pipe.

Each sheet was tested on the organ pipe and the results are shown in Figure 3.

$$y = 1{\cdot}05 + \frac{88{\cdot}9}{x} \qquad\qquad R^2 = 96{\cdot}1\% \qquad (2)$$

Equation 2 is a linear fit to the points in Figure 3 where y is the SPL and x is the open area. The R^2 correlation coefficient shows a close fit that can be used to predict the performance of a particular sheet with a particular open area. When

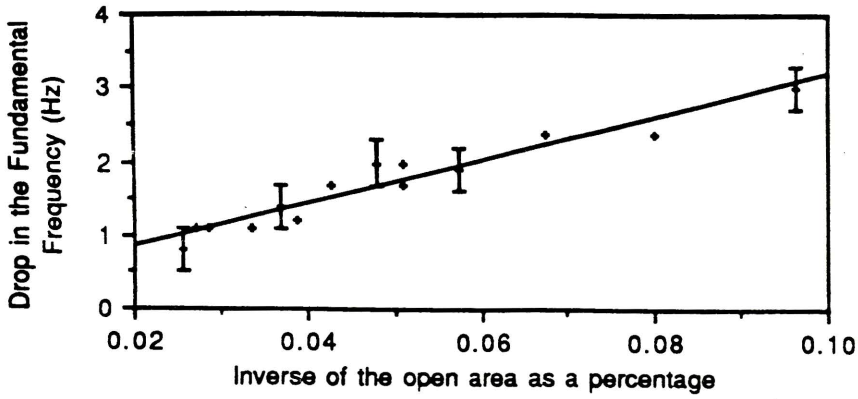

Figure 4: Relation between the drop in the fundamental frequency of the organ pipe and the open area of the applied mute.

carrying out the measurements it was noticed that the performance of a sheet with a particular open area was drastically affected by the way in which it was attached to the open end of the organ pipe. Firmly fixing the sheet to the pipe was found to give the most consistant results.

$$y = 0{\cdot}28 + \frac{29{\cdot}1}{x} \qquad R^2 = 92{\cdot}3\% \tag{3}$$

Figure 4 shows how the fundamental frequency changes with the open area of the perforated sheets and Equation 3 gives the line of best fit for the points with y as the fundamental frequency and x is the open area. Again. a close fit is obtained.

The application of the sheets tended to decrease the levels of all the harmonics. bar the fundamental. in a similar fashion. The higher harmonics were generally attenuated by 5dB more than the fundamental. A possible explanation[5, 6] for this is that the fundamental is generated by the input of airflow from the jet whereas the harmonics are the result of the jet profile.

In conclusion. the method of muting the organ pipe by perforated sheets can be used to obtain drops in the SPL of between 2·9dB and 9·1dB with drops in the fundamental frequency of up to 3Hz.

5 Muting by Absorptive Materials.

The second method of muting organ pipes was to place absorptive materials over the open end. There were a total of 16 materials investigated including varying thicknesses of foam rubber. steel. rubber. carpet and fibreglass. Some of these materials such as the steel did not let the fundamental mode resonate when they were applied directly to the open end of the organ pipe. The strategy adopted. then, was to measure the muting qualities of each sample as it was positioned at various distances from the open end. This was done because some of the materials did not let the organ pipe resonate when they were fixed to the open end and so they were moved a few millimeters away from the end of the organ pipe until the pipe was resonating again. The criteria that were used to establish which was a "good" mute were a large drop in the SPL and a relatively small drop in the fundamental frequency. It was found that all the materials tended to have the same maximum SPL drop of around 4 to 5dB but the fibreglass and the open-cell foam rubbers had the smallest drop in the fundamental frequency of typically 1 to 2Hz. A more quantitative description of the mutes was required and so each material was tested for it's specific normal acoustic impedence. The method used[1] employs an impedence tube with the testing material at one end and a loud speaker at the other. From measurements of the positions and magnitude of maxima and minima the values of R_s and X_s, the components of the specific normal acoustic impedence Z_s (where $Z_s = R_s + jX_s$), can be calculated. The results showed that there was no correlation with the specific normal acoustic impedence of a material and it's performance as a mute.

It was decided to test the materials for their airflow resistance to see if a correlation could be established between this and the material's performance as a mute.

The apparatus consisted of a 'rotameter', to measure the volume velocity flow, connected to a long tube of diameter 10·3cm containing foam rubber inserts to steady the air flow. At the far end of the tube was the sample to be tested. The static pressure was measured on both sides of the sample to measure the pressure difference across the material.

The rotameter was calibrated for volume velocity by measuring the air velocity at the exit tube with a pitot tube and calculating the volume velocity from knowing this and the cross-sectional area of the exit tube. The rotameter was calibrated for volume velocities between $8 \times 10^{-4}\ m^3s^{-1}$ and $3{\cdot}6 \times 10^{-3}\ m^3s^{-1}$.

Only the fibreglass and the foam rubber samples were tested for airflow resistance as the other materials (such as steel) would have effectively infinite airflow resistance as they are non-porous.

Unfortunately, airflow resistance measurements were made at air velocities of $0{\cdot}1 - 0{\cdot}4\ ms^{-1}$ which is out of the specified range[2] of $0{\cdot}5 - 50\ mms^{-1}$. It was thought that the results were unaffected as consistant values for the airflow resistance were obtained.

The method of muting organ pipes by using absorptive materials has shown to reduce the total SPL by up to 5dB with no more than a 2Hz drop in the frequency of the fundamental mode.

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[1]As described in Australian Standard 1935–1976

[2]ASTM Designation: 522 - 80. '*Standard test for airflow resistance of acoustical materials*'

6 Muting by a 'Double Organ Pipe'

The initial motive behind developing a double organ pipe stems from the driving mechanism of the single organ pipe. The driving mechanism is the interaction between the resonance of the pipe and the action of the jet. As the jet flicks into the pipe, it drives the resonating column of air. If one were to place another column of air next to the first so that the jet alternately blows into each pipe, then both pipes should be excited similarly. The major difference in the two pipes will be that they resonate 180° out of phase. The open ends of each pipe, being physically next to each other, should destructively interfere and cancel each other out.

The double organ pipe was made by enclosing the mouth of the organ pipe with another column of the same dimensions. The acoustic output was drastically reduced in this case. To check that the output was in fact being reduced by the active cancellation of the two open ends, a physical barrier such as a wooden board could be introduced between the two open ends and the acoustic output was observed to increase.

Another method to check whether the mechanism stated above was in operation was to look at the output from each pipe. When this was done simultaneously, it was observed that the phase of the the two resonating columns was in fact 180° out of phase.

When the double organ pipe was compared to the original single organ pipe, there was a reduction of 26.8dB(A) in the total SPL. This is a very large drop but it must be noted that there is not much control over this value as in the other two methods of muting the organ pipe. Also, with such large reductions in the SPL, it was noticed that the effects of noise from the air supply became significant in the overall sound produced. Analysis of the effect on the levels of the harmonics showed that the double organ pipe tended to act as high pass filter since it reduced the fundamental mode by 27dB and reduced the sixth harmonic by only 15dB. The reduction in the harmonics inbetween tended to lie on a straight line between these two points.

7 Conclusion

The three methods used to mute an organ pipe were the application of perforated sheets and absorbing materials to the open end of the pipe as well as a double organ pipe being used.

The muting by perforated sheets showed to be the most useful as any value for the reduction in the SPL could be obtained by altering the open area of the perforations. The reduction in the fundamental frequency can also be easily and reliably predicted using this method. Practical limits on the construction of the perforated sheets have shown to constrain the reduction of the SPL to between

2·9 and 9·1dB.

The method of muting by using absorbing materials has also shown to be of use with SPL reductions of around 5dB. There were attempts to correlate the effectiveness of each mute with it's airflow resistance but the results are not as conclusive as for the muting by perforated sheets. Nevertheless, there does seem to be a reasonable correlation evident, allowing some predictions of the muting qualities of each material to be made, given the airflow resistance of the material.

The construction of the double organ pipe has shown to produce SPL reductions of 26·8dB. Unfortunately, this method does not let one alter this value very easily. The double organ pipe has also shown to act as a high pass filter as it tends to make the harmonics more comparable in their levels.

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LIFT-OFF CHARACTERISTICS OF ELECTROMAGNETIC ACOUSTIC TRANSDUCER RECEPTION

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In many ultrasonic applications the intimate contact required between conventional piezoelectric transducers and the product under inspection can prove to be a major problem. An example of such an application is the on-line internal inspection of hot steel, where hot, moving and unfinished surfaces are encountered, and operation across an air gap of at least a few millimetres is required [Baharis, 1991]. Electromagnetic acoustic transducers (EMATs) are one solution to this non-contact requirement, but where the temperatures may approach 1000 degrees Celsius, it is necessary to cool and protect the transducer head. This leads to increased separation (or lift-off) from the product under test, and diminishes the sensitivity of the system. It is important to understand this effect and to be able to predict the rate at which the sensitivity drops off as it will determine whether EMATs are a practical solution to an application. The lift-off characteristics of EMAT reception have been grouped into three separate components and an experiment to isolate and measure one of these has been designed. Some mathematical modelling provides a theoretical base and its predictions are compared with the experimental results. The ramifications of these results are discussed in the context of the ultrasonic internal inspection of hot steel.

1. Introduction

One of the major problems with EMATs is that they provide only a few millimetres of lift-off from the surface before the signal becomes so weak as to be useless. Improvement in the lift-off characteristics of such devices is of prime importance in being able to produce a practical instrument that is capable of withstanding the harsh environmental conditions typically encountered.

The main purpose of this paper is to discuss one of the main limitations to increased lift-off, namely the secondary ac magnetic field produced in the region above the isonified area. Some experimental evidence of lift off characteristics will be presented to show the influence of this major factor alone.

2. Principles of EMAT Operation

Many types of EMATs which operate into the Megahertz range of acoustic frequencies, can be constructed and each type is generally optimised to generate and detect one or more classes of acoustic waves. The elemental transducer shown in Figure 1, although not a practical device, serves as an illustration of the principle of EMAT longitudinal wave ultrasound generation.

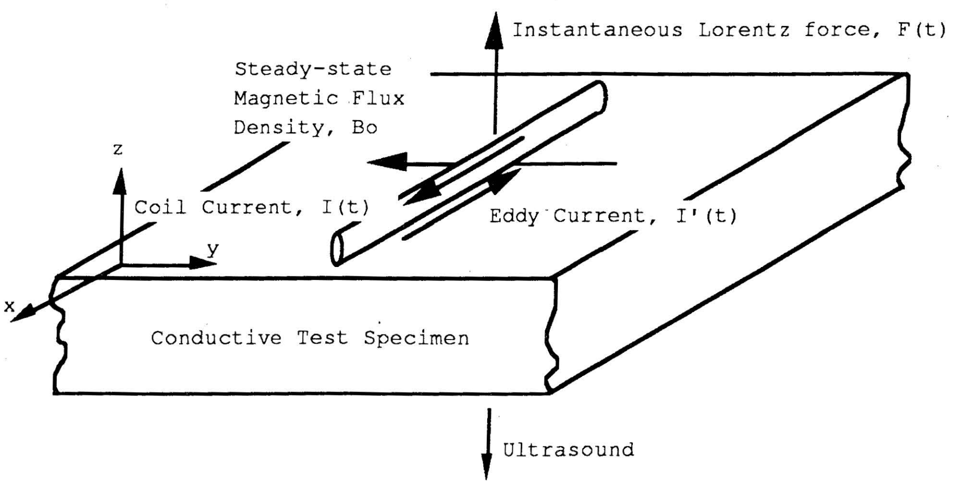


Figure 1: Longitudinal wave generation

An alternating current, I(t), is fed into the wire which is in close proximity to the surface of the test specimen. This in turn induces an opposing dynamic eddy current, I'(t), on the surface of the test specimen and, in the presence of the external strong magnetic field bias, B_o, causes a deflection of the moving electrons in the direction of the cross product of I'(t) and B_o.

The resultant Lorentz force generates a pressure wave that propagates into the bulk of the material. By changing the orientation of the magnetic field such that it is perpendicular to the surface of the specimen, bulk transverse waves may be generated.

The detection of ultrasound is a reciprocal process and a simple configuration, shown in Figure 2, is used to demonstrate the principle. In the presence of the external steady state magnetic field, an acoustic wave, reflected from the specimen surface, will produce a current sheet on the surface of the conductive specimen. In the body of the material, secondary electric current flows are set up which counteract the main induced currents. On the surface however, complete cancellation is not possible and a net current sheet results. This net current sheet has an effective depth given by the skin depth of the material at the frequency of interest.

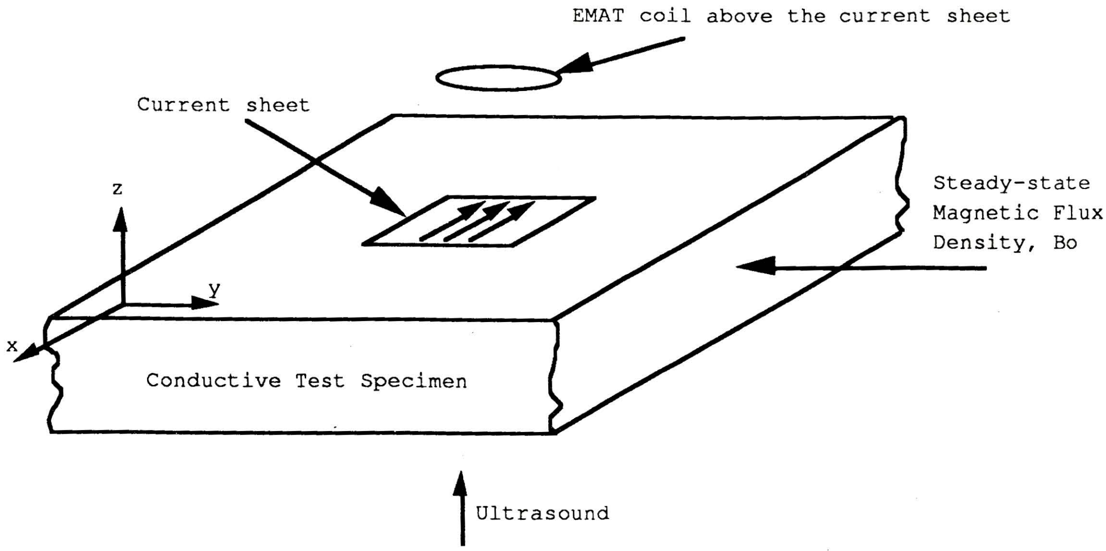


Figure 2: Longitudinal wave reception

This current sheet at the surface creates a secondary magnetic field in the air above the sample surface which is detected by an adjacent coil attached to an amplifier.

2. EMAT on Reception

The following analysis and the experimental work considers the reception of ultrasound by EMATs, although much of what is expressed is applicable to the transmission case.

The interpretation of EMAT behaviour is simplified for the cases where the skin depth δ is smaller than the acoustic wavelength, λ, in the material.

The skin depth is $\delta = \sqrt{\dfrac{2}{\mu\omega\sigma}}$

where μ and σ are the permeability and the conductivity of the medium respectively.

If the skin depth becomes comparable with the acoustic wavelength, then the net current sheet will show a phase variation with depth and this phase variation leads to a diminution of the field strength. Hence EMAT operation is restricted to frequencies where the skin depth is small in comparison with the acoustic wavelength. For steels at room temperature, this ratio is small even at 5 MHz. However for hot steel, the resistivity increases and the permeability decreases to unity, both factors increasing the skin depth This effectively limits EMAT transduction to frequencies less than 4 MHz for longitudinal waves and less than 1 MHz for shear waves.

For the case of a longitudinal acoustic wave striking a plane steel surface normally, the magnetic field produced at the surface by the EMAT process is

$$B_{out} = B_0 \zeta \delta \omega \sigma \mu$$

where δ is the skin depth
ω is the angular frequency
μ is the permeability of the steel
and ζ is the acoustic displacement amplitude.

This can be reduced to

$$B_{out} = B_0 \zeta \sqrt{2\omega\sigma\mu}$$

In this case, B_{out} increases with frequency. However in practical situations ζ generally also decreases with frequency.

In terms of power, the efficiency of conversion, E, whether in transmission or reception, is given as [Beissner, 1976]

$$E = \frac{B_0^2}{\delta\omega\mu Z}$$

where Z is the acoustic impedance.

With typical values for steel at 1000 °C

$Z = 45.10^6$ kg.m^{-2}.s^{-1}
$\mu = 4\pi.10^{-7}$ T.m.A^{-1}
$\sigma = 10^6$ m^{-1}.Ω^{-1}

and $B_0 = 1$ T the following values are obtained

	f = 1 MHz	**f = 4 MHz**
δ	0.5 mm	0.25 mm
λ **(@ 5,000m.s^{-1})**	5 mm	1.25 mm
E	5.6×10^{-6}	2.8×10^{-6}

A wave amplitude of 10^{-10} m and a frequency of 1 MHz will produce a secondary magnetic field at the surface of 0.4 μT.

3. Spatial Extent of the Induced Secondary Magnetic Field

The above calculations describe the transduction mechanism for the case of plane waves of infinite extent striking the surface at right angles. As such, the values should be considered as maximum achievable values. However, any system has limited extent and geometric factors will reduce the effective field which can be detected.

The limiting geometric factors are:

- the amplitude and phase of the ultrasonic wave
- the extent of the static magnetic field
- the position of any conducting surfaces.

All three factors contribute to the overall efficiency of transduction. However, to simplify the analysis, the last two factors were reduced in importance by only considering a special case, that of a uniform magnetic field in the specimen over the region at which the ultrasonic wave impinges. Hence the effect of the conducting surface of the magnet was eliminated as well as the limited extent of the magnetic field.

4. Computer Simulation

A number of computer simulations were undertaken of various current sheet configurations, but only one will be presented here, namely the complete 3D field in the half space above a rectangular active surface. The current density was considered uniform over this area. Although the current density distribution in a real situation depends on the spatial extent of the bias magnetic field and the acoustic field the simplified simulation was still considered to be indicative of the real situation.

The usual quasi-static approximations were taken for this study, so the Biot-Savart Law was employed.

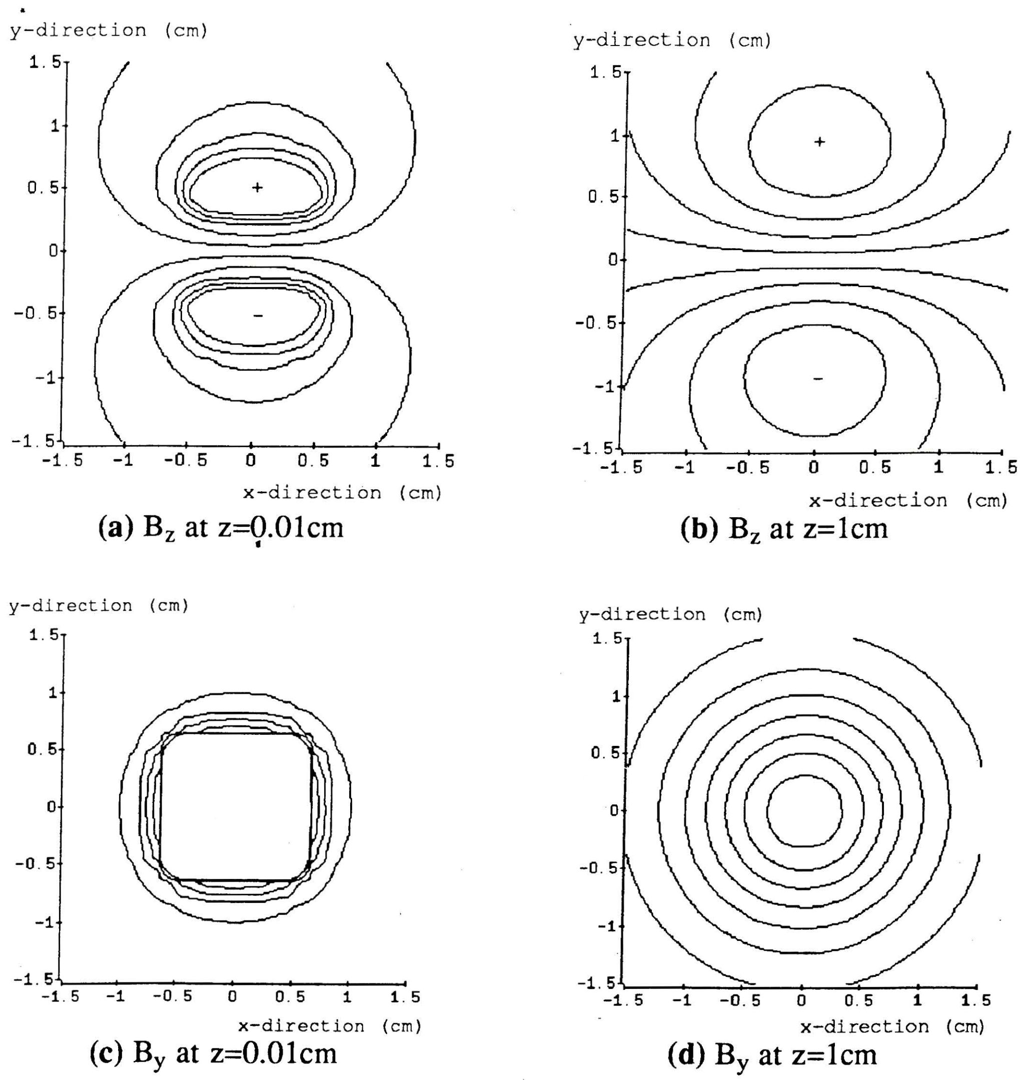
The active area is a square with sides of 1 cm and the amplitude of the effective current at the surface, $J\delta$, is 1 A/m. The contour plots are given with arbitrary contour levels, primarily to indicate the character of the field so produced. However the field strength at the surface is about 10^{-7} T.

Figures 3a and 3b show the z component of the field (which is normal to the surface) at two planes in the z axis. Figures 3c and 3d shows the corresponding y component of the field, that along the surface but at right angles to the current sheet, again for the same two distances from the surface.

Figure 4 illustrates the variation in the y component of the field in the yz plane, along the central x axis.

These figures demonstrate a number of points:

- the z field component has different directions on either side of the plane given by y=0.
- near the surface, the z component reaches a maximum just at the edge of the active surface. As the plane of interest, z, is increased, so these maxima occur at larger values of y. (In other simulations of current sheets showing a smoother spatial distribution, these peaks in the z component are not so pronounced.)
- When the lift-off is less than half the unit length of the active area, the field drops off linearly. So creating a large active area will help increase the lift-off capability.
- as expected, for values of z at which the active area does not subtend a large angle, then the field falls off similarly to that from a point source, namely as $1/z^2$.



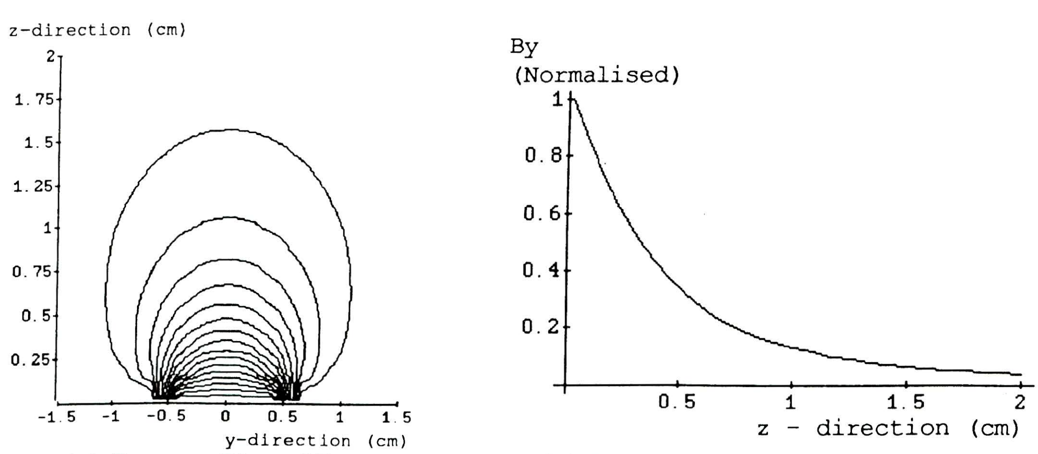
(a) B_z at z=0.01cm

(b) B_z at z=1cm

(c) B_y at z=0.01cm

(d) B_y at z=1cm

Figure 3: Contour plots of the magnetic field components in the xy plane above a 1cm square current sheet



(a) Contour Plot of B_y

(b) B_y as a function of z for x=y=0

Figure 4: Magnetic field y component in the zy plane above a 1cm square current sheet

5. Reception Coil Geometry

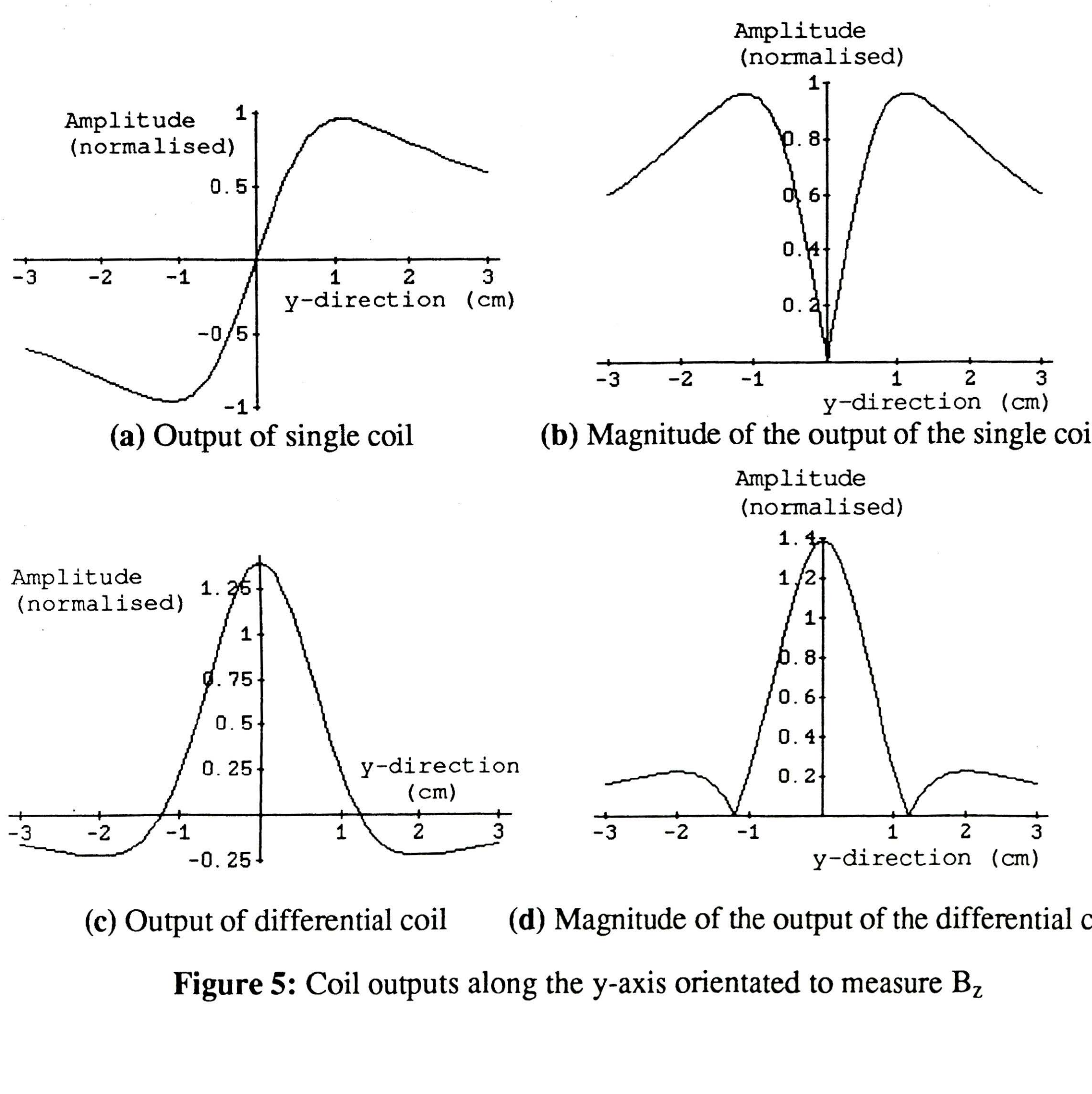
The emf that is produced at the reception coil depends primarily on the flux that is enclosed by the coil. The net flux through a coil is a convolution of the magnetic field and the size, and direction of the winding, of the coil. Normally the coils are laid flat, so that the z component of the field is detected.

Two coil shapes are considered:

- Single Coil type - This may be spread out in a pancake style or more tightly wound as with a normal coil.
- Differential coil - This is a pancake coil, wound as in a figure of eight configuration.

The following figures are outputs of computer simulation for the cases of the two coil types scanned across the surface. In this case, a 2D current sheet with Gaussian amplitude distribution was considered.

Figure 5a illustrates the response of a simple coil of unit length, aligned to be sensitive to the z component of the field, scanned across the surface of the sample. Experimental peak-to-peak measurements however, record only magnitude, taking no account of phase, and this is shown in Figure 5b.



(a) Output of single coil **(b)** Magnitude of the output of the single coil

(c) Output of differential coil **(d)** Magnitude of the output of the differential coil

Figure 5: Coil outputs along the y-axis orientated to measure B_z

Similarly, Figures 5c and 5d show the responses of a differential coil (both sections having unit length) as it is scanned across the surface. Note that the finite dimensions of the coil tend to smear out the spatial response.

6. Experiments

Experiments were conducted to test the dimensions of the secondary field produced by the simplified arrangement shown in Figure 6.

The two basic coil types were scanned in the half space above the surface. The coils were aligned parallel to the surface so that the z component of the magnetic field was detected. The single coil used for monitoring B_z had a face with dimensions 12 by 8 mm. The differential coil was a square of side 10 mm, each coil of the differential arrangement having a width of 5 mm.

A conventional piezoelectric transducer was continuously pulsed by a standard pulse-echo NDT unit. The EMAT signals were detected, amplified and captured on a digitising CRO. The amplitude of the first arrived, single transit pulse was recorded.

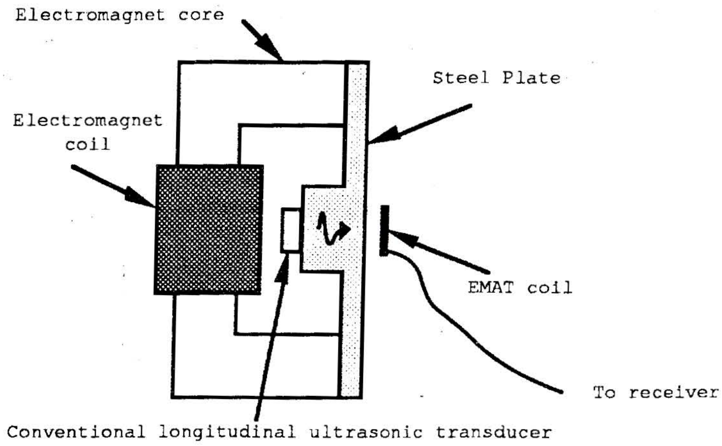
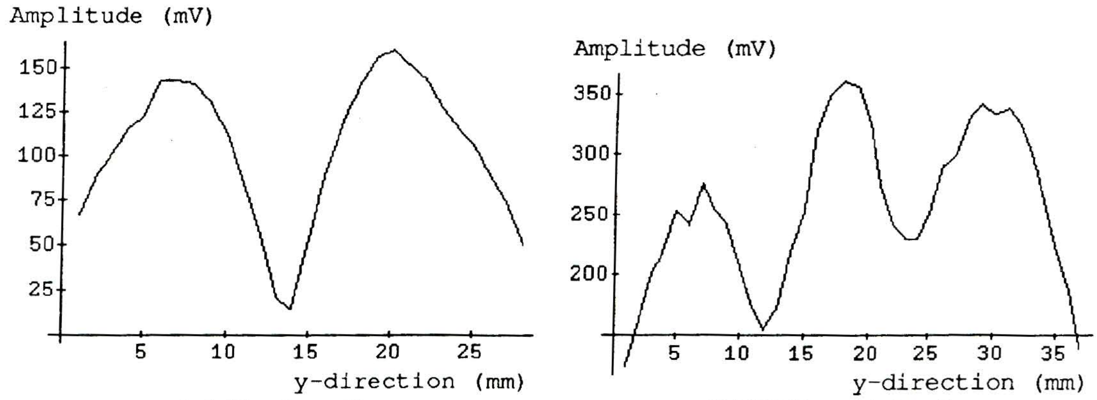


Figure 6: Experimental configuration

Figures 7a and 7b were obtained using a 4 MHz, 11 mm diameter piezoelectric transducer by scanning the coils in the y direction. These experimental results show similar characteristics to those predicted in Figures 5b and 5d.



(a) Single coil **(b)** Differential coil

Figure 7: Coil output along the y-axis orientated to measure B_z

The single coil which was used to measure B_y had a face of dimensions 14 by 5 mm and its output, which is shown in Figure 8, was comparable to the predictions shown in Figure 4b. It should be noted however, that the simulations results are for a infinitesimal coil, whilst the experimental results are obviously not. This explains the absence of a linear region for small values of z.

The experiments qualitatively confirm the overall geometric factors found in the simulations, such as the fall off in signal amplitude with lift-off and the variation in signal amplitude while scanning across the surface, using the two types of transducers mentioned.

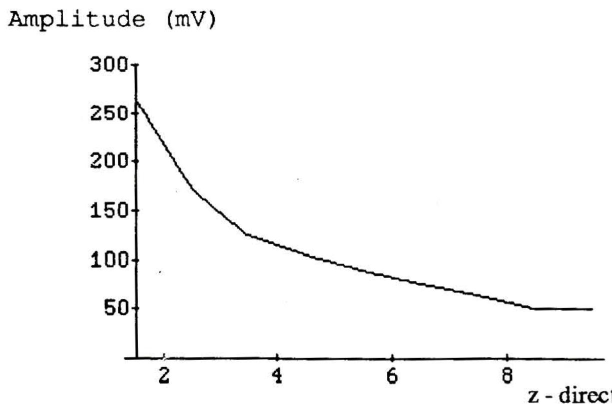


Figure 8: Coil output along the z-axis orientated to measure B_y

7. Discussion

This study has generally confirmed that for a planar acoustic wavefront, lift-off characteristics are improved by having a larger active area. It also shows that a differential coil arrangement can create a larger signal output than a single coil.

However to produce a large active area, the surface and acoustic wavefronts must be coplanar over this area. This requires the specimen surface roughness to be significantly less than the acoustic wavelength, otherwise a larger active area will be of little benefit.

EMATs are been used at the exit of the Bloom Caster at BHP Rod and Bar Products Division in Newcastle, to internally inspect the blooms. Here, a transverse wave is injected at the top of the bloom and is detected underneath the bloom. In this case the wavefronts involved are normal to the surface of the steel. However other applications call for the generation (and/or detection) of ultrasound at angles away from the normal. For these applications, a meander coil is often employed and in these geometries, the active area will not have phase coherence along its length. The lift-off characteristics are therefore poorer in these circumstances.

The other major factor in producing a large active area is the strength of the bias magnetic field at the surface layer. There is a trade off between the strength and the extent of the magnetic field which can be realistically be produced. The strength of the bias magnetic field is also strongly dependent on the temperature of the steel, through the effects on the permeability of the steel.

8. Conclusion

One of the main purposes of this work was to understand the reasons for the marked signal diminution as EMATs are pulled away from the surface.

The main reason for this effect is the limited strength and area of the current sheet produced by the ultrasonic beam. Ideally a large planar wavefront interacting with a strong DC magnetic field, also of large extent, will create a secondary magnetic field with a relatively small lift-off diminution.

9. Acknowledgments

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RESOLVING COMMUNITY NOISE FROM A STEELMAKING PLANT

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ABSTRACT

In 1984, the absorptive silencer in the induced draft fan duct at the No 3 Basic Oxygen Steel plant, Port Kembla Steelworks catastrophically failed after only 6 months operation. The resultant down-time on the Steelmaking plant cost approximately 3 million dollars in lost production. The cause of the failure: abrasive/corrosive wear of the splitters.

Requirements of the Regulatory body for New South Wales, the State Pollution Control Commission, was that a silencer be proven in this plant and then retrofitted to the 2 older plants. These requirements were caused by public complaint of the tonal emission from the stack, audible for up to 5 kilometres from the plant and affecting several thousand people.

A new method for the reduction of noise was sought. After several false starts, a tuneable, active attenuator was installed at the fan blade cut-off. The expectation was 11 dB reduction at 200Hz. After two and a half years and an investment of $250,000 the system was discarded as unworkable.

By the start of 1991, following total investment of $1.2 million dollars, the noise problem on all three BOS plants had been solved, not by a new technology in acoustics but by rethinking the original noise reduction method. A 25dB(A) reduction was achieved and community complaint has dropped to nil.

Apart from discussing the problems encountered, this paper is intended to show that hi-tech solutions are not the answer to all "difficult" noise problems.

INTRODUCTION

In 1984, the Port Kembla Steelmaking plant installed its third Basic Oxygen Steelmaking vessel. The environmental protection authority for the state of New South Wales, the then State Pollution Control Commission, required that a silencer be installed following the Induced Draft fan on the Waste Gas system for this new vessel. A further requirement was, that once the silencer had proven effective, the other two vessels would be similarly retrofitted with equivalent silencing systems. Since the construction of the first two vessels residential complaints were regularly being made concerning the tonal emission from the plant. Complaints were received from as far distant as 5 kilometres. The emission was audible by several thousand people and potentially offensive to many hundreds. The noise emission was a pure tone at 200 Hz caused by the blade passing at the fan cut-off. A reduction of approximately 15dB at this frequency was needed to reduce the level to below the set requirement. An absorptive silencer was designed for a 25dB reduction at 250 Hz octave band. The 10db excess attenuation was to allow for expected deterioration in the silencer performance caused by dust deposition in the perforations of the facing material. Following commissioning of the plant the silencer worked efficiently for around three months until the splitters in the silencer disintegrated in the duct during operation. The resultant down-time on the steelmaking plant while the debris was removed from the duct resulted in an approximate $3 million (Australian) in lost production. The cause of the failure was determined to be the abrasive/corrosive wear caused by the gases/dust particles in the waste gas duct.

THE BOS PROCESS

To understand the problems inherent in resolving this noise emission it is necessary to understand the basis of the BOS steelmaking process. To convert molten iron into steel it is required to reduce the undesirable elements in the iron. This is done in the BOS vessel by burning them out by injecting oxygen into the vessel. The primary contaminant in iron is carbon. The carbon is removed principally as CO - carbon monoxide - which is extracted from the vessel by the induced draft fan, the cause of the noise problem. Along with the gaseous waste is a large quantity of dust particles, primarily iron oxide that is scrubbed from the waste gas prior to the ID fan.

Carbon monoxide is highly flammable and, when mixed with oxygen, forms an explosive mixture. At the start of each steelmaking operation some unused oxygen can pass through the duct. This means that between each "heat" of steel the duct needs to be flushed by an inert gas to remove the residual CO from the system. Nitrogen and steam are used for this purpose. One obvious requirement is that there be no pockets in the ductwork where the CO can be trapped and cause a potential for explosion at the start of the next production cycle.

Deterioration of the silencer occurred because of :

a. Particle impingement on the perforated steel surface
b. High moisture content in the duct
c. Temperature variations in the duct

These elements caused the carbon steel to be rapidly corroded and break up. The decision was taken to find an alternative means of attenuating the noise emission from these fans.

Tuneable, Active Attenuator

The use of a reactive type silencer was investigated but because of the potential for explosive gas mixtures the concept was not acceptable. An active system was also evaluated however the high moisture content and the corrosive/abrasive nature of the gases in the duct indicated that the system would not survive for an adequate period.

Pressure from the public and statutory areas continued and the failure of normal methods to provide an answer to the problem required that the Company investigate less conventional methods of noise control. A joint venture was commenced with the Australian Commonwealth Scientific Investigation and Research Organisation (CSIRO). A tuneable, active attenuator was recommended and two and a half years of investigation resulted in the installation of an attenuator at the fan blade cut-off. Preliminary trials indicated an 11dB reduction at 200Hz. However, in practice, no measurable reduction in the emission could be attained. After an investment of $250 000 the system was discarded as unworkable in this situation.

Reactive-Absorptive Silencer

While the Company, by these steps, had satisfied the Statutory Authorities that it was dedicated to resolve the problem, the community affected was seeing no resolution to a long-term problem. A visit to the 1989 InterNoise conference by the author found a company that had experienced a similar problem (See InterNoise 89, proceedings pp 409-412)

The Company had resolved the problem using an absorptive/reactive silencer mounted at the exit of the waste gas stack. This seemed a potentially acceptable solution to the BOS problem since it removed the explosion problem by having the gaseous pockets at the end of the duct.

The principle difficulty with this solution was the problem of maintenance of the unit on top of the 70 metre stack. Any problems encountered with the silencer could cause significant delays on the vessel while repairs were carried out. However, even with this limitation, the viability of the system was excepted and costing and engineering evaluation was commenced.

Concurrent with this investigation a further study was conducted to determine whether a solution could be reached to reduce the corrosion if the original splitters were replaced. The use of the splitter silencer was seen to be preferable in terms of maintainability. The results of this investigation provided a means whereby the original system could be replaced with an unknown life but a reasonable surety that the original catastrophic destruction of the splitters would not recur.

The primary change to the construction was the type of steel used for the perforated cladding. The type chosen was 3mm 3CR12 corrosion resistant steel. This steel had been proven particularly abrasion resistant in coal rail wagons outperforming both mild steel and aluminium in the same situation. In addition, the high Chromium content made the steel resistant to corrosion.

The splitters were installed and the noise emission level was reduced by an average 25dB at the blade passing frequency when measured at residential areas 1.0 to 1.6 kilometres from the source. This, in effect, meant that the tonal 200Hz component was inaudible at any position outside the Company boundary.

The on-going problem with the system is the build-up of dust carried over from the gas cleaning system. A loss in efficiency of the silencer by 10dB has been measured and has necessitated grit-blasting the splitter faces approximately every 3 months. Life expectancy of the splitters is approximately 18 months to 2 years. The primary cause of deterioration of the splitters appears to be abrasion from the dust and buckling of the perforated steel caused by the continual temperature variations in the duct.

Overall from the time the original splitter silencer destructed until a resolution was achieved, a total investment of $1.2 million (Australian) was made. Many avenues of high-technology noise reduction were explored but the conclusion was that the conventional solution, modified to suit the application, was the superior means of attenuation.

A PROCEDURE FOR COMPUTING THE INSERTION LOSS PRODUCED BY A PIPE LAGGING

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ABSTRACT

Pipe laggings are formed of porous jackets such as fibreglass blankets and impervious jackets such as metal cladding sheets. Sometimes air spaces are used to separate these jackets. A procedure which enables the prediction of the insertion losses produced by a lagging formed of these jackets is described. The models used to consider the interaction of acoustic waves with these jackets are based on the fundamental parameters which define the jackets. These include their dimensions, flow resistivities, Young's Modulii and densities. The insertion losses associated with the breathing, bending and ovalling modes of pipe vibration are considered. The procedure is used to determine the insertion losses produced by a typical lagging applied to a typical pipe. The results show the importance of the lagging in controlling the sound radiated by the bending mode.

INTRODUCTION

The usual way of attenuating the noise radiated by pipes is to lag them with constructions formed of porous jackets such as glasswool blankets and impervious jackets such as metal cladding sheets. Papers in the readily accessible literature relating to the acoustic performance of pipe laggings generally have been concerned with presenting experimental results [1] [2]. It appears that few attempts have been made to predict the insertion losses produced by pipe laggings. This is not surprising in view of the difficulty which has been encountered in satisfactorily predicting the transmission of sound from the inside to the outside of a pipe. It has been found with regard to this latter problem that theoretical predictions of the sound transmission loss for a plane wave from the inside to the outside of a pipe can significantly overestimate (by 30 dB) the actual sound transmission loss [3] [4]. It has been suggested that this situation arises because unexpected vibrations of the pipe in the beam bending mode can be induced by a variety of mechanisms.

THE COMPUTATIONAL PROCEDURE

A typical construction of the lagging applied to a pipe is shown in Figure 1. The outer surface of the infinitely long pipe has a radial velocity defined by $v_r = V_r \cos n\phi \exp[j(\omega t - k_z z)]$. The value of n, which is an integer, describes the motion of the pipe cross-section. When $n = 0$ the pipe is supporting a breathing mode wave which is travelling in the positive z direction with a wave number of k_z. When $n = 1$ the pipe is supporting a bending mode wave and when $n = 2$ the pipe is supporting an ovalling mode wave.

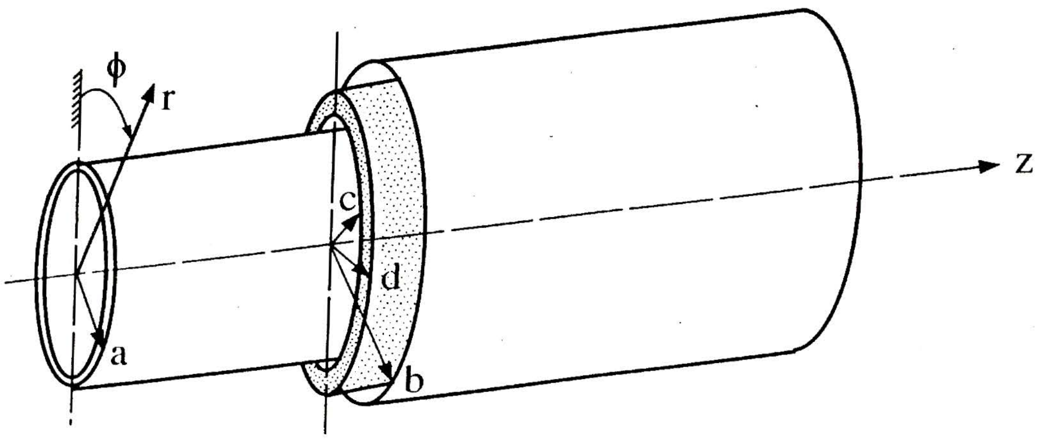
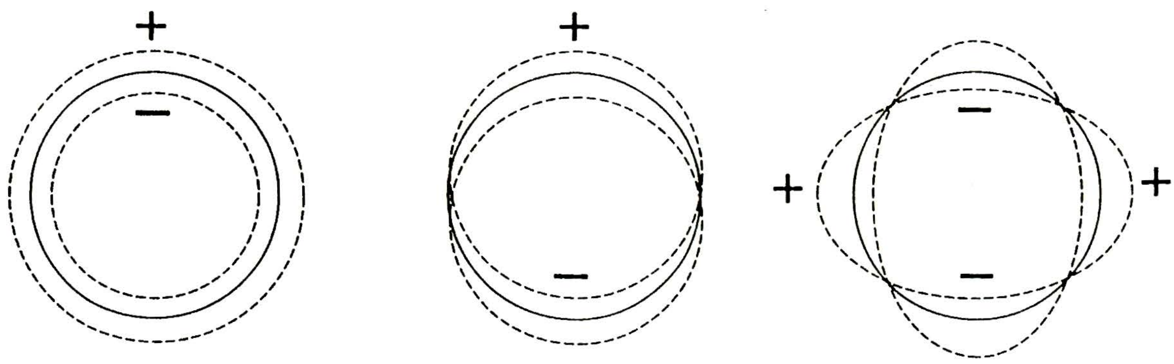


Figure 1: Lagged Pipe Showing Air Space, Porous Jacket and Impervious Jacket

The radial displacement of a cross-section of the pipe in the modes associated with $n = 0$ to 2 is shown in Figure 2. Sound energy can be radiated from such a vibrating pipe so long as k_z, the axial wavenumber which describes the wave motion of the pipe surface, is less than k, the acoustic wavenumber of the fluid which surrounds the pipe. Generally, there is a non-linear relation between ω and k_z for the pipe; for example, when $n = 1$, $k_z \propto \omega^{1/2}$. However, there is a linear relationship between k and ω, that is, $k \propto \omega$ Thus a particular pipe cannot radiate energy in a particular mode until some critical frequency is reached. When $k_z = 0$ the wave propagation direction is normal to the pipe axis. As k_z increases the wave propagation direction becomes inclined to the pipe axis and "conical" waves are radiated by the pipe. When $k_z = k$ the wave propagation direction is parallel to the pipe axis and no sound power is radiated from the pipe. It will be assumed in the remainder of this paper that $k_z < k$ so that sound is radiated by the pipe.



$n = 0$ $n = 1$ $n = 2$

Figure 2: Motions of Pipe Cross Sections Associated with Breathing (n=0), Bending (n=1) and Ovalling (n=2) Modes

It can be seen from Figure 1 that the lagging is constructed of a number of types of jackets. These jackets are the airspace(s), the porous layer(s) and the impervious barrier(s). When the pipe is vibrating in a particular mode the waves in all of these jackets have the same values of ω and k_z. It is assumed that the acoustic pressure and the radial particle velocity on the inner and outer surfaces of these jackets vary with cos $n\phi$ like the radial velocity, v_r on the pipe surface. It is also assumed that the acoustic pressures and radial particle velocities at the interfaces between adjacent jackets are equal. Thus the radial impedance on the surface of one jacket is equal to the radial impedance on the surface of the adjacent jacket.

The insertion loss produced by the lagging in a particular mode at a particular value of k_z and at a particular frequency ω can be found by comparing the sound powers radiated by a unit length of the bare and the lagged pipes. Because the acoustic pressure and radial particle velocity are assumed to vary with cos $n\phi$, the ratio of the sound powers, which gives the insertion loss, can be obtained by determining the ratio of the radial intensities at a particular point. The point just on the outside of the lagging, that is, at $r = b$ and $\phi = 0$ is convenient.

Appendix A presents the development of an expression which enables the radial intensity at the point $r = b$, $\phi = 0$ to be determined in terms of the acoustic pressure and the radial impedance at this point. Appendix B presents the development of expressions which give the radial impedance at this point and the pressure produced at this point by the unlagged vibrating pipe. Appendices C and D present formulae which allow the radial impedances on the inner surface of the different jackets to be found in terms of the fundamental physical quantities which describe the jackets and the radial impedances on the outer surface of the jackets. These appendices also include formulae which enable the acoustic pressures at $\phi = 0$ on the outer surfaces of the jackets to be found when the acoustic pressures at $\phi = 0$ on the inner surfaces are known as are the radial impedances on the inner and outer surfaces.

Consider first the unlagged pipe. The radial intensity at $r = b$, $\phi = 0$ (that is, at the position of the outer surface of the lagging without the lagging present) can be found from the pressure and impedance formulae given in Appendix B. The radial intensity can then be found as described in Appendix A.

A similar procedure is followed when the lagging is present. The radial impedance on the outer surface of the lagging, that is, at $r = b$, $\phi = 0$ is found first as before. The radial impedance formulae in Appendices C and D are then successively used to find the radial impedance at the pipe surface. The pressure at $\phi = 0$ at the pipe surface can be found as the pipe radial velocity and the radial impedance at this point is known. The pressure formulae in Appendices C and D are then successively applied to give the pressure on the outer surface of the lagging at $\phi = 0$. This pressure, along with the original radial impedance at this point, gives the radial intensity at $\phi = 0$ on the outer surface of the lagging. The insertion loss can be obtained from this intensity and that found at the same point without the lagging.

TYPICAL RESULTS

Suppose that a 200mm Schedule 40 steel pipe is lagged with a 50mm thick fibreglass blanket spaced 25mm from the pipe. The fibreglass has a flow resistivity of 15000 Rayls/m. The outer surface of the fibreglass is itself covered with 0.3mm thick steel sheet. The dispersion curves which relate the axial wavenumber, k_Z and the frequency for the structural waves in the pipe can be determined by the procedure described in Appendix E. These curves for n = 0, 1 and 2 are plotted in Figure 3. These curves show that wave propagation in the breathing (n = 0) and ovalling (n = 2) modes cannot occur until critical frequencies are reached. The breathing mode wave cannot propagate until a frequency of 8260 Hz is exceeded and the ovalling mode wave cannot propagate until the frequency exceeds 750 Hz. The dotted line shown on Figure 3 is the dispersion relationship for acoustic waves. The pipe can only radiate sound when the structural axial wavenumber, k_Z is greater than the acoustical wavenumber, k. This criterion is always satisfied for the breathing (n = 0) and ovalling (n = 2) modes once they are "cut on". Only at frequencies below 50 Hz is this criterion not satisfied for the bending (n = 1) mode.

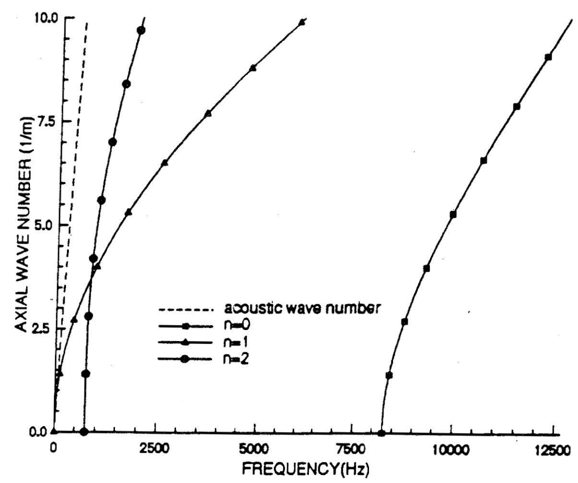
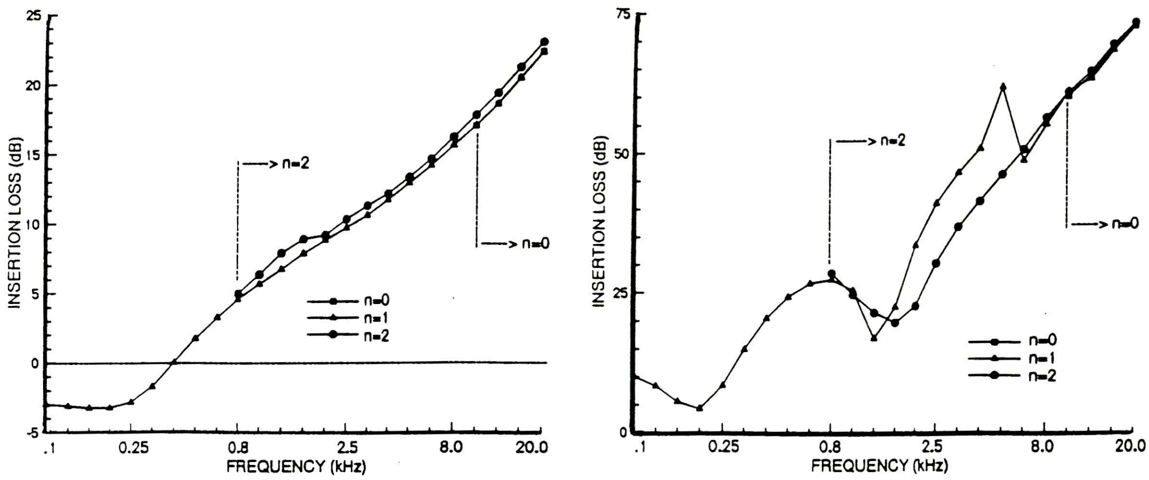


Figure 3. Dispersion Curves

The axial wavenumber - frequency relationships shown in Figure 3 were used in the computational procedure already described to determine the insertion losses. The insertion losses were found in 1/3 octave bands by finding the intensities at the point $r = b$, $\phi = 0$ with and without the lagging at eleven frequencies in each 1/3 octave band. The average intensities and the insertion losses were then found. The insertion losses for the three modes for the fibreglass jacket alone and the complete lagging are shown in Figure 4.

COMMENTS

The results shown in Figures 3 and 4 indicate the importance of having a lagging which is effective in controlling the sound radiated by the bending (n = 1) mode of pipe vibration. The importance of controlling the sound radiated by this mode is mainly due to the fact that



(a) **(b)**

Figure 4. 1/3 Octave Band Insertion Losses for (a) Porous Jacket Only and (b) Complete Lagging

the pipe can vibrate in the bending (n = 1) mode at all frequencies. It cannot do this with the other modes such as the breathing (n = 0) and ovalling (n = 2) modes. In many applications a pipe lagging which is designed to produce satisfactory attenuation of the sound radiated by the bending (n = 1) mode will be a satisfactory pipe lagging.

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APPENDIX A

EXPRESSION FOR RADIAL INTENSITY IN AIR SURROUNDING PIPE

The computational procedure described in the body of the paper is based on the use of pressure and radial impedance and so it is convenient to derive the expression for the radial intensity terms of these quantities. At a point defined by r, ϕ and z in the air surrounding the lagged pipe the complex representations of the acoustic pressure, $p = P\cos(\omega t + \phi_p)$ and the radial particle velocity, $u_r = U_r \cos(\omega t + \phi_u)$ are $\mathbf{p} = \mathbf{P}\exp[j\omega t]$ and $\mathbf{u_r} = \mathbf{U_r}\exp[j\omega t]$ where $\mathbf{P} = P\exp[j\phi_p]$ and $\mathbf{U_r} = U_r\exp[j\phi_u]$. The radial impedance, $\mathbf{z_r}$, is given by $\mathbf{z_r} = \mathbf{P}/\mathbf{U_r} = |\mathbf{P}/\mathbf{U_r}| \angle \phi_p - \phi_u = P/U_r \angle \phi_p - \phi_u$. The instantaneous radial intensity, I_r is given by pu_r and the average radial intensity, $\overline{I}_r$, obtained by averaging I_r over one cycle, is given by $\overline{I}_r = 0.5\,PU_r\cos(\phi_p - \phi_u)$. This expression for $\overline{I}_r$ can be rewritten as equation (A.1) by use of the results $P = U_r|\mathbf{z_r}|$ and $\mathrm{Re}\{\mathbf{z_r}\} = |\mathbf{z_r}|\cos(\phi_p - \phi_u)$.

$$\overline{I}_r = 0.5|\mathbf{P}/\mathbf{z_r}|^2\,\mathrm{Re}\{\mathbf{z_r}\} \qquad (A.1)$$

APPENDIX B

EXPRESSIONS FOR PRESSURE AND RADIAL IMPEDANCE IN SURROUNDING AIR

Suppose that the outer surface of the infinitely long pipe of radius a whose axis is aligned with the z axis has a radial velocity $\mathbf{v_r}$ whose complex representation is $\mathbf{v_r} = \mathbf{V_r}\cos n\phi\,\exp[j(\omega t - k_z z)]$. The acoustic pressure, whose complex representation is $\mathbf{p}$, produced by this pipe motion at a point defined by r and ϕ can be found by expressing the wave equation $\nabla^2\mathbf{p} + k^2\mathbf{p} = 0$ in cylindrical coordinates, assuming a separable solution $\mathbf{p} = \mathbf{P}R(r)\,\Phi(\phi)\exp[j(\omega t - k_z z)]$ and solving the resulting differential equations B(1) for R(r) and $\Phi(\phi)$ to give the solutions defined by equation B(2).

$$\frac{d^2R}{dr^2} + \frac{1}{r}\frac{dR}{dr} + (k^2 - k_z^2 - \frac{n^2}{r^2})R = 0, \quad (a) \qquad \frac{d^2\Phi}{d\phi^2} + n^2\Phi = 0 \quad (b) \qquad (B.1)$$

$$R(r) = A\,J_n(k_r r) + B\,N_n(k_r r), \quad (a) \qquad \Phi(\phi) = C\cos n\phi \quad (b) \qquad (B.2)$$

A, B and C are constants and J_n and N_n are the n^{th} order Bessel & Neumann functions in which $k_r^2 = k^2 - k_z^2$. The radial particle velocity, u_r, can be found from the linearized Euler equation $\partial p/\partial r = -\rho\partial u_r/\partial t$. This radial particle velocity can be made equal to the radial velocity, $\mathbf{v_r}$ of the pipe surface. The radiation condition provides a further boundary condition and so the constants AC and BC can be determined. The pressure at the point r=b, ϕ=0 then can be found as can the radial impedance. They are as follows:

$$\mathbf{p} = -j\mathbf{V_r}\,\rho c\,\frac{k}{k_r}\frac{J_n(k_r b) - jN_n(k_r b)}{J_n'(k_r a) - jN_n'(k_r a)}\exp[j(\omega t - k_z z)] \quad (a)$$

$$\frac{\mathbf{z_r}}{\rho c} = -j\frac{k}{k_r}\left[\frac{J_n(k_r b) - jN_n(k_r b)}{J_n'(k_r b) - jN_n'(k_r b)}\right] \quad (b) \qquad (B.3)$$

APPENDIX C

EXPRESSIONS RELATING THE RADIAL IMPEDANCES AND PRESSURES AT THE INNER AND OUTER SURFACES OF A RIGID POROUS JACKET

A rigid porous jacket whose inner and outer radii are a and b respectively is made of a homogeneous and isotropic material with a flow resistivity of R_1 and porosity of Ω. The arrangement is shown in Figure C.1. The propagation of sound waves in this material is governed by the wave equation. $\nabla^2\mathbf{p} + \mathbf{k}^2\mathbf{p} = 0$. The term $\mathbf{k}$ is the complex wave number. The acoustic pressure and radial particle velocity in this jacket are assumed to vary with ϕ according to $\cos n\phi$.

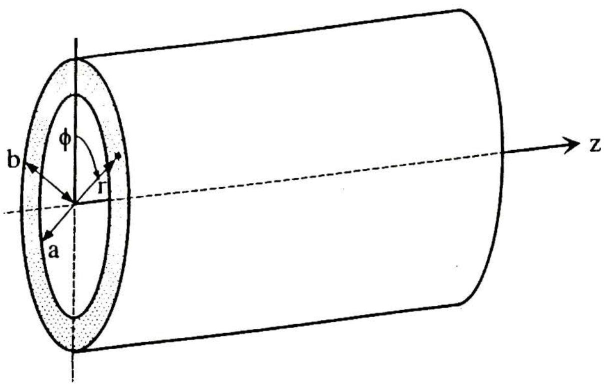


Figure C.1 Porous Jacket Model

The wave equation can be expressed in cylindrical coordinates and solved to derive expressions for the acoustic pressure and the radial particle velocity. It is then possible to derive an expression which allows the impedance in the radial direction on the inner surface of the porous jacket, ie at r = a, to be found when that on the outer surface, ie at r = b, is known. This expression is given by equation C.1(a). It is also possible to derive an expression which enables the pressure on the outer surface, ie at r = b, to be found when that on the inner surface, ie at r = a, is known as are the radial impedances on these surfaces. This expression is given by equation C.1(b).

$$\frac{\mathbf{z}_{ra}}{\mathbf{Z}_o} = -j\frac{\mathbf{k}\, J_n(\mathbf{k}_r a) + \alpha N_n(\mathbf{k}_r a)}{\mathbf{k}_r\, J_n'(\mathbf{k}_r a) + \alpha N_n'(\mathbf{k}_r a)} \quad \text{(a) and} \quad \mathbf{P}_b = \mathbf{P}_a \frac{J_n(\mathbf{k}_r b) + \alpha N_n(\mathbf{k}_r b)}{J_n(\mathbf{k}_r a) + \alpha N_n(\mathbf{k}_r a)} \quad \text{(b)} \qquad (C.1)$$

where

$$\alpha = -\frac{[\mathbf{z}_{rb}/\mathbf{Z}_o\, \mathbf{k}_r/\mathbf{k}\, J_n'(\mathbf{k}_r b) + jJ_n(\mathbf{k}_r b)]}{[\mathbf{z}_{rb}/\mathbf{Z}_o\, \mathbf{k}_r/\mathbf{k}\, N_n'(\mathbf{k}_r b) + jN_n(\mathbf{k}_r b)]} \qquad (C.2)$$

The quantity $\mathbf{k}_r$ which appears in these equations is given by $\mathbf{k}_r^2 = \mathbf{k}^2 - \mathbf{k}_z^2$. Various models can be used to determine $\mathbf{k}$ and $\mathbf{Z}_o$ which are needed in the evaluation of equations (C.1) and (C.2). The semi-empirical formulae of Delany and Bazely [5] and Mechel [6] are convenient. These formulae are given below in forms compatible with the notation used in this paper. They are expressed in terms of the dimensionless quantity $C=R_1/\rho f$ where R_1 is the flow resistivity of the porous material, ρ is the density of the gas involved and f is the frequency. Equations (C.3) apply when $C\geq 60$ and equations (C.4) apply when $C<60$.

$$\mathbf{k} = -jk[-1.466 + j0.212C]^{1/2} \qquad \frac{\mathbf{Z}_\Omega}{\rho c} = [(C/2\pi + j1.403)/(-1.466 + j0.212C)^{1/2}] \qquad (C.3)$$

$$\mathbf{k} = -jk[0.189C^{0.618} + j(1+0.0978C^{0.693})] \qquad \frac{\mathbf{Z}_\Omega}{\rho c} = [1+0.0489C^{0.754} - j0.087C^{0.731}] \qquad (C.4)$$

Equations C.1 and C.2 can be used also when the jacket is an air space rather than a porous layer. In this case $\mathbf{k}$ is replaced by ω/c and $\mathbf{Z}_o$ is replaced by ρc.

APPENDIX D

EXPRESSIONS RELATING THE RADIAL IMPEDANCES AND PRESSURES AT THE INNER AND OUTER SURFACES OF A CYLINDRICAL IMPERVIOUS JACKET

A thin walled cylindrical shell of nominal radius a and wall thickness h is made of a material with a Young's Modulus of E, Poisson's Ratio ν , density ρ and loss factor of η . The arrangement is shown in Figure D.1. Donnell's formulation of the equations of motion for such a shell is expressed in terms of w, v and u, the displacements of the mid-surface of the shell in the r, ϕ and z directions as shown in Figure D.1. His equations are given in [7].

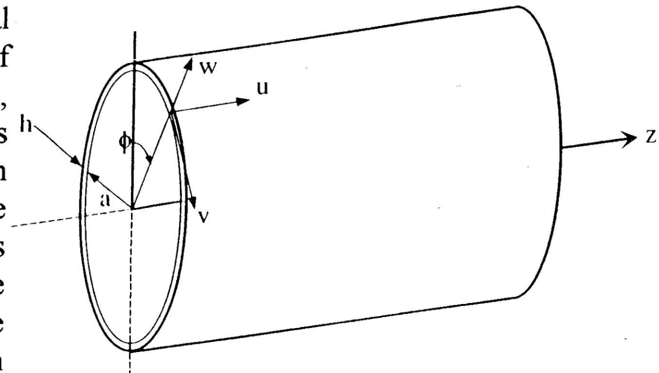


Figure D.1 Cylindrical Impervious Jacket Model

$$\frac{\partial^2 u}{\partial z^2}+\frac{1-\nu}{2a^2}\frac{\partial^2 u}{\partial \phi^2}+\frac{1+\nu}{2a}\frac{\partial^2 v}{\partial z \partial \phi}+\frac{\nu}{a}\frac{\partial w}{\partial z}-\frac{\ddot{u}}{c_p^2}=0 \qquad \text{(D.1a)}$$

$$\frac{1+\nu}{2a}\frac{\partial^2 u}{\partial z \partial \phi}+\frac{1-\nu}{2}\frac{\partial^2 v}{\partial z^2}+\frac{1}{a^2}\frac{\partial^2 v}{\partial \phi^2}+\frac{1}{a^2}\frac{\partial w}{\partial \phi}-\frac{\ddot{v}}{c_p^2}=0 \qquad \text{(D.1b)}$$

$$\frac{\nu}{a}\frac{\partial u}{\partial z}+\frac{1}{a^2}\frac{\partial v}{\partial \phi}+\frac{w}{a^2}+\beta_o^2\left(a^2\frac{\partial^4 w}{\partial z^4}+2\frac{\partial^4 w}{\partial z^2 \partial \phi^2}+\frac{1}{a^2}\frac{\partial^4 w}{\partial \phi^4}\right)+\frac{\ddot{w}}{c_p^2}=\frac{p_a(1-\nu^2)}{Eh} \qquad \text{(D.1c)}$$

The terms β_o^2 and c_p^2 are given by $\beta_o^2 = h^2/12a^2$ and $c_p^2 = E/\rho(1-\nu^2)$.

The term p_a which appears in equation (D.1c) is a function of time, ϕ and z. It is the net pressure which is applied to the cylindrical shell and it is assumed here that it has the complex representation given by equation (D.2) in which the subscripts i and o relate to the pressures inside and outside the cylindrical shell.

$$\mathbf{p}_a = (\mathbf{P}_i - \mathbf{P}_o)\cos n\phi \exp[j(\omega t - k_z z)] \qquad \text{(D.2)}$$

In view of the assumed form of loading on the cylindrical shell, that is, equation (D.2), the complex representations of the u, v and w displacements are given by equations D.3.

$$\mathbf{u} = \mathbf{U}\cos n\phi \exp[j(\omega t - k_z z)], \mathbf{v} = \mathbf{V}\sin n\phi \exp[j(\omega t - k_z z)], \mathbf{w} = \mathbf{W}\cos n\phi \exp[j(\omega t - k_z z)] \qquad \text{(D.3)}$$

Substitution of equations D.3 into equations D.1 leads to the radial impedance and pressure equations D.4 and D.5.

$$z_i = z_o + j\gamma/\omega \qquad (D.4) \qquad\qquad \mathbf{P_o} = \mathbf{P_i} - j\gamma\mathbf{U}/\omega \qquad (D.5)$$

γ is given by equation (D.6). $\mathbf{P_o}$, $\mathbf{P_i}$ and $\mathbf{U}$ are the complex representations of the pressures and radial particle velocities at ϕ=0.

$$\gamma = \frac{\mathbf{E}h}{(1-\nu^2)}\left[\frac{\mathbf{L(QC-BR)C+M(AR-PC)C}}{\mathbf{A(QC-BR)+B(AR-PC)}} - \mathbf{N}\right] \qquad (D.6)$$

E in this equation is the complex Young's Modulus. It is given by **E** = E(1+jη). E is the conventional Young's Modulus and η is the loss factor for the material of the cylindrical shell. The terms **A, B, C, P, Q, R, L, M** and **N** are given by the equations:

$$\mathbf{A} = (\omega/c_p)^2 - k_z^2 - n^2(1-\nu)/2a^2; \qquad \mathbf{B} = -jnk_z(1+\nu)/2a; \qquad \mathbf{C} = -jk_z\nu/a;$$

$$\mathbf{P} = jnk_z(1+\nu)/2a; \quad \mathbf{Q} = (\omega/c_p)^2 - (1-\nu)k_z^2/2 - (n/a)^2; \qquad \mathbf{R} = -n/a^2; \quad (D.7)$$

$$\mathbf{L} = -jk_z\nu/a; \quad \mathbf{M} = n/a^2 \text{ and } \mathbf{N} = 1/a^2 + \beta_o^2(a^2k_z^4 + 2n^2k_z^2 + n^4/a^2) - \omega^2/c_p^2 .$$

APPENDIX E

DERIVATION OF THE DISPERSION RELATIONSHIPS FOR A THIN WALLED CYLINDRICAL SHELL

The dispersion relationships which relate the axial wavenumber, k_z and the frequency, ω for the various modes of vibration, that is, n = 0, 1, 2 can be obtained by substituting the solutions given by equations (D.3) into the homogeneous forms of equations (D.1). The resulting three equations involving **U**, **V** and **W** have non-trivial solutions for **U**, **V** and **W** only if the determinant of the coefficients of **U**, **V** and **W** is equal to zero. Thus, in terms of **A**, **B**, **C** etc as defined by equations (D.7), it is necessary that:

$$\det\begin{vmatrix}\mathbf{A,\ B,\ C}\\ \mathbf{P,\ Q,\ R}\\ \mathbf{L, M, N}\end{vmatrix} = 0 \qquad (E.1)$$

Since **A**, **B**, **C** etc. are expressed terms of k_z and ω, equation (E.1) defines the dispersion relationship which must exist between k_z and ω for a given thin walled cylindrical shell (as defined by a, c_p and ν) and a given mode (as defined by n).

John Bowers. Paper for AAS Annual Conference 1992

Title: Acoustic Reaction, to aviod Legal Action:

I received a call one day, from the manager of a Freezer Company in a large country town about 90 Km. away. He stated that as a result of pending legal action by housing occupants in a nearby residential area, certain steps had been taken to reduce noise from the freezer plant. He wanted me to measure sound levels so that the affect of the treatment could be evaluated. It was arranged to do this on a Saturday afternoon when other noise in the area would be low.

Figure 1.

On arrival at the plant, I was shown that partial enclosure of the number 2 blast freeze compressor had been carried out, and the internal walls of this enclosure were lined with slagwool type material. He also pointed out that the complaints were received from two corner residences across the street, and another further away along a side street. He also said that there was a 6 metre high fence still to be built along the boundry of the freezer property. Number 2 blast freeze compressor was the only compressor of concern but overnight loading and unloading of trucks was also a subject of complaint.

Figure 2.

I measured sound levels with No. 2 Blast freeze compressor on and all other equipment shut down. The pattern indicated structural borne noise from the building as well as airborne noise from the compressor. I pointed out that as the building was higher than the proposed 6 metre fence that this would help with forklift movements and truck loading but would not be of any great use if the compressor noise is still a problem. Compressor noise levels at the footpath across the street were barely detectable above noise from a busy highway some 1/2 Km. away and were determined at around 45 dBA (Fast) during very brief lulls in the traffic noise. I stated that I did not see that a complaint could be justified. Any previous sound level measurements were in the hands of the company's legal advisors and were not available on a Saturday afternoon, so an immediate evaluation of any noise reduction achieved could not be made.

Figure 1.

Several weeks later I had an urgent call from the Freezer Company. The two complainants closest to the Plant were satisfied that improvements had been made and had dropped out from further action. The other, some distance away from the plant, had checks made by the E.P.A. and were pressing claims against both the Company and the Local Council through the courts. The claims were on the basis that a 5 dB penalty added for tonal components put the level from No. 2 Blast Freeze compressor above the permitted 45 dB overnight limit for an R4 area.

Figure 3.

I was not permitted any direct contact with the people making the complaint so this meant working in the dark to some extent. There was a noticable tonal component at 490 Hz. from the fans in the blast freeze chamber at the opposite end of the building to the compressor. I measured levels using the 500 Hz octave filter around this area and expressed some doubt in the ability of this tone to carry sufficiently to cause severe problems. The Company had a set of lower speed fans available and decided to fit them. Meanwhile I arranged to measure overnight sound levels around the residential area.

Figure 4.

These charts were taken from the footpath outside of the complainants residences in the early hours of the morning. No. 2 Blast freeze compressor was running along with other units. The noise from the freezer plant was 43 dBA (Fast). The first chart shows that in 11 minutes of continuous monitoring the noise from the highway was above this level all of the time. The second chart included noise from a train leaving the nearby railway station together with highway noise, with only several very brief drops in level to the freezer plant noise.

After fitting of the new blast freeze fans the freezer company, through their legal represenatives, contacted E.P.A. to request a recheck of levels at the home of the Plaintiffs. This was carried out and the advice was that there was no change. I then contacted the relevant E.P.A. officer myself and was told that the sound level inside of the home was higher than the levels at the footpath outside, and that the problem tone was in the 1/10th octave band centred at 315 Hz. This now gave me something to work on.

Another report on work done earlier, including measurement of ground vibrations, was made available. This was sufficient to rule out a direct underground transmission path so structural borne sound from the compressor to the building became the subject for investigation.

Figure 5.

Narrow band frequency analysis showed a peak at 320 Hz. This was a harmonic of the compressor piston frequency at 40 Hz.

Figure 6.

Vibration frequency analysis from the compressor, through the refridgerant delivery and return lines and in the building structure, showed a strong pulsation in the refridgerant return line from the evaporators to the compressor. This line was a 75mm diameter pipe clamped ridgidly to the wall columns of the building. Vibration measurement on the wall showed velocity measurements of of 0.25mm/sec at 320 Hz.

Figure 7.

I worked out dimensions for a single expansion chamber muffler based on the theory in Chapter 12 of the Beranek book "Noise & Vibration Control" for installation in the 75mm return line close to the compressor. I also specified a suitable vibration isolating mounting system that would be required if the pipe vibration was due to direct transfer from the compressor and not a pulsation as I suspected. The company elected to do both. The vibration isolating mountings installed by their own maintenance people were ready first. This made a reduction in the building vibration but the tonal sound component although reduced, was still there.

Figure 8.

The following Saturday morning I received a phone call from the manager of the Freezer Company. He asked was it possible for me to come straight up to the plant and not to worry about bringing any instruments. About one hour and 90 Km later I was greated by the Plant Manager with a beaming smile. The filter had been installed and the tonal sound had disappeared completely. He invited me to come for a walk and we went to meet the people who were making the complaint. When the door knock was answered by a very distraught looking young lady he asked what she thought of the freezer plant noise now. She said straight off that "No 2 Freeze compressor is not operating" he answered that it is on now and has been running all night and that this man, indicating me, has fixed it. The lady turned accusingly to me and said "Well, why didn't you fix it years ago". She then settled down and explained that the walls of the house all hummed with this sound and that it was sufficient to wake them up at night every time the compressor kicked in. The vibration was so strong that you could even feel it in the verandah posts.

Conclusions:

1. Most of the houses in the area were brick on solid footings. This particular house was a prefabricated construction. The area and the directivity of the roof were such as to make a good receptor for airborne sound from the Freezer Plant and a condition of resonance was set up between the structure of the house and the particular driving frequency from the Freezer Plant.

2. It was just on 3 months from the time I first visited the Plant untill the solution to the problem was in place. Earlier access to the complainants would have greatly simplified tracking the source of the complaint, with significant savings in time and cost.

Instrumentation Used:

Impulse precision sound level meter.	Bruel & Kjaer type 2209
12.5mm condenser microphone	Bruel & Kjear type 4163
Octave band filter set	Bruel & Kjaer type 1613
Acoustical calibrator	Bruel & Kjaer type 4320
Vibration Meter	Bruel & Kjaer type 2511
Accelerometer	Bruel & Kjaer type 4370
Tunable band pass filter	Bruel & Kjaer type 1621
Portable graphic level recorder	Bruel & Kjaer type 2306
Tripod & Interconnecting cables	

Bibleography:

Beranek: Noise & Vibration Control.

C M Harris: Handbook of Noise Control.

Bruel & Kjaer: Acoustic Noise Measurements.

Bruel & Kjaer: Mechanical Vibration & Shock Measurements.

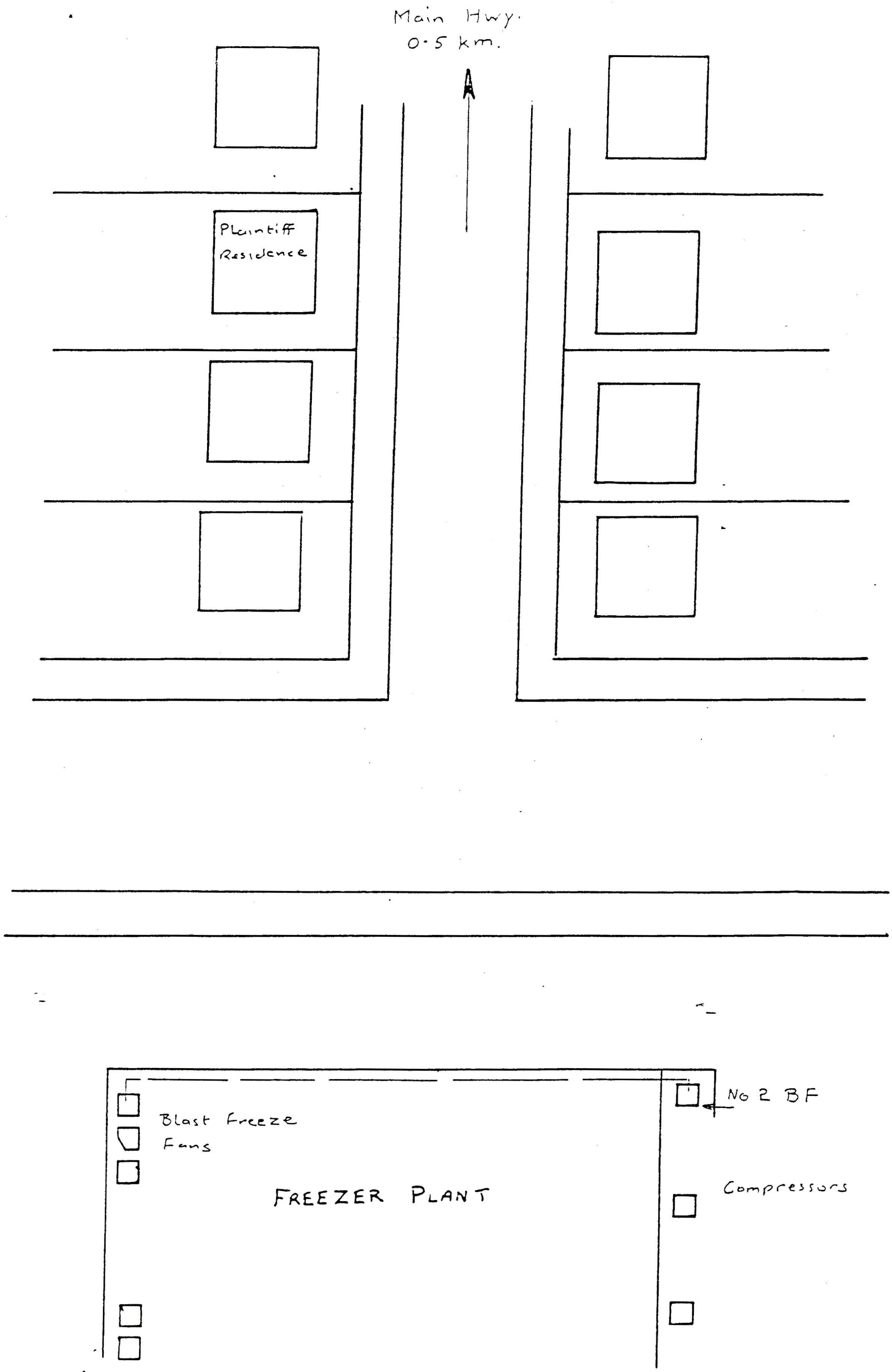
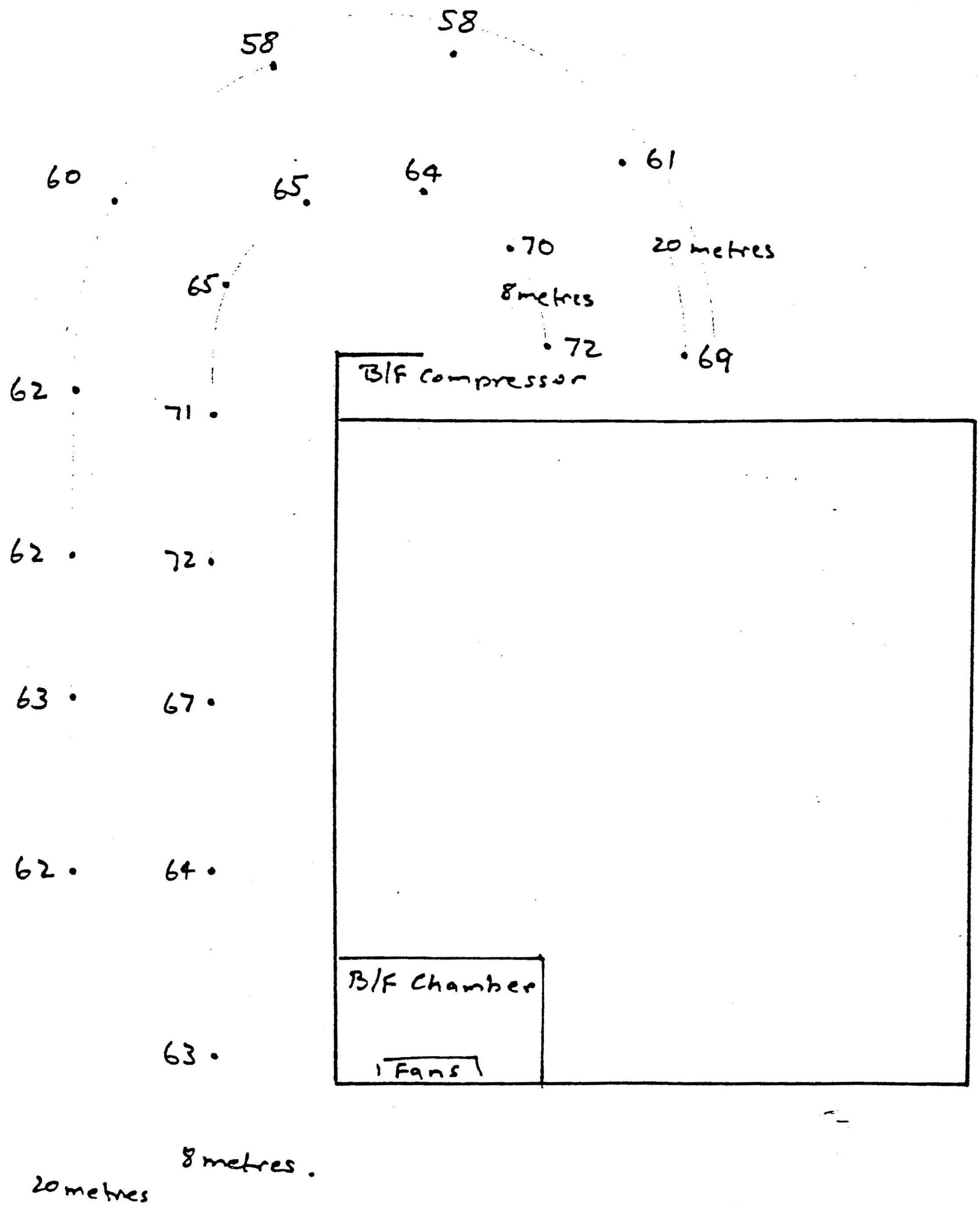
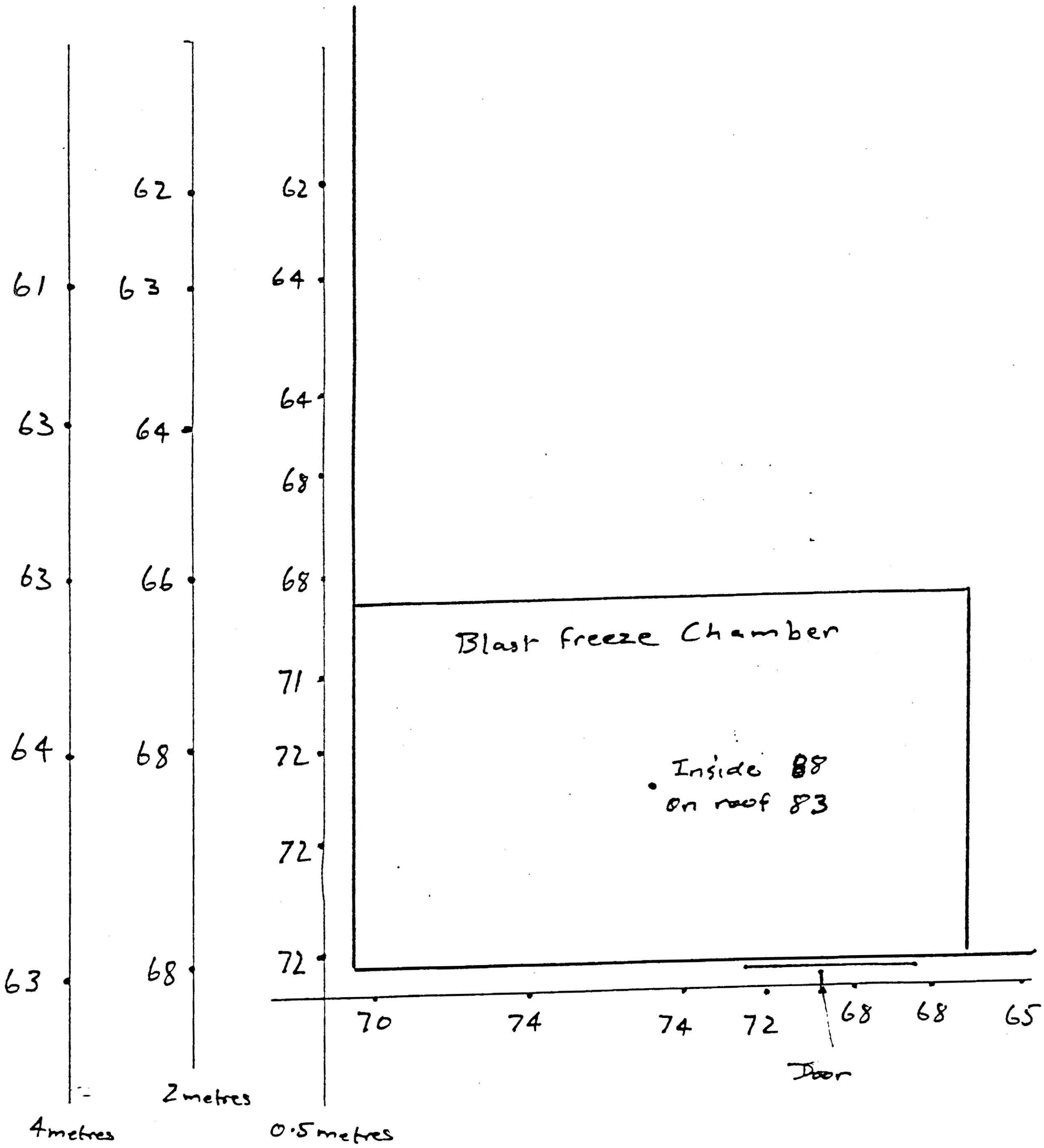

Main Hwy.
0·5 km.
Plaintiff Residence
Blast Freeze Fans
FREEZER PLANT
No 2 BF
Compressors

FIGURE 1



Sound Levels dBA at 8 metres & 20 metres from building

FIGURE 2.



BLAST FREEZE FANS.
dB in 500 Hz Octave Filter.

FIGURE 3.

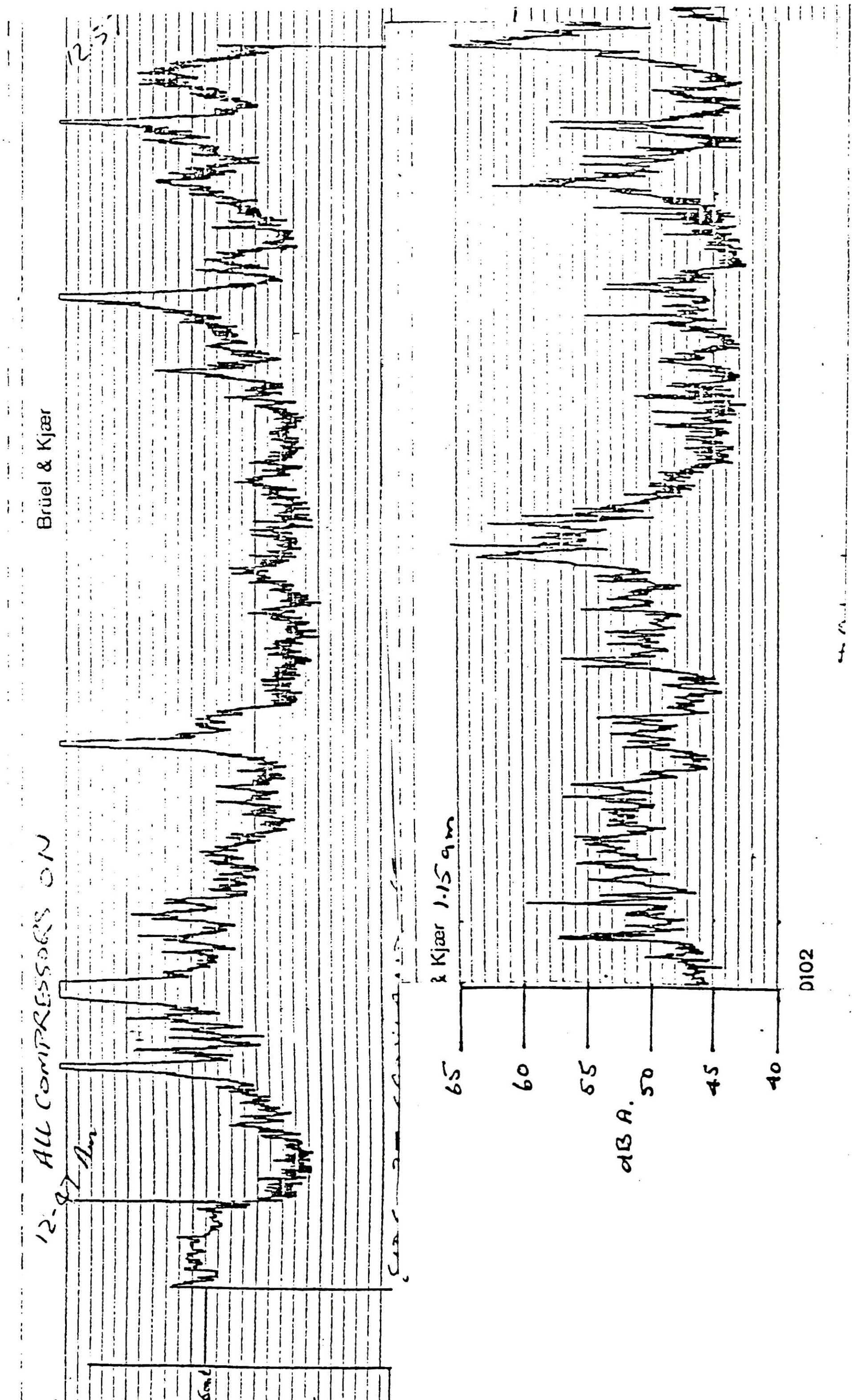

Brüel & Kjær
ALL COMPRESSORS ON
12-27 Am
12.57
65
60
55
50
45
40
& Kjær 1.15 am
dBA.
0102

FIGURE 4

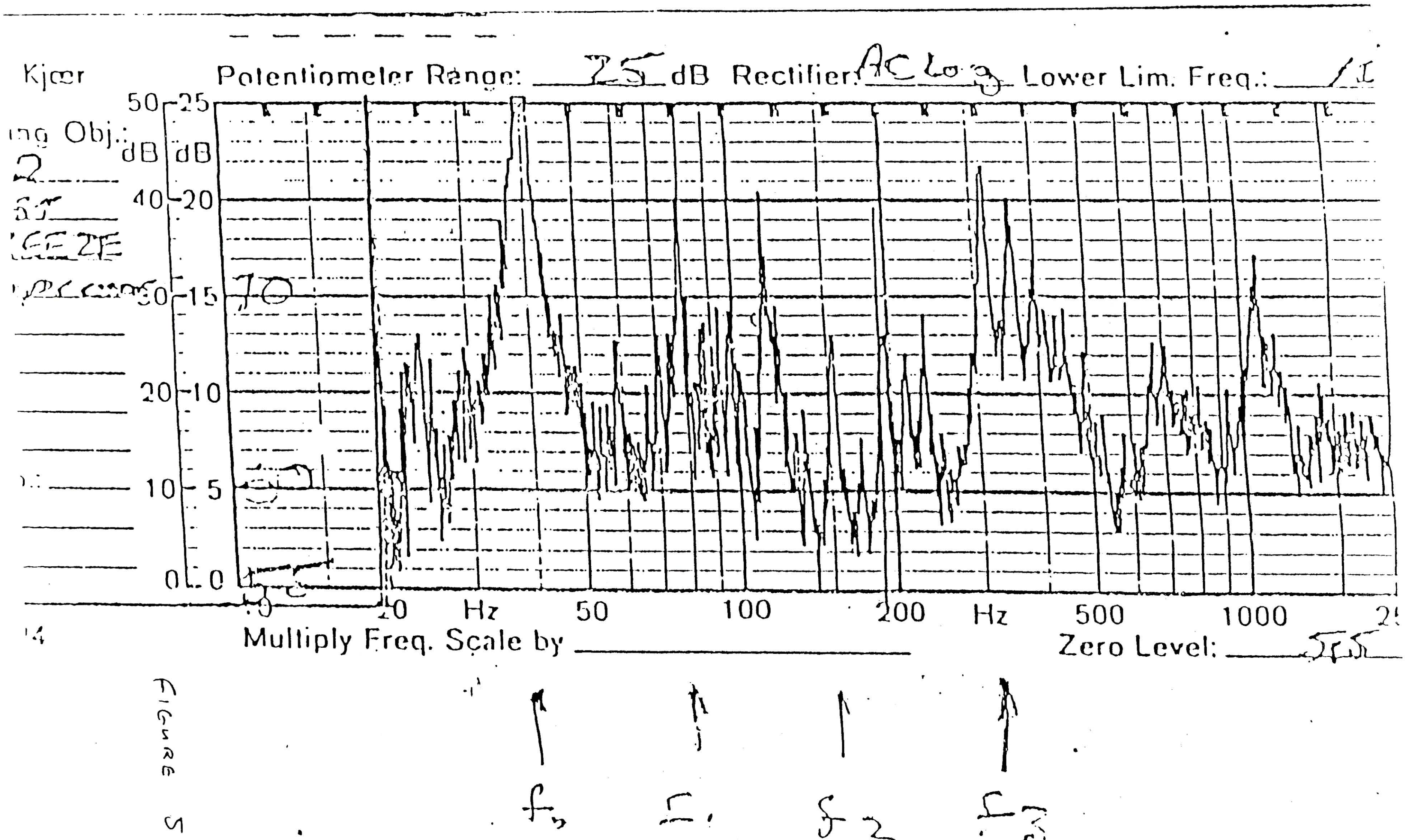
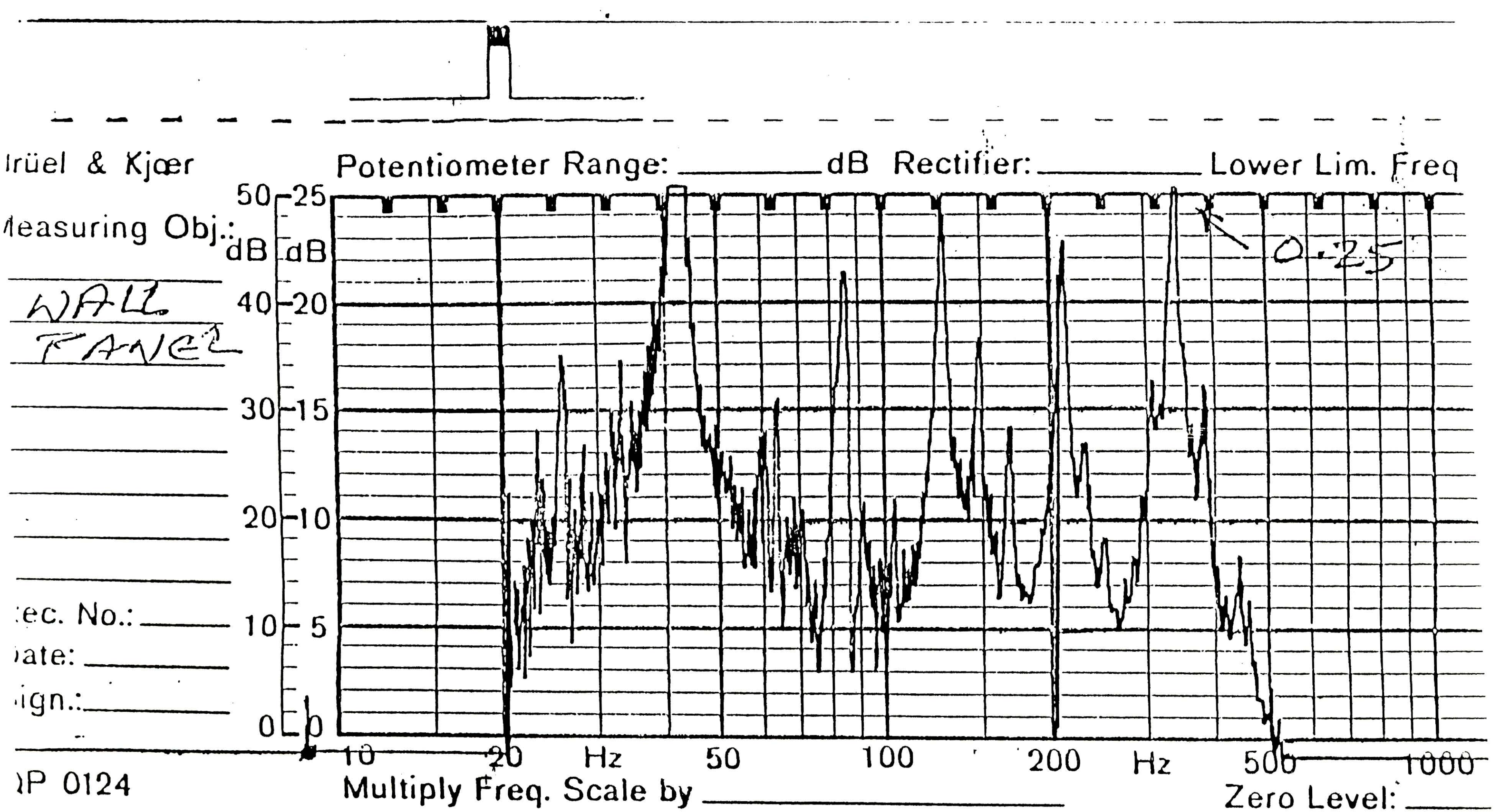
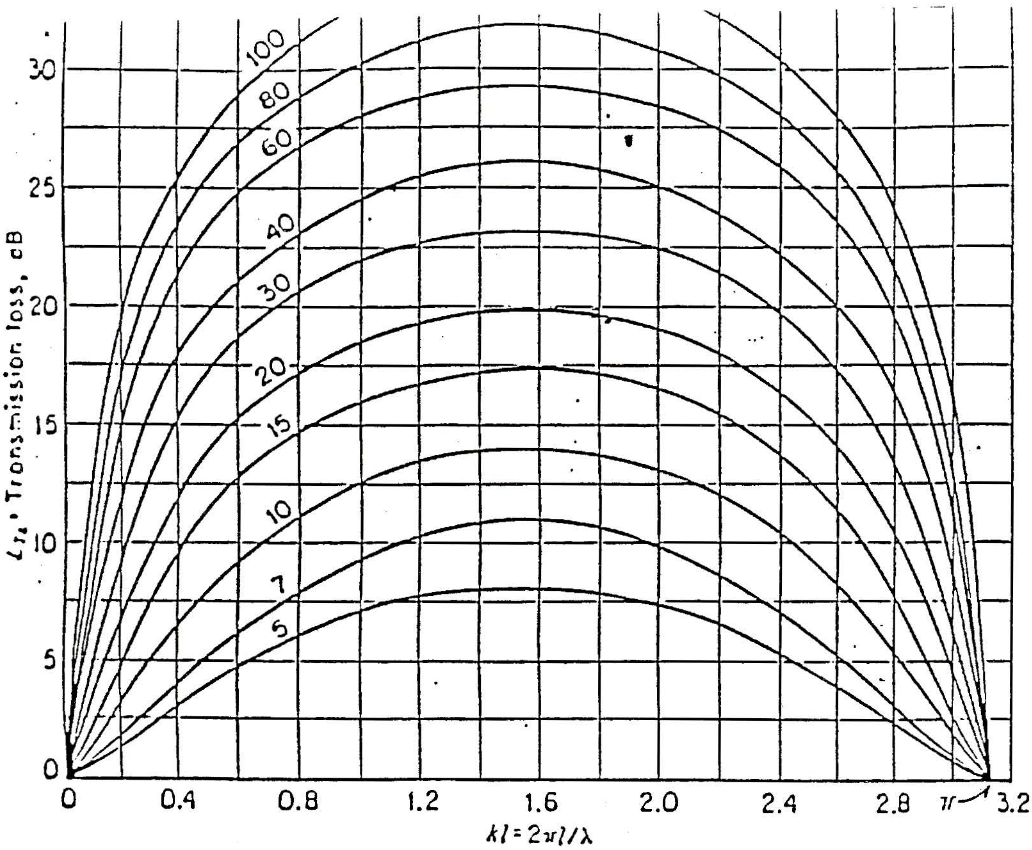

Kjær
Potentiometer Range: 25 dB
Rectifier: AC Log
Lower Lim. Freq.:
Obj.:
50 25
dB dB
40 20
30 15
20 10
10 5
0 0
20
Hz
50
100
200
Hz
500
1000
Multiply Freq. Scale by
Zero Level:
f_0
f_1
f_2
f_3

FIGURE 5



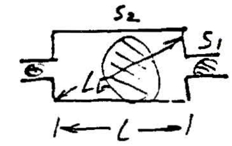
Brüel & Kjær
Potentiometer Range: ______ dB Rectifier: ______ Lower Lim. Freq
Measuring Obj.:
WALL PANEL
Rec. No.:
Date:
Sign.:
QP 0124
50 25 dB dB
40 20
30 15
20 10
10 5
0 0
10 20 Hz 50 100 200 Hz 500 1000
0.25
Multiply Freq. Scale by
Zero Level:

FIGURE 6



$f = 160$ $\quad$ $°C = -20$ to -30

$\lambda = \frac{345}{f}\sqrt{\frac{°C + 273}{295}} = 1.98$ m.



Select M. $20 = \frac{\text{Area } S_2}{S_1}$ $\quad$ dia @ S_1 is 75 mm

$\therefore$ for $m = 20$ $\quad$ dia @ S_2 = 325 mm.

for $kL = 1.6$. $\quad \frac{2\pi L}{\lambda}$ $\quad L = 448$ mm

Lt to be $< 0.8\lambda$. $\quad$ Lt $= < 1.58$ $\quad$ OK

$L_{TL} = 10 \log \left[1 + \frac{1}{4}\left(m - \frac{1}{m}\right)^2 \sin^2 kL\right]$

$= 10 \log \left[1 + \frac{398}{4} \times 1.6\right]$

$= \underline{\underline{22 \text{ dB}}}.$

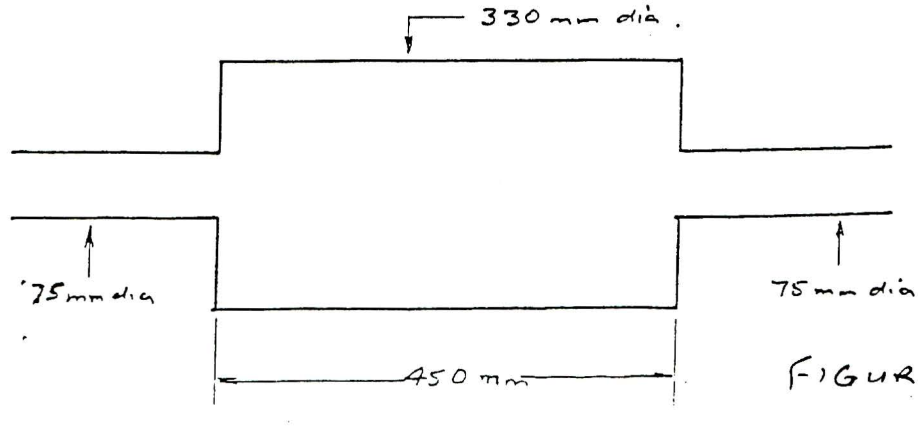


FIGURE 7.

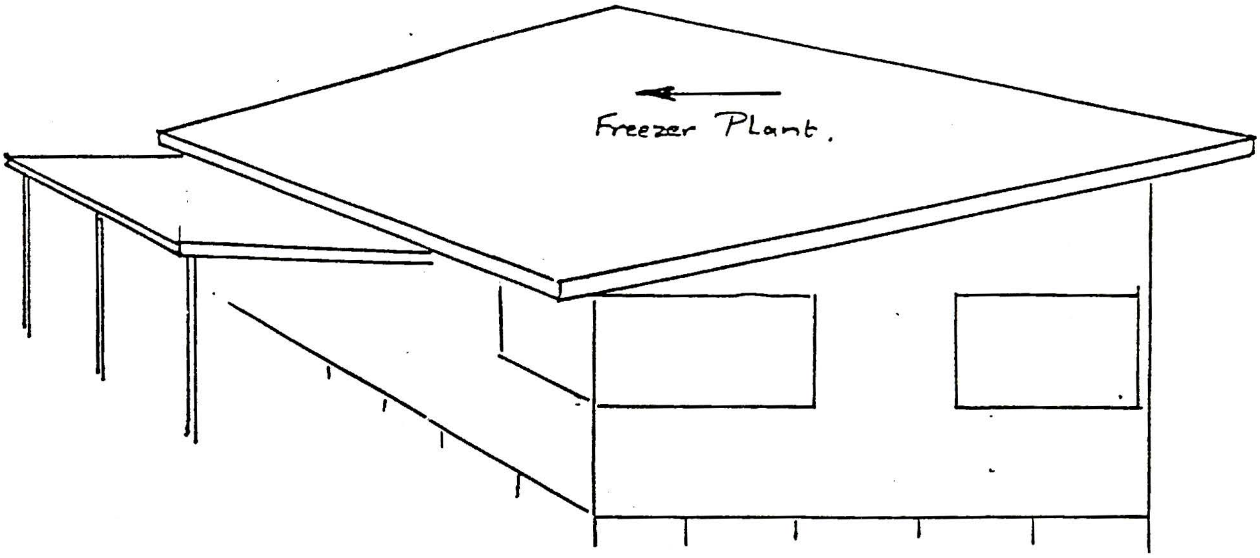


FIGURE 8.

THE DEVELOPMENT OF ACOUSTIC VOLUME VELOCITY SOURCES

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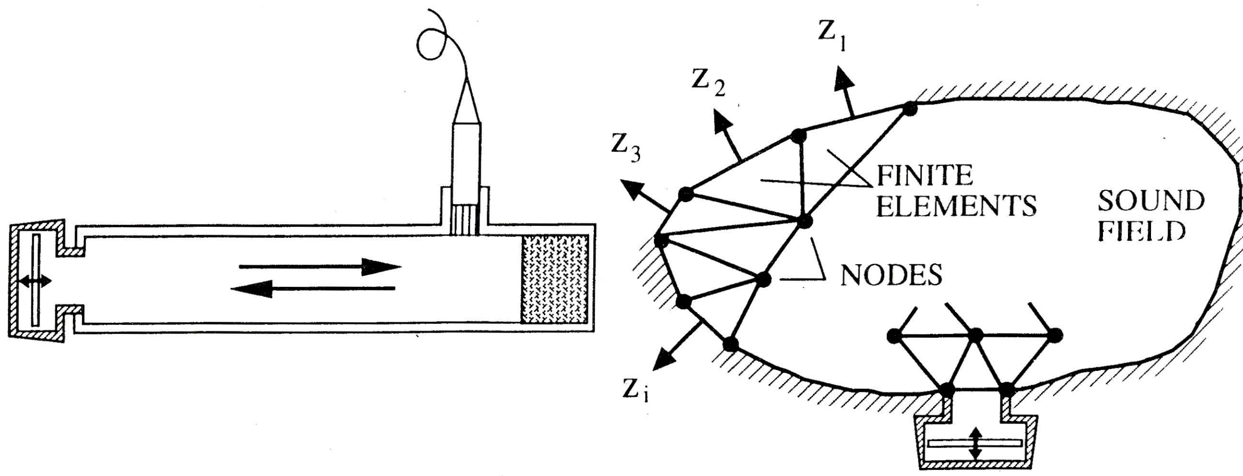
ABSTRACT

Acoustic volume velocity sources can be useful in experimental acoustics. They can be used, for example, to determine the particle velocity information needed in measuring acoustic impedance. The development of modern signal processing instrumentation has allowed volume velocity sources to be used conveniently. Two volume velocity sources developed at The University of New South Wales are described. One is based on a modified acoustic horn driver and the second is a specially constructed device. The methods used for calibrating the devices are described and the results of a performance test based on measuring the input and transfer impedances of a closed end tube are given.

INTRODUCTION

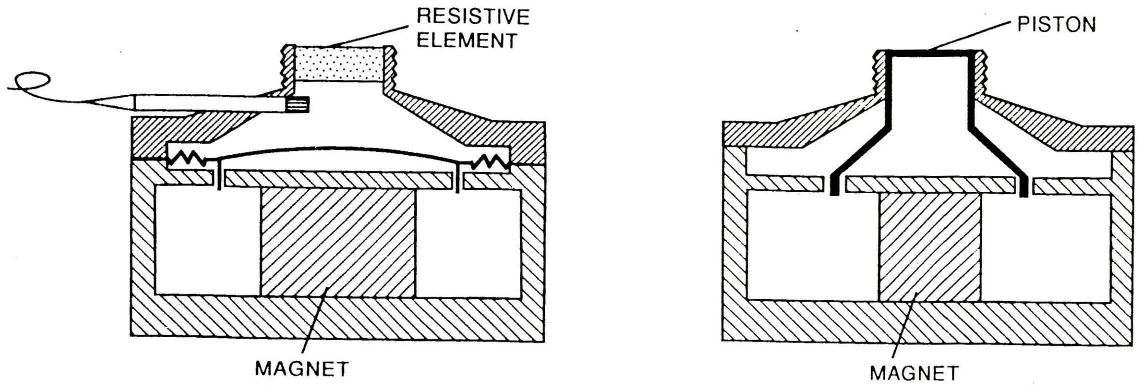
Certain acoustic measurements can be undertaken conveniently if a source of known volume velocity is available. Examples of such measurements are given in references [1] and [2]. Reference [1] describes how a volume velocity source can be used to measure the normal incidence specific acoustic impedance of acoustic materials. The physical arrangement of the device is shown in Figure 1(a). Reference [2] describes how a volume velocity source can be used in experiments relating to the finite element modelling of an acoustical cavity. The "driving force" in many mathematical models of acoustical systems is usually one or more volume velocity sources. The physical arrangement of the system referred to in [2] is shown in Figure 1(b).



(a) Impedance Measurements **(b) Verification of Finite Element Model**
Figure 1. Applications of Volume Velocity Sources

Historically, the attraction of volume velocity sources for use in acoustic measurements has been their ability to obviate the need to measure acoustic particle velocity which is a difficult quantity to measure directly. Volume velocity sources and in particular constant volume velocity sources have been developed with the aim of being useful in measuring the acoustic impedance in systems as diverse as vocal tracts, musical instruments, machinery manifolds and acoustical materials. The development of phase matched microphones has enabled acoustic particle velocity to be found easily and so has reduced the role of volume velocity sources as a means of establishing known particle velocities. However, modern digital signal processing has allowed volume velocity sources to be used in a more versatile manner than before. The technique described in [1] is an example of how modern digital signal processing can be used with a volume velocity source.

Salava [3] has described some of the types of volume velocity source which have been constructed. Briefly there are two basic types of volume velocity source and they are shown in Figure 2. The first type, shown in Figure 2(a), involves a driver, a microphone and a resistive element such as a disc of sintered metal, whose resistance is much greater than the impedance of the system attached to the volume velocity source. The volume velocity produced by the source is obtained from the pressure measured by the microphone and the flow resistance of the resistive element. An obvious disadvantage is the requirement that the system which is attached to the volume velocity source must have a much lower impedance than that of the resistive element. The second type, shown in Figure 2(b), incorporates a diaphragm or piston whose motion is measured. The motion can be measured directly by devices such as capacitive displacement transducers, accelerometers, and velocity coils or indirectly by measuring the pressure fluctuations in a closed cavity behind the piston or diaphragm.

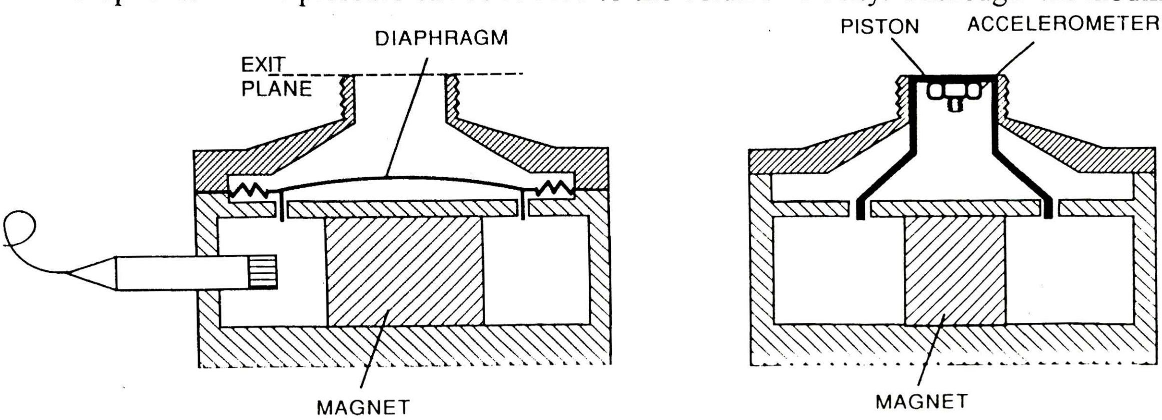


(a) Resistive Element Type **(b) Piston Type**

Figure 2. Types of Volume Velocity Sources

THE UNSW VOLUME VELOCITY SOURCES

The two types of volume velocity source which have been developed at The University of New South Wales are variations of the piston type shown in Figure 2(b). The main features of the two types are shown in Figure 3. The main difference is in how the motion of the diaphragm or piston is measured. The horn driver type of volume velocity source shown in Figure 3(a) was constructed by modifying a horn driver so that a microphone could be inserted into the sealed cavity behind the diaphragm. As the diaphragm moves and gas is displaced into and out of the throat of the horn driver the pressure in the sealed cavity alters and this can be sensed by the microphone. The pressure can be related to the volume velocity. Although the modified horn



(a) Horn Driver Type **(b) Piston Type**

Figure 3. UNSW Volume Velocity Sources

driver type of volume velocity source can be made readily in that all that needs be done is to machine a hole into the body of the horn driver for the microphone, it has several major disadvantages. Firstly, it has a large coupling volume between the diaphragm and the exit plane of the device. Little can be done to reduce this volume. Secondly, the microphone in the closed cavity which backs the diaphragm may not be able to sense accurately the pressure changes in the cavity because the cavity is partitioned into several volumes by the gap in which the coil moves. However, this can be overcome to some extent by greatly widening sections of the gap annulus.

The compliance of the coupling volume can cause serious measurement errors when the impedance presented by the load at the exit plane of the volume velocity source is high. Under such a circumstance, the volume velocity delivered to the load will be small despite large excursions of the diaphragm of the horn driver. In principle this problem can be overcome by measuring the acoustic pressure near the exit plane of the volume velocity source. The arrangement of the volume velocity source to allow this to be done is shown in Figure 4(a). The compliance of the coupling volume can be established by blocking the exit plane of the volume velocity source and measuring the transfer function which relates the pressure measured by the microphone in the cavity to the pressure measured by the microphone close to the exit plane. The model of the modified horn driver is shown in Figure 4(b). This model can be used to consider both the effect of the compliance volume and the calibration procedure.

The piston type of volume velocity source is intended to overcome the preceding shortcomings of the modified horn driver type of volume velocity source. The device shown in Figure 3(b) was constructed from standard components which included a ceramic magnet, a loudspeaker drive coil and a low mass accelerometer. The accelerometer allows the motion of the piston to be determined. A detail of the device which is important to its functioning is the sealing between the piston and the fixed throat parts. Several sealing techniques were used. The one which ultimately was found to be the most successful used an "O" ring.

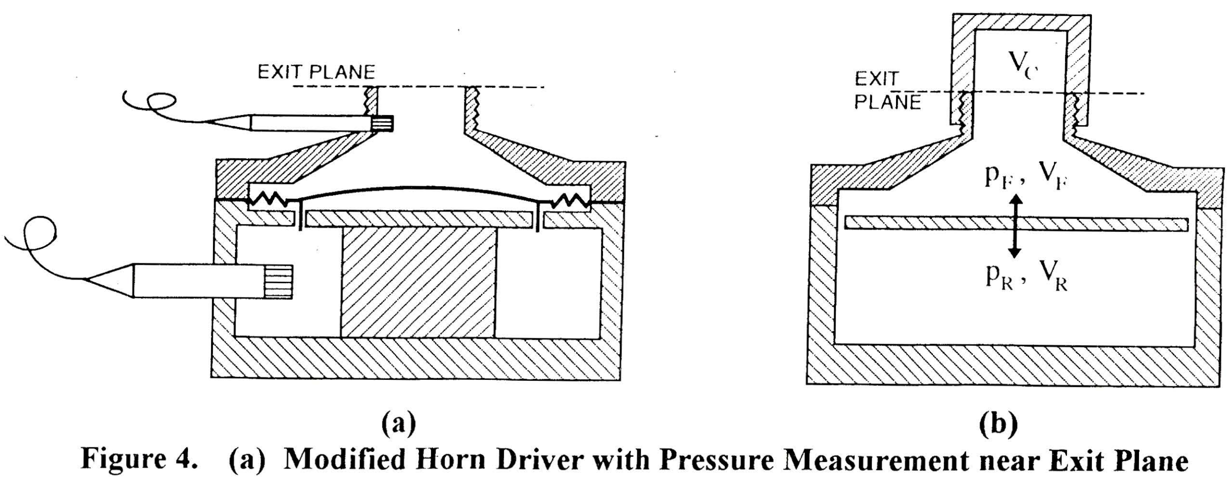


Figure 4. (a) Modified Horn Driver with Pressure Measurement near Exit Plane
(b) Model Used for Analysis of Modified Horn Driver

EQUATIONS FOR MODIFIED HORN DRIVER

The equations which govern the behaviour of the modified horn driver can be determined from the model shown in Figure 4(b). The equilibrium absolute pressure in the regions of volume V_R and V_F on either side of the diaphragm is P_0. The region of volume V_F extends up to the exit plane of the device. If the diaphragm is moved a small distance upwards so that it displaces a small volume ΔV, the acoustic pressures p_R and p_F associated with each of the volumes are as given by equations (1).

$$p_R = -(P_0\gamma / V_R)\Delta V \qquad p_F = (P_0\gamma / V_F)(\Delta V - A\Delta x) \qquad (1)$$

These equations are based on the assumptions that the gas in the regions of volumes V_R and V_F behaves adiabatically and that ΔV is small compared to both V_R and V_F. γ is the specific heats ratio of the gas. A is the cross-sectional area of the exit plane of the device and Δx is the movement of the gas particles at the exit plane. Equations (1) can be used to derive equation (2), the required volume velocity equation, which shows that if V_R and V_F are known and $\mathbf{p_F}$ and $\mathbf{p_R}$ are measured, the volume velocity, $\mathbf{U}$, can be found.

$$\mathbf{U} = -j\omega[(V_F/P_O\gamma)\mathbf{p_F} + (V_R/P_O\gamma)\mathbf{p_R}] \tag{2}$$

$\mathbf{p_F}$, $\mathbf{p_R}$ and $\mathbf{U}$ are the complex representations of the harmonically varying pressures and volume velocity. ω is the angular frequency associated with this harmonic motion. In view of the fact that the microphone in the closed cavity which backs the diaphragm may not be able to sense accurately the pressure changes in the cavity because of the complex shape of this cavity, it is advantageous to consider this volume to be a frequency dependent complex quantity denoted $\mathbf{V_R}(\omega)$. Thus equation (2) can be written as:

$$\mathbf{U} = -j\omega[(V_F/P_O\gamma)\mathbf{p_F} + (\mathbf{V_R}(\omega)/P_O\gamma)\mathbf{p_R}] \tag{3}$$

Usually it is required to find the pressure, $\mathbf{p}$, at some point in an acoustical system per unit volume velocity of the source and so from equation (3):

$$\mathbf{p/U} = (\mathbf{p/p_R})/-j\omega(\mathbf{V_R}(\omega)/P_O\gamma)[1+(V_F/\mathbf{V_R}(\omega))(\mathbf{p_F/p_R})] \tag{4}$$

It can be seen that to determine $\mathbf{p/U}$ it is necessary to determine two transfer functions. The first, $\mathbf{p/p_R}$, relates the pressure at the point of interest to the pressure in the cavity behind the diaphragm. The second, $\mathbf{p_F/p_R}$, relates the pressure at the front of the diaphragm to that in the cavity behind it. This transfer function allows the effect of the load impedance to be considered.

CALIBRATION PROCEDURE FOR MODIFIED HORN DRIVER

The calibration procedure, which essentially involves determining V_F and $\mathbf{V_R}(\omega)$, was undertaken by attaching five small cavities of volumes V_C to the source as shown in Figure 4 (b) and measuring the transfer functions $\mathbf{p_R/p_F}$. Since the volume velocity entering the calibration volume is $j\omega(V_C/P_O\gamma)\mathbf{p_F}$, equation (3) gives

$$(\mathbf{p_R/p_F}) = -(V_F + V_C)/\mathbf{V_R}(\omega). \tag{5}$$

When $V_C = 0$ the corresponding transfer function, denoted $(\mathbf{p_R/p_F})_0$, is given by equation (6).

$$(\mathbf{p_R/p_F})_0 = -V_F/\mathbf{V_R}(\omega) \tag{6}$$

Transfer functions for non-zero values of V_C can be divided or "equalised" by equation (6) to give equation (7).

$$(\mathbf{p_R/p_F})/(\mathbf{p_R/p_F})_0 = \frac{1}{V_F} x V_C + 1 \tag{7}$$

At each frequency the five values of the equalised transfer functions measured with the five calibration cavities were used to least squares fit a straight line of the form of equation (7). The reciprocal of the gradient of this line gives V_F. The calibration cavities had a common diameter of 25 mm and nominal lengths of 0, 3, 8, 13 & 18 mm. The values of the real and imaginary components of the equalised transfer functions at 1000Hz. for the five volumes are shown in Table I.

TABLE I

EQUALISED TRANSFER FUNCTION COMPONENTS AT 1000Hz WITH VARIOUS CALIBRATION CAVITIES

Nominal Cavity Length (mm)	Cavity Volume (m^3)	Real Component	Imaginary Component
0	0	1.00	0.000000
3	1.7328×10^{-6}	1.05	0.000290
8	4.0644×10^{-6}	1.13	-0.000025
13	6.5483×10^{-6}	1.22	-0.000725
18	8.9045×10^{-6}	1.30	-0.002820

A feature of this table is the relatively small magnitudes of the imaginary components of the equalised transfer functions. This is of course expected from equation (7) and it supports the validity of the model which leads to equation (7).

Figure 5 shows the values of V_F determined at 2Hz intervals by the least squares process. Several features of Figure 5 are noteworthy. The major spike evident at about 550Hz is probably associated with resonances in the individual corrugations of the corrugated circular

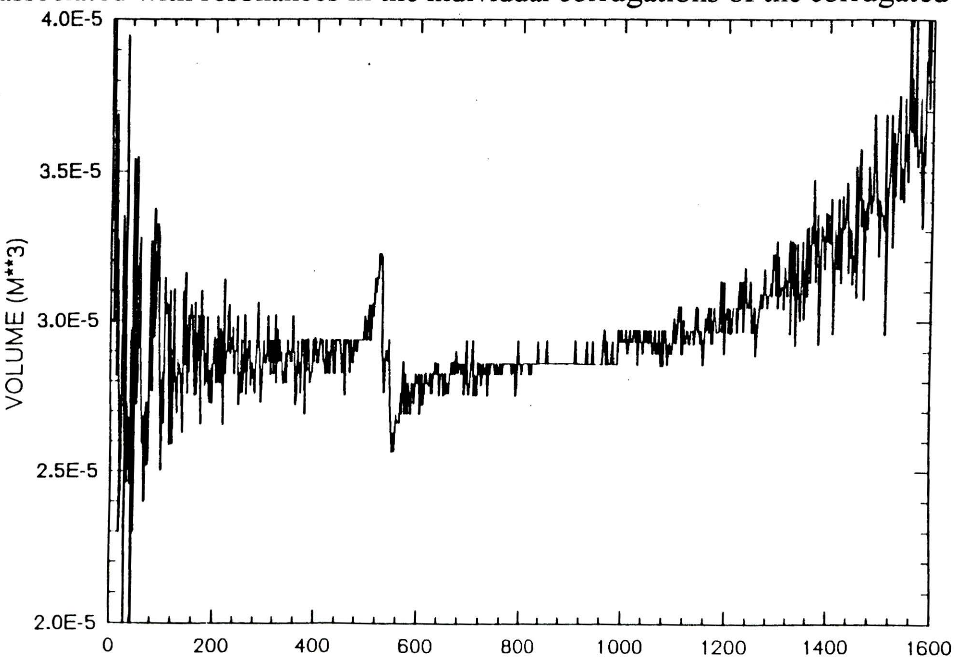


Figure 5. V_F Versus Frequency

flexure which supports the diaphragm. It can be seen that the average value of V_C is slightly different below and above 550Hz. The rising trend of the graph at frequencies above about 1200Hz is associated with wave effects in the volume in front of the diaphragm and the calibration volumes.

Figure 6 shows, at 2Hz intervals, the real and imaginary components of $\mathbf{V_R}(\omega)$ which were obtained from equation (6). The values of V_F given in Figure 5 were used in equation (6).

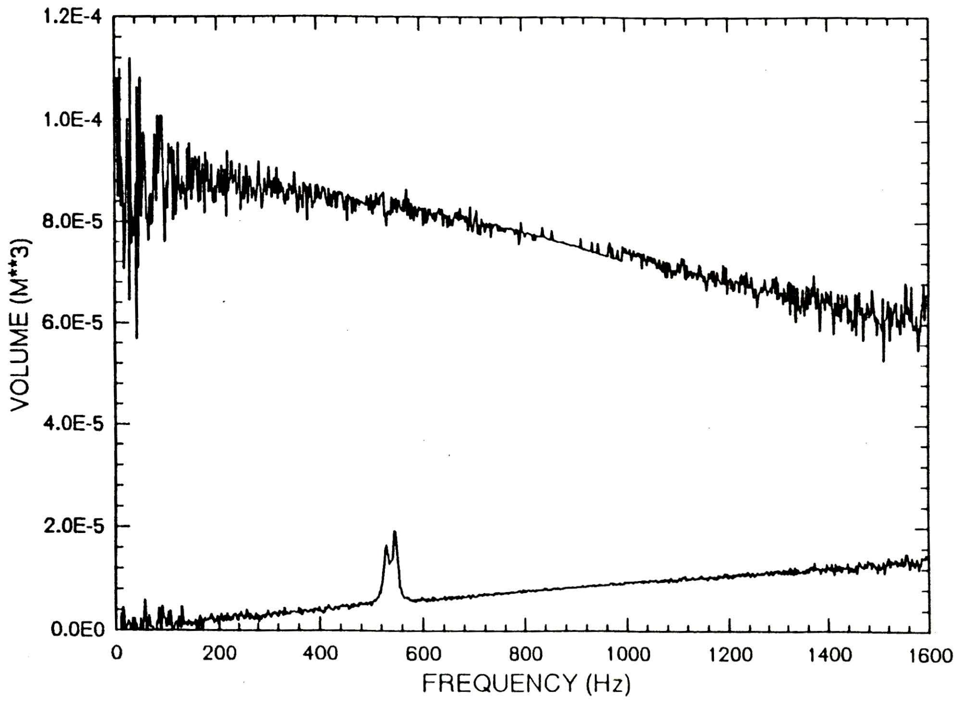


Figure 6. Real (Upper) and Imaginary (Lower) Components of $\mathbf{V_R}(\omega)$

The major spike evident at about 550Hz in Figure 5 is again evident in the imaginary component in Figure 6. It will be seen subsequently that this spike has little effect.

EQUATION FOR PISTON TYPE

The attraction of the piston type source when compared with the modified horn driver type source is that the coupling volume is very small and in fact will be zero if the mean position of the piston can be adjusted so that it lies in the exit plane of the source. Thus the complication of compensating for the effects of this volume is avoided. Further, the volume velocity can be determined directly from the motion of the piston which is sensed by the accelerometer. The volume velocity, **U**, is simply given by equation (8). A is the piston area and **a** is its acceleration.

$$\mathbf{U} = \mathrm{A}\mathbf{a}/j\omega \qquad (8)$$

Since it is usually required to find the pressure, **p**, at some point in an acoustical system per unit volume velocity of the source, equation (8) gives

$$\mathbf{p}/\mathbf{U} = j\omega\mathbf{p}/\mathrm{A}\mathbf{a} \qquad (9)$$

CALIBRATION PROCEDURE FOR PISTON TYPE

It can be seen from equation (8) that, if there is no leakage around the piston, the calibration of the volume velocity source depends upon the calibration of the accelerometer. The exposed outer face of the piston, as shown in Figure 3(b), allows the accelerometer calibration to be checked by conventional back-to-back calibration with a reference accelerometer.

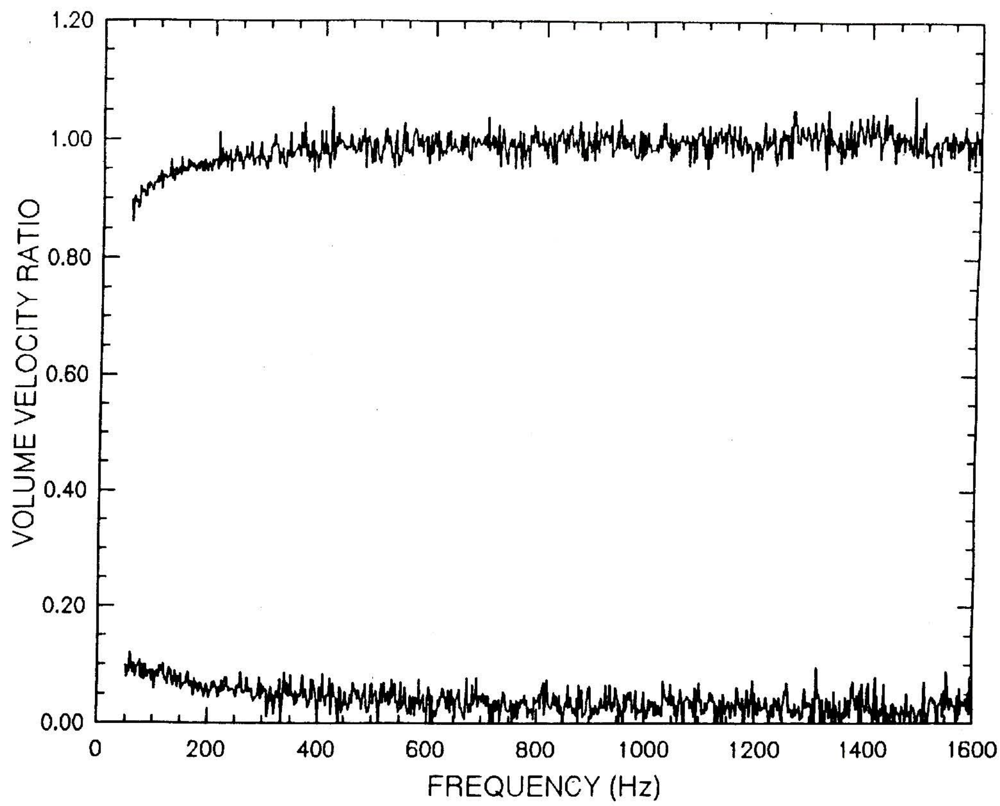


Figure 7. Real (Upper) and Imaginary (Lower) Components of the Ratio of the Volume Velocities Determined by Equations (8) and (10)

The volume velocity, **U**, produced by the source also can be determined by measuring the pressure, **p**, in a small volume V_C attached to the source in the manner shown in Figure 4(b). The analysis used to give equations (1) can be applied again to give equation (10).

$$\mathbf{U} = j\omega(V_C/P_0\gamma)\mathbf{p} \tag{10}$$

The ratio of the volume velocities determined by equations (8) and (10) should be unity. The real and imaginary components of this ratio are plotted in Figure 7.

EVALUATION OF PERFORMANCE

A useful test to evaluate the performance of volume velocity sources is to use them to determine the driving point and transfer impedances for a column of gas in a tube. Singh and Schary [4] used such a test to assess the accuracy of a technique for measuring the acoustic impedance of flow manifolds. Pratt et al [5] adopted a similar test to evaluate a procedure used for measuring the acoustic impedance of brass instruments. The significant features of the test system used here are shown in Figure 8.

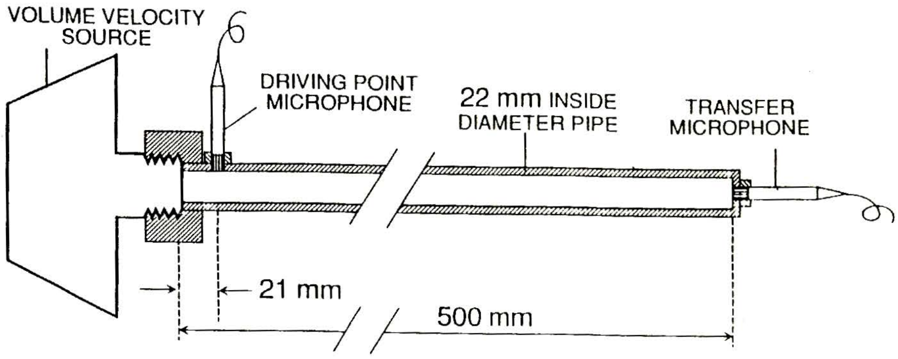


Figure 8. Test System for Assessing the Performance of the Volume Velocity Sources

The acoustic waves in the tube are attenuated as they travel along the tube largely as result of viscous and thermal effects at the tube walls. The attenuation rate can be expressed in terms of the Helmholtz-Kirchhoff wall-attenuation coefficient α_w and so the complex representation of the acoustic pressure the positive and negative travelling plane waves can be written as:

$$\mathbf{p}_+ = \mathbf{P}_+ \exp[-\alpha_w x]\, \exp[j(\omega t - kx)] \text{ and } \mathbf{p}_- = \mathbf{P}_- \exp[+\alpha_w x]\, \exp[j(\omega t + kx)]. \qquad (11)$$

Temkin [6] gives the expression defined by equation (12) for α_w. It involves the speed of sound c_o, the angular frequency ω, the kinematic viscosity v_o, the tube radius R, the specific heats ratio γ and the Prandtl Number P_r.

$$\alpha_w = \frac{1}{c_o}\left(\frac{\omega v_0}{2R^2}\right)^{1/2}\left(1 + \frac{\gamma - 1}{P_r^{1/2}}\right) \qquad (12)$$

The complex representation of the longitudinal particle velocities associated with each wave are given by $\mathbf{u}_+ = \mathbf{p}_+/\rho c$ and $\mathbf{u}_- = -\mathbf{p}_-/\rho c$. The boundary conditions are $\mathbf{u} = \mathbf{u}_+ + \mathbf{u}_- = \mathbf{U}/A$ at x = 0 and $\mathbf{u} = \mathbf{u}_+ + \mathbf{u}_- = 0$ at x = L. These boundary conditions lead to the following equation which relates the pressure at $x = L^*$, $\mathbf{p}_L{}^*$, to $\mathbf{U}$, the volume velocity at x = 0.

$$\mathbf{p}_L{}^*/\mathbf{U} = -j(\rho c/A)(\cos[(k-j\alpha_w)(L - L^*)]/(\sin[(k-j\alpha_w)L]) \qquad (13)$$

Equation (13) gives the driving point impedance when $L^* = 0$ and a transfer impedance when $0 < L^* < L$.

Figure 9 shows the modulus and phase of the "driving point impedance" computed from equation (13) with $L^* = 0.021$ m. Ideally, the driving point impedance would be obtained with $L^* = 0$. However, the microphone could not be located at the plane defined by $L^* = 0$. The measured values are shown as points on Figure 9. The tube internal diameter 2R was 0.0222 m and the tube length, L was 0.500 m. The measurements were made with an air temperature of 20°C for which the velocity of sound, c_o is 343 m/s, the density, ρ is 1.21 kg/m^3, the kinematic viscosity, v_o is 1.5×10^{-5} m^2/s, the specific heats ratio, γ is 1.4 and the Prandtl Number P_r is 0.709. The measured values were obtained with the modified horn driver volume velocity source.

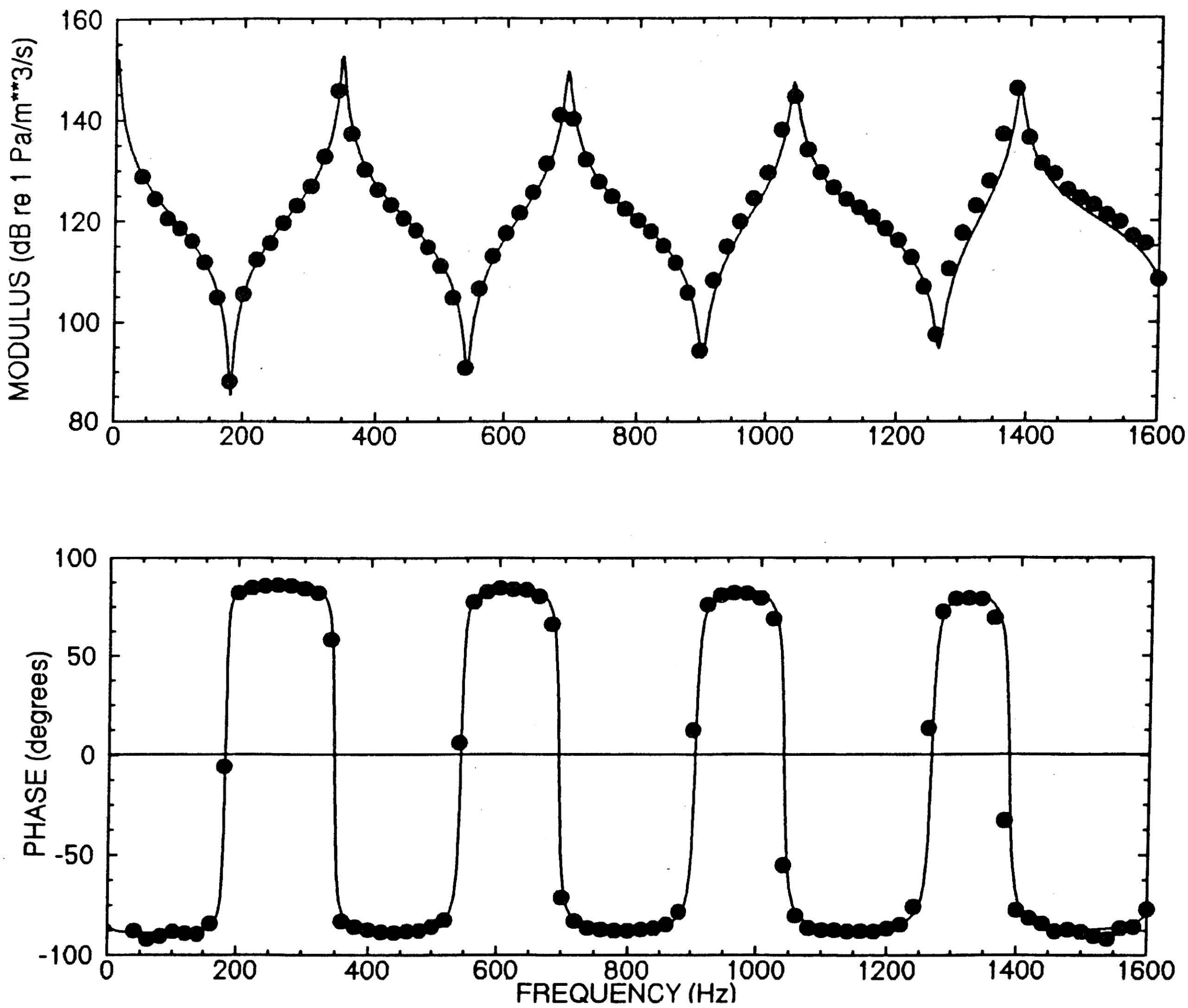


Figure 9. Modulus and Phase of the Computed (–)and Measured (•) "Driving Point Impedance": (Modified Horn Driver Source)

Figure 10 shows the modulus and phase of the transfer impedance computed with $L = L^* = 0.500$ m. The parameter values used in evaluating equation (13) were as previously given. The measured values, which are shown as points on Figure 10, were obtained with the piston type volume velocity source.

CONCLUDING COMMENTS

It can be seen from Figure 9 that the agreement between the computed and measured values of both the modulus and the phase is excellent. The dynamic range of the modulus is approximately 70 dB. The somewhat inferior agreement between the computed and measured values evident in Figure 10 is related more to the fact that a transfer impedance is involved than that the piston type volume velocity source was used to make the measurements.

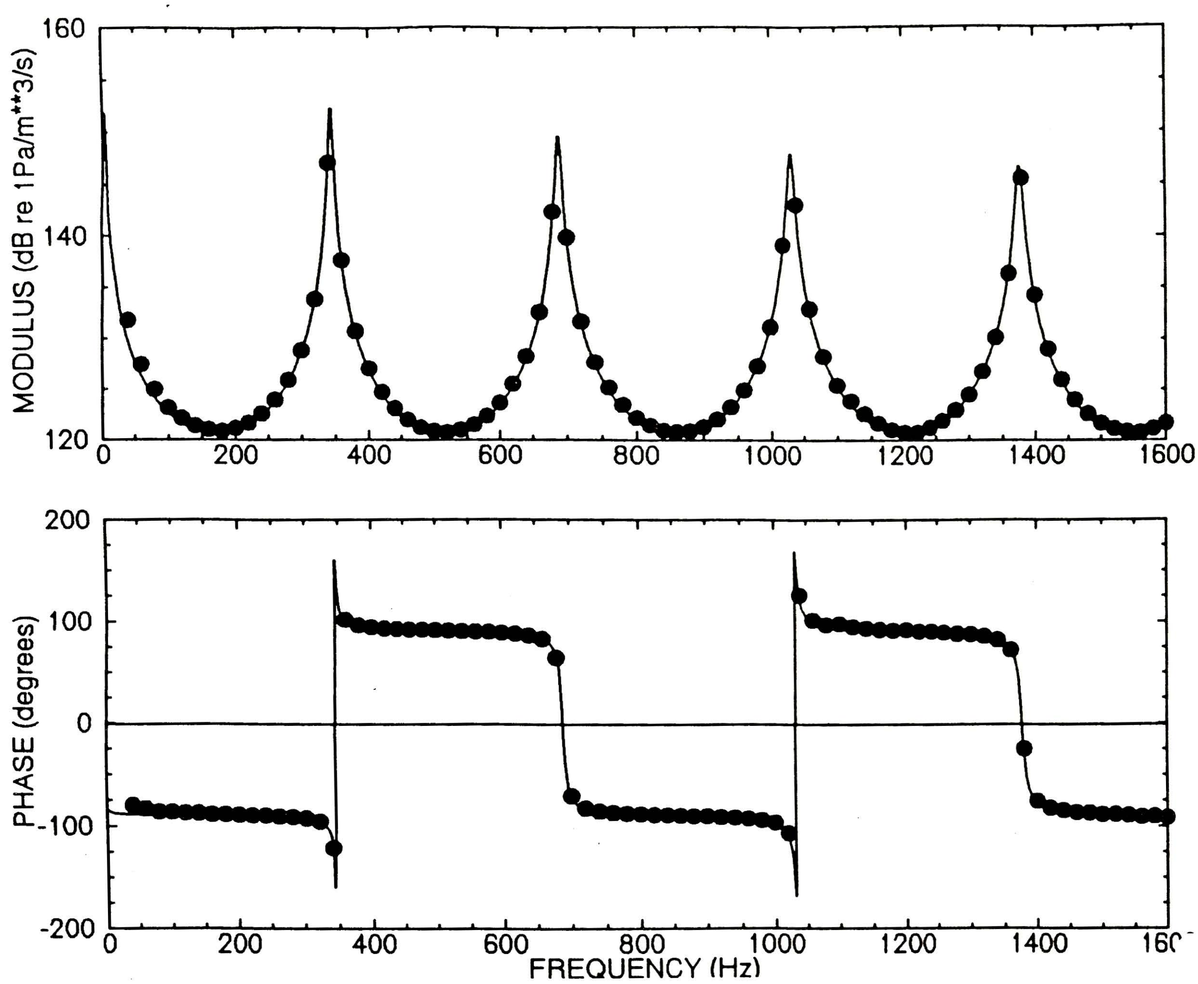


Figure 10. Modulus and Phase of the Computed (–) and Measured (•) Transfer Impedance (Piston Type Source)

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NOISE PATH DETERMINATION FOR A SHIP SERVICE DIESEL GENERATOR

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The auxiliary generator on the FFG-7 class of ships has been identified as a significant source of underwater radiated noise. An investigation was undertaken to determine the major noise transmission path. Field measurements of the airborne and structureborne noise were made and an SEA program was used to estimate the underwater radiated noise from each of these.

INTRODUCTION

Low acoustic signatures of vessels are basic to many operational requirements of the RAN. With the rapid increase in sensing capabilities in the region of interest to the RAN, it is necessary to provide the Navy with the capability to evaluate and reduce machinery acoustic emissions from vessels to the sea.

The FFG-7 class of ships form the current front-line of the RAN, but have been acknowledged as having significant shortcomings in their acoustic signature. Acoustic ranging reports for the FFG-7 have identified the Ship Service Diesel Generators (SSDG's) as a major noise source. Sell and Riley, 1987, report that "The most significant noise sources in serials up to and including *x* knots are the SSDG's and the Gas Turbine Compressor."

As part of the work on quietening the FFG-7's an investigation was conducted into the underwater radiated noise from the SSDG and the transmission paths from the SSDG into the water.

This paper details the results of experimental measurements of the source strengths of the SSDG and the use of Statistical Energy Analysis (SEA) to rank the various sound transmission paths. The analysis shows that two paths are of roughly equal importance and hence both need treatment.

SSDG DESCRIPTION

The Ship Service Diesel Generator sets aboard the FFG-7 provide the ship's electrical power. There are four SSDG's on each FFG-7. Each set is capable of providing 1 MW of power and is driven by a 2389 cu in., V-16, 1425 bhp, 2 stroke diesel engine.

Each set is mounted on a welded steel sub-base designed to form a rigid, self-supporting structural unit suitable for installation on its foundation in the ship. The total weight of each unit is approximately 18 tonnes. Each sub-base is mounted via eight resilient mounts to the ship's foundation. A sketch of an SSDG set and sub-base is shown in Figure 1.

The four SSDG's are located in separate enclosures throughout the ship and isolated from other machinery spaces by an acoustic enclosure on three sides. The fourth side and base of the enclosure is formed by the hull. Each SSDG sub-base is resiliently mounted on a rigid welded steel foundation. A typical foundation is shown in Figure 2, while Figure 3 shows a typical enclosure.

An initial investigation was conducted where the sound pressure levels inside the SSDG enclosure and velocity levels above and below the mounts were measured. The mounts provided in excess of 20dB attenuation, with velocity levels in the order of 10^{-4} m/s measured below the mount on the foundation. The sound pressure levels exceeded 110dB in one third octave band, and 100dB in half the third octave bands measured. While the enclosure walls separating the SSDG from other machinery spaces had acoustic treatment, there was no acoustic treatment on the hull.

In order to determine the relative importance of the various paths it was decided to calculate the radiated sound pressure levels in the water due to each path using Statistical Energy Analysis (SEA). Three major paths were considered, vertical structureborne, horizontal structure borne and airborne.

SEA CALCULATIONS

For SEA calculations it is necessary to divide the system into subsystems which represent a group of similar modes capable of energy storage, Lyon (1975). Physically independent substructures of the overall structure may include several subsystems, also boundaries of the subsystems do not necessarily represent physical boundaries of the substructure.

In this case the foundation was divided into the main beams, on which the isolators are connected, the interconnecting cross beams, and the legs and side beams connecting these to the hull. Each main beam has both vertical and horizontal bending, thus there are two SEA subsystems for each beam, one for each direction of motion. The cross beams, legs and side beams are either in bending or in-plane axial motion, depending upon the mode of the main beam. The hull was considered as a curved, ribbed plate in flexure. The SSDG enclosure was considered as a 3-D acoustic volume coupled to the hull, while the water was represented as a semi-infinite acoustic fluid.

A schematic of the SEA system is shown in Figure 4.

The inputs for the system are the vertical and horizontal velocities of the main beams and the sound power from the SSDG set. The spatial averaged velocity levels for the beams were determined in one third octave bands over the frequency range of interest. Sound intensity was used to determine the sound power from the SSDG in one third octave bands. In addition the spatial averaged sound pressure levels inside the enclosure were measured. These values are shown in table 1.

INBOARD VERTICAL VELOCITY dB re 1m/s	OUTBOARD VERTICAL VELOCITY dB re 1 m/s	INBOARD HORIZ. VELOCITY dB re 1m/s	OUTBOARD HORIZ. VELOCITY dB re 1m/s	SOUND PRESSURE LEVEL dB re 2.10^{-5} Pa	SOUND POWER LEVEL dB re 10^{-12} w
-89.0	-90.2	-92.4	-92.2	80.0	85.0
-81.4	-88.7	-86.4	-85.2	80.0	89.2
-84.0	-87.0	-87.1	-87.1	86.6	91.2
-83.0	-86.7	-84.9	-85.0	88.5	97.0
-79.4	-83.7	-79.9	-81.4	93.5	105.6
-79.8	-85.9	-83.3	-83.5	94.7	102.0
-84.9	-87.1	-89.4	-90.1	91.0	103.0
-82.3	-84.7	-88.3	-89.9	91.2	104.0
-81.2	-82.5	-87.6	-88.8	101.3	107.0
-83.5	-85.6	-91.4	-92.2	110.3	114.0
-84.8	-87.1	-90.9	-93.0	101.6	110.6
-87.7	-87.4	-90.1	-90.0	103.1	114.6
-90.5	-93.4	-93.4	-95.0	103.9	114.0
-87.9	-92.2	-93.5	-94.3	105.6	116.4
-91.7	-95.3	-98.3	-100.0	102.1	111.4
-93.0	-97.0	-99.6	-101.3	102.1	110.2
-97.5	-102	-106.8	-104.8	97.6	106.3

Table 1
Spatial Averaged Velocity, Sound Pressure and Sound Power Levels For SEA Model Input

The SEA calculations were performed using AutoSEA, developed by Vibro-Acoustic Sciences Ltd.. Predicted sound pressure levels in the water due to each of the three paths are shown in Figure 5.

The analysis shows that the horizontal structureborne path is not important, as it is atleast 10dB lower than either of the other two paths. The vertical structureborne and airborne paths are both significant and neither can be considered as the dominant path. There are only two or three frequency bands where one path is 10 dB or more higher than the other path.

The predicted total sound pressure level in the water due to all three path corresponds well with measured levels. Peaks in the curves occur at corresponding frequencies and levels are within 3 to 5 dB.

The next step in the investigation is to carry out a series of "what if" studies , for example, varying the velocity input for the structureborne noise (simulating improved resilient mounts), increase the absorption coefficient in the enclosure, varying the transmission loss for the hull, to see how each of these influences the sound pressure level in the water.

CONCLUSIONS

SEA provides a quick, convenient method to assess the relative importance of the various transmission paths. Of the three paths analysed, no one path can be considered dominant and only the horizontal structureborne path can be discounted from further analysis. The other two paths, the airborne and the vertical structureborne, both need to be treated if significant reductions in levels are to be achieved.

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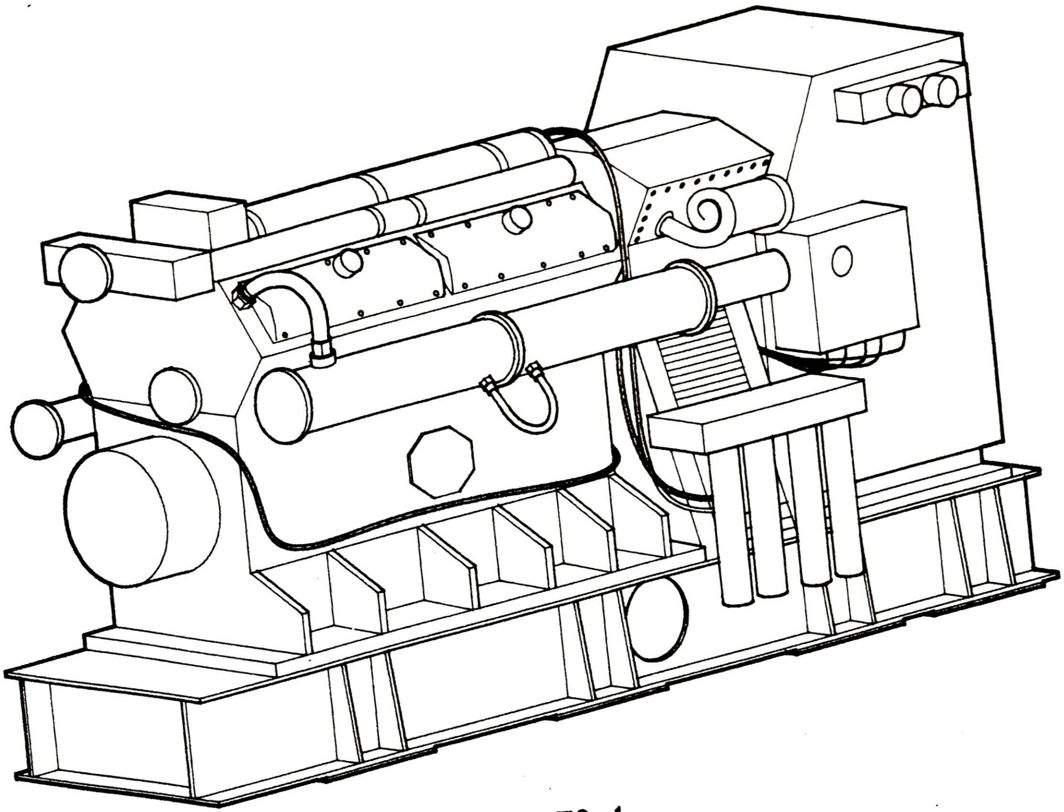

FIG. 1
DIESEL GENERATOR SET ON SUB-BASE

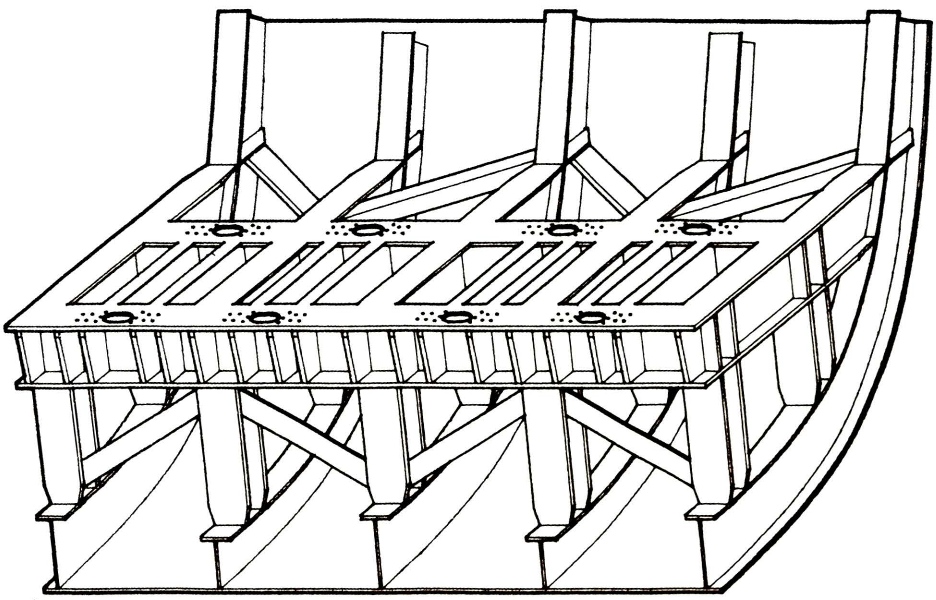

FIG. 2
SSDG FOUNDATION

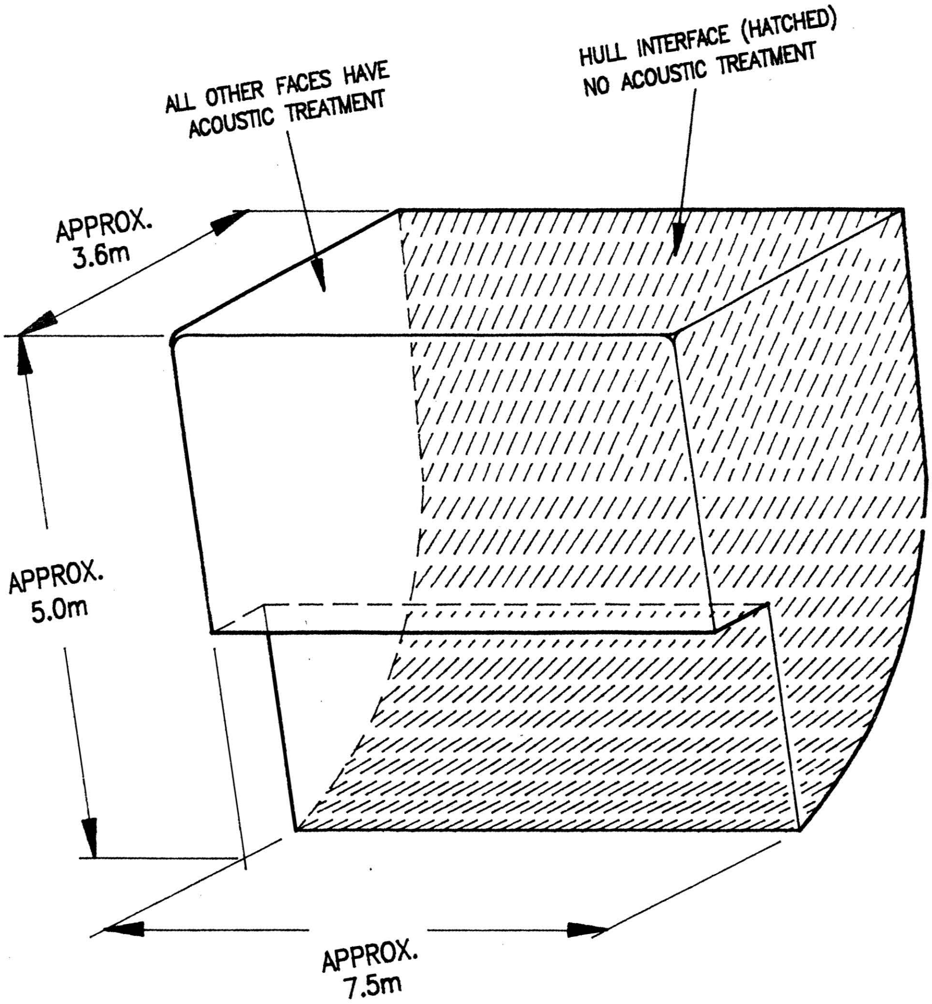

FIG. 3
SSDG ENCLOSURE

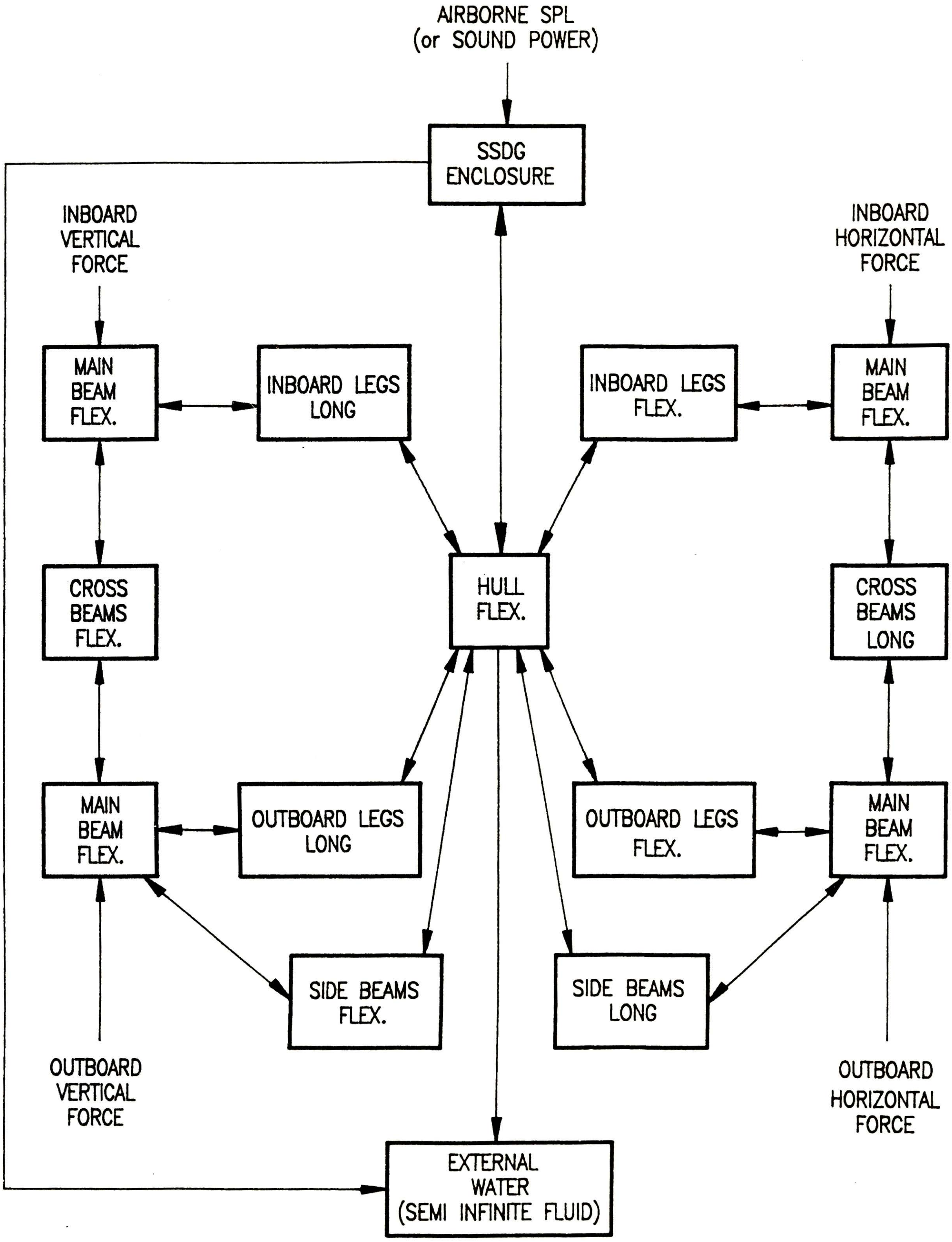

FIG. 4
SCHEMATIC OF SEA SYSTEM

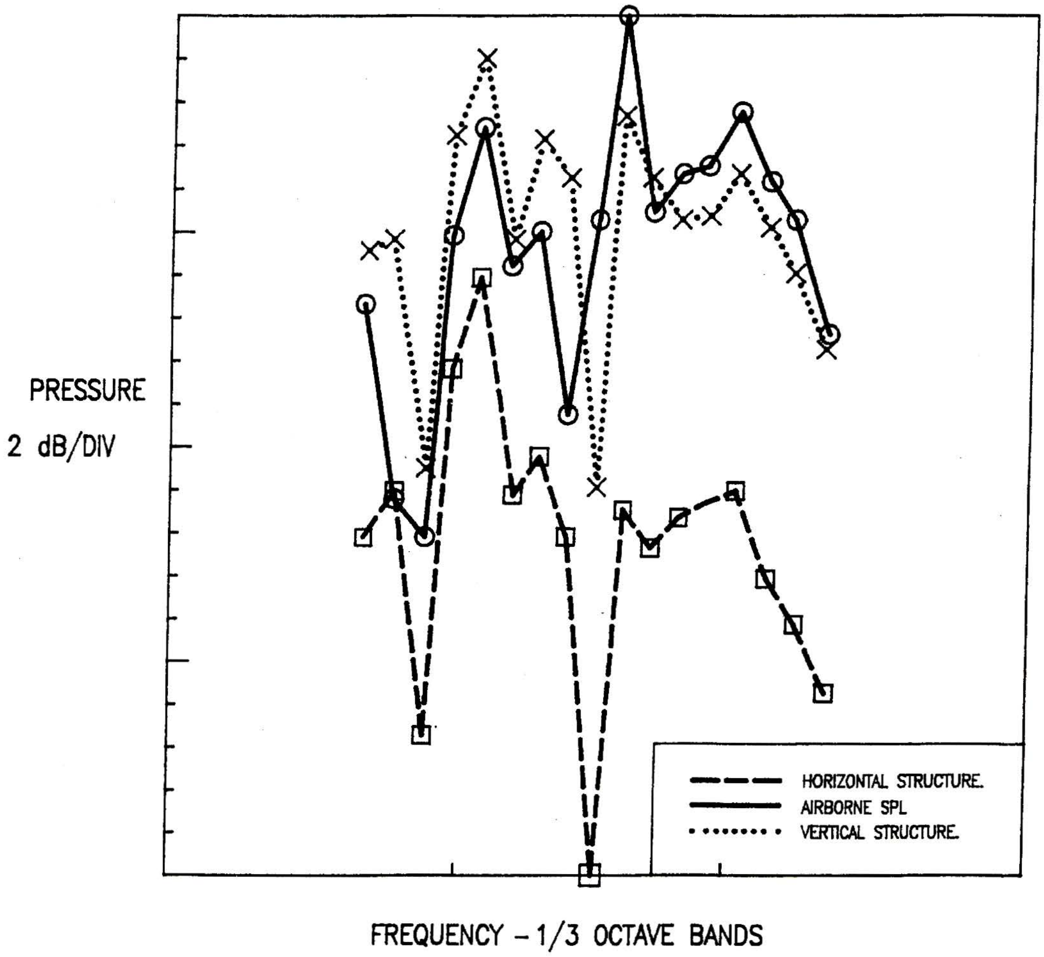

FIG 5.
PRESSURE LEVELS IN WATER

BLAST EMISSIONS FROM EXCAVATION AND TUNNELLING

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Abstract

Many major building and roadway projects require excavation of large quantities of hard rock. Over recent years the high cost of vacant urban land has encouraged the use of underground construction for road and rail transport links, services tunnels and parking stations. Such projects include the Sydney Harbour Tunnel, the Eastern Distributor, the Bennelong and Queen Victoria Building Parking Stations in Sydney, Duplication of the Brisbane Inner City Rail Tunnels and potential projects such as the Park Street Road Tunnel, the Metro-West Rail Link, and a possible rail link to Sydney airport.

Limits placed on the emission of vibration (and noise and overpressure) from blasting for excavation and tunnelling projects in urban areas, are often a major factor controlling the selection of excavation methods and rates of progress in removing rock. They therefore have critical implications for the duration and cost of many projects. If the emission limits are inappropriate, adverse environmental impacts can increase rather than decrease, as they will result in the duration of the project being unnecessary prolonged.

This presentation reviews the nature of the emissions from the various blasting techniques available for the bulk excavation of rock, and their effects on buildings and their occupants, on historic structures and on other services are described. Australian and overseas emission limits, in terms of human disturbance and structural damage, are reviewed and the implications of compliance for excavation projects are discussed.

A justification for some rationalisation of the current limits is presented.

1 INTRODUCTION

Airborne emissions from blasting are caused by rock heave, stemming release, gas venting and inadequately covered surface detonating cord. High frequency, audible noise is almost invariably radiated by a surface blast, and is perceived as a "crack", "boom" or "rumble", depending primarily on the type of blast and the distance to the receiver. This high frequency energy is superimposed on lower frequency airblast overpressure energy, the frequencies of which (approximately 20 Hz or lower) are usually below the audible frequency range of human hearing.

On reaching a dwelling, the low frequency airblast overpressure from a blast causes the structure to vibrate. If the induced vibration is of sufficient magnitude, it will be noticed or felt by the occupants, and even greater amplitudes could cause damage to the building or its contents.

Large, flexible building elements (walls, floors, windows, etc) and building contents connected to them (pictures, blinds, light fittings) are most susceptible to airblast, and if they (or objects on shelves, tables, cupboards etc) are loose, they may audibly rattle or be seen to vibrate.

Audible noise is also re-radiated by the vibrating surfaces of the building, and sometimes by the ground surface outside the dwelling (as the ground borne vibration passes beneath the building). For blasting in underground mines and in tunnels, it is often this re-radiated noise which causes the characteristic "boom" or "rumbling" sounds associated with such blasting (as airborne audible noise from the underground mine or tunnel portal has been effectively attenuated).

The low frequency *ground borne* vibration from a blasting can also cause buildings to vibrate in a similar manner to that described for low frequency airblast overpressure, although vibration of the structural frame of the dwelling (rather than mid-wall vibration) is more affected by ground-borne energy than airblast emissions. Secondary noise radiation from vibrating surfaces, rattling of crockery, loose windows and other objects can also occur (as for airblast), and the vibration induced in floors may be felt by the occupants.

If building occupants are unprepared for the blast, airborne (direct and re-radiated) noise can have a significant "startle" effect. The presence of rattling and visible movement increases the concern of residents, and the extent of and degree of disturbance they may experience (Heggie and Godson Report 902-R3 1988).

2 NATURE OF AIRBLAST OVERPRESSURE EMISSIONS

The use of explosives at quarrying, surface mining and excavation and demolition sites gives rise to airborne pressure fluctuations over a wide range of frequencies. The higher-frequency portion of this energy is sometimes audible (perceived as "noise" accompanying the blast), but most of the energy occurs at inaudible frequencies of less than 20 Hz.

The detonation of the explosives, the sudden expansion of gases, and the almost instantaneous fracturing and movement of the rock mass, all cause vibration and movement of the ground. Typical traces corresponding to the airblast overpressure emission, and vibration time histories in the ground and in various parts of a structure from a coal mine high wall shot appear in **Figure 2.1** (Siskind, Stagg et al 1980).

The various ground vibration waves travel through the ground at a relatively high speed (compared to the airborne wave), and when they reach the receiver or an airblast overpressure monitoring location, the vertical component of the ground movement excites the air and creates the first airborne disturbance at the monitoring location. This stage of the airblast wave is identified as the Rock Pressure Pulse (RPP). Although it is the first to arrive, this part of the overpressure waveform is usually lower in amplitude than the remainder of the components making up the total airblast waveform.

The second part of the airblast wave is the Air Pressure Pulse (APP), and is generated by the vertical ground movement in the immediate vicinity of the blast. It arrives at the monitoring position some time after the RPP, as its speed in air (344 m/s) is usually lower than that of the ground-transmitted Rock Pressure Pulse (RPP).

Other, sometimes quite significant pulses can be superimposed on the RPP and APP, due to various release mechanisms of the gases generated during the detonation of the explosive. These gases can vent rapidly through fissures or cracks in the rock face or through the collar of the blast hole. The escape of gases through the face is called the Gas Venting Pulse (GVP), whilst that through the collar is described as the Stemming Release Pulse (SRP).

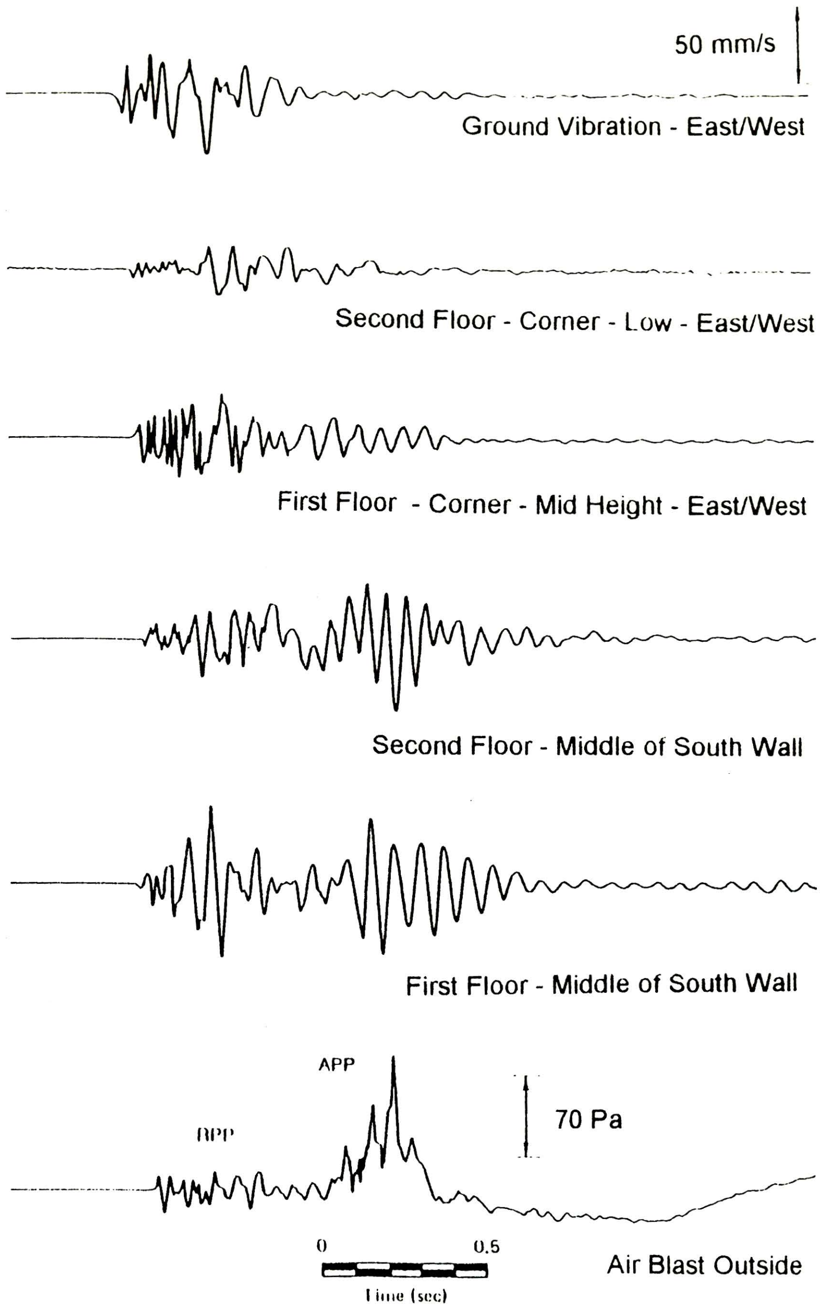


Figure 2.1 **Ground Vibration Structure Vibration and Overpressure Waveforms**

Figure 2.2 presents data (Siskind, Stachura et al 1980) relating to a large number of airblast overpressure measurements conducted by the US Bureau of Mines. The upper and lower limit predictions of airblast overpressure versus cube root scaled distance are shown by the free-air (unconfined) and Rock Pressure Pulse (RPP) lines. The difference between total confinement and free-air blasts is 41 dB ± 5 dB over a relatively wide range of cube root scaled distances. The measuring instrumentation for these data had a response down to 5 Hz or lower.

The US Bureau of Mines (Siskind, Stachura et al 1980) collated data from many investigations of airblast emission. By performing regression analysis on the data points, the researchers were able to develop relationships for the emission levels from various mining and shot types and cube root scaled distance. **Figure 2.3** presents some of their results.

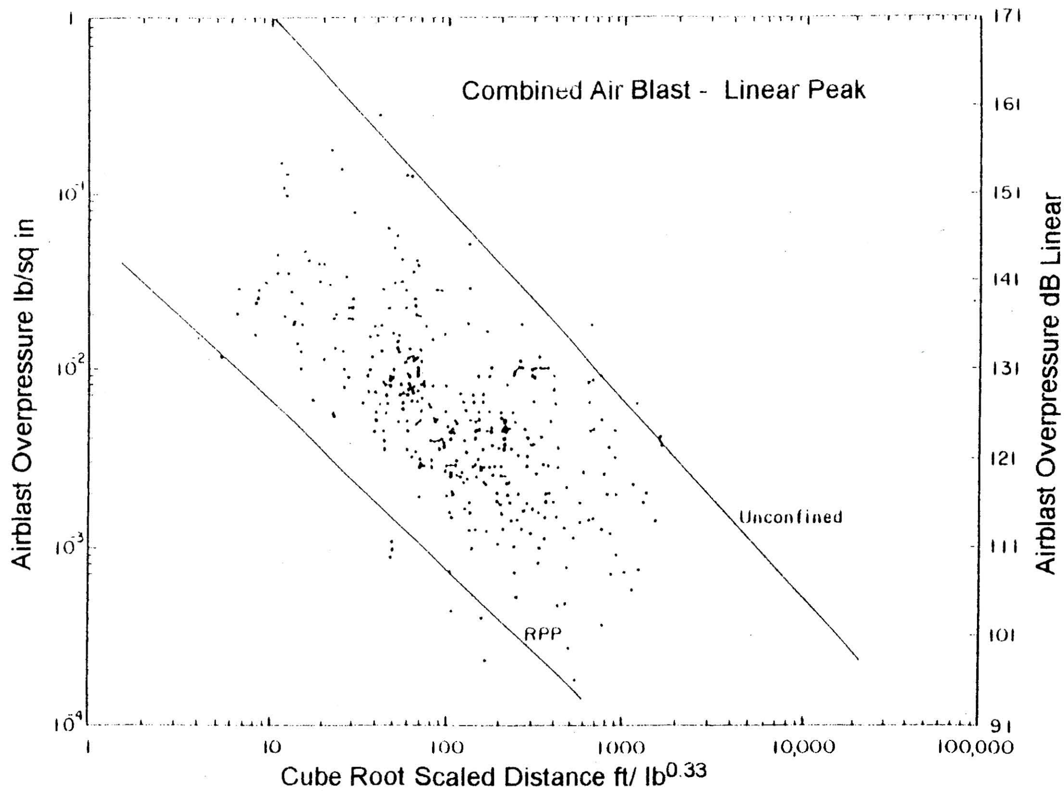


Figure 2.2 Range of Airblast Overpressure Emission Levels (All Sites)

The left hand graph shows the results of two studies of emissions from various industries. Although the results of the two studies differed, they both indicated higher *measured* emission levels in quarrying and construction (excavation) than in coal mining. This is probably due to the shorter *absolute* distances from blast to receiver for the quarry and excavation blasting, and to the greater relative attenuation of higher frequency energy components over the larger distances associated with coal mine blasts. Some researchers (Dowding 1985) advocate the use of a square root scaled distance for short blast receiver distances.

The differences may also possibly have be partly attributed to atmospheric effects, which are difficult to quantify over long distances in field studies. Such atmospheric effects, (eg wind gradients and air temperature inversions) can significantly affect received airblast overpressure levels, particularly over large distances. Schomer (1973) has shown that for propagation distances of 3 km to 60 km, inversions can produce zones of intensification of up to 3 times average values (approximately 10 dB), with an average increase of 1.8 times (5 dB). One must assume that similar degrees of excess *attenuation* would occur upwind.

The right hand graph in **Figure 2.3** illustrates the effects of using instrumentation with various low frequency cut-off limits. The use of the C-scale frequency weighting network clearly results in much lower measured levels of emission, however, the use of 0.1 Hz, 2 Hz or 5 Hz cut-off with linear response had comparatively little effect (less than 5 dB) on measured airblast overpressure levels, at least for coal parting and high wall shots.

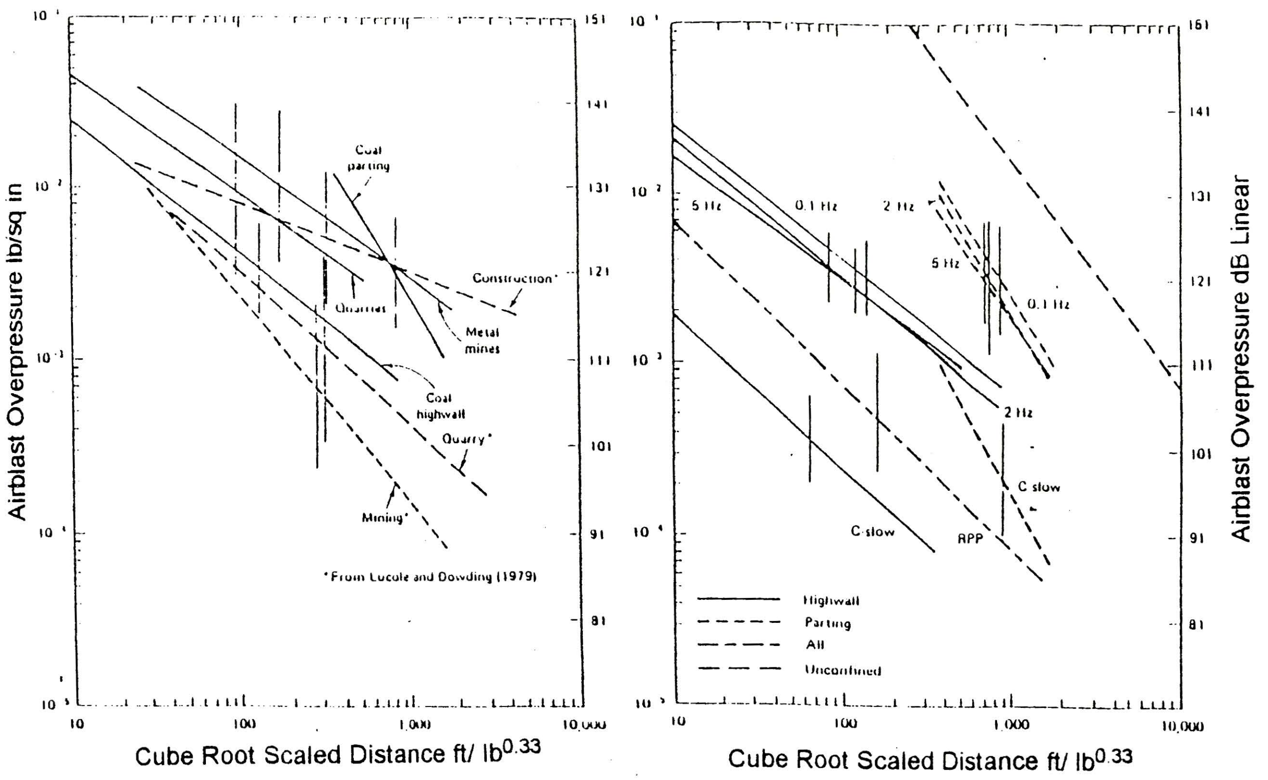


Figure 2.3 Airblast Propagation from Various Mining and Shot Types

3 SUBJECTIVE RESPONSE TO BLAST EMISSIONS

Of all the readily perceived phenomena associated with blasting, the most noticeable is usually the audible sound component, and it is interesting to note that no regulating authorities place specific limits on this component of the emissions (Godson and Heggie Report 902-R1 1988). Some research and regulation in this direction have been taken in regard to small arms fire and target shooting (Hede et al 1981, EPA 1985, Peploe et al 1989) and to artillery ranges and munitions test facilities (Makarucha 1987, Peploe et al 1987).

The second most noticeable component, but the one which probably has most influence on degree of disturbance to residents, is the rattling or visible movement of walls, windows crockery and loose objects (Schomer and Neathammer 1987, Schomer, Hottman and Eldred 1987, Schomer and Averbuch 1989). This rattling can be caused by the ground vibration or the airblast overpressure.

People seem to be particularly sensitive to sources of building vibration (and induced rattling) from outside their dwellings. Slamming doors, closing windows sharply, the playing of children or even normal walking across a loose timber floor often cause vibration and rattling much greater than that from blasting, however they are accepted by most home-owners as "natural" occurrences.

When the source of the rattling is from outside their houses however, and particularly when it is associated with a relatively loud, characteristic noise (with a possible "startle" effect) and all the connotations of potential for damage commonly perceived to be associated with blasting, then the same effects are often highly disturbing to the occupants.

4 RELATIVE SIGNIFICANCE OF AIRBLAST EMISSIONS

The predominant frequency content of airblast energy from blasting is usually less that 5 Hz to 20 Hz, and the sound "wave" propagates effectively in all directions and passes over quite large obstructions with little of the "barrier" attenuation that would be expected of higher frequency (audible) noise (sometimes even if the blast face is oriented away from residences or other sensitive receivers).

Blast designs are usually controlled by airblast emission criteria, rather than by ground vibration limits, when residences are close to the blast. This is particularly the case for blasting in excavations and tunnels in urban areas.

In general, ground vibration from blasting is a much more significant influence on the damage response of structures than airblast overpressure. Levels of ground vibration and airblast overpressure that produce equivalent structure motions (Siskind, Stagg et al 1980) are shown in **Figure 4.1**.

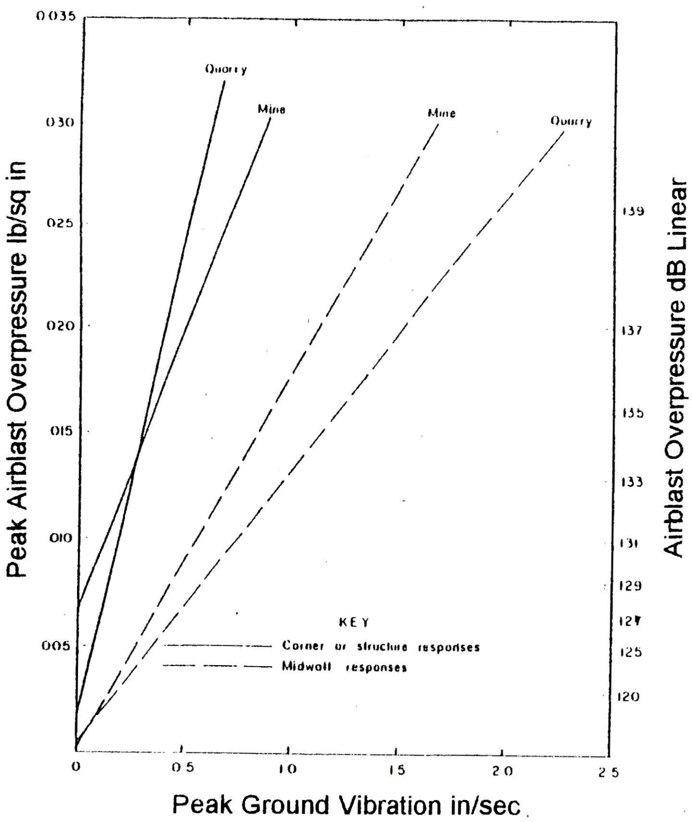


Figure 4.1 Ground Vibration and Airblast for Same Structural Response

The airblasts were measured with 0.1 Hz low frequency response systems. Typical 2 Hz to 5 Hz systems would give airblasts with sound levels in the order of 1 dB to 5 dB lower.

Between 137 dB and 138 dB of airblast overpressure is required to produce the same structural corner response as a peak ground vibration level of 12 mm/s. Building corner response is the most relevant motion in respect to potential to cause damage. Mid-wall responses relate primarily to induced rattling and secondary noise. Differences between mine and quarry blasting are not significant for the critical range of airblast overpressure levels, 131 dB to 135 dB, equivalent in effect (on structural corner response) to approximately 3 mm/s to 7 mm/s of ground vibration.

Relatively strong mid wall vibrations are produced by airblast overpressure at levels of only 128 dB to 130 dB, equivalent in effect (on mid-wall response) to ground vibration in the order of 12 mm/s.

5 DEVELOPMENT OF COMFORT CRITERIA FOR BLASTING EMISSIONS

The low frequency components of the airborne emissions from blasting, which are generally recognised as having the greatest potential to cause damage to structures and building elements, has historically (and probably quite appropriately) been quantified using the overall peak linear sound pressure level. "Blast monitors" often used for these measurements usually have an upper frequency limit of about 200 Hz, and lower frequency limits extending down to 1 Hz or below. Precision sound level meters, which are also used to measure airblast overpressure, may extend the upper frequency limit to as high as 20 kHz.

Studies have established reasonable correlation between this measurement index and risk of damage to building elements, eg window glass, the part of structures usually considered to be most susceptible to airblast damage effects (Windes 1943, Duvall and Devine 1966, Siskind, Stachura et al 1980, Schomer et al 1980).

As various regulating authorities developed "comfort" criteria (Godson and Heggie Report 902-R1 1988), it became the norm to use the same peak linear sound pressure level index to quantify the potentially disturbing effects of the blast, due partly to the vast body of emissions data that had already been gathered. To gain confidence in the use of these comfort criteria, it would seem prudent to examine the degree of correlation between this index and the subjective response of people to the most perceptible effects of the blast; direct airborne audible noise, audible re-radiated noise and the occurrence and sound level of rattling.

Human comfort criteria and building damage criteria are usually regulated by separate authorities. The setting of reasonable and workable comfort limits for blasting needs to take into consideration the manner in which the community perceives the building damage risk. The thresholds at which the community is "disturbed" by blast emissions are clearly affected by the very widespread misconception that building vibration from blasting has a high probability of causing damage. There is an argument therefore, that by setting "comfort" emission limits for airborne emissions that are significantly lower than damage risk limits, the regulating authorities are assisting to perpetuate this public misconception.

Due to the strong nexus between degree of disturbance and fear of damage from blasting, regulating authorities and those responsible for writing standards are in a unique position to share in modifying public opinion by including strong and informative comment on damage risk in their "comfort" regulations and standards, and by setting comfort emission limits that are possibly only slightly lower than damage risk criteria. While such an approach may not be socially (or politically) acceptable for long term blasting programmes at established mines and quarries, there is clear justification (and precedence) for its adoption in respect to the short term emissions from excavation blasting associated with building and transportation construction works in urban areas.

6 EXISTING COMFORT CRITERIA FOR BLAST EMISSIONS

In Australia, the limits for blast emissions in respect to disturbance to building occupants are presently set well below the levels required to cause damage to structures (Godson and Heggie 1987). In NSW, the Environment Protection Authority (formerly the State Pollution Control Commission) is the regulating body which recommends human comfort limits for ground vibration and airblast overpressure emissions from blasting in mines, quarries and construction (EPA 1985).

Although different limits apply for various times of day, these guideline criteria presently take the general form shown in **Table 6.1** (for blasting between 9.00 am and 3.00 pm Monday to Saturday):

Emission Parameter	**Maximum Levels for 95% of Blast Events**	**Emission Levels Never to be Exceeded**
Airblast Overpressure	115 dB Linear	120 dB Linear
Ground Vibration (Peak Vector Sum)	5 mm/s	10 mm/s

Table 6.1 NSW EPA Blast Emissions Comfort Criteria (Guidelines)

Criteria in a similar probabilistic form are also recommended by the Australia and New Zealand Environment Council (ANZEC 1990), and in the current committee draft of the Standards Association of Australia's proposed Standard "Ground Transmitted Vibration and Airblast from Explosions" (SAA AV/9/3/91-3 1991). The current draft revision of the SAA Explosives Code AS2187 (SAA DR 91249:S 1991) provides guidance in the form of fixed emission limits, without specific permissible probabilities of exceedance.

Due to the variability of blast emission levels, it is imperative that the quantities used to define blast emissions be measured, analysed and expressed in an appropriate manner, and this obviously requires a statistically consistent approach.

7 STATISTICAL SCATTER OF BLAST EMISSION LEVELS

In order to set reasonable criteria and to provide a confident basis on which to design production blasts to meet certain "probability of exceedance" limits (similar to those promulgated by the NSW EPA), consideration must be given to the statistical variability of blast emission levels. An approach to this aspect of blasting is described by the following example.

The Sydney Harbour Tunnel project required several small tunnels to be blasted to divert existing underground drains around the proposed alignment of the road tunnels for both north and south shore works. In addition to these small tunnel diversions, the two main northern road tunnels from Bradfield Park to the Warringah Expressway were to be driven by a combination of road headers, rockhammers and conventional heading and bench blasting. All the tunnelling contracts which involved blasting were in close proximity to residences and/or historical buildings.

Trial blasts were therefore carried out using a range of (usually reduced) charge weights and measurement distances to establish the particular emission and propagation characteristics at each contract site. When sufficient data points were available, the results could be statistically analysed to determine (with a known confidence) the blast design parameters (maximum instantaneous charge MIC and distance to receiver) required to ensure that no more than 5% of ongoing blasts exceeded the specified emission levels. This 5% exceedance limit conformed to the style of the criteria set for the project by the NSW Environment Protection Authority. After the trial blast periods, the results of monitoring of on-going production blasts were progressively added to the blast site laws to improve the reliability and confidence of their predictions.

Figure 7.1 presents the peak vector sum (PVS) ground vibration velocity levels (V in mm/s) measured over a series of blast events for the Sydney Harbour Tunnel. These levels are shown plotted (on log-log axes) against the "square root scaled distance" (SD) for each blast event, where:

$$(\text{SD in m.kg}^{-1/2}) \quad = \quad (\text{D in m})/(\text{M in kg})^{1/2}$$

and,

	D	=	Slant distance from effective charge to receiver
	M	=	Effective maximum instantaneous charge weight

Examination of **Figure 7.1** clearly illustrates the high variation in emission levels for a given blast design and distance. For the data points with scaled distances of 209 m.kg$^{-1/2}$, the measured vibration levels ranged from 1.4 mm/s to 15 mm/s, a factor of more than ten to one.

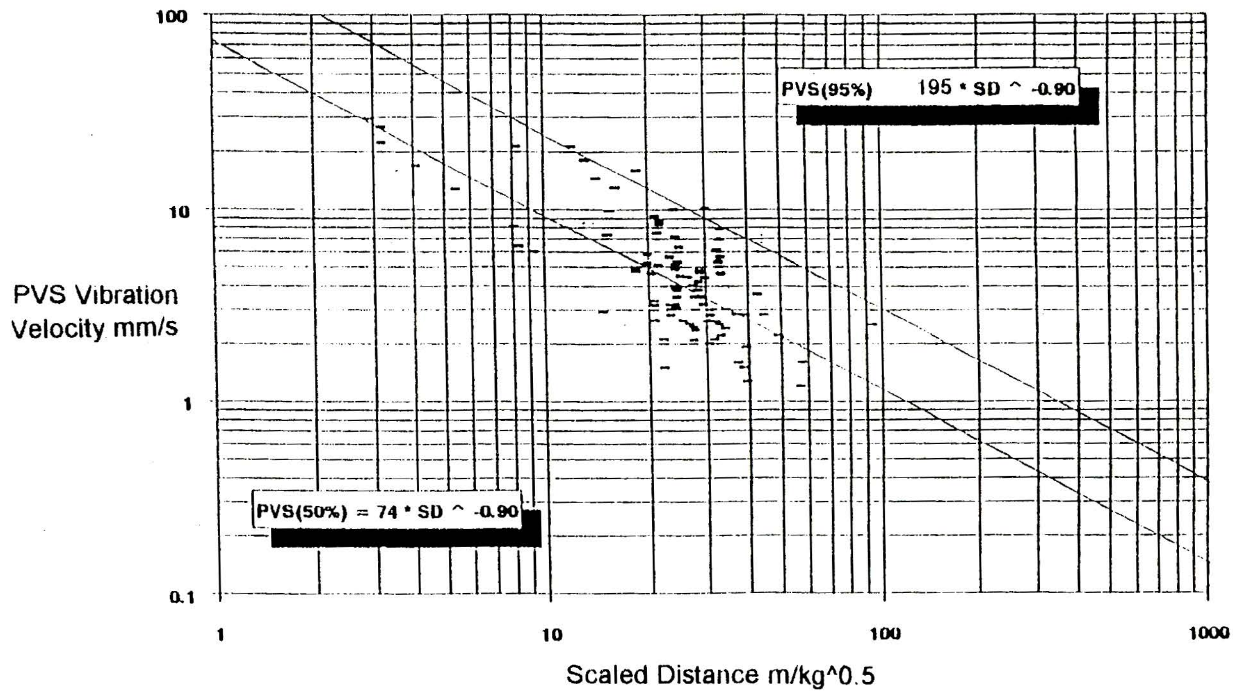


Figure 7.1 Statistical Site Law (Vibration) for Blast Emissions

A "Mean Site Law" for the total population of on-going blasting (traditionally assumed to be a log-normal distribution) can be determined by linear regression analysis of the 96 data points contained in the sample using the least squares, line of best fit method. This relationship is also shown in **Figure 7.1**.

If a regulatory authority's requirement that a certain percentage of blast emission levels (eg 5% of blasts, or 1 in 20) shall not exceed a certain limit is to be properly addressed, a statistical single-sided "tolerance" interval (*not* a confidence interval for the mean) is required (Heggie and Thomas). Such a tolerance interval indicates (with a certain level of confidence), the limit above which a certain percentage of the data points will lie.

Although potentially somewhat confusing, it is therefore essential to consider two "percentages" for a tolerance interval - the percentage of data points to be included below (or above) the limit and the percentage degree of confidence in stating that limit. For the 96 vibration levels in this data set, the upper 5% tolerance interval (determined with 95% confidence) is also shown in **Figure 7.1**.

Strictly speaking, there is also a level of confidence associated with both the slopes and the intercepts of these lines, however for convenience, the slope of the upper 5% tolerance interval has been set to be the same as the mean slope of the mean regression line. This is the usual practice, but it should be remembered that the validity of the statistical calculations is greatest for blast designs nearest the medians of the ranges of *actual* distances (D) and charge weights (D) used in the trial blasting. Caution should be exercised in assuming the results will also apply to blasts with significantly different *absolute* distances or charge weights, even though the *scaled distances* are similar.

Experience is now indicating that a considerable number of blast monitoring data points are often required to determine (let alone predict) the 5% exceedance level for a series of blasts with a reasonable degree of confidence. Consideration should therefore be given to using a lower probability of exceedance figure, for example 10%, as per the proposed draft revision of British Standard 6472 , and the current committee draft of the Standards Association of Australia's proposed Standard "Ground Transmitted Vibration and Airblast from Explosions" (SAA AV/9/3/91-3 1991).

A similar probabilistic approach should be adopted for developing a site law for airblast overpressure emissions. **Figure 7.2** shows such a relationship for a series of trial blasts for a large hard rock quarry.

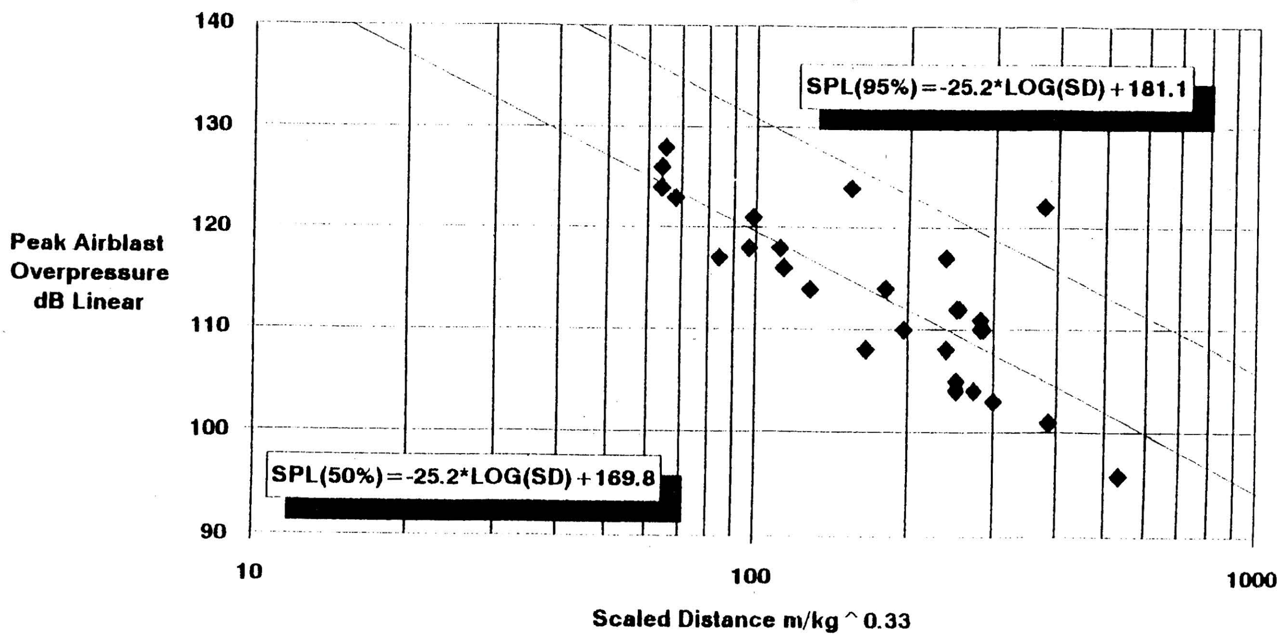


Figure 7.2 Airblast Overpressure Site Law for Large Hard Rock Quarry

8 DAMAGE CRITERIA FOR AIR BLAST EMISSIONS

The current (1983) version of the Standards Association's Explosives Code AS 2187 contains no quantitative guidance in regard to comfort or damage limits for airblast overpressure emissions. While cracked plaster is the type of damage most frequently mentioned in airblast complaints, research suggests that window panes, particularly large plate glass windows, fail before any damage to the remainder of the structure occurs (Siskind, Stachura et al 1980, Windes 1943, Duvall and Devine 1966, Rosenthal and Morlock 1987). Minor cosmetic cracking of internal surface finishes may occur in conjunction with extensive window glass breakage.

Whether a window will be cracked by the airblast overpressure wave from an in-ground blast depends primarily on the peak overpressure level, the area, thickness and strength of the glass, the orientation of the window with respect to the blast, and the condition of the surface of the glass (Redpath 1976, ANSI 1983).

Redpath (1976) reports that the strength of a typical pane of glass is reduced by approximately a factor of two from when it is manufactured to when it is installed, simply because of the minute surface scratches incurred in shipping, handling, installing and cleaning.

Other factors such as edge defects, the methods of glass installation and any pre-stressing from building settlement or warping of the frame, appear to make little contribution to risk of breakage from airblast pressure loading, a result predicted from plate theory (ANSI 1983).

In summary, the probability of damage to windows exposed to a single airblast overpressure event is as shown in **Table 8.1**.

Airblast Overpressure Levels	**Probability of Damage**		**Effects and Comments**
140 dB Lin	0.2 kPa	0.01%	No damage - windows rattle
150 dB Lin	0.6 kPa	0.5%	Very occasional failure
160 dB Lin	2.0 kPa	20%	Substantial failures
180 dB Lin	20 kPa	95%	Almost all fail

Table 8.1 Probability of Window Damage from Airblast Overpressure

In assessing the impact of airblast overpressure on the community, it is worth noting that many thunderstorms generate overpressure levels in excess of 140 dB, but the literature has revealed no circumstances of window breakage due to a thunderclap alone (ie excluding wind effects and flying debris).

The US Bureau of Mines (Siskind, Stachura et al 1980) has undertaken considerable research into levels of air overpressure and their possible effects, and its recommendations are almost invariably taken as the standard to be applied to open pit workings in the United States. **Table 8.2** presents these recommendations for various instrument frequency responses.

The levels in **Table 8.2** are based on the minimum probability of the most superficial type of damage in residential property. The weakest parts of a structure exposed to airblast overpressure are the windows, so they are the most likely to suffer damage.

To maintain the probability of damage at 1 : 100 000 it is further recommended that the levels given in **Table 8.2** be reduced by 9 dB per doubling of window size in excess of 7.5 m^2. To exceed the recommended levels will clearly increase the probability of damage, with pre-stressed windows breaking at about 150 dB, and most other windows breaking at 170 dB and structural damage being expected at about 180 dB.

Instrument Response	Maximum Level
0.1 Hz high pass	134 dB
2.0 Hz high pass	133 dB
5.0 or 6.0 Hz high pass	129 dB
C - slow	105 dBC

Table 8.2 USBM Recommended Airblast Overpressure Limits

The levels in **Table 8.2** are based on the minimum probability of the most superficial type of damage in residential property. The weakest parts of a structure exposed to airblast overpressure are the windows, so they are the most likely to suffer damage.

Based largely on the work carried out by the US Bureau of Mines, **Table 8.3** shows the regulatory limits for limiting the risk of damage by airblast overpressure from blasting (depending on the low frequency limit of the measuring system) promulgated by the US. Office of Surface Mining, Reclamation and Enforcement.

Instrument Low Frequency Limit	Peak Airblast Level Limit
2 Hz or lower	132 dB
6 Hz or lower	130 dB

Table 8.3 OSMRE Regulatory Limits for Airblast Overpressure

9 PROPOSED SAA CRITERIA FOR BLAST EMISSIONS

Table 9.1 shows the revised criteria for airblast emission levels currently under consideration for the current committee draft of the Standards Association of Australia's proposed Standard "Ground Transmitted Vibration and Airblast from Explosions" (SAA AV/9/3/91-3 1991).

The proposed comfort limit figures shown in **Table 9.1** (but *not* the damage risk limits) may be exceeded by 10% of blast events, up to an additional 5 dB.

Instrumentation for measuring airblast emission levels is required have a bandwidth of 2 Hz to 250 Hz (3 dB down points). The response at 0.5 Hz must be 15 dB or more below the response in the pass band. This frequency response characteristic and the damage limits are based on the recommendations of the US Bureau of Mines (see also **Table 8.2**).

The draft standard's 120 dB comfort limit is 5 dB more lenient than the Environment Protection Authority's current weekday daytime criterion of 115 dB. The revised standard has also changed the permissible percentage of exceedances from 5% to 10%, which is statistically more workable and is in keeping with the proposed revision to BS 6472.

Time of Blast	"Comfort" Limit	"Damage" Limit
Monday to Saturday 9.00 am to 5.00 pm	120 dB	133 dB
Monday to Saturday 5.00 pm to 9.00 am	115 dB	133 dB
Sundays and Public Holidays	115 dB	133 dB

Table 9.1 Proposed Revised SAA Airblast Overpressure Criteria

All the approaches using comfort limits based on probabilities of exceedance place an absolute upper absolute limit on permissible emission levels. The concept of a "never to be exceeded" level is however, statistically invalid. The 5 dB margin between the permissible 10% exceedance level and the "never to be exceeded" level is far to small, given the wide spreads in emission levels that occur in practice. Further study of the statistical distributions of blast emission levels is required, so that a more mathematically consistent approach can be developed

The permissible percentage exceedance levels should also be qualified by a statement regarding the confidence interval required for calculating the nominated emission exceedance level from required a set of blast emission records.

10. QUANTITATIVE EFFECTS OF THE PRESENCE OF RATTLING

Although structural damage is unlikely, airblast overpressure does play a most important role in the annoyance aspect of blasting. Relatively low levels of airblast (ie compared to damage limits) can be sufficient to cause the rattling of loose ornaments or windows and hence give the impression of a significant ground vibration shaking the property.

Impulsive ground vibrations as low as 0.5 mm/s can cause complaints when accompanied by such secondary noise effects. This is because the average person forms a judgment based largely on his or her psychoacoustic responses, and is usually unaware of the important distinction between the characteristics and effects of the building motion alone, and the sound effects that accompany it.

A guide to the subjective effects of airblast overpressure levels on the community is shown by **Figure 10.1**, which relates the percentage of people *highly annoyed* by the rattling of homes and objects by sonic booms from supersonic aircraft.

For the linear peak levels shown in **Figure 10.1**, 5% of the community would be very annoyed by house rattles at a mean sonic boom level of about 125 dB. The levels not exceeded for 95% of the time in a study by Borsky (1965) show that the tolerable (5% very annoyed) level is in the order of 130 dB.

Figure 10.2, from the work of Schomer et al (1989) using a "test house" built inside a laboratory, and exposed to artificially produced "blast sounds", suggests that the presence of audible rattling significantly affects the degree of annoyance caused indoors by airblast from artillery.

At low blast levels (eg 112 dB linear outside the test dwelling) the presence of rattling caused an "annoyance offset" of 13 dB. The effect was less pronounced for higher airblast levels, with an "annoyance offset" due to rattling of 6 dB at a blast level of 122 dB.

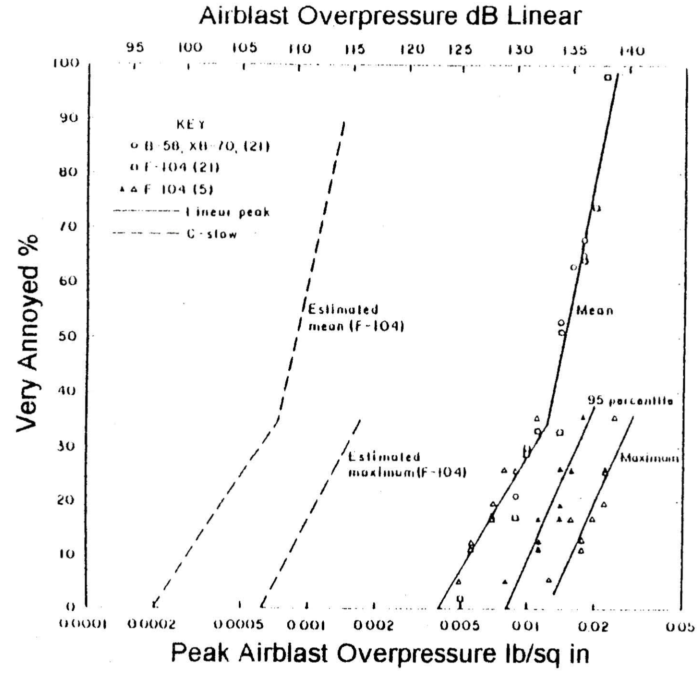


Figure 10.1 Population Very Annoyed by House Rattles

Figure 10.3 shows the spectrum of a typical test blast used during Schomer's tests, indicating an apparent absence of the high level, very low frequency energy (ie below 10 Hz) that would be expected of many blasting situations in mining and quarrying. Further study is therefore required to investigate the validity of applying Schomer's results to blasting in mines, quarries and excavation works.

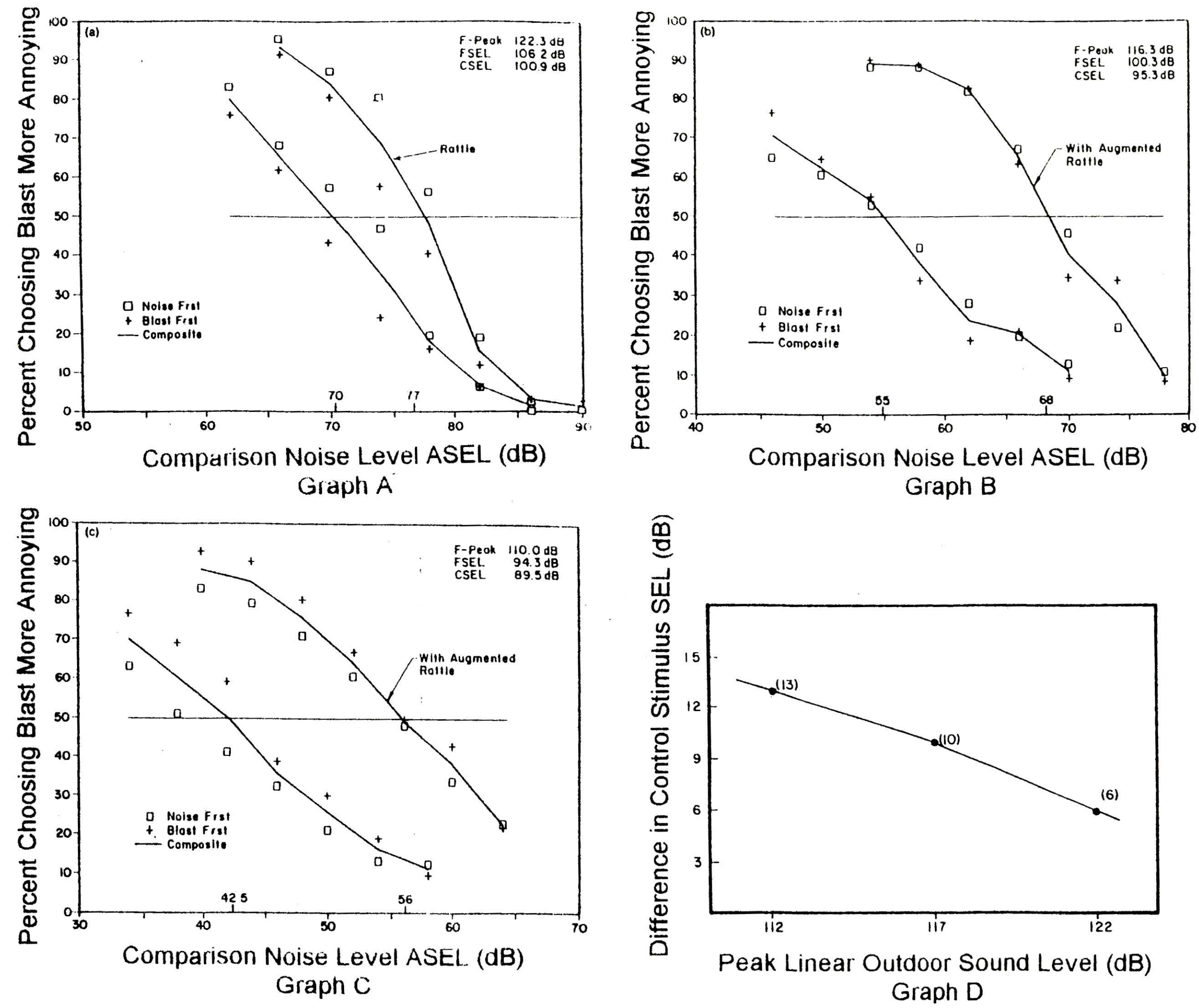
11. CONCLUSIONS

Careful and controlled blasting for tunnels and excavations in urban areas (Vuolio and Jonsson 1984) is under-utilised in Australia, and has the potential to significantly reduce environmental impacts of the alternative noise- and vibration- producing mechanised techniques that are presently in widespread use.

There is a pressing need for more awareness of the potential short term benefits of excavation blasting within the general community. Increased knowledge and awareness is also required by state and federal regulating authorities, Local Councils and by developers, builders, excavation contractors and their consultants.

In view of the strong relationship between complaint thresholds ("comfort" or "disturbance" criteria) and the general community's perception of risk of damage from blasting, Australian Standard 2187 should be considerably revised so that is reflects the current state of knowledge, particularly in relation to the effects of airblast overpressure and the significance of the frequency content of airblast and ground vibration.

Further study of the statistical distributions of blast emissions should also be carried out, and be reflected in the Standards Association of Australia's proposed Standard "Ground Transmitted Vibration and Airblast from Explosions".



Graphs A to C compare the results for the occupants of test house. The simulated blast linear peak levels were 112 dB, 116 dB and 110 dB, respectively. At the 50% point, the corresponding control level difference represents the increase in annoyance which results from the presence of rattles. Graph D shows the differences in equivalent control stimulus ASEL as a function of peak linear outdoor blast level, for the high to low rattle cases.

Figure 10.2 Quantitative Effects of Rattles on Airblast Perception

With sensible damage risk criteria in place, and a concerted effort to educate and reassure involved parties and potentially affected community groups, one would hope that the tendency for the community to over-react to emissions from blasting will diminish.

Attention should then be directed towards a revision of the current "comfort" criteria promulgated by various Australian regulatory authorities for mines and quarries. Special focus should be brought to bear in modifying the current limits and regulatory procedures to suit the special circumstances of short term excavation, demolition and tunnel blasting in urban areas. Particular study is required to define more clearly the relationships between audible re-radiation and rattling caused within dwellings by blast vibration and airblast overpressure.

In view of the high sensitivity of building occupants to re-radiated noise and rattling, blast emission limits should be based on (and would probably be largely controlled by) potential to cause these effects, as well as on tactile perceptibility. Unfortunately, documented research into the former is minimal. All of the discussion relating to vibration limits in the current ISO Standard 2631, for example, relates only to the *tactile* perception of vibration. The guidance from ISO 2631 specifically excludes taking into account auditory perception of re-radiated sound or rattling.

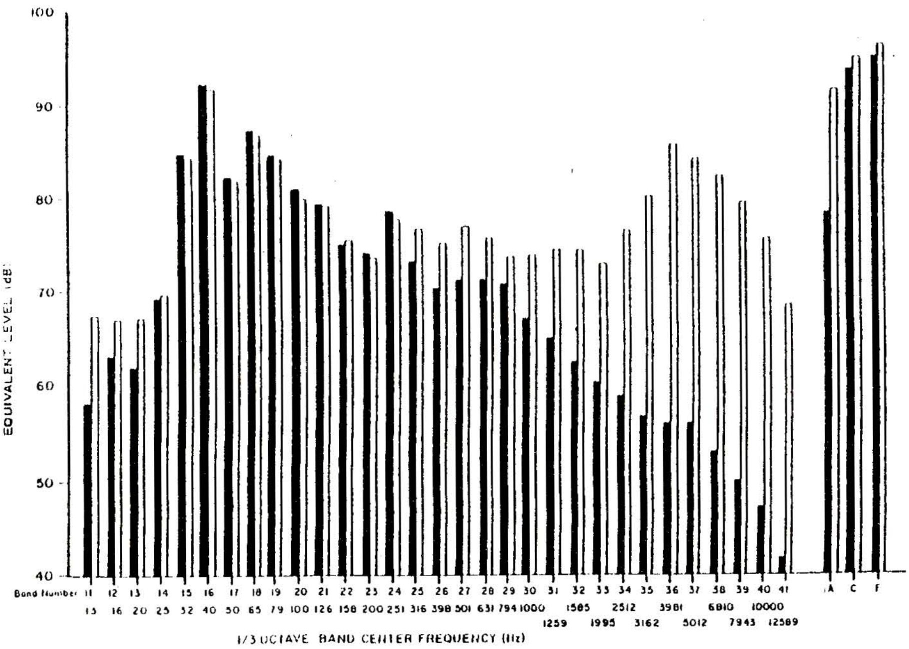


Figure 10.3 Spectrum of Indoor Blast Noise (With and Without Rattles)

The short term nature of excavation blasting in urban areas also needs to be taken into consideration, particularly where blasting can also result in environmental benefits by reducing the extent and duration of the use of dust- and noise-producing rockbreakers and other noisy equipment.

Annex A of ISO 2631-2:1989 provides tentative guidance on the magnitudes of long-term vibration at which adverse comment may arise. ISO 2631 also states in regard to shorter term works (page 1) that:

> *"adjustments and variances may be allowed for short-term engineering works (for example foundation excavation and tunnelling) where good public relations practices are followed and prior warning is given."*

Note 7 of Annex A also states the following in regard to transient vibration excitation with several occurrences per day (eg blasting):

> *"In hard rock excavation, where underground disturbances cause higher frequency vibration, a factor of up to 128 has been found to be satisfactory for residential properties in some countries."*

ISO curve 128 for vertical vibration (the most sensitive direction for people sitting or standing) corresponds to an RMS velocity level of 12.8 mm/s, or (for a sine wave) a peak velocity level of 18 mm/s. Blast vibration waveforms (particularly for tunnelling), with their higher crest factors, would correspond to even greater peak velocity levels. These vibration levels are considerably higher than the highly restrictive comfort criteria currently being applied by regulatory authorities to short term blasting in urban areas. Similar leniency is also applicable to acoustic emissions from blasting on short term excavation projects.

Measurement and analysis techniques also need to be reviewed to ensure that the measures used to define emissions correlate satisfactorily with the effects which disturb or cause discomfort to building occupants.

Finally, some "path finding" will be necessary, and those within the industry will be required to demonstrate the courage of their convictions in applying their knowledge and experience to the advantage of their projects, and for the benefit of the community and the environment.

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"NIGHTMARE ON LONSDALE"

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BACKGROUND

The Commonwealth Office building is on the corner of Lonsdale and Spring Streets, Melbourne. As part of the air conditioning/smoke control systems, there was a requirement for eight fans per floor, each to exhaust 900 L/s @ 250Pa.

The fans were in two banks of four fans, sited immediately above the acoustic false ceiling, with the requirement that with all fans running, the level on the floor below should not exceed NR40.

The building has 29 floors, meaning that 232 fans were required.

The space to house these fans was limited to a depth of approximately 600mm.

FAN/ATTENUATOR SELECTION

A number of different types of arrangements and fan selections were considered:-

1. Backward-curved in-line centrifugal fan
2. 400mm diameter contra-rotating axial at 1440 r.p.m.
3. 315mm diameter single-stage axial, running at 2880 r.p.m.

Selections 1 and 2 were eliminated due to there being insufficient room to attenuate the fan noise. Option 3 was adopted as the preferred solution for the following reasons:-

1. It was the most compact selection possible.
2. It was the most economical selection.
3. Its measured sound power, particularly in the low frequencies, was less than the other two options.

After discussion with staff at Fantech and an acoustic consultant, an arrangement was conceived as a likely solution (Figure 1) and a prototype was built for testing.

A test rig was set up in Fantech's factory where noise measurements using a Bruel & Kjaer sound level meter with octave filters were taken one metre from the fan assembly. Results of these measurements are shown in Figure 2.

Following development of the prototype, tests were conducted at Enersonics to check our readings and confirm that the fans would meet the required specification. These readings showed that the fans exceeded requirements by approximately 5dB to the 125 and 250Hz octave bands.

The prototype was returned to Fantech where various improvements and modifications were made, reducing the sound level by 6, 5 and 7dB in the octave bands 63, 125 and 250 respectively.

FIGURE 1

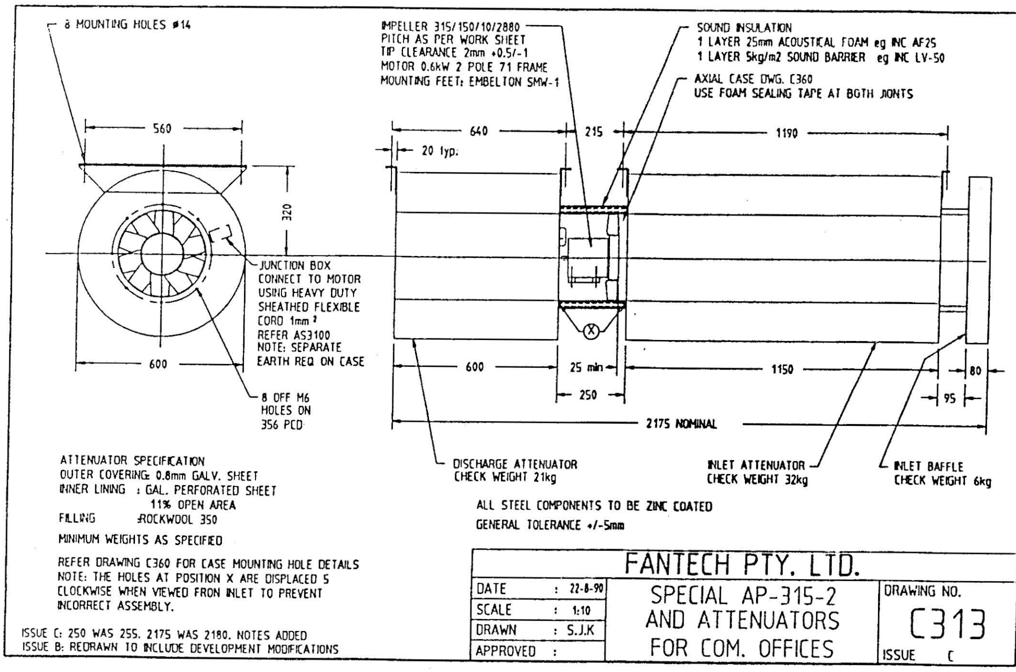
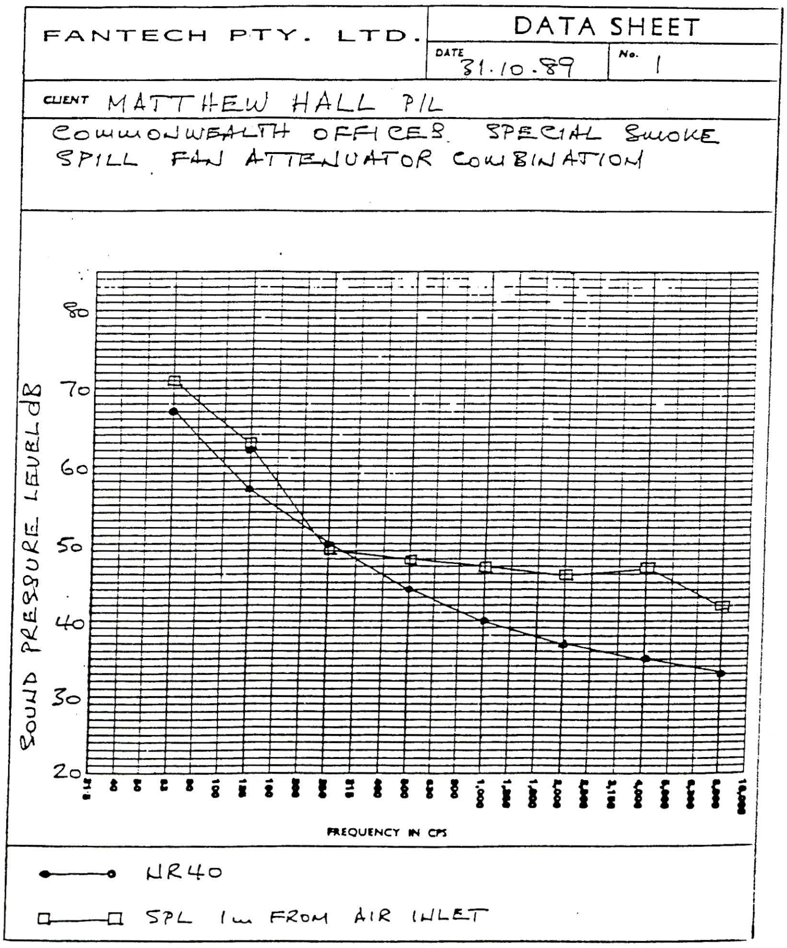


FIGURE 2

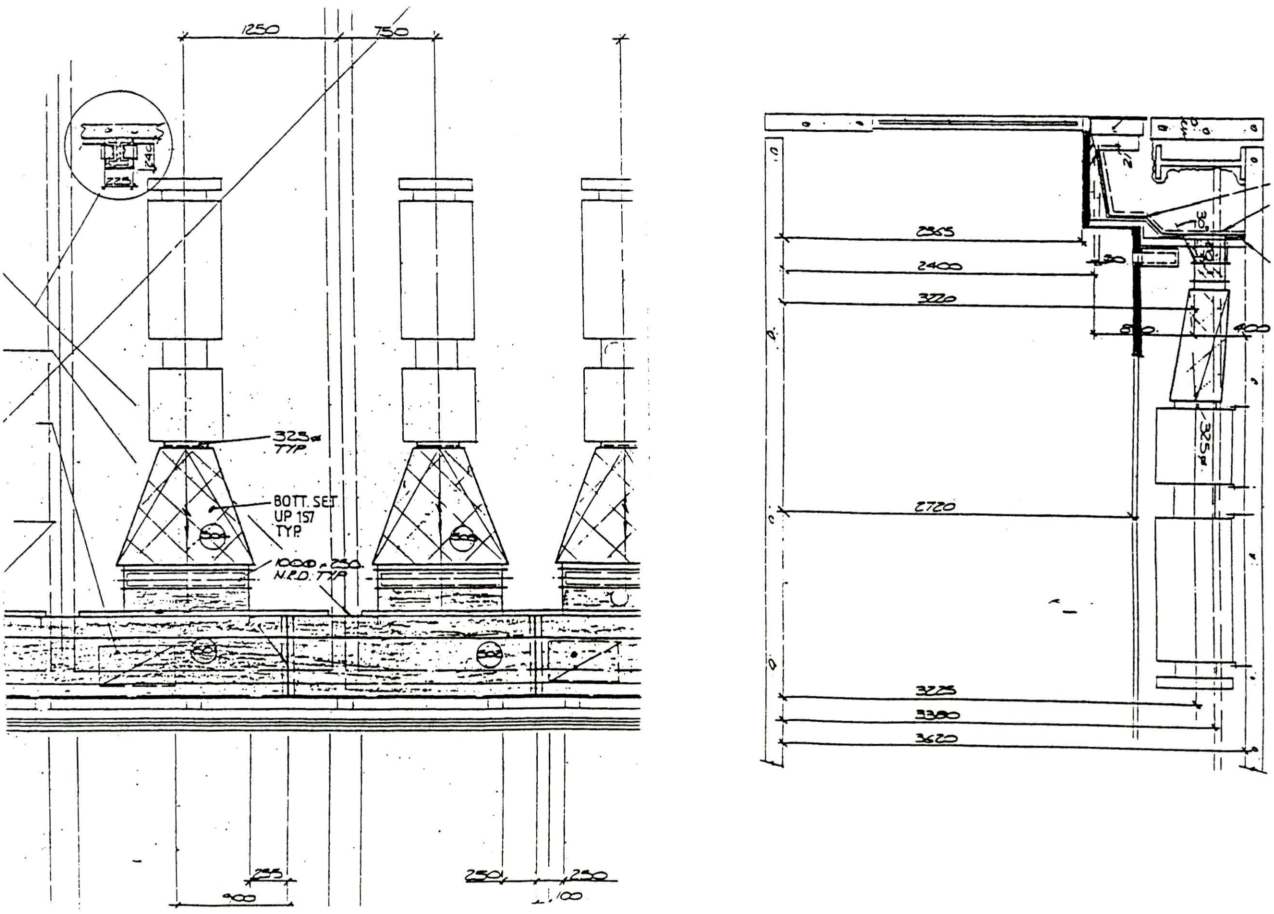


Following these improvements the client's acoustic consultant confirmed our measurements and agreed that the fans were now suitable to achieve the specified level, whilst warning that little or nothing was to spare and that quality control during production was of paramount importance.

In order to ensure all fans met QA, the tested fan was used as a benchmark. All production fans were tested for noise at one metre from the fan inlet and any fans found to be noisier than the test fan were rejected or re-worked prior to despatch.

The layout of the fans on a typical floor are shown in Figure 3. You will note there are no flexible connections between the fan/attenuator assembly and ductwork. One of the strategies in the fan design to minimise break-out, was to mount the impeller/motor assembly on rubber isolators.

FIGURE 3



THE SITE: THE NIGHTMARE

After all of the precautions and testing, we were confident of a successful installation. However, this was not to be!

On start-up, it was noticed that the noise levels with the relief/smoke spill fans running appeared intrusive. At this stage the false ceilings had not been installed. Further investigation revealed that levels in the 63Hz octave band were the most critical, with the 125Hz band being marginal.

It was also noted that with all fans running, there was a distinctive and intrusive beat note in the 63Hz octave band. A typical level recorded with all fans funning was 78dB, some 11dB over that allowed by the specification.

To further complicate matters, there was a wide variation in individual fans. With one fan only running, the levels at 63Hz could vary on a particular floor from 61 to 77dB.

As all the fans had been measured prior to despatch, it was possible to compare the readings taken on site with those recorded in the factory. There was absolutely no correlation. The factory tests showed that a bare fan without any attenuation had a SPL at one metre from the intake in the 63Hz band of between 67 to 71dB, with the noisiest fans on site often being the quietest in the factory, and vice versa.

A narrow band analysis was taken on site and this revealed a distinctive peak at 47Hz, which exactly coincided with motor speed. In other words, with a ten-bladed fan we were experiencing a phenomenon which occurred once every revolution! Strange enough; however, worse was to follow.

It was discovered that if fans were swapped in location; i.e. if a noisy fan was changed with a quiet fan, then the noisy fan became quiet, and vice versa. This led us to suspect that what we were witnessing was a resonant condition.

By making adjustments to the tension on the mounting brackets, it was possible on some fans to effect a significant improvement. However, on other fans, it made either no difference, or sometimes the noise got worse!

Time does not allow me to discuss all the solutions which were tried. By late in the job there were two acoustic consultants; one employed by the contractor and one by Fantech. In addition, the contractor invited an acoustic supplier to experiment on site.

Countless theories were postulated and investigated, with little or no success.

The bad news kept coming - someone noticed that when the fans were running on a particular floor, the noise level on the floor above was quite noticeable, even though no fans were running on that floor. Even worse, the whole floor was vibrating!

In other words, four fans weighing approximately 10 kg each, with plastic-bladed impellers finely balanced, and which were resiliently mounted, were shaking the whole building.

The natural frequency of the slab was measured and found to be 43Hz - very close to the fan resonant frequency.

THE SOLUTION

After many trials it was discovered that the air flow on site far exceeded design - by various means we established that the system pressure drop was about 25Pa - not the 250Pa specified. We had been testing the fans, and indeed selecting the fans, at the wrong pressure!

The acoustic contractor had experimented by inserting a 150 diameter pod in the inlet attenuator and to our surprise, the level improved by approximately 2dB in the 63Hz band. Experiments in our factory confirmed this, but also revealed that a non-acoustic pod worked even better, which would indicate that the improvements were achieved by improved air flow onto the impeller and not by absorption.

We tried changing impellers on one floor from ten to five blades and the improvement was dramatic. The critical frequency moved from the 63Hz band to the 250Hz band (not exactly what you would expect).

Finally we mounted the assembly on spring hangers having a 25mm deflection, which not only eliminated the noise and vibration on the floor above, but also reduced the beat note to an acceptable level.

In order to finally solve the problem, four things were done:-

1. All fan assemblies were spring-mounted (no flexible connectors were found to be required).
2. All fan blades were changed from ten blades 27^0 to five blades 25^0.
3. 150mm diameter non-acoustic pods were inserted in the inlet attenuator.
4. The attenuator inlet baffles were removed.

With all measures taken, the levels just met the NR40 criteria, with the critical frequency being in the 250Hz octave band.

NOISE CONTROL CRITERIA

by

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ABSTRACT

Noise Control Criteria have been developed for establishing the fact of a noise problem and the responsibility of the noise maker; and for applying precise acoustical principles at the design stage of buildings, equipment and machinery.

With any noise problem, criteria are applied to the appropriate noise measurements.

These criteria are fundamental criteria based on physio- and psycho-acoustic tests, or statutory criteria and regulations derived from them. They include

(a) Sound Levels and Loudnesses of sounds and noises classed as 'quiet', 'loud', 'deafening', etc,
(b) 'Speech Interference Level' and associated criteria, and
(c) miscellaneous criteria relating measured Sound Levels, frequency spectra, durations, etc to the degree to which people exposed to the noise find it annoying.

Many of these criteria have now been incorporated in Australian, International and other standards, and in statutory regulations.

Some case histories illustrate their application.

INTRODUCTION

1. Noise, variously defined as "sound which is undesired by the recipient" (ref 12, defn 1003), as "disagreeable or undesired sound or other disturbance" (ref 16, defn 308 1008(2)), or, more complexly, as

"(a) sound which a listener does not wish to hear,
(b) sound from sources other than the one emitting the sound it is desired to receive, measure or record,
(c) a class of sound of an erratic, intermittent or statistically random nature" (ref 9),

is, as indicated by these selected standard definitions, a complex matter having both well-defined physical (objective) and psychological (subjective) characteristics. Though often difficult to resolve, noise problems can be, and are successfully dealt with when all relevant physical and psychological factors are taken into account.

2. From the physical point of view of noise as "sound undesired by the recipient", it is possible with the many measurement techniques now available, though most were only beginning to be developed sixty years ago, to determine the characteristics of the source of the sound -- its location, duration, and frequency and wave-form characteristics -- and of the air-borne and structure-borne paths by which it is propagated and transmitted. From the psychological point of view it has been possible to measure many aspects of our human responses to sound and noise -- the sensitivity of the human ear to sounds of different pitch throughout the audible spectrum of 20 Hz to 20 kHz, its responses to sounds of different intensity in terms of loudness and annoyance, and the levels of sounds of different intensity, duration, pitch and waveform likely to interrupt tasks dependent on hearing other desired sound, or likely to cause temporary or permanent hearing damage. As yet, much work remains to be done to determine precisely the levels of impulsive and transient sound likely to cause permanent hearing damage.

3. The results of all these continuing researches and investigations have been used to develop quantitative Noise Control Criteria, which, as generally accepted criteria, are used as standards both for establishing the responsibility of noise-makers for reducing their noise, and also in the design of quieter building interiors, equipment, machinery and vehicles. Because vibration is often closely associated with noise because noise problems often include significant vibration, in that noise is generated by vibrating surfaces, and in that ground-borne and structure-borne vibration have to be taken account of in the design of buildings, Vibration Control Criteria are quite often also included in the consideration of Noise Control Criteria. In what follows there is both a brief outline of the nature and development of many Noise and Vibration Control Criteria, and also some specific illustration of their application in a selected group of case histories.

NOISE AND VIBRATION MEASUREMENT

4. The possibility of measuring the characteristics of sounds by means of a not too bulky portable instrument, that is, able to be used outside a physics or acoustics laboratory, became possible in the late 1920s with the development of microphones and the electronic amplification of electrical signals. Amongst the earliest instruments to be developed was a 'noise meter', soon to become standardized as the 'Sound Level Meter', comprising a microphone, amplifier, calibrated attenuator, frequency-weighting networks and output meter (normally giving 'rms' rather than 'peak' indication). This instrument was designed to provide a measure of the oscillatory sound pressure -- either with equal response to all frequencies within the audible band from 20 Hz to 20 kHz, or with selective response according to a

frequency-weighting network -- in the noise propagation path at the location of the microphone, with the oscillatory sound pressure measured as a ratio or level in decibels (dB) with respect to the standard reference pressure of 0,0002 dyn/sq cm, now described as 20 micropascal (μPa).

5. The history of the 'Vibration Meter' is similar, with portable Vibration Meters, now capable of measuring vibration acceleration, velocity or displacement within a frequency band of 0 Hz or around 0,2 Hz to 15 kHz, having begun to be developed at about the same time as the Sound Level Meter, because of their common use of electronic operation.

6. Other instruments to have become associated with Sound Level and Vibration Meters in the measurement of noise and vibration are various pure tone and other noise signal and vibration generators, frequency analyzers (filter and FFT types) using either constant percentage bandwidth -- from broader band one octave (71% bandwidth) through half octave (35%) and one-third octave (23%) band analyzers down to the narrow one-twentyfourth octave (3%) band analyzer -- or constant bandwidth narrow band analyzers with a bandwidth between 0,025 and 50 Hz depending on the selected overall bandwidth, cathode ray oscilloscopes, graphic level and signal (analog or digital) recorders, and statistical level analyzers.

7. In addition, the Sound Level and Vibration Meters themselves have been considerably developed to include the measurement of both rms and peak values, usually with a 'maximum hold' control so that maxima of either rms or peak levels can be captured and displayed for convenient observation. Modern meters usually include, as well as the 'fast' (F) and 'slow' (S) time responses, a now standardized 'impulse' (I) response (with short rise, long decay time) to enable the maximum value of a very short duration impulsive type sound to be captured and conveniently observed. As noted elsewhere, there is much still to be learned about the nature of impulsive sound, and of our human responses to it (both physical and psychological).

8. Care has been taken here to carefully distinguish between 'peak' and 'maximum' values. In this use, 'peak' is used only in its specifically mathematical sense to contrast, and distinguish between the 'rms' and 'peak' values of the waveform of any sound or vibration signal. For precision and clarity of meaning, 'peak' needs to be kept for this use only. When the varying amplitude of a continuing waveform from time to time reaches a maximum value, for clarity it is also best referred to as a 'maximum', not a 'peak'. Unfortunately, some writers of articles, standards, etc, and instrument manufacturers, in not being careful in this, cause confusion. What, for example, is a 'peak hold' control? a control specifically for measuring 'peak' values, or (more properly) a 'maximum hold' control?

9. The most recent edition of AS 1259 on Sound Level Meters, AS 1259.1-1990, 'Sound level meters: non-integrating' (ref 4), is still, unhappily, the source of some confusion over use of the word 'peak', especially in paragraphs 4.3.2 and 5.5 on 'Time-weighting', and 8.1, 8.3 and 8.5 on 'Detector and Indicator Characteristics'. The fundamental error seems to arise from including 'peak' as a time-weighting characteristic instead of confining it, with 'rms', to discussions on detector and indicator characteristics where it properly belongs. Where, as in paras 4.3.2, end of 8.1, and beginning of 8.5, there are references to 'peak' (in contradistinction to 'rms') values of the sound waveform, the word 'peak' would be better expanded to 'instantaneous peak value'. And where, in the middle of paras 8.1 and 8.3, there are references to a 'peak detector', it would be more clearly referred to as a 'maximum indicator (peak detector)'. This problem arises over discussions

of 'P characteristic' and 'P detector-indicator'. As a time-weighting the 'P characteristic', though not as precisely defined as the 'I characteristic', also possesses its short averaging and long fall times, doing for 'peak' values what the 'I characteristic' does similarly for 'rms' values of the sound waveform under measurement. The whole matter would be properly clarified by using 'rms' and 'peak' in connexion with detector characteristics, by having, as far as possible, F, S and I time-weighting characteristics available for both 'rms' and 'peak' values of the sound level, and ensuring that controls for obtaining maximum values of either 'rms' or 'peak' levels are described as 'maximum hold' (NOT 'peak hold'!!). Our language is a precise medium for communicating our thought and feeling; clear thinking produces precise statement, muddled thinking (such as here over 'peak') generates unnecessary and untold confusion.

THE DEVELOPMENT OF NOISE CONTROL CRITERIA

10. Although the Sound Level Meter is an 'objective' meter in that it measures physical sound pressure levels independently of any human auditory estimate of the strength of a sound or noise, it was originally hoped that it might also provide a measure of Loudness -- "an observer's auditory estimate of the strength of a sound" (ref 12, defn 3010) -- because in much noise measurement work during the 1930s, 40s and 50s, attention was focussed on the problem of obtaining a numerical measure of the loudness of sounds and noises. While it was appreciated that the sensation of loudness was not the only significant subjective characteristic of a sound, experience had shown that the average person's degree of tolerance, or aversion to an unwanted sound was in general more closely related to the loudness of that sound than to any other factor easily susceptible of measurement (ref 13). Portable Loudness or Phon Meters experimented with during this period -- in which the observer listened to the sound with one ear, and with the other listened through an earphone to a standard 1 kHz tone whose level was then adjusted by means of a calibrated attenuator to be of equal loudness -- were finally deemed unsatisfactory for this purpose because of the too great difficulty of replicating in the field the precise laboratory conditions required for this comparative measurement (ref 2, section 3).

Noise criteria based on Loudness

11. As part of this work of measuring the loudness of sounds and noises, a number of Noise Control Criteria based on Loudness were developed. Early in this period were the 1933 Fletcher-Munson 'Equal loudness contours for pure tones', and the 'Loudness function' (relation between Loudness Level in phons and Loudness in loudness units or sones) which were standardized, tentatively in 1936, and confirmed in 1942 in American Standard Z 24.2, 'Noise Measurement' (ref 1). From the 40, 70 and 90 phon pure tone loudness contours were developed the corresponding 'A', 'B' and 'C' (= low to medium fidelity 'flat') frequency weighting networks, which were then standardized and subsequently adopted in the Sound Level Meter, in the hope (now known to be vain except in the particular case of single pure tones) that it could be used as a Loudness Meter. The more recent (1969) 'D' weighting, now also standardized (ref 4), was developed by K D Kryter as part of his researches into the measurement of the annoyance of aircraft noise (ref 32, p 14).

12. While the term 'sound pressure level' has been, and is, used to describe the measurements made with Sound Level Meters set to the 'flat' frequency weighting, the term 'sound level', in dB(A), dB(B), dB(C) or dB(D), refers, respectively, to the use of the A-, B-, C- or D-weighting networks in measuring sound pressure levels. From this was developed the procedure widely used during the 1930s, 40s and 50s of measuring and quoting sound levels

below 55 in dB(A), between 55 and 85 in dB(B), and above 85 in dB(C), a fact to be remembered when interpreting noise measurements made in those earlier years. TABLES 1A and 1B, as early alternative forms of this type of Noise Control Criterion, show the sound levels of numerous common sounds, mostly measured under specified conditions, and usefully rate the levels on a qualitative loudness scale from 'very faint' to 'deafening'. TABLES 1C and 1D show other groups of typical sounds with their levels given as Loudness Levels in phons and Loudnesses in sones.

13. From 1933 there was continuing development of methods of measuring the loudness of sounds and noises, and of the psycho-acoustic factors underlying them. In 1937, two English researchers, Churcher and King, published another set of 'Equal loudness contours for pure tones', and a 'Loudness function', slightly different from the Fletcher-Munson findings. Finally, in the immediate post-World War II period, all this earlier work was superseded by the researches, now considered sufficiently definitive to be internationally standardized, of D W Robinson, R W Dadson and others, who established the 'Loudness function' now given in, for example, British Standard (BS) 3045:1958 (ref 14) and AS 1047-1971 (ref 2, Table 1), and the 'Equal loudness contours for pure tones' given in BS 3383:1961 (ref 15) and AS 1047-1971 (ref 2,Appendix A). Related work, by S S Stevens and E Zwicker in determining the loudness of octave or one-third octave bands of noise, has also been standardized, in ISO 532-1975, 'Method for calculating loudness level' (ref 29), so enabling the development of values of 'Computed' or 'Calculated' Loudness Level of sounds and noises from their octave (ref 29, method A) or one-third octave (method B) analyses, which, particularly in the case of steady sounds, are a reasonably close estimate of their Loudness Level as measured under laboratory conditions when this is also available. Work by K D Kryter on the 'Annoyance' of bands of noise, similar to that of Stevens' loudness investigations, has also been accepted, and is used principally, though not solely, for the measurement and computation of the 'Perceived Noise Level' of aircraft noise; but so far as is currently known is not yet standardized. Perceived Noise Levels are quoted in 'PNdB' (a unit analogous to the phon for Loudness Level), with the unit of annoyance on an arithmetic scale being the 'noy' (analogous to the sone for Loudness).

14. The measurement of the Sound Level of a noise, as a single number estimate of its loudness, has culminated in the work of R W Young, who found in 1964 that, for a group of common noises of different spectrum shapes, their A-weighted Sound Level was the equal best (with dB(B) and AS 1469 NR no.) of 13 measurement criteria for ranking these noises according to their loudness (ref 38). From Young's Tables I and II, and also from TABLE 1D below, it can be concluded that a sound's Loudness Level in phons can be satisfactorily estimated from its Sound Level in dB(A), the Loudness Level estimate being dB(A) + 13 ($\pm$ 1, for 95% confidence). As a further result, the A-weighted Sound Level of a noise, as the most readily measurable, is the most widely recommended and used single number noise measurement for noise criteria, specifications and measurements (as originally recommended in the American Tentative Standards for Sound Level Meters, Z 24.3-1936!!). Now also, earlier measured Sound Levels of noises obtained according to the pre-1964 "less than 55 in dB(A), between 55 and 85 in dB(B), and above 85 in dB(C)" method of measurement can, if their octave or narrower band frequency spectra are known and therefore able to furnish a satisfactory estimate of the relationships between their dB(A), dB(B) and dB(C), be all converted with reasonable accuracy to their corresponding Sound Levels in dB(A) to enable their readier comparison.

15. As an important Noise Control Criterion the Loudness criterion is a general one covering the whole of the audible intensity spectrum from 'very

faint' to 'deafening' in 20 dB wide Sound Level bands from 0 to 140 dB (and beyond). As a criterion its main function, apart from indicating and locating the two hearing thresholds (of hearing at the 'very faint lower limit, and of pain at the 'deafening' upper limit), is that it indicates preferences -- that 'quieter' noise is preferable to louder noise. By virtue of its derivation from the logarithmic decibel scale of Sound Levels, the phon scale of Loudness Level does not give scale numbers proportional to the Loudness. This is provided by the 'Loudness function' (see BS 3045:1958 (ref 14) and AS 1047-1971, Table 1 (ref 2)), in which the sone scale of Loudness -- an arithmetic rather than logarithmic scale -- gives scale numbers proportional to loudness, from which it is readily derived that a 10 phon change in Loudness Level is equivalent to a two-fold change in Loudness. However, it did not take workers in the field of acoustics dealing with specific noise problems very long to find that the qualitative Loudness criteria provided by the descriptions of noises according to their overall Sound Levels as, for example, 'faint', 'moderate' or 'very loud' over these 20 dB wide ranges, lacked precision in that these loudness descriptions did not necessarily indicate exactly what was required and desirable, and lacked authority in that, in not being sufficiently clear and forceful, they still allowed noise-makers to evade their responsibility of reducing the noise.

Criteria based on speech intelligibility

16. Considerable advances in the establishment of more precise and authoritative Noise Control Criteria have been made as a result of the researches of those who have investigated the acoustic conditions under which reliable speech communication at various voice levels from 'normal' or 'conversational', through 'raised' and 'very loud' or 'stage voice', to 'shouting' is or is not possible against different levels of background noise, for different distances apart of those speaking to each other. Two procedures have considerable current use: measurement of the 'speech interference level' of the background noise in a space; and computation of the 'articulation index' of the environment, room or area in which the speech communication is to occur. These, and several other methods of assessing speech intelligibility and privacy, are included and described in detail in AS 2822-1985 (ref 11).

17. Of these two procedures, computation of the Articulation Index (AI) of an unoccupied room or enclosed space, though the more detailed and complex, can to be used to estimate the degrees of both speech intelligibility and speech privacy possible against the existing background noise in the room. The AI of a room is computed by measuring the one-third octave spectrum of its background noise (through the 200 to 6300 Hz bands) and plotting these one-third octave levels on an Articulation Index Dot Field Chart (such as at AS 2822-1985, Figure 2) and counting the number of dots above the background noise spectrum line, the AI being computed as this number of dots as a fraction of the total of 200 dots (each dot above the noise spectrum therefore contributing 0,005 to the AI). The AI Dot Field Chart is constructed on a one-third octave analysis chart including band centre frequencies from 200 to 6300 Hz and band levels from 10 to 80 dB. The 200 dots, allocated to the various frequency bands according to AS 2822-1985, Table 1, and then uniformly distributed within each band according to AS 2822-1985, Figure 2, are enclosed within a 30 dB wide space on this chart, the space having its lower and upper boundary lines meeting the 400 Hz abscissa at 40 and 70 dB. Below 400 Hz the boundary lines have a slope of + 13 dB/decade; above 400 Hz their slope is - 18 dB/decade. The degrees of speech intelligibility or privacy possible in the room under test are then derived from AS 2822-1985, Table 2 according to the computed AI. Details of the conditions which apply to this procedure and under which they are valid are given in AS 2822-1985, which in its Figure 1 gives a series of Maximum Permissible Speech Interference Levels

(SILs) for reliable conversation as Noise Control Criteria for comparison with a measured SIL of the background noise in the room or space under test.

18. Calculation of the SIL of the background noise in an area (exterior or interior) is a simpler procedure and requires an octave band analysis of the background noise at the location where the speech communication is to occur. This measured and calculated SIL is then compared with the results of psycho-acoustic tests in the form of a Table or Graph showing 'Maximum Permissible SILs for Reliable Communication' between speaker and listener(s) for several speaker's voice levels and various distances between persons. Notes appended to the Table or Graph specify the conditions under which these maximum permissible SILs can be validly used.

19. The earliest form of the SIL of a background noise, due to L L Beranek (1947), was calculated as the arithmetic mean of the sound pressure levels measured in the older 850, 1700 and 3400 Hz (that is, 600-1200, 1200-2400 and 2400-4800 Hz) octave bands. With the introduction of the preferred series, the new SIL (here called SIL3) was calculated as the mean of the sound pressure levels in the 500, 1000 and 2000 Hz octave bands. In the later 1960s, J C Webster extended Beranek's work and derived a new set of 'maximum permissible SILs' (ref 36, pp 36-38) to correspond with the SILs measured from the levels in this preferred series of octave bands. More recent work has indicated that, with the preferred series of octave bands, a SIL (here called SIL4) calculated as the mean of the levels in the four octave bands from 500 to 4000 Hz (covering an audible band from 350 to 5600 Hz) gives better coverage of this human speech frequency band. The use of a SIL based on these four octave bands was suggested in ISO Technical Report TR 3352-1974 (ref 30), and has been adopted in AS 2822-1985 (ref 11) for the measurement and calculation of SIL4s. However, their values of 'Maximum Permissible SIL4 for Reliable Communication' differ noticeably, with those in AS 2822 being similar to Webster (1969), and those in the ISO report being about 5 dB lower and therefore more conservative. TABLE 2 included here is derived from AS 2822-1985, Figure 1 and Webster (1969), with some extrapolations on the basic principle of a 6 dB change per step in voice level or per two-fold change in distance between persons.

20. Because of the precision -- a precision significantly greater than is possible with the criteria based on loudness -- with which these Noise Control Criteria based on speech intelligibility can be used, noise problems can be much more precisely defined. With background noise having a particular SIL, speech communication against it is fairly clearly defined as either satisfactory or unsatisfactory, the band of uncertainty being about 3 to 5 dB. And with a room and its existing background noise having a particular value of AI, the degrees of speech intelligibility and privacy likely are reasonably clearly defined as, for example, according to AS 2822-1985, Table 2.

21. In addition, it has been found that, for a variety of typical background noises, the difference between their A-weighted sound level and SIL4 lies between 5 and 10 dB with a mean of 8 dB. The scope of the SIL criterion has in AS 2822-1985 therefore been extended to allow SIL4s to be estimated from their A-weighted sound level and this mean difference. TABLE 2 here has been similarly extended.

22. The conditions for reliable speech communication against background noise thus determined as a result of these psycho-acoustic researches have proved most fruitful in enabling the development of a large group of Noise Control Criteria for limiting the background noise levels in areas in which

the expected large variety of human activities occur -- from sleeping to eating, from work to recreation and travel, and from concerts, theatrical performances, and worship services to business and conferences, etc.

23. An early example (probably resulting from Beranek's work in the late 1940s) of a collection of such 'Criteria for Noise Control' for different room uses, in terms of maximum permissible SIL for reliable speech communication, measured when the room is unoccupied (ref 34, old SIL, p 75), is included here as TABLE 3 with old SILs converted to equivalent SIL4s. Corresponding estimated A-weighted sound levels (numerically greater than SIL4 by 8 dB as in AS 2822-1985) have been included for convenient comparison with other more recently published and similar criteria.

24. AS 2107, first published in 1977 as 'Ambient Sound Levels for Areas of Occupancy within Buildings', and revised in 1987 as 'Recommended Design Sound Levels and Reverberation Times for Building Interiors' (ref 10), gives a very comprehensive set of Noise Control Criteria for virtually every type of building and interior use, in terms of two recommended design sound levels for limiting background noise -- a "satisfactory" level, and a "maximum" level (usually 5 dB greater, but occasionally 10 or even 20 dB) which ought not to be exceeded. Most of these criteria are given in terms of A-weighted sound levels, but with several particular groups given in terms of AS 1469 (= ISO) Noise Rating (NR) no. (ref 8), the NR no. allowing stricter control over the background noise because it also specifies a limiting spectrum shape. Also, because of the determined spectrum shape with any NR no, there is a unique relationship between NR no. and SIL4, the NR no. being always greater than SIL4 by 1 dB.

25. For comparison purposes, TABLE 3 includes the corresponding recommended levels from AS 2107-1987, and shows that, though similar to the older criteria, the AS 2107 recommendations are sometimes somewhat more conservative. The 1987 revision of this Standard also gives much useful information on the control to be exercised -- either at the design stage of new buildings, or in the control of noise in existing buildings -- over the amounts of reverberation to be allowed in many of these areas. To indicate something of the comprehensiveness of the scope of AS 2107, TABLE 4 gives a brief selection of typical Recommended Design Sound Levels and, where given, Reverberation Times for different areas of occupancy in buildings. Where provided, the recommended reverberation times are very useful additional Noise Control Criteria. In some interior areas, reverberation times of the order of 0,6 to 1,0 s provide useful amplification of the sound without being long enough to cause confusion by reducing speech intelligibility. It is, however, surprising that some interior areas in, for example, industrial buildings (TABLE 4, group 3) and public buildings (post offices and general banking areas) are not provided with maximum desirable reverberation times of the order of 0,5 s in order to minimize the possibility of noise problems in a reverberant area of the type sometimes referred to as 'cocktail party acoustics', in which one group of people begins talking, but when joined by another, then another, and further groups, find they must talk the more loudly, until eventually all groups are shouting!! Large lunch areas with too much reverberation are especially prone to this noise problem; so also in a different way are reverberant factory areas having noisy operations such as hammering, and machines such as punch presses within them. There is no longer any need for these areas to have 'cavernous' acoustics, with the wide range of acoustic absorbent materials (including concrete panels) now available.

26. The comprehensive Noise Control Criteria included in this Australian

Standard serve two major purposes. In the case of rooms and interior spaces in existing buildings they are criteria for determining whether or not the current background noise levels (which may arise from either occupational noise inside the building, or traffic or other noise coming in from outside) are a noise problem, and if so, by how much the excess noise must be reduced. For proposed buildings they provide design criteria to be used by architects and engineers -- particularly those for whom acoustics are not of special interest, and who are therefore not likely to be conscious of the real and pressing need to take proper account of acoustical considerations -- so that mistakes, normally requiring costly subsequent correction, are avoided and not made. The case histories to be discussed below include several such 'mistakes' made through complete neglect of acoustical (and vibration) considerations at the initial design stage of a building.

27. AS 2107 and similar standards are today a very necessary part of the equipment required by all building designers, whether architect or engineer. However, it must be also recognized that they represent, as it were, only one side of the coin. The other side requires building designers to be thoroughly familiar with the acoustic properties of their building materials, so that they know when to use acoustic absorbent materials for reducing the reverberation in a room or enclosure, and when to use barrier materials of mass per unit area of wall, floor or ceiling sufficient to adequately limit the transmission of noise between rooms, or between a room and the exterior environment (either to prevent loud noise generated inside from getting out, or to protect an interior room from loud noise generated outside, such as from traffic or other industry). When in doubt, building designers should always confer with an acoustical consultant!! The following general formula enables calculation of the Sound Reduction Factor (SRF) in dB at any frequency, f kHz, of a single thickness barrier of building material of surface density, D_s kg/sq m, but does not allow for co-incidence effects (which can reduce a partition's SRF at its critical co-incidence frequencies by 10 to 15 dB).

$$SRF = 18{,}5460\ (D_s)^{0{,}17937} + 4{,}2651\ (D_s)^{0{,}14201} \cdot \ln f \quad \text{dB}$$

28. In addition to the two Australian Standards (2822 and 2107) already referred to, which provide the basic criteria for noise control inside buildings, there are three further groups of Australian Standards concerned with aspects of architectural and building acoustics: Standards describing methods of testing building materials for acoustic absorption or transmission loss, Standards describing acoustical tests in buildings, and Standards concerned with the impact of environmental noise on human activity inside buildings. Those currently available as Australian Standards are listed in Appendix A. These standards then require to be supplemented by the Acoustic Absorption and SRF data provided by the manufacturers of acoustic and building products.

Criteria based on human hearing loss

29. The two groups of Noise Control Criteria already described cover, firstly, in the case of the Loudness criteria, the whole audible intensity spectrum from 0 to 140 phons, and secondly, in the case of the Speech Intelligibility criteria, a critical intensity band at the centre of this spectrum. The other critical area of this intensity spectrum is that already pointed to by the Loudness criteria as the 'deafening' region -- comprising those noises of high audible intensity which are highly likely to cause permanent hearing and hearing loss. Hearing damage and loss from this cause are described as

'noise-induced' to distinguish them from, for example, hearing loss due to age (presbycusis) or ear disease.

30. As can be imagined, Noise Control Criteria designed to reduce or prevent noise-induced human hearing damage and loss are contentious. Not only do such criteria as have been already been determined depend on the duration of our exposure to these intense noises, and on their frequency spectrum and wave shape (for example, steady or impulsive), but they depend also on the decisions of two groups of people sitting on opposite sides of a table, over whom governments, government committees and other statutory organizations have to arbitrate as best they can! On one side of this table are all those many groups of people concerned with human health and welfare; while on the other we have the noise-makers -- the administrators, manufacturers, etc who find it costly, often enormously so, to reduce their noise. But reduce it they must; and here we must acknowledge and pay tribute to those who already have achieved much noise reduction. The classic example comes from the construction industries where rivetting, as a truly deafening process for both rivetters and adjacent workers, was replaced by electric arc welding. A more recent example is that of mobile and stationary air compressors, which up to about 1960 were likely to generate a noise level of 90 dB(A) or more at 1 metre, but which by around 1975 had, through various quietening means, this level down to 70 to 75 dB(A) at 1 metre. The noise problems of, for example, circular saws and pneumatic paving breakers remain among those more stubborn ones still to be solved.

31. The results of many and varied researches have greatly contributed to our knowledge of noise-induced hearing damage and loss, and to the setting of such Hearing Damage Risk Noise Control Criteria as we currently have. Though not there precisely defined, hearing damage risk noise levels are indicated in the Loudness criteria (TABLES 1A, 1B, 1C) through sounds having sound levels above 100 dB (= 100 dB(C)) being 'deafening', and through the threshold of feeling or pain being located at somewhere above 120 dB or at 130 phons.

32. More precise hearing damage risk criteria have been developed as a result of studies of noisy work environments, and of regular audiometric tests on those working in them. These criteria are variously defined: in terms of maximum permissible noise exposure level, maximum exposure duration, and of frequency, for both pure tones, and also bands (usually octave or one-third octave) of broad band noise. Some published criteria combine in one Table or Graph the effects of all three variables: maximum permissible level, exposure duration, and frequency. Others include only sound level and duration.

33. As the sensitivity of the human ear varies with sounds of different pitch, so also does its proneness to suffer noise-induced hearing damage and loss, its greatest proneness to hearing damage being around 3 kHz. Although numerous attempts have been made to develop a set of hearing damage risk spectra below which no hearing damage can be reasonably expected to occur, none, except perhaps the most conservative, appear able to inspire any great confidence in them. One of the earlier damage risk criteria, and still the most conservative, is that sounds of level above 100 dB(B) should be regarded as probably unsafe for long term everyday exposure, and ear protection or noise reduction is necessary; levels below 80 dB(B) are probably safe even with pure tones, with no hearing damage likely; for sound levels above 80 dB(B) more detailed investigation and frequency analysis are necessary (ref 34, p 76).

34. A group of typical hearing damage risk criteria incorporating maximum

permissible noise levels in terms of both exposure duration and frequency spectrum are given, for bands of noise, in TABLES 5A, 5B, 5C and 5D, and for pure tones in TABLE 6. Because the frequency contours for broad band noise can represent whole spectra of wide band noise, the corresponding overall sound levels for these contours have been included in TABLES 5A to 5D to facilitate comparisons between them.

35. Of the various Damage Risk Contours for broad band noise, the 1953 levels in TABLE 5A (from American Standards Association (ASA, now ANSI) Committee X2) represent an earlier assessment of this problem; the 1956 Damage Risk Levels (with mean values similar to those of ASA Committee X2 above) have additional 'serious risk' and 'negligible risk' contours 10 dB above and below the mean contour. The Kryter contours of TABLE 5B extend those of TABLE 5A to take account also of daily exposure duration, the 8-hour contour being fairly close in sound level to the 1956 'negligible risk' contour of TABLE 5A. TABLE 5C gives the results of some French investigations to re-determine the 8-hour 'serious risk' and 'negligible risk' contours. The two French contours are significantly more conservative than any of the others shown here, in that the French 'serious risk' contour is similar to the 1956 'negligible risk' of TABLE 5A and Kryter's 8-hour contour of TABLE 5B; and in that the French 'negligible risk' contour of 82 dB(A) is 4 to 17 dB below the 1956 92 dB(A) 'negligible risk' contour, and 10 to 17 dB below Kryter's 94 dB(A) 8-hour contour, the greatest differences with both being at frequencies above 1 kHz. TABLE 5D shows Kryter's 8-hour and the French 'negligible risk' contours relocated to give overall sound levels of 90 dB(A), 85 dB(A) and 74 dB(A) = 80 dB(B), to correspond, respectively, with the two 8-hour 'Daily Noise Dose' = 1,0 criteria of the Victorian 1978 and 1992 'Noise' regulations, and the earlier conservative criterion of 80 dB(B) for 'probably safe' daily 8-hour noise exposures.

36. Further criteria concerned with noise-induced Hearing Damage Risk are given in the 1983 and 1988 (current) editions of AS 1269, 'Hearing conservation' (refs 5,6), AS 1270-1988, 'Hearing protectors' (ref 7), and the Victorian statutory 'Occupational Health and Safety (Noise) Regulations 1992 (ref 25) which now supersede the original 'Health (Hearing Conservation) Regulations 1978'(ref 24). The main thrust of these Noise Control Criteria is that the environmental noise levels in all places of employment are to be such that any person's 'Daily Noise Dose' does not exceed 1,0. The primary aim is for noise exposure levels to be reduced by engineering means as the long-term measure. Where this is not immediately feasible, other methods become necessary, such as the wearing of hearing protection devices, though these must be regarded as only a short-term measure, not as a long-term measure to replace eventual noise reductions by engineering means.

37. As an almost universal practice, the Daily Noise Dose of 1,0 has been defined in terms of an 'Equivalent Continuous A-weighted Sound Pressure Level' over an 8-hour day (L_{Aeq8h}) of 90 dB(A), with some allowance for exposure to higher levels for shorter daily periods on the basis of an additional 3 dB (the internationally accepted increment, but smaller than the US OSHA increment of 5 dB) for each halving of the exposure duration, up to (as in the original Victorian regulations of 1978) a maximum exposure level of 115 dB(A)S (that is, using the Sound Level Meter 'slow' time weighting). In the new 1992 now current Noise Regulations, the Daily Noise Dose of 1,0 has had its basis reduced from 90 to 85 dB(A) for an 8-hour daily exposure, and the maximum permissible exposure changed from 115 dB(A)S to 140 dB(lin)$_{pk}$, using a 'P' time weighting function. The serious confusion in the statement of this last provision has already been discussed (para 9 above).

38. AS 1269-1983, as well as a Partial Noise Dose calculation chart (Fig 3.1), also gives (in Appendix D) Tables of 'Calculated Incidence and Degree of Hearing Loss' in various groups of noise-exposed people, in terms of the percentages of people likely to suffer noise-induced hearing loss, and their mean percentage loss of hearing, for 8-hour Exposure Levels from 75 to 115 dB(A) and exposure durations from 5 to 45 years. These indicate how the expected incidences of hearing loss begin to increase more sharply for 8-hour Exposure Levels above 90 dB(A), and provide at least some reasonable basis for what is otherwise described as the arbitrary selection of 90 dB(A) as the 8-hour Exposure Level criterion. With the reduction of this level to 85 dB(A) in the new Victorian government statutory 'Occupational Health and Safety (Noise) Regulations 1992', there has been now provided a somewhat greater protection against noise-induced hearing damage for employees in noisy work environments.

39. The Tables of 'Calculated Incidence and Degree of Hearing Loss' in AS 1269-1983, Appendix D also indicate that, even for equivalent continuous 8-hour noise exposure levels as low as 75 dB(A), some people will begin to experience small hearing losses after 20 and more years' duration of regular exposure. This gives considerable point to the wisdom of the earlier judgment (para 33 above) that noise exposure levels up to 80 dB(B) (approximately equivalent to 74 dB(A)) "are probably safe even with pure tones, with no hearing damage likely".

40. Noise regulations such as the Victorian 'Occupational Health and Safety (Noise) Regulations 1992', because they are government statutory regulations, are therefore legally enforceable and have to be acted on. This has important implications, not only for existing work places where noisy plant and machinery will need to be quietened, or replaced with new and quieter equipment, and employees acoustically shielded from intractably noisy locations, but also for the design of new work places, which would use, for example, progressive flow systems involving employee isolation from noisy processes, and in which machinery and plant would be considered as at least potentially noisy and therefore also to be acoustically isolated. Such design work should be expected to proceed on the acoustical basis that, if possible, noise should be reduced at its source; that, where this is not readily possible, significant reduction is required in the sound transmission paths between noise source and nearby persons; and that regulations such as the Victorian 'Occupational Health and Safety (Noise) Regulations 1992' provide the Noise Control Criteria.

Noise Control Criteria for machinery and plant

41. Reducing the noise of plant and machinery at its source is always a task calling for much ingenuity -- both in properly identifying the noise sources, and then in locating them. There is ample evidence that much has already been achieved through, for example, properly designed anti-vibration mounts, substituting double-helical (or similar) gears and pinions for straight spur type gear drives, reducing fan speeds (if possible, to below about 700 r/min) and increasing the pitch of their blades, higher hydraulic or pneumatic force in impact processes (such as hammering), selected use of greater machine mass to more effectively absorb impact energy, and the careful acoustic shrouding of highly localized noisy operations. And there is still the challenge for machinery and plant designers to produce, for example, quieter circular saws and pneumatic paving breakers.

42. Some of the noise reductions already achieved have been effected

through organizations purchasing new machinery and plant inserting noise limiting clauses in the purchase specification, and then insisting that they be adhered to, in order to obtain lower than previously customary operating noise levels. In addition, of the numerous Australian Standards specifying methods of noise testing (see Appendix B), some, such as AS 2726-1984, 'Chain saws -- safety requirements', include for the guidance of purchasers maximum permissible noise and vibration limits for the equipment under normal operation. Many of these types of industrial noise will be effectively reduced only through concerted attack on the problem from all quarters, an attack now also having the legal force of statutory regulations behind it.

43. As indicated above, there are two levels of Noise Control Criteria required for the control of machinery and plant noise where operators and other employees are present. The higher of these two levels are determined by considerations of occupational health, and depend on the statutory limits set to minimize noise-induced hearing damage. The lower of these two limits are determined by considerations of occupational safety, and so depend mostly on the need for safety requirements depending on speech intelligibility. These latter, though the harder to achieve, are none the less necessary.

44. Rail, road, sea and air transport vehicles, though their noise has environmental implications, are, when considered singly, a part of this machinery and plant group of engineering equipment. As with other machinery and plant, so also with transport vehicles is it necessary to design them to operate quietly. The Australian Standards listed in Appendix B also include several which describe methods of determining the noise emitted by railbound (AS 2377) and road transport vehicles (AS 2240, AS 3713), by earth-moving machinery and agricultural tractors (AS 2012), and by sea-going vessels and platforms (AS 1948, AS 1949 and AS 2254). ISO 3891-1978, 'Procedure for describing aircraft noise heard on the ground' provides a standard method for measuring some aspects of aircraft noise. While the Australian government Australian Design Rules (ADR) nos. 28, 28A and 28B for motor vehicle noise, that is, for the noise emitted by individual vehicles, have provided Noise Control Criteria for limiting their noise emitted as measured according to AS 2240-1979, and AS 2254 furnish some recommendations for limiting the noise in various areas of occupancy in sea-going vessels and offshore platforms, it has been left so far to purchasers of other types of vehicle to set appropriate Noise Control Criteria in their purchase specifications and contracts. Such criteria as are current are very likely to be influenced by existing environmental noise criteria.

Environmental Noise Control Criteria

45. As Noise Control Criteria for machinery, plant and individual transport vehicles are influenced by both hearing damage and speech intelligibility considerations, environmental noise criteria are influenced by speech intelligibility considerations, and by considerations associated with promoting general human and community health and welfare. In this connexion, environmental noise includes the acoustic environments around our homes, schools, our various places of community activity, and near transport ways (railways, main roads and airports) bordering residential areas. Noise problems might thus arise from community, commercial, industrial or transport activities and operations.

46. The progress of industrilization in our communities and nations over the last 200 years or more has been accompanied by a continuing increase in the amounts of noise generated by commercial, construction and industrial

operations, and by transport vehicles. Whereas until about the time of World War II this increasing noise was largely accepted as an inevitable, though unfortunate part of our industrial 'progress', in more recent years we have begun in our communities to take a firmer line and to protest more vigorously that this cacophony of noise is a source of serious annoyance, and is therefore no longer necessary and must be reduced.

47. An early method of rating the degree of annoyance to noises experienced by communities in residential environments was that developed in the early 1950s by W A Rosenblith and K N Stevens, and also worked on by H O Parrack (ref 26, ch 36). A description of the method was included in the third (1956) edition of the General Radio (GR) Handbook of Noise Measurement, this being a significant addition to the contents of the previous (1953) edition, which then continued to be included through the fourth (1961) and fifth (1963) editions of this Handbook, after which (from 1967) it was no more than briefly referred to. In principle, the rating of a noise according to this method depended primarily on its absolute octave band levels measured at the point of complaint. These measured octave band levels were assigned a Level Rank -- developed from a chart (analogous to an AS 1469 NR no. chart) with 14 Level Ranks each from 4 to 6 dB wide (along the ordinate) extending from below 30 dB (850 Hz band) to above 93 dB -- which was then modified by means of 'Correction Numbers' (better and more accurately to be described as 'Adjustment Numbers' (see para 51 below)) to allow for differences in noise spectrum character, impulsiveness, rate of repetition, background noise level, time of day, and conditioning to exposure, to obtain the Composite Noise Rating for use with a second chart relating this rating to an estimate of expected community response advancing in five steps from 'No observed reaction' through 'sporadic' and 'widespread complaints' to 'threats of' and 'vigorous community action'. The environmental noise rating method described here was intended mainly as a guide. It was therefore expected that, as more experience was gained in using it, some revision of its numerical values would be found desirable (ref 35, pp 67-69).

48. In fact, more than revision of its numerical values has occurred. Other methods have been developed which are based on different fundamental assumptions. One quite different approach was adopted by the English committee set up under Command 2056 in April 1960 "to examine the nature, sources and effects of the problem of noise and to advise what further measures can be taken to mitigate it", and which produced its 'Noise: Final Report' (ref 37) in 1963. Typical of the type of criteria which this committee produced are the Subjective Assessments of Motor Vehicle and Aircraft Passby Noise given in TABLE 7, and the Criteria for Noise within Buildings given in TABLE 8. Of interest in this latter TABLE are that the maximum permissible sound levels are in terms of statistical L_{10} (= 'near maximum') levels, and that the night time levels for bedrooms are similar to those of AS 2107. With the assessments of vehicle passby noise of TABLE 7 it can be concluded that, although they are not legally enforceable as there given and stated, they indicate that it would desirable for these vehicle noises (as measured under the stated conditions) to be 'quiet', or at worst, no more than 'moderately noisy'. It is probable that some of the results of the work of this Wilson committee found its way into BS 4142:1967, 'Method of rating industrial noise affecting mixed residential and industrial areas'.

49. A method having some resemblances to the US method of obtaining Composite Noise Ratings for noise intruding into residential areas, in order to estimate the expected community response to them, and which was also based on BS 4142: 1967, was developed for inclusion in the first (1973) edition of

AS 1055, 'Noise assessment in residential areas' (ref 3). But, whereas in the US Composite Noise Rating method the Rating was developed from the absolute octave band levels of the intrusive noise as modified by several penalty 'Adjustments' (there called 'Corrections') depending on the character of the noise and its background, the basic noise assessment principle adopted in AS 1055-1973 used the relative differences in A-weighted sound levels of the intrusive noise and its background, both levels being modified by 'Adjustments' depending on their character. In the words of the Standard, "the method of assessing noise annoyance described in this Australian Standard Code of Practice is based on a comparison of the measured noise level" (of the intrusive noise) "with an acceptable noise level for the particular circumstances being investigated". But whereas in AS 1055-1973 "this code recommended rules for measuring and assessing noise at residential sites", later editions have clarified this further that the code "applies primarily to noise emitted from industrial, commercial and residential premises".

50. In making these comparisons, the two basic noise levels are the 'Adjusted Noise Level' of the intrusive noise (which depends on its actual measured level, then modified by 'Adjustments' in accordance with its particular waveform, frequency spectrum and duration characteristics), and the 'Acceptable Noise Level' (which may be derived from a measured Ambient Noise Level, or calculated from a Base Level of 40 dB(A), then modified by 'Adjustments' in accordance with the time of day, and the type of suburban area and transportation density of the location in question). In this method both forms of 'Acceptable Noise Level' are to be obtained (provided the ambient level can be measured when the intrusive noise is absent), the lower of the two being the adopted 'Acceptable Noise Level'. If the 'Adjusted Noise Level' exceeds the corresponding 'Acceptable Noise Level', the intrusive noise is likely to be annoying, with differences of 5 dB(A) or less likely to be of marginal significance, and differences of 10 dB(A) or more likely to be the source of sporadic or widespread complaint or of even stronger public reaction for larger differences (as suggested by AS 1055-1973, Appendix E).

51. The 'Adjustments' of AS 1055-1973 in Table 1 and Figures 1 and 2 which are used for modifying the Measured Level of the intrusive noise to obtain its 'Adjusted Noise Level', and the Ambient Noise and Base Levels to obtain the 'Acceptable Noise Level' with which the 'Adjusted Noise Level' is compared, are of the nature of estimated penalties derived from psycho-acoustic tests, by which the 'Adjusted Noise Level' is then expected to give a better measure of the annoying quality of the intrusive noise, because such effects of increased annoyance cannot be directly measured with a Sound Level Meter. It is therefore both inaccurate and wrong to refer to these 'Adjustments' as 'Corrections'. We in fact meet here a rather loose use of the term 'correction', which in mathematics and the applied sciences has the precise meaning of "the amount by which a number, quantity or instrument measurement or reading which contains a known and identifiable error has to be changed to make it true". Thus, these modifications and adjustments to a measured value cannot be 'corrections' in the sense that the original measurement was incorrect and therefore needed to be corrected because of a known and identifiable instrument or meter error; they cannot be 'corrections' in the further sense that there is no precise criterion or standard by which they can be judged less or more correct than the original measurement; and to call the 'adjusted' values 'corrected' values gives what are in fact estimates based on not very precise human responses and judgments an air of authority which they do not and cannot possess. The compilers of Australian Standards and similar documents have acted wisely in referring to these modifications as 'Adjustments'.

52. · Several revisions of AS 1055-1973 have since been made: in 1978, to clarify difficulties experienced by users of the 1973 Standard, and in 1984 and 1989 to take account of the development of the more complex Sound Level Meters now available for evaluating environmental noise problems (especially those meters furnishing a statistical analysis of the varying sound levels measured over a specific duration). The procedure of modifying measured sound levels by means of 'adjustments' has been further clarified. "The measurements described in this Standard are designed to give a reliable physical description of the environmental noise. For assessment of human reactions to noise, it is sometimes necessary to make adjustments to the measured levels in order to arrive at a more meaningful basis for the assessment" (AS 1055.1-1989). Of the three parts of the current (1989) edition of AS 1055, parts 1 and 2 generally replace and amplify the 1973 and 1978 editions and parts 1 and 2 of the 1984 edition; AS 1055.3-1984 and AS 1055.3-1989 have extended the coverage of the use of this Standard to a more comprehensive relating of land uses with existing and possible future environmental noise levels.

53. In Victoria, the Environment Protection Authority (EPA) adopted and adapted the noise assessment procedures of AS 1055-1973 and AS 1055-1978, and issued its own statutory regulations for the control of noise emanating from commercial, industrial or trade premises within the Melbourne metropolitan area, in 1978 as Draft Environment Protection Policy no. 59/78, in 1981 as State Environment Protection Policy no. N-1:1981, and in 1987 as a revision of N-1 (ref 21). This policy is currently undergoing further revision. With this method the intrusive noise has its sound level (in dB(A)) measured and adjusted to give its 'Effective Noise Level' for comparison with the 'Zoning Permissible Noise Level' for the area in which the intrusive noise is being complained of. Noise Levels are to be measured and expressed as Equivalent Continuous Sound Levels; adjustments provide for indoor measurement, tonal components, impulsive type noise, and the duration and intermittent character of the intrusive noise. The presence of tonal components in the noise require to be confirmed by one-third octave frequency analysis; the presence of a tonal component is confirmed if the level (A-weighted in this Policy) in the one-third octave band suspected to contain the tonal component exceeds the arithmetic mean of the two adjacent one-third octave band levels by over 3 dB. Zoning Permissible Noise Levels vary with the time of day and the distribution of heavy and light industrial, commercial and residential land uses within a 200 m radius of the measurement point in the Noise Sensitive Area. Complainants of intrusive noise in the vicinity of their homes thus have a means of having the excessive noise reduced if its Effective Noise Level exceeds the corresponding Zoning Permissible Noise Level. Such revisions as have been made to this EPA Policy have been basically to improve the workability of the noise assessment method.

54. In addition to these EPA Noise Control Criteria for the control of noise emanating from commercial, industrial or trade premises, the EPA and other government authorities concerned with environmental noise arising from transport operations and activities such as the movement of trains, and the flow of motor traffic on expressways and other heavily trafficked roadways in built-up urban areas, have established several Noise Control Criteria for controlling the levels of rail and motor traffic noise transmitted to neighboring residential areas. For rail traffic there is a maximum permissible 24-hour Equivalent Continuous Sound Level of 65 dB(A), while for motor traffic there is a maximum permissible value of the 18-hour L_{10} (= near maximum) noise level (from statistical analysis) which, formerly at 68 dB(A), has been recently reduced to 63 dB(A). Both of these criteria, for the noise measure-

ment out-of-doors, are available in case of complaint about railway or motor traffic noise penetrating into residential areas, though as yet guidelines.

55. The Noise Control Criteria described here -- based on Loudness, Speech Intelligibility, Hearing Damage Risk, or more general health and welfare considerations, with some included in statutory regulations, and others in Standards -- though not completely comprehensive, provide many useful noise criteria for two basic groups of circumstances: for confirming the existence of a noise problem and establishing the responsibility of the noise-maker, and for use at the design stage of buildings, equipment, machinery, fixed and mobile plant, and transport vehicles, so that through taking proper account of relevant acoustical considerations, new noise problems are not created, problems which, if once perpetrated, are costly to remedy. In this country, the Standards Association of Australia has amply supplied our need for Noise Control Criteria to help deal with all these noise problem and design situations.

Vibration Control Criteria

56. Just as excessive noise can cause damage to human hearing and interfere with speech intelligibility, so also can excessive vibration cause damage to the human body or building structures, and interfere with the proper operation of scientific and other equipment. As a result of experience gained with these types of problem, a number of Vibration Control Criteria have been established. From the point of view of the human body, it is most sensitive to vibration around 5 Hz, the mean threshold of perception being 0,02 m/s^2 rms, with vibration at this frequency above 0,28 m/s^2 being 'unpleasant' and above about 2,0 m/s^2 rms being 'intolerable' (ref 27, fig 44.20). Another field of the study of human susceptibility to vibration is that of 'motion sickness', which occurs at frequencies around 0,5 to 1,0 Hz, and has particular application to quality of vehicle ride. In connexion with human susceptibility to vibration, AS 2763-1988, 'Hand-transmitted vibration' provides 'Guidelines for measurement and assessment of human exposure' (ref 10b).

57. From the point of view of damage to buildings from ground-borne vibration, while some codes have allowed a vibration velocity as high as 50 mm/s pk before it is considered that any building damage will result, a more conservative and safer code is that of DIN Standard no. 4150:Part 3:1977 (ref 20) in which, with the rms magnitude of the ground velocity vector below 2,5 mm/s, it is stated that "structural damage is not possible", between 2,5 and 6,0 mm/s damage is "very unlikely", between 6,0 and 10,0 mm/s damage is "unlikely", and above 10,0 mm/s "damage is possible and the structure should be investigated". Other Vibration Control Criteria are determined by the particular conditions of operation and susceptibilities of the equipment, instrument or machine in question. For example, with the condition monitoring of machines, the criteria are determined by the performance of the machine when and in optimum working condition. Chainsaws have safety requirements for their use, including some concerned with vibration and noise, specified in AS 2726-1984, 'Chainsaws -- safety requirements' (ref 10a). Ultra-sensitive measuring instruments have their particular vibration criteria. Electron microscopes, for example, cannot be expected to give satisfactory performance at their highest magnifications if there is any vibration present below 5 Hz, and above 5 Hz if the peak-to-peak vibration displacement exceeds 3 micrometre. Such Vibration Control Criteria provide a proper guide to the type and scope of vibration measurement required when a problem is suspected.

THE APPLICATION OF NOISE AND VIBRATION CONTROL CRITERIA TO SPECIFIC PROBLEMS

58. Noise problems are many and varied, as is illustrated by the Brüel and Kjaer booklet, 'Noise Control: Principles and Practice' (ref 19), which contains a description of 49 noise problems and their solutions, together with notes on the factors which have influenced the generation of the noise and its propagation in materials, structures and rooms. However, these problems all begin with the assumed and confirmed existence of the problem. In many cases it is first of all necessary to actually confirm and conclusively establish the existence of the noise problem and the responsibility of the noise-makers before it can be agreed that the problem must be remedied. It is at this point that agreed and accepted Noise Control Criteria become necessary -- either criteria such as statutory regulations, which are legally enforceable, or criteria having agreed authority as are to be found in Standards. The following case histories illustrate the need for, and use of, some of these Noise Control Criteria.

59. However, before these case histories are discussed in detail, it is necessary here to discuss one further general aspect of identifying and solving noise problems. With noise measurements which indicate that noise levels at a particular location exceed a relevant noise control criterion, and that, as a result, some noise reducing measure must be undertaken to reduce the noise by a specified amount by either reducing the noise at its source, or reducing the noise transmitted between source and hearer, there is clear indication of what is required. In other cases where noise reduction is necessary because the noise is annoying, but where there is no precise indication of the amount of sound level reduction required, it is helpful to consider the decibel sound level scale in relation to the phon Loudness Level and sone Loudness scales (that is, in relation to the Loudness Function). With this Function, a two-fold change in Loudness corresponds to an interval of 10 phons, and therefore, because Loudness Levels in phons and A-weighted sound levels are reasonably closely related, to an interval of approximately 10 dB(A). Thus, because a two-fold change in Loudness is clearly discernible, a noise reduction of 10 dB(A) is significant and worthwhile. Smaller reductions in sound level are therefore less significant, and below about 4 dB(A) not particularly worthwhile unless as part of the noise reduction procedure the frequency character of the noise is significantly changed, such as by greatly reducing its high frequency content, or reducing or eliminating tonal components, but without greatly changing the overall sound level. Case history no. 7, for example, illustrates this, where a remedial measure removed several annoying tonal components from a noise, but without significantly changing its overall sound level.

Speech intelligibility in meeting halls

60. Case history no. 1 concerns the problem which faced the users of a public meeting hall situated on an arterial road also carrying a tram route, who, when the tram track, previously of paved ballast construction, was reconstructed in concrete, found that the noise of passing trams had increased somewhat and was proving to be an increased interference to the speech intelligibility of their meetings. This type of problem -- which then (mid 1950s) arose because of the high incidence of noise from tram wheel flats, but which nowadays no longer occurs through the conversion from cast iron to composition brake shoes in older type trams, and the use of resilient instead of solid wheels in the newer all-electric types -- is of interest still because, owing to a lack, then, of suitable Noise Control Criteria, and to the older method of measuring low sound levels in dB(A), medium in dB(B) and high in

dB(C)(which actually inhibited their direct comparability), the problem was not satisfactorily solved. However, the noise measurement data obtained at the time (Appendix C below) can, by means of the now added corresponding sound levels in dB(A) and speech interference levels (SIL4) estimated from the octave band spectra of similar noises, be re-examined and the noise levels therefore more readily compared with each other in the light of the appropriate Noise Control Criteria from AS 2107 (see TABLE 4) or TABLE 3. Several measurements and calculations were also recently made to obtain estimates of the variation in vehicle noise levels during a passby period of ± 12 s for vehicles at a minimum distance of 18 m from the noise measurement location (Appendix D).

61. With the noise measurements given in Appendix C, including the estimated SIL4 and sound levels in dB(A), a re-examination of them shows, firstly, that there was, on average, an increase in tram passby noise of 4 dB(A) after reconstruction of the track. With a Noise Control Criterion for this type of building interior of 30 to 35 dB(A)(TABLE 4, for Assembly Halls up to 250 seats, and Conference Rooms) or a maximum permissible SIL4 of 28 dB (TABLE 3), this re-examination shows secondly that, with a background noise level of 54 dB(A) or estimated SIL4 of 46 dB in the meeting hall, its acoustic environment was rather more noisy than desirable, by of the order of 20 dB, even without the additional noise of any passing vehicles, let alone the noisier heavier vehicles. However, with maximum passby SIL4s of 55 to 59 dB for groups of motorcars, 63 dB for individual larger vehicles and trams on the then new concrete track, and an estimate of 59 dB for trams on the previous paved ballast track, the Maximum Permissible SIL4s for Reliable Speech Communication (TABLE 2) indicate thirdly that, with 'very loud' (or 'stage') voice level, reliable speech communication would still be possible at up to 5,5 m for an SIL4 of 55 dB (minimum for motorcars), up to 3,5 m for an SIL4 of 59 dB (maximum for motorcars, mean for trams on former paved ballast track), but reduced to up to only 2,2 m for an SIL4 of 63 dB (mean for larger vehicles, and trams on the reconstructed concrete track). Thus, the passby noise of any vehicle in the street outside would have caused a temporarily but significantly reduced speech intelligibility.

62. If, in this situation, the SIL4 of 55 dB (minimum for groups of motorcars) is assumed to be what might be reasonably considered as still just tolerable, then any noise with SIL4 greater than 55 dB begins to be intolerable, the degree of intolerability depending on the duration of its SIL4 above 55 dB. With motorcars and trucks at speeds of around 40 to 50 km/h, the passby SIL4 of 59 dB could be expected to exceed the tolerance SIL4 of 55 dB for about 4,4 s, while the passby SIL4 of 63 dB might exceed it for up to about 9 s, or an extra 4,6 s. With trams at the then normal average speed of around 30 km/h, the corresponding mean passby SIL4 of 59 dB (paved ballast track) could be expected to exceed the tolerance SIL4 of 55 dB for about 7,5 s, while the mean passby SIL4 of 63 dB (concrete track) could be expected to exceed it for up to 16,5 s, or an additional 9 s over the noise of trams under the former track condition. Such considerations as this, even though taken with the help of estimated values, help put this whole noise problem into clearer perspective, and indicate where the serious problems were. As stated above, this serious problem of noise from tram wheel flats no longer exists. A further conclusion is that, with the benefit of hindsight, we would now take frequency analyses (at least octave band) of all the noises directly involved with such a problem.

Noise transmitted in ducts

63. Some noise problems arise as a result of otherwise well-intentioned

action, but action taken without thought of acoustical considerations. Case history no. 2 provides an example of this. At a vehicle and plant maintenance depot, the mechanical workshop was to be provided with a new stationary reciprocating air compressor. Because of its exterior location right next to the workshop, the compressor was to be enclosed, with adequate provision, also, for it to be air cooled. Thus, for winter weather there would be a good supply of heated air available for warming the workshop interior. Appropriate ducting was therefore provided for the heated air from inside the compressor casing to be either diverted to an exterior outlet or, in cold weather, to be fed into the workshop. After installation and commissioning of the compressor and its ducting, the noise levels in the workshop were considered intolerable, with the heated air either diverted outside or fed into the workshop. Also, because the wall opening carrying the duct had not been then completely sealed, some compressor noise was expected to be entering the workshop by this path.

64. Noise levels inside the workshop were measured under both conditions of heated air diversion. The main measurements consisted in taking octave band analyses of the workshop interior noise to obtain A-weighted sound levels and speech interference levels (SIL4). With average levels in the workshop of 87 dB(A) and 80 dB (SIL4) for warm air 'on', 76 dB(A) and 68 dB (SIL4) for warm air 'diverted', and 69 dB(A) and 60 dB (SIL4) for warm air 'diverted' plus a 10 mm thick timber board placed over the air inlet grill, and corresponding levels generally 4 dB higher at a point close to, and 1 m out from the air inlet grill, noise from the compressor through the warm air duct was considered highly objectionable, especially as, when judged by the SIL4s of 60 to 84 dB, the noise was sufficiently loud to seriously interfere with reliable speech communication (see TABLE 2). Thus, from the point of view of speech intelligibility in this workshop area, the SIL4 of 80 dB with the warm air 'on' meant that reliable communication would be possible only up to distances between persons of 0,63 m and by shouting. With the warm air and the lower SIL4 of 68 dB, conditions for reliable speech communication were somewhat improved, with shouting up to distances of 2,5 m between persons, very loud voice up to 1,2 m, or raised voice up to 0,6 m. With the additional board over the air inlet grill and the SIL4 in the workshop further reduced to 60 dB, reliable communication would be possible with very loud voice up to 3,2 m, raised voice up to 1,6 m, or normal voice up to 0,8 m.

65. The existence of a serious noise problem was confirmed by these high SILs, and decisively established by the relevant Noise Control Criteria of AS 2107, in which the highest recommended design sound levels inside industrial buildings are 50 dB(A) as 'satisfactory' and 70 dB(A) as 'maximum', so that for satisfactory use of the warm air duct a noise reduction of at least 17 dB(A) was required. Consideration of the problem as a whole indicated that acoustic lining of the existing warm air outlet duct from the compressor case was a necessary first step. This duct was a straight duct about 2,5 m long with a 90 degree bend at the compressor case exit. Octave band analysis of the noise measured in the mechanical workshop near the air inlet grill, showed that, at 91 dB(A), octave band levels of 85 dB or more occurred in the 125 to 1000 Hz bands, indicating that any duct lining material to be seriously considered required an as high as possible acoustic absorption in this frequency zone. An available material with acoustic absorption coefficients of 0,25 (125 Hz), 0,50 (250 Hz) to 0,95 (2000 Hz) was therefore selected as suitable, with these coefficients as the basis of the noise reduction design.

66. Because calculations of the noise reduction possible through lining of the existing duct with this material gave a calculated noise reduction of

8 dB(A), some form of extended duct having an additional minimum reduction of 10 dB(A) was considered necessary and therefore designed. The additional ducting, to have a noise reduction of 10 dB(A), was designed in the form of a simple rectangular lined plenum chamber with, to minimize any direct propagation of higher frequency noise, the inlet and outlet at opposite ends and on opposite sides of the plenum chamber. The appropriate design formulas for a lined plenum chamber were obtained from Beranek's 'Noise Reduction', in the chapter on dissipative mufflers (ref 18, ch 17). Noise measurements made after the acoustic lining of the existing duct, installation of the lined plenum chamber, and sealing of the exterior brick wall around the air duct, showed the installation to have satisfactorily achieved its aim of preventing the noise generated within the compressor case from being transmitted to the workshop. Overall reductions in workshop interior noise level and SIL4 near the air inlet were from 91 down to 57 dB(A), and 84 down to 43 dB in SIL4 with warm air 'on', and from 81 down to 54 dB(A), and from 72 down to 43 dB in SIL4 with warm air 'diverted', the actual noise reductions effected through the lined ducts and sealing of the brick wall around the duct being, respectively, 34 and 41 dB(A). Thus, the acoustic environment in the workshop had been brought within the design criteria of AS 2107. The corresponding reductions in SIL4 of 41 and 29 dB also showed significant improvement in the conditions for speech intelligibility, with reliable communication using normal voice now possible up to distances between persons of 5,6 m. There were several ways of commenting on this noise problem: firstly, that the noise in the workshop was very loud and annoying; secondly, that the general noise levels exceeded the AS 2107 maximum recommendations; or thirdly, that the noise emanating from the duct was so intense as to make reliable speech communication extremely difficult, as just described by the SIL4 measurements. Of these, the last is the most convincing because the speech intelligibility Noise Control Criterion is both precise and reasonable.

Noise Control Criteria for lunch rooms

67. Case history no. 3 concerns two lunch rooms, a small one of volume of around 200 m^3, the other a larger one of volume around 1000 m^3, both of which gave rise to complaints of excessive noise resulting from significant reverberation. Typical of the noise problem of the small room was that any noise, as of a chair being moved across the floor, was greatly magnified and often aurally painful. In the larger room, conversation became difficult at peak times, with everybody needing eventually to shout at their neighbors in order to be understood. The reverberation times in each room were measured (according to the method of AS 2460-1981, 'Method for the measurement of reverberation time in enclosures') and found to be, respectively, 2,2 and 2,6 s. By means of the formula

$$T_r = \frac{0{,}161\ V}{\alpha . S}$$

where T_r = the reverberation time in s,
V = room volume in m^3,
S = room surface area in m^2,
α = mean coefficient of acoustic absorption of these surfaces,
$\alpha . S$ = sum of the acoustic absorptions of all room surfaces,

the required amounts of acoustic absorptive material of known characteristics were calculated, to reduce the reverberation times of these rooms to a desirable value of the order of 0,5 s.

68. After the calculated treatments had been effected, the reverberation times were again measured. In the small room, with its reverberation time reduced from 2,2 to 0,5 s, the acoustic environment was greatly improved, with noise, for example, from a chair being moved over the floor being heard as only a slight scraping, and no longer aurally painful. In the larger room, with its reverberation time reduced from 2,6 to 1,0 s, conversation at peak times was found to be much more comfortable, with little or no need for shouting. Though the details of the noise measurements and results are not described here, reverberation times were measured using octave band analyses.

69. One feature of interest with these lunch room problems is that AS 2107-1987, in its recommended reverberation times for building interiors, does not provide recommendations for, for example, the interiors of industrial buildings (including factory and manufacturing areas and lunch rooms), and public areas such as post offices and general banking areas (see TABLE 4). In all of these (and other) areas -- which, if they are too reverberant, can become unnecessarily and uncomfortably noisy, and hence difficult for reliable speech communication -- their acoustic environments are much more satisfactory with minimum reverberation. Recommended design reverberation times of the order of 0,5 to a maximum of around 1,0 s, depending on room volume, appear to be satisfactory limiting amounts.

<u>Auxiliary plant in lecture rooms</u>

70. Although, as just discussed, AS 2107 does not include all the recommended design sound levels and reverberation times for building interiors which we might need, the design criteria which it does provide make it a most useful Australian Standard. <u>Case history no. 4</u> illustrates a further use for its Noise Control Criteria in a case which arose in the first place from a complaint about a noisy air conditioner in a lecture room having about 60 seats. As a result, the various aspects of the acoustic environment of this room were examined.

71. The lecture room, which had a unit air conditioner installed near a back corner, was also situated near a moderately trafficked main road, and next to a lane along which heavy vehicles were regularly but not frequently driven (the frequency being of the order of one per hour). The actual noise complaint was that the air conditioner had usually to be turned off if those at the back of the room were to satisfactorily hear the lecturer; there had been no specific complaints about other noise. Ambient noise levels in the lecture room while unoccupied were measured with the air conditioner off, then on, at a location 3 m out from the front of the air conditioner, 1,2 m above floor level, as in the air conditioner purchase specification. According to this specification the maximum noise at this location from the air conditioner while operating was to be 40 NR.

72. Noise measurements made at the location described with the air conditioner at maximum operation showed its noise output to be significantly greater than specified, and equal to 51 NR (at 250 and 500 Hz) or 55 dB(A), with the corresponding SIL4 being 44 dB, enabling satisfactory speech intelligibility with 'raised' voice at distances between persons of up to only 10 m, or with 'very loud' (or 'stage') voice up to 20 m. Thus, the air conditioner, which did not comply with its noise purchase specification by + 11 NR, was somewhat of a noise problem when judged by SIL4 criteria (TABLE 2), and even more of a problem when judged by the AS 2107 criterion of 30 dB(A)(satisfactory) to 35 dB(A)(maximum) for assembly halls with up to 250 seats and conference rooms, or by the even more exacting AS 2107 criterion of 25 to 30 NR for a conference room (without speech reinforcement) for 50 to 250 persons

(TABLE 4), the noise levels near the air conditioner operating being 55 dB(A) and 51 NR. With the air conditioner off, minimum ambient noise levels were 44 dB(A), 40 NR and an SIL4 of 30 dB, all of these still being above the AS 2107 recommendations. These ambient levels rose slightly to 47 dB(A), 43 NR and 35 dB (SIL4) with the added exterior noise of birds twittering coming through a roof vent, and to 47 dB(A), 45 NR and 30 dB (SIL4) for bursts of exterior traffic noise. The existence of the AS 2107 and AS 2822 (speech intelligibility, TABLE 2) Noise Control Criteria thus raised the necessity for a serious re-consideration of the room in question as a lecture room, quite apart from the air conditioner noise problem. This latter problem, also raised in acute form the need for much tighter control of their maximum noise emissions, and for manufacturers to acoustically improve their designs.

Rooms for audiometry

73. Most uses of rooms and spaces inside buildings require to be covered by Noise Control Criteria if the required use of the interior space is to be satisfactory. For rooms to be used for audiometric testing, the 'Maximum Acceptable Background Noise Levels' are specified in AS 1269, 'Hearing Conservation' (refs 5, 6). AS 1269 also includes the requirement that "to achieve the background noise levels it will be necessary in many circumstances to use a sound-isolating booth". This Standard has given several groups of acceptable background noise levels, depending on the type of audiometric earphone and cushion or enclosure combination, the most exacting in, for example, AS 1269-1983 being, for octave band levels at frequencies from 125 through 8000 Hz, 52, 35, 15, 14, 29, 36 and 28 dB (see Appendix E). Case history no. 5 gives examples of several rooms tested for this purpose, the corresponding measured levels being as in Appendix E. These measured octave band levels, typical of 'quieter' rooms not specially treated for this purpose, show that Room A, a usually quiet room, would have been satisfactory for audiometric testing (without a sound-isolating booth) only with its air conditioner and the nearby exterior plant and equipment not operating. Room B, a medical consulting room, could only be suitable when it was equipped with a suitable sound-isolating booth, the remaining levels indicating that such booths themselves need to be well constructed and fitted with specially quietened fans for them to comply with the most stringent requirements of AS 1269 (in the 500 and 1000 Hz octave bands), the measurements here showing that success was not achieved until the fourth attempt. Room C, another 'quiet' room, again shows the difficulty of complying with the requirements of AS 1269, especially in the 500 and 1000 Hz octave bands.

The tolerability of factory interior noise

74. In AS 2107, the recommended design sound levels for such operations as assembly lines in industrial buildings are 50 dB(A) as satisfactory and 70 dB(A) as a maximum (see TABLE 4). Case history no. 6 describes an experience which indicates that the provisions of this Standard are realistic rather than idealistic or conservative. In such a factory assembly line it became necessary to instal a dust collection system, the assembly line itself being housed in a large building of floor area of the order of 25 by 20 m, covered by a saw-tooth roof about 15 to 17 m above the floor. As initially installed just under the roof, the dust collector fan exhausted its air directly into the factory area to provide a continuing movement of air. The resulting air turbulence at the fan outlet generated a general noise level inside the building of 90 dB(A), an intolerable noise under continuous conditions, and 20 dB(A) above the AS 2107 recommended maximum of 70 dB(A). The fitting of an

outlet muffler reduced the general level of this loud background noise from 90 to 78 dB(A), the lower frequency spectrum being close to 70 NR (see Appendix F). Because this somewhat reduced level of background noise could be described as 'just tolerable', it was concluded that 70 NR represents the very maximum tolerable level for continuous background noise. This noise spectrum, with an overall level of 77 dB(A), is noisier than the AS 2107 recommended maximum of 70 dB(A), the 7 dB decrease representing a 35 to 40 percent reduction in loudness. The noise level of 77 dB(A) also represents a level just above the 80 dB(B)/74 dB(A) considered relatively safe for everyday continuous exposure. The final outcome of this noise problem was that, as a further alteration, the fan exhaust was directed outside the building, with a significant reduction in the interior background noise.

Commercial premises' air conditioning plant near dwellings

75. Numerous noise complaints arise as a result of plant and machinery noise emitted by commercial and industrial premises being transmitted to adjacent residential areas. State Environment Protection Policy no. N-1: 1987 is often invoked in such cases. Case history no. 7 concerns a situation in which the noise from three exterior fans as part of a commercial building's air-conditioning plant was sufficiently annoying to the occupiers of an adjacent dwelling that a complaint was registered. With fan noise at the building's property line giving rise to an Effective Noise Level of 59 dB(A) in comparison with Day, Evening and Night Zoning Permissible Levels, respectively, of 54, 48 and 43 dB(A) as calculated according to Policy no. N-1, there was need for up to 16 dB(A) noise reduction to achieve complete compliance with the statutory policy. Because the intrusive noise included tonal components in the 63 and 125 Hz one-third octave bands, the levels of 43 and 49 dB(A) were adjusted to 49 and 54 dB(A) to obtain the Effective Noise Level of 59 dB(A), which otherwise would have been 57 dB(A) without tonal adjustment.

76. Several frequency analyses of the fan noise were undertaken -- octave, one-third octave, and narrow band -- to obtain detailed information about the frequency composition of the noise which, though vaguely tonal, was not obviously so (see details in Appendix I). While the octave band analysis showed the highest band levels to be in the 63 and 125 Hz bands, the one-third octave band analysis confirmed this through the 63 and 125 Hz bands appearing as low frequency 'tonal' bands with their levels exceeding those of the means of their adjacent bands by 14 and 10 dB, each with an excess greater than 3 dB, so confirming the presence of tonal components in the fan noise. The narrow band analysis (with 1 to 2 Hz bandwidth) showed prominent noise components at around 60, 120, 180, 245, 305, 362, 426 and 487 Hz: components which appeared directly related to the fan speed (16 r/s) and the number of fan blades (4). With the existing fans having metal-bladed impellers, a noise reduction plan of replacing these with plastics-bladed impellers, reducing fan speed, and increasing the height of the wall around them to increase their acoustic barrier function, was considered and put into effect.

77. Following the substitution of plastics-bladed impellers for the original metal fans, advice was received that there was no further complaint. Frequency analyses of the plastics-bladed fan noise showed that, although there had been no actual overall sound level reduction, the octave band levels in the 63 and 125 Hz bands had decreased by 19 and 4 dB, and the one-third octave band levels at these same frequencies had decreased by 23 and 7 dB. In addition, the one-third octave band analysis showed that there were no longer 'tonal' bands at these frequencies. Because those making these measurements did not request access to the complainant's dwelling to confirm

that the noise nuisance was due to the sound from the metal-bladed fans exciting a room resonance at about 60 or 120 Hz, this could only be surmised to have been the real nature of the problem, particularly because the change from metal-bladed to plastics-bladed fan impellers virtually eliminated these prominent tonal components from the fan noise, as shown by the frequency spectra in Appendix I. Use of the formula for calculating the natural frequencies of a room of dimensions l, w and h (in m, and as quoted in ref 26, ch 29, p 21),

$$f = \frac{c}{2} \cdot \sqrt{\frac{A^2}{l^2} + \frac{B^2}{w^2} + \frac{C^2}{h^2}}$$

where A, B and C are positive integers, c the velocity of sound in air (in m/s), and f (in Hz) represents a series of frequencies depending on the values given to A, B and C.

A room of dimensions 5 x 4 x 2,7 m would thus have its first ten natural frequencies at 84, 103, 112, 127, 129, 139, 148, 149, 151 and 158 Hz; while to have its lowest natural frequency at 60 Hz, a room would need to be at least as large as 8 x 6 x 3,6 m. In this particular case, the acoustical technical information available on plastics-bladed fan impellers, though it suggested an overall noise reduction of 5 dB when compared with equivalent metal-bladed impellers, was insufficiently detailed to indicate the larger noise reductions possible (such as the 23 and 7 dB measured at 63 and 125 Hz in this problem) at resonance frequencies of the blades of equivalent metal fans.

Rotating machinery mountings

78. The matter of resonances giving rise to noise (or vibration) problems is not one which is or even can be readily covered by existing or possible Noise Control Criteria. It is, however, a matter of great importance in design situations, whether of rooms or enclosures, or of rotating machinery and of machinery and other mountings. Case history no. 8 concerns such a situation in which a mass-produced assembly containing an electric motor mounted on a panel was too frequently found to be unduly noisy. This problem, though obviously making its presence felt as a noise problem, was as much a problem of vibration causing the excessive noise, and shows once again the pressing need for noise and vibration considerations to be fully taken into account at the design stage of any building, equipment, plant or machinery. In this case, the noise problem resulted from the near concurrence of several resonances. Firstly, vibration tests on a sample of typical motors showed that their designed operating speed was just below a critical shaft speed, which therefore generated greater vibration than at other operating speeds, and made the problem of any slack in the shaft bearings much more critical than it would be at speeds away from from a critical motor speed. Secondly, the problem of mounting such an excessively vibrating motor would become more critical. Thirdly, the problem of mounting such a motor on a panel would be further compounded if the motor speed corresponded closely to a panel resonance. The series of vibration tests on a sample of motors and panels confirmed that all these difficulties were in fact being encountered, the panel resonances having been also identified by noise test.

79. With this problem, the suggested remedy included several steps, the first of which was to assume that it was redesign of the motor and its operating speed, rather than of the mounting panel, which was the more feasible. Thus, the primary requirements were redesign of the motor operating conditions to a speed between major mounting panel resonances; redesign of the motor shaft (possibly by a slight increase in its diameter, or reduction of

the distance between bearings) to move its critical speed to a greater margin above its normal operating speed; and higher level quality control of the motor shaft bearings.

Noise in railway subways

80. In a paper on 'Public Transport System Noise and Noise Zoning' delivered at this Society's 'Noise-Zoning Conference', Warburton, Vic, 1971 (ref 22, paper no. 5), one of its conclusions was that "further reduction of noise is both desirable and possible, particularly in the fields of diesel engine noise and the operation of trains in tunnels". Concerning the operation of trains in tunnels or subways, as in 'Metros' or City Underground railways, the principles of quiet operation were considered to be stringent, and to include "the need for smooth wheels and rails, resiliently-supported rails (with supports -- for example, of rubber rather than ballast for space economy -- having a resilience at least as low as 40 kN/mm), rails treated to reduce vibration, tunnels with acoustically treated walls to reduce reverberation to a minimum, and adequate carriage body insulation".

81. The primary noise generated by railbound vehicles, such as suburban electric trains and trams, having steel wheels running on steel rails, is their wheel-on-rail noise, or wheel rumble, which is minimum with smooth rail and wheel tread surfaces, and increases as wheel and rail surface roughnesses increase -- the noise level increasing by up to 10 dB(A) under the more usual circumstances, or by 15 dB(A) or more under the more exceptional circumstances of noticeable wheel flats or worn rail joints. Wheel-on-rail noise also varies with the resilience of the rail support, and increases with its increasing stiffness. Increased resilience of the rail support also better reduces transmission from the rail to surrounding structures of the vibration generated by wheels moving over the track. Other noise such as from the transmission gears is minimized through the use of helical, double helical, hypoid or similar gears to replace the noisier straight spur gearing. While in the open air the wheel rumble and other noise of railbound vehicles is freely dispersed; in subways the noise is contained. If a subway tunnel is highly reverberant, which is the natural condition of a concrete- or similarly-lined structure, the noise is significantly amplified (sometimes by at least 20 dB), and the occupants of passenger vehicles suffer accordingly. Proper acoustic treatment of tunnel interiors -- with maximum resilience of the rail support to minimize the wheel rumble noise at its source, and sufficient acoustic absorption to keep the tunnel reverberation time down to about 0,6 to 0,8 s -- can be, and is used to minimize the noise levels to which subway passengers are exposed.

82. Case history no. 9 includes several groups of noise measurements made inside electric trains and trams travelling at their normal service speeds in the metropolitan subways of 13 different American, Australian and European cities, and shows that these measurements enable us to distinguish those incorporating noise reduction treaments from those without them. In general, though not invariably, the more recently constructed tracks and tunnels have been properly treated to minimize both noise and also the transmission of vibration from the track to the tunnel wall and adjacent structures and buildings. Some typical levels of the noise inside trains and trams travelling in subways at their normal service speeds are included in Appendix G.

83. Of the subway trains and trams running in older tunnels, London's 'tube' trains are amongst the quieter running because of the measures taken by London Transport to minimize tunnel reverberation. Of the more recently constructed

subways, Chicago has achieved quieter running with ballast tracks, while the Melbourne city underground rail loop, the San Francisco BART system, Toronto Transit and, more recently than these 1973 overseas noise measurements, the Washington, DC metropolitan transit authority have achieved quiet running of the trains in their subways by means of a special design of resiliently supported track incorporating rubber pads between rail and concrete sleeper and rubber blocks between concrete sleeper and tunnel wall, together with acoustic absorption to minimize tunnel reverberation. In each system the results are noticeably quiet running of trains (which sometimes become more noisy because of rough wheel tread surfaces), and effective vibration insulation of tracks from the tunnel structure. The Montreal design, following its introduction in the Paris metro, using rubber-tired wheels provides a different, though not any quieter solution to the problems of minimizing train noise in subways, with the noise of the rubber-tired wheels on a concrete running surface being a type of 'swishing' sound.

84. The operation of trains and trams in subways is another area in which there appear to be no specific Noise Control Criteria apart from what the systems' designers consider to be a reasonable compromise between passenger comfort and the added capital costs of incorporating quiet operation into the initial design. The Australian examples of higher operating noise levels, as for example in some sections of Sydney's Eastern Suburbs underground line to Bondi Junction with, apparently, 'concrete' track and reverberant tunnels, seem to have occurred through a too great emphasis on reduced capital costs. But such 'economies' are self-defeating because the cost of subsequent remedial treatment is invariably greater than incorporating the anti-noise and -vibration treatemts into the initial design. Of the noise measurements given in Appendix G, several show that, with proper design and care, the quiet running of suburban electric trains and trams in subways can be achieved, with sound levels inside vehicles of 85 dB(A) and less. Sound levels above 90 dB(A) indicate poor design (or dirty ballast in New York!) with little or no noise reducing treatments.

Traffic noise evaluation

85. The problem of traffic noise, especially from heavily trafficked urban expressways, working its way into noise-sensitive locations such as residential areas is of relatively recent provenance, and no more than around 50 years old. Various methods have been proposed for reducing it either by reducing it at its sources (engine, tyres and exhaust), or by modifying the noise transmission path between source and hearers through acoustic insulation of dwellings, noise zoning (separating heavily trafficked roadways from noise-sensitive areas: a matter of effective and efficient land use planning), the use of roadway cuttings, noise barriers or road tunnels, and the reducing of traffic flows and speeds (for example, by the diversion of people to metropolitan public transport (which is more energy and space efficient), and of freight to rail) (ref 23).

86. One of the chief problems of the introduction into a previously quiet area of a new arterial road or expressway which soon becomes heavily trafficked, is that residents are taken completely by surprise at the high intensity of the traffic noise, something which they had never imagined possible. It is then too late to make noise measurements in the absence of such heavy traffic flows!! What is perhaps even more deceptive is that the traffic noise guideline of a maximum value of the 18-hour L_{10} of 63 dB(A), formerly 68 dB(A), does not appear all that high. There are, however, two reasons why the noise giving rise to L_{10} noise levels of 68, or even 63 dB(A), appears

louder (and therefore more annoying) than these numbers might suggest, particularly in view of the experience (not normally annoying, or at least quite tolerable) that the maximum passby noise level of a passing motorcar heard near the adjacent kerb is around 65 to 75 dB(A), depending on the make of car and its speed. Firstly, the noise of passing vehicles at low traffic flows (10 to 250 veh/h) is occasional foreground noise, but which above about 300 to 500 veh/h changes to being intrusive background noise (ref 23). Secondly, some knowledge and experience are required of how, at a given distance from a roadway, L_{10} varies with traffic flow. Thus, for a distance of the order of 20 m, L_{10} is likely to increase by 15 dB(A) for an increase in traffic flow from 10 to 100 veh/h, to increase by a further 3 dB(A) to 500 veh/h, and then by about 2 dB(A) for each doubling of the flow to 4000 veh/h. As with many other situations, so also with traffic noise problems, a before and after study as part of the introduction of a new main roadway into an area, with environmental noise measurements in the noise-sensitive area before the high traffic flows occur, as well as afterwards, are necessary.

87. The most common remedy used at the present time to shield residential areas from arterial road and expressway traffic noise is the noise barrier. The noise reduction of such barriers depends primarily on the barrier's excess height above the direct noise transmission path between noise source and hearer, as well as on the distances between source and barrier, and barrier and hearer. Formulas are available for predicting this noise reduction when these distances and the frequency spectrum of the noise source are known, one such formula being given in BS CP3:Chapter III:1972, 'Sound insulation and noise reduction', Fig 10, 'Noise reduction by screens in open air' (ref 17). Case history no. 10 showed that, after tests with several barriers of 2m in total height giving measured noise reductions of 11 and 14 dB(A), there was good agreement with the calculated noise reduction of 12 dB(A) predicted by the BS CP3 formula used in conjunction with an average octave band frequency spectrum for motor vehicle noise. Because travel by motorcar is significantly less energy- and space-efficient in metropolitan areas than travel by public transport (refs 22b, 23a and 22a), there is considerable scope for further reducing the serious environmental noise problem of arterial road and expressway traffic by diverting people to public transport.

Loud equipment and machinery noise

88. Case history no. 11 involved the measurement of equipment and machinery noise at the operator's ear position. Appendix H gives the octave band frequency spectra of a group of some of the noisiest pieces of equipment in existence, to indicate something of the problems faced by those trying to reduce this noise, especially of reducing it at its source or sources, in order that operators can work in an acoustic environment involving no or, at worst, minimum hearing damage risk, and without having for ever to resort to hearing protection devices as at present. With the need for hearing protection against the noise of the equipment and machinery listed in Appendix H, devices having an SLC_{80} rating of at least 25 dB would be generally required. Our basic problem here is two-fold: not only of how to reduce this equipment and machinery noise at its source(s), but by how much in order to reduce operators' hearing damage risk to a minimum.

89. A survey of the operations, and equipment and machinery noises listed in Appendix H will show that all, except perhaps for the operation of dressing grinding wheels, are operations and equipment in common and widespread use. Pneumatic paving breakers and 'jack hammers' of one sort or another are used on many construction sites, circular saws (of both the larger fixed and smal-

-ler portable types), planers and moulders can be expected in most carpentry and wood-working shops, furnace burners are likely on any industrial site where metals are melted for casting, etc, smoke testing is a required operation in diesel engine maintenance locations, and pneumatic torque wrenches (with their characteristic 'clatter') can be found in virtually every motor vehicle maintenance shop. Very many employees are therefore exposed to intense noise while operating these pieces of equipment, of which none of those listed here had noise levels at the operator's ear above the 1978 Victorian Regulations' maximum of 115 dB(A)S. With the operations and pieces of equipment and machinery listed, the risk of hearing damage is further accentuated by the fact that, mostly, they have the greater part of their acoustic energy at high (1 to 2 kHz) or very high (4 to 8 kHz) frequencies, the notable exceptions being diesel engine noise at smoke test speed, and furnace burner noise. One simple warning of this prominence of high frequency noise (for which the human hearing damage risk is greatest), which can be simply measured by a sound level meter having 'A' and 'B' frequency weightings (without need for frequency analyzers), is that the sound level in dB(A) exceeds that in dB(B) by 1 dB or, in the worst cases, by 2 dB; the Table of 'Frequency-weighting characteristics' shows why. Other characteristics of this equipment and machinery noise can be noted here. Firstly, the noise from a machine in need of maintenance can be significantly greater than when in properly maintained condition (as, for example, with the wood-working moulder with blunt or sharp blades). Secondly, the noise from a machine when idling can be greater than when it is working (as with the planer). Thirdly, in a reverberant location the decrease in noise level with distance can be noticeably less than 6 dB with each doubling of the distance between source and hearer (as with the large circular saw noise at 1 and 20 m). In this case, whereas a decrease of about 25 dB would be expected under free-field conditions, the actual decrease was (on average above 125 Hz) 17 dB, indicating somewhat reverberant conditions, with a Room Constant (R) of around 2000 m^2, instead of infinity as for free-field conditions, as estimated from the formula

Sound pressure level at r m from a noise source of Directivity Factor, Q

$$= \text{Power level} + 10 \lg \left(\frac{Q}{4\pi r^2} + \frac{4}{R} \right) + 0{,}2$$

In addition, $R = \frac{\alpha \cdot S}{1 - \alpha}$ where S is the total room surface area (m^2), α the mean absorption coefficient, and α S the sum of the partial αS for the individual surfaces.

90. Noise from the operations, equipment and machinery listed in Appendix H has so far proved difficult to reduce at the sources. With pneumatic paving breakers, for example, in which four possible noise sources can be identified (pneumatic engine noise, air exhaust noise, engine to moil steel hammering, and the moil steel pavement breaking), noise at all four needs to be reduced before any significant reduction can be effected. For, with a total noise level of 113 dB(A), and, assuming all four sources to produce equal noise (at 107 dB(A)), it becomes understandable why the attempts to date to reduce this noise by shrouding the pneumatic engine and its exhaust outlet have effected only small noise reductions. Greater reductions are necessary if the sound level at the operator's ear is to be reduced to 85 dB(A) for long term exposure in accordance with the current Victorian 'Noise' Regulations of 1992. However, if as is suspected, the real safe or no-risk noise level for long exposure, for which there is (and probably also was) considerable evidence, is as low as the 80 dB(B) criterion (approximately equivalent to 74 dB(A)), even

greater noise reductions will be necessary. Again, we must conclude that the proper taking account of acoustic considerations at the design stage of any equipment, machinery and plant is absolutely necessary. It is a sobering thought that, if 200 years ago there had been stringent environment protection regulations in existence and enforced, many of today's polluting industries begun then under much more lax conditions might never have come into existence because of prohibitive costs for keeping industry 'clean'; for which we would well be the better off.

91. The development, formulation and use of Noise Control Criteria both implies and requires the making of noise measurements if noise problems are to be properly identified and satisfactorily solved. The basic noise measuring instrument is still the Sound Level Meter, though today's meters provide considerably more facilities than the early ones, which, with the capability of measuring rms sound levels from 24 to 140 dB, provided only A, B and C frequency-weightings and 'fast' and 'slow' time-weightings as accessories. Today's meters are very likely to be also able to measure 'peak' as well as 'rms' sound levels, have additional 'D' and 'flat' (or 'lin') frequency-weightings (the earlier C-weighting being only nearly flat), additional 'I' (impulse) and 'P' time-weightings for measuring, respectively, the short-term instantaneous 'rms' or 'peak' maxima of impulsive sounds, and a 'Maximum-hold' control for capturing them for convenient observation. A Sound Level Meter's ability for measuring the characteristics of impulsive sounds is today a necessary requirement for our clearer understanding of, and better evaluating their potential for causing noise-induced hearing damage. Some meters also are provided with integrating facilities for measuring over a defined duration of time the L_{eq} and SEL (formerly L_{AX}) of sounds of continually varying level. But for most purposes, especially in assessing hearing damage risk and speech intelligibility, in applying the appropriate Noise Control Criteria, and in then obtaining sufficient detailed information about the noise problem to enable its solution, other instruments such as Frequency Analyzers, and Graphic Level, Data and Tape Recorders are an essential part of any acoustical investigator's equipment.

92. The main purpose of quoting the equipment and machinery operating noise levels included here has been to show that, in spite of the noise reductions already achieved, there are still many noisy pieces of equipment and machinery in existence which require significant redesign if they are to operate noticeably more quietly than at present, a requirement which has behind it the authority of both AS 1269-1988,'Hearing Protection', and the Victorian government statutory 'Occupational Health and Safety (Noise) Regulations 1992'. But even these provisions are still an interim compromise between human welfare and monetary expense, with the ultimate aim being to reduce long-term noise exposure levels to no more than 80 dB(B)/74 dB(A)$_{rms}$.

SOME USES OF VIBRATION CONTROL CRITERIA

93. In our daily lives we encounter many sources of vibration and their resultant effects, some from 'natural' activity such as earthquakes, and many from humanly organized activities such as in manufacturing or in transport. Much of this vibration is of magnitude low enough to cause no problems. Some, however, is severe enough to interfere with instrument operation, to excite resonance vibration (often at considerable distance from its source), or to cause some damage to buildings. Some vibration, as from hand-held tools and equipment such as chain saws and pneumatic paving breakers, is sufficient to cause human bodily damage (such as Vibration White Finger or worse). In these cases there is need to apply the criteria of AS 2763-1988

(ref 10b), and of ISO/DIS 5349.2 on which they are based. The earlier discussion (paras 81ff above) about the need to prevent the vibration due to subway train movements from being transmitted from track to tunnel structure is at least partly to minimize transmission of this vibration into buildings close to the tunnels, so that the movement of trains doesn't excite resonant vibrations in doors and windows, causing them to rattle without obvious or apparent cause. In another earlier discussion (para 56 above), on Vibration Control Criteria, the possible effects of ground-borne vibration (from whatever cause, earthquake, quarry blasting, or rail or road traffic movement) on instruments and buildings was considered. Two final case histories illustrate the use of Vibration Control Criteria in these circumstances.

Vibration control criteria and instrument operation

94. Case history no. 12 concerns an organization which as part of its operations used several electron microscopes. The organization was concerned that vibration from traffic on a roadway adjacent to its building, especially from heavy vehicles, might be disturbing the operation of these microscopes, particularly when set to maximum magnification. The relevant Vibration Control Criterion for these instruments, as supplied by their manufacturer, was that below 5 Hz they should be subject to no measurable vibration, and that above 5 Hz the vibration displacement should not exceed 3 micrometre peak-to-peak. Several vibration tests were therefore made, firstly at other roadside locations of the magnitude of the ground-borne vibration emanating from moving motor vehicles of various size on the roadway, and transmitted to locations within 20 to 30 m; and secondly of the structure-borne vibrations existing in this organization's building. All vibration was measured in terms of its displacements along the three orthogonal axes.

95. The results of the vibration tests showed a number of interesting features. Firstly, because the building in which these microscopes were situated contained on its uppermost floor various items of building services plant (for air-conditioning, etc), considerable amounts of structure-borne vibration were being transmitted from this plant throughout the whole building, primarily in measurable (and therefore deleterious) amounts below 5 Hz. Secondly, ground vibration measurements just outside the building also detected this vibration, which was of generally sufficient magnitude to mask any ground-borne vibration transmitted from vehicle movements on the adjacent roadway. Thirdly, the exterior ground vibration measurements, because they were made with simultaneous use of visual graphic level as well as of magnetic tape recording, also enabled the vibratory effects of wind gusts passing around a corner of the building to be detected and identified as well as physically observed and felt on the occasions on which they occurred. This showed that there is always a need, when either vibration or noise signals are being recorded for subsequent laboratory analysis, for an indicating instrument or chart to be in use as the measurements are being made so that the effects of unusual or otherwise unsuspected occurrences (such as the effects of these wind gusts) can be noted and identified at the time they occur and are observed.

96. The organization concerned here was therefore faced with a problem quite different from the one which it had initially expected. Under the circumstances prevailing, a possible measure to isolate the electron microscopes from the structure-borne vibration to which they were then subject was to mount them on a special floor at the base of the building, on a foundation completely independent of the building structure, and itself not subject to significant vibration. This case provides an example to show that, designers

of buildings which will contain building plant and machinery for air-conditioning and other building services, need to explore the possibility of isolating this plant from the main structure of the building in order to minimize transmission of its vibration into the building. Alternatively, buildings which are being designed to contain sensitive instruments requiring to be free of vibration, need to be designed with proper provision for their isolation from any vibration. Under these circumstances, when the vibration is of low frequency, such as below 5 Hz, effective and practical anti-vibration mounts for individual instruments -- apart from a special floor slab and vibration-free foundation independent of the main building structure -- are difficult to design and cumbersome to use.

Vibration control criteria and traffic movement

97. All moving traffic, whether on rail or road, generates some vibration which is then propagated as ground-borne vibration. At times, residents who live near railways or heavily-trafficked roadways including those carrying a tramway, finding that their dwelling is suffering minor damage such as wall or ceiling cracks, immediately suspect the rail, road or tram traffic movements. One of the purposes of the Vibration Control Criteria of the German Standard DIN 4150, Part 3, 1977 (ref 20) is to provide a satisfactory and accepted criterion for arbitrating disputes of this kind. For, once measurement of the 3-dimensional vibration velocity vector has been made at the appropriate location, comparison of its magnitude with the criteria of DIN 4150, Part 3 will establish beyond reasonable doubt whether the building damage observed can or cannot be considered as resulting from rail or road traffic movements nearby. The DIN 4150 criteria (see para 57 above) are especially clear here, with categories of "structural damage not possible" (for vibration velocities below 2,5 mm/s rms), "damage is very unlikely" (velocities between 2,5 and 6,0 mm/s rms), "damage is unlikely" (velocities between 6,0 and 10,0 mm/s rms), and "damage is possible and the structure should be investigated" (velocities above 10,0 mm/s rms).

98. When it is found to be virtually impossible that the building damage could have been due to ground-borne vibration generated by nearby rail or road traffic because the vibration velocity arising from it as measured near the dwelling was less than 2,5 mm/s rms, other reasons for the damage have then to be sought. One quite common cause is from non-uniform movement of the building foundations resulting from clay shrinkage during an unusually dry summer season. In addition, such shrinkage in dry summers can be aggravated by the presence of Australian native trees and bushes too near a dwelling. The ease with which appropriate vibration measurements can be made and compared with the corresponding Vibration Control Criteria saves much needless argument in these circumstances.

The usefulness of control criteria

99. The case histories set out above have been designed to illustrate that appropriate noise and vibration measurements used with the corresponding noise or vibration control criteria are an essential part of the process of establishing the existence or non-existence of a noise or vibration problem, and indicating the lines along which it might be solved. However, it is usual that, once the existence of a noise or vibration problem has been established, the measurements required for this purpose are not always adequate for its subsequent solution, and more measurements such as more detailed frequency or time analyses are required. While this aspect of noise and vibration problems has not been stressed or covered in detail here, enough data from these case histories (in Appendixes D to I) have been given to suggest

this. For example, the sound level, measured at the operator's ear, of 113 dB(A) for pneumatic paving breaker noise (Appendix H) is enough to indicate that, according to the appropriate regulations (refs 6, 24 and 25), direct exposure of a person (operator, nearby worker or bystander) to this level of noise without hearing protection must be limited to a daily duration of 44 s (formerly 140 s on the basis of the 90 dB(A) 8 h exposure for a Daily Noise Dose = 1,0). However, any attempt to find a solution to the problem of pneumatic paving breaker noise would require not only the measurement of the octave band noise levels (needed for selecting an appropriate hearing protection device) but additional one-third octave or narrower band analysis to check for resonances, etc, as well as careful measurement, with a meter employing the 'I' or similar short time response, of the maximum instantaneous sound levels (either rms or peak, but preferably peak) as those aspects of such intense sound which are likely to cause noise-induced hearing loss.

100. In addition, two case histories -- those concerned with excessive reverberation in lunch rooms (no. 3), and with the noise inside subway trains in tunnels (no. 9) -- were not adequately provided with published Noise Control Criteria; and another (no. 1) had an inconclusive outcome because of apparently insufficient criteria available at the time of its occurrence, although part of the difficulty with this case was also the unavailability at the time of any instrument other than a Sound Level Meter. It may be hoped that continuing researches and investigations will result in the provision of adequate Noise and Vibration Control Criteria where none exist at present.

CONCLUSIONS

101. The increased industrialization and mechanization which have been effected over the last 200 years or more, while they have brought us many benefits, have also brought about a considerable increase in the levels of noise to which we are daily exposed. Much of this noise and, often, associated vibration has brought to light many human and equipment problems which must be solved and remedied. This need for solution and remedy has initiated and stimulated much development

- in the production and use of measuring instruments, many of them portable for field use, to enable highly detailed analysis and study of noise and vibration problems;
- in much research into, and quantification of the many human responses to noise and vibration; and
- in the development over the last 60 years of Noise and Vibration Control Criteria for use both in clearly establishing the existence of a noise or vibration problem and the responsibility of its originators, and also for use at the design stage of buildings, equipment, machinery, plant and vehicles so that possible future noise or vibration problems (and their normally expensive resolution) are not created, through proper initial consideration by designers of all relevant acoustic and vibration factors, an activity whose importance cannot be too greatly emphasized.

102. <u>Practical noise and vibration control</u> begin at the design stage of buildings, equipment, machinery, plant and vehicles by taking account of all relevant acoustical and vibration considerations, many of which are embodied in the noise and vibration control criteria developed over the last 60 years, for human health and welfare, and the efficient operation of buildings, machinery, etc. Such control also begins with establishing and confirming the existence of a noise or vibration problem, and identifying those respon-

sible for it, by means of initial measurements and their comparison with the appropriate control criteria, followed by additional and more detailed measurements to further investigate the problem in order to develop its resolution.

103. Noise Control Criteria have several basic forms, being based on loudness considerations, which cover the whole audible range of levels from 'very faint' to 'deafening'; on speech intelligibility considerations, with quantitative measures for specifying the various conditions under which reliable speech communication is or is not possible against backgrounds of more or less noise; on more general considerations of human health and welfare against a background of various acoustic environments; and on hearing damage risk considerations, to determine the levels of broad band noise, pure tones and impulsive sound likely to cause permanent hearing damage for different daily and yearly durations of noise exposure associated with a maximum permissible 'Daily Noise Dose' (DND) of 1,0 (or 100 percent).

104. Of special concern are the hearing damage risk criteria, particularly because over the last forty years or more they have varied considerably, with, for broad band noise, the 8-hour per day long-term maximum exposure level for 'negligible risk' decreasing from 101 dB(A) in 1953, to 92 dB(A) in 1956, to the French criterion of 82 dB(A) of the mid-1960s. While some of this variability can be considered due to the added knowledge and experience gained over a continually increasing time in which audiometric measurements of noise-induced human hearing loss have been available, some of it must also be considered due to the varying states of compromise between human health and safety, and monetary expenditure for ensuring a quieter environment. However, behind all these considerations lies the 80 dB(B) maximum noise criterion (pre-1953) which is "probably safe even with pure tones, with no hearing damage likely". Recent data on noise-induced hearing loss published in AS 1269-1983 and AS 1269-1988 tend to confirm this. The other aspect of noise-induced hearing damage, which is currently the subject of much research, is a concern to determine, in contrast to the previous work mostly on average hearing damage risk exposure levels, the instantaneous maximum values of sound level (both 'rms' and 'peak') of the more intense sounds to which our ears are exposed, and which are likely to cause permanent hearing damage.

105. Vibration Control Criteria have been similarly developed, so that there is now a considerable body of criteria covering safe exposure and tolerability levels in human beings, damage risk and safe exposure levels for buildings, equipment, machinery and plant, including those developed for the condition monitoring of equipment and machinery.

106. The many researches discussed here have resulted in these Noise and Vibration Control Criteria being embodied in numerous publications of the Standards Association of Australia, of the International Standards Organization (ISO) and International Electrotechnical Commission (IEC), of numerous other national Standards Institutions, and several Victorian government statutory regulations such as the 'Occupational Health and Safety (Noise) Regulations 1992' (a revision of the original 1978 regulations) and 'Environment Protection Policy no. N-1 of 1981/1987. This country is fortunate in having available a comprehensive group of Australian Standards covering many aspects of noise and vibration control criteria.

107. These Standards, though comprehensive, with good coverage of hearing damage risk, speech intelligibility, environmental noise and vibration concerns, have several areas where the scope of their criteria could be somewhat

enlarged, as in recommended reverberation times (AS 2107) for lunch areas, factories, and post office and banking areas.

108. Practical noise and vibration control, from one point of view, begin, it has been said, with identifying and assessing the problem so as to resolve it. That noise and vibration problems, difficult though some may be, can mostly be remedied is illustrated in the several case histories given here. Such remedies, however, are usually costly; the problems are likely to have been avoidable. Practical noise and vibration control, from another point of view, therefore begin at the design stage of a project, by taking proper account of the relevant acoustical and vibration factors, including the already existing and available Noise and Vibration Control Criteria developed over the last 60 years, so not causing avoidable and unnecessary problems, and avoiding the consequent expense of having to remedy them.

ADDENDUM: Between the times of writing and publishing this paper, preliminary tests were made of the impulsiveness of the noise of a 2-stroke petrol-engined lawn mower and mechanical typewriter. Appendix J gives the results.

The results give us cause to re-assess our methods of measuring noise when possible hearing damage risk is involved. Lawn mower noise (at the operator's ear position) is unquestionably loud, and a risk with an L_{eq} of 89 dB(A)F_{rms}, and maxima at 90 dB(A)F_{rms} and 103 dB(A)F_{pk}, the Crest Factor and ($L_{max.pk}$ - L_{eq}) being 13 and 14 dB(A). Typewriter noise is, however, much more problematic. While such noise appears reasonably safe with an L_{eq} of 75 dB(A)F_{rms}, and probably still reasonably so with a maximum of 80 dB(A)F_{rms} or 85 dB(A)I_{rms}, this apparent margin of safety has disappeared at a maximum level of 104 dB(A)F_{pk} or 112 dB(A)I_{pk}, the Crest Factor and ($L_{max.pk}$ - L_{eq}) being 24 and 29 dB(A).

These types of results suggest that a more consistent practice with noise exposure levels likely to cause hearing damage would be to measure L_{eq} in dB(A)$_{rms}$ as previously, with also the corresponding maxima in dB(A)$_{rms}$ and dB(A)$_{pk}$ so as to obtain the Crest Factor (=$L_{max.pk}$ - $L_{max.rms}$) and $L_{max.pk}$ - L_{eq}, which indicate, respectively, the degrees of impulsiveness and intermittency of the noise under test.

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TABLE·1A SUBJECTIVE ASSESSMENT OF NOISE LEVELS OF COMMON SOUNDS

SOUND (near source)	SOUND LEVEL in dB	SOUND (General environment)	SUBJECTIVE ASSESSMENT
	140		
37 kW victory siren (30 m) F-84 at takeoff (24 m behind) Hydraulic press (0,9 m) Lge pneumatic riveter(1,2 m)	130	Boiler shop (max)	
Pneumatic chipper (1,5 m) Multiple sand blast unit(1,2m) Trumpet car horn (0,9 m)	120	Engine room of submarine:full speed Jet engine test control room	Deafening
Automatic punch press(0,9m) Chipping hammer (0,9 m) Cut-off saw (0,6 m) Annealing furnace (1,2 m)	110	Woodworking shop I/s DC-6 airliner;weaving room	
Automatic lathe (0,9 m) Subway train (6 m) Heavy trucks (6 m)	100	Metal container factory I/s Chicago subway car	
Train whistles (150 m) 7,5 kW outboard motor(15 m) Small trucks accelerating (9 m)	90	I/s motor bus I/s sedan car in city traffic	Very loud
Light trucks in city (6 m) Motorcars (6 m)	80	Office with tabulating m/cs Heavy traffic (7,5 to 15 m)	
	70	Average traffic (30 m) Accounting office	Loud
Conversational speech (0,9 m)	60	Chicago industrial areas	
15 MVA, 115 kV transformer (60 m)	50	Private business office Light traffic (30 m) I/s average home	Moderate
	40		
	30	Suburban areas at night Broadcasting studio(speech) Broadcasting studio(music)	Faint
	20	Studio for sound pictures	
Whisper (1,5 m)	10		Very faint
Threshold of hearing: 1 kHz	0		

NOTE: Sound levels (from the 1953 GR Handbook (ref 34, p 2)) below 55 are in dB(A), from 55 to 85 in dB(B), above 85 in dB(C).

TABLE 1B SUBJECTIVE ASSESSMENT OF NOISE LEVELS OF COMMON SOUNDS

SOUND	SOUND LEVEL in dB (Note 2)	SUBJECTIVE ASSESSMENT
Jet aeroplane take-off (at 60 m) Threshold of pain		Deafening
	120	
Thunder, artillery Nearby riveter Pneumatic hammer (at 2 m) Boiler factory		
	100	
Lorry unmuffled Noisy factory interior Loud street noise Police whistle		Very loud
	80	
Noisy office Average street noise Average wireless (radio set) Average factory interior		Loud
	60	
Noisy home Average office Average conversation Quiet wireless		Moderate
	40	
Quiet home Quiet private office Average auditorium Quiet conversation		Faint
	20	
Rustle of leaves Whisper Sound proof room Threshold of audibility		Very faint
	0	

NOTES: (1) This TABLE (derived from 'Sound and Vibration Analysis' - ENGG, 29 Aug 1952, Table I, pp 286-88) gives the normal subjective assessment of sounds in relation to their sound level, and suggests sounds with levels of the order shown.

(2) Sound levels below 55 are in dB(A), from 55 to 85 in dB(B), and above 85 in dB(C).

TABLE.1C LOUDNESS LEVELS OF TYPICAL SOUNDS

SOUND	LOUDNESS LEVEL in phons	LOUDNESS in sones
Threshold of feeling or pain	130	512
Nearby aeroplane engine	110 to 120	128 to 256
Nearby pneumatic drill	105 to 110	90 to 128
Nearby loud motor horn	100 to 105	64 to 90
Interior of tube train, windows open	90 to 95	32 to 45
Interior of noisy motor vehicle; loud radio set	90	32
Interior of main line train, windows open	80	16
Interior of quiet motorcar; medium radio set	70	8
Conversation (average to loud)	60 to 75	4 to 11
Suburban residential district	40 to 50	1 to 2
Quiet country residence	20 to 30	0,25 to 0,5
Threshold of audibility	0	0,06

NOTES: (1) English data quoted in Langford Smith F (ed), The Radiotron Designer's Handbook (ref 33, p 89)
(2) Corresponding Loudnesses in sones (on an arithmetic scale with scale numbers proportional to the loudness, in contrast to the phon scale which is logarithmic, have been added according to AS 1047-1971 (ref 2, Table 1).

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TABLE 1D NOISE LEVELS OF SOME TYPICAL SOUNDS

NOISE SOURCE OR ENVIRONMENT	SOUND LEVEL in dB(A)	LOUDNESS LEVEL in phons	LOUDNESS in sones
Heavy diesel-engined vehicle (at 7,5 m)	92	108	111
Printing press plant (medium size automatic)	86	98	56
Loudly reproduced orchestral music in large (room	82	-	-
Ringing alarm clock at 0,6 m	80	91	34
Inside suburban electric train (60-70 km/h)	76	87	26
Inside small sports car at 80 km/h	75	-	-
Inside small sports car at 50 km/h	72	-	-
Inside small saloon car at 50 km/h	70	86	24
Vacuum cleaner in private residence (at 3 m)	69	81	17
Typing pool (9 typewriters in use)	65	79	15
Busy restaurant or canteen	65	77	13
Household department of large store	62	75	11,3
Self-service grocery store	60	74	10,6
Men's clothing department of large store	53	68	7
Soft whisper at 1,5 m	34	47	1,6
Room in a quiet London dwelling at midnight	32	-	-

NOTE: Data are from the Wilson 'Final report on Noise' (ref 37, Table I, p 3) In this Table, Loudness Levels in phons have been added to correspond to the Loudnesses in sones, according to AS 1047-1971 (ref 2, Table 1). The mean difference of Phon - dB(A) is 13 (± 1 for 95% conf)

TABLE 2 MAXIMUM PERMISSIBLE SPEECH INTERFERENCE LEVELS FOR RELIABLE COMMUNICATION FOR VOICE LEVELS (men) AND DISTANCES SHOWN

DISTANCE BETWEEN PERSONS in m	MAXIMUM PERMISSIBLE SPEECH INTERFERENCE LEVELS (SIL4) IN dB FOR VOICE LEVEL						EQUIVALENT MAX PERMISSIBLE SOUND LEVEL IN dB(A) FOR VOICE LEVEL	
	Normal	Raised	V Loud	Shout	Max Effort	Max for Amplif	Normal	Raised
0,10	78	84	90	96	-	-	86	92
0,12	76	82	88	94	-	-	84	90
0,16	74	80	86	92	-	-	82	88
0,20	72	78	84	90	-	-	80	86
0,25	70	76	82	88	-	-	78	84
0,32	68	74	80	86	103	124	76	82
0,40	66	72	78	84	101	122	74	80
0,50	64	70	76	82	99	120	72	78
0,63	62	68	74	80	97	118	70	76
0,80	60	66	72	78	95	116	68	74
1,0	58	64	70	76	93	114	66	72
1,2	56	62	68	74	91	112	64	70
1,6	54	60	66	72	89	110	62	68
2,0	52	58	64	70	87	108	60	66
2,5	50	56	62	68	85	106	58	64
3,2	48	54	60	66	83	104	56	62
4,0	46	52	58	64	81	102	54	60
5,0	44	50	56	62	79	100	52	58
6,3	42	48	54	60	77	98	50	56
8,0	40	46	52	58	75	96	48	54
10,0	38	44	50	56	73	94	46	52
11,2	37	43	49	55	72	93	45	51
12,5	36	42	48	54	71	92	44	50
14,1	35	41	47	53	70	91	43	49
16,0	34	40	46	52	69	90	42	48
18,0	33	39	45	51	68	89	41	47
20,0	32	38	44	50	67	88	40	46
22,4	31	37	43	49	66	87	39	45
25,0	30	36	42	48	65	86	38	44
29,0	29	35	41	47	64	85	37	43
32	28	34	40	46	63	84	36	42
36	27	33	39	45	62	83	35	41
40	26	32	38	44	61	82	34	40
45	25	31	37	43	60	81	33	39
50	24	30	36	42	59	80	32	38
56	23	29	35	41	58	79	31	37
64	22	28	34	40	57	78	30	36
72	21	27	33	39	56	77	29	35
80	20	26	32	38	55	76	28	34
90	19	25	31	37	54	75	27	33

NOTE: These Maximum Permissible SILs are from AS 2822 (ref 11) and J C Webster (ref 36, pp 37-40) for use with measured SIL4s (see para 19). Conditions for their use are given in AS 2822 and GR Handbooks of Noise Measurement (ref 34, p 74; ref 35, p64; ref 35a, p 62)

TABLE 3 CRITERIA FOR NOISE CONTROL IN ROOMS

TYPE OF ROOM	MAX PERMISS LEVEL FOR SPEECH INTELLIGIBILITY		
	SIL4 in dB Room unoccupied	EQUIV LEVEL in dB(A) (Note 2)	COMPARATIVE LEVEL FROM AS 2107
Small private office	43	51	35 to 40 dB(A)
Conference room for 20	33	41	30 to 35 dB(A)
Conference room for 50	28	36	30 to 35 NR
Cinema	33	41	25 to 30 NR
Theatre for drama (500 seats, no amplification)	28	36	20 to 25 NR
Sports coliseum (with amplification)	53	61	50 to 55 dB(A)
Concert hall (no amplification)	23	31	20 to 25 NR
Assembly hall (no amplification)	28	36	30 to 35 dB(A) 25 to 30 dB(A) (Note 3)
School classroom	28	36	35 to 40 dB(A)
Home bedroom	28	36	25 to 30 dB(A)

NOTES: (1) This Table was earlier included in the 1953 GR Handbook of Noise Measurement (ref 34, Table 7-II, p 74). It is reproduced here with the original SILs of that Table now converted to equivalent SIL4s.

(2) The equivalent A-weighted sound levels (col 3) are added, and were derived from the AS 2822 relationship of mean dB(A) - SIL4 = 8 dB. The comparative levels (col 4) have been added from AS 2107 for purposes of comparison.

(3) AS 2107 gives two recommendations for Assembly halls with no amplification: the first for halls with up to 250 seats, the second for halls with over 250 seats.

TABLE 4 SOME AS 2107-1987 RECOMMENDED DESIGN SOUND LEVELS AND REVERBERATION TIMES FOR DIFFERENT AREAS OF OCCUPANCY IN BUILDINGS

TYPE OF OCCUPANCY/ACTIVITY	RECOMMENDED DESIGN SOUND LEVELS		RECOMMENDED REVERBERATION TIMES in s
	Satisfactory	Maximum	
1 EDUCATIONAL BUILDINGS			
Assembly halls up to 250 seats	30 dB(A)	35 dB(A)	0,7 to 1,44
Assembly halls over 250 seats	25 dB(A)	30 dB(A)	0,6 to 0,8
Class rooms -- single cell	35 dB(A)	40 dB(A)	0,6 to 0,7
2 HEALTH BUILDINGS			
Corridors and lobby spaces	40 dB(A)	50 dB(A)	-
Casualty areas, consulting rooms	40 dB(A)	45 dB(A)	-
Delivery suites	45 dB(A)	50 dB(A)	-
Operating theatres, 1-bed wards	30 dB(A)	35 dB(A)	-
3 INDUSTRIAL BUILDINGS			
Assembly lines - light machinery	50 dB(A)	70 dB(A)	-
Foremen's offices	45 dB(A)	50 dB(A)	-
Laboratories or test areas	40 dB(A)	50 dB(A)	-
Lunch areas	40 dB(A)	55 dB(A)	-
4 INDOOR SPORTS BUILDINGS			
-- with coaching	45 dB(A)	50 dB(A)	1,4 to 2,9
-- without coaching	50 dB(A)	55 dB(A)	1,4 to 2,9 (max)
5 OFFICE BUILDINGS			
Board and conference rooms	30 dB(A)	35 dB(A)	0,6 to 0,8
Private offices	35 dB(A)	40 dB(A)	0,6 to 0,8
Reception areas, rest rooms	40 dB(A)	45 dB(A)	- , 0,4
Toilets and tea rooms	50 dB(A)	65 dB(A)	-
6 PUBLIC BUILDINGS			
Conference and convention centres			
up to 50 persons (w/o speech reinf)	30 NR	35 NR	0,7 to 1,44
from 50 to 250 (" " ")	25 NR	30 NR	0,7 to 1,44
up to 250 persons (with sp reinf)	35 NR	40 NR	0,7 to 1,44
more than 250 (" " ")	25 NR	35 NR	0,7 to 1,44
Place of worship - up to 250 cong	30 dB(A)	35 dB(A)	1,6 to 2,8
Post offices and general banking areas	45 dB(A)	50 dB(A)	-
7 RESIDENTIAL BUILDINGS			
Sleeping (rural and outer suburb)	25 dB(A)	30 dB(A)	-
Sleeping (inner suburb)	30 dB(A)	35 dB(A)	-
8 SHOP BUILDINGS			
Specialty shops (needing speech)	40 dB(A)	45 dB(A)	-
Supermarkets	50 dB(A)	55 dB(A)	-
9 STUDIO BUILDINGS			
Film or TV studios	20 NR	25 NR	0,25 to 0,9
Music recording; sound stages	15 NR	20 NR	0,8 to 2,36

NOTES: Data extracted from AS 2107-1987 (ref 10, Table 1, Appendix A), which gives all details of use, and of notes to this Table.

TABLE 5A 1953, 1956 HEARING DAMAGE RISK CONTOURS FOR OCTAVE BANDS OF NOISE (LONG EXPOSURE)

CONTOUR	MAXIMUM PERMISSIBLE EXPOSURE LEVELS IN dB FOR OCT BANDS OF NOISE FOR FREQ (Hz)										CORRESPONDING OVERALL SOUND LEVELS			
	31,5	63	125	250	500	1k	2k	4k	8k	16k	dB	dB(C)	dB(B)	dB(A)
1953 ASA Comm Z24-X-2	114	107	100	95	94	94	94	94	94	94	115	113	105	101
1956														
Serious risk	124	117	110	106	105	105	105	105	105	105	125	123	115	112
Mean	114	107	100	96	95	95	95	95	95	95	115	113	105	102
Negligible risk	104	97	90	86	85	85	85	85	85	85	105	103	95	92

NOTES: (1) The 1953 contour was derived from the 1953 GR Handbook (ref 34, p 76).
(2) The 1956 contours were derived from the 1956 GR Handbook (ref 35, p 66).
(3) Levels in the 31,5, 63 and 16 000 Hz octave bands were obtained by extrapolation from each graph.

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TABLE 5B HEARING DAMAGE RISK CONTOURS FOR OCTAVE BANDS OF NOISE

NOISE DURATION (per day)	MAXIMUM PERMISSIBLE EXPOSURE LEVELS IN dB FOR OCT BANDS OF NOISE FOR FREQ (Hz)										CORRESPONDING OVERALL SOUND LEVELS			
	31,5	63	125	250	500	1k	2k	4k	8k	16k	dB	dB(C)	dB(B)	dB(A)
up to 1,5 min	135	135	135	135	135	135	124	121	130	130	-	-	-	138
3 min	135	135	135	135	135	125	113	111	123	128	-	-	-	133
7 min	135	135	135	135	122	115	106	104	113	118	-	-	-	128
15 min	135	135	135	122	112	106	100	98	106	109	-	-	-	121
30 min	135	135	126	114	105	100	95	93	101	103	-	-	-	113
1 h	135	133	118	107	98	95	92	90	97	97	-	-	-	110
2 h	135	123	110	101	95	92	88	87	91	89	-	-	-	103
4 h	117	110	103	96	92	88	86	85	88	85	-	-	-	96
8 h	110	103	96	92	88	86	85	85	87	83	111	109	100	94

NOTES: (1) These data are from Kryter (ref 32, p 173)
(2) The above levels are 5 dB lower for noise measured in $\frac{1}{3}$-oct bands
(3) Band levels at 31,5, 63 and 16 000 Hz are by extrapolation; at low frequency from Kryter's graph, at high freq from the standard Equal Loudness Contours for pure tones.

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TABLE 5C HEARING DAMAGE RISK CONTOURS FOR OCTAVE BANDS OF NOISE (French)

8-hour CONTOUR	MAXIMUM PERMISSIBLE EXPOSURE LEVELS IN dB FOR OCT BANDS OF NOISE FOR FREQ (Hz)										CORRESPONDING OVERALL SOUND LEVELS			
	31,5	63	125	250	500	1k	2k	4k	8k	16k	dB	dB(C)	dB(B)	dB(A)
Serious risk	100	100	95	90	85	85	85	85	85	85	104	103	97	93
Neglig risk	100	90	85	80	78	76	74	72	70	68	101	98	88	82

NOTE: These data are also from Kryter (ref 32, p 180)

TABLE 5D THE KRYTER AND FRENCH 8-HOUR HEARING DAMAGE RISK CONTOURS ADJUSTED TO GIVE OVERALL SOUND LEVELS OF 90 and 85 dB(A), AND 80 dB(B)

8-HOUR CONTOUR	MAXIMUM PERMISSIBLE EXPOSURE LEVELS IN dB FOR OCT BANDS OF NOISE FOR FREQ (Hz)										CORRESPONDING OVERALL SOUND LEVELS			
	31,5	63	125	250	500	1k	2k	4k	8k	16k	dB	dB(C)	dB(B)	dB(A)
Kryter	106	99	92	88	84	82	81	81	83	79	107	105	96	90
Fr neg risk	108	98	93	88	86	84	82	80	78	76	109	106	96	90
Kryter	101	94	87	83	79	77	76	76	78	74	102	100	91	85
Fr neg risk	103	93	88	83	81	79	77	75	73	71	104	101	91	85
Kryter	90	83	76	72	68	66	65	65	67	63	91	89	80	74
Fr neg risk	92	82	77	72	70	68	66	64	62	60	93	90	80	74

NOTES: (1) The frequency contours used in this Table are the 8-h Kryter contour (TABLE 5B) and the French 'negligible risk' contour (TABLE 5C).
(2) The above levels must be reduced by 5 dB for noise measured in $\frac{1}{3}$-octave bands.

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TABLE 6 HEARING DAMAGE RISK CONTOURS FOR PURE TONES

NOISE DURATION (per day)	MAXIMUM PERMISSIBLE EXPOSURE SOUND PRESSURE LEVELS IN dB FOR PURE TONES AT FREQUENCIES (Hz) OF														
	63	100	125	250	500	700	1k	1,5k	2k	3k	4k	5k	6k	7k	8k
up to 1,5 min	130	130	130	130	127	124	123	122	118	115	116	119	122	125	125
3 min	130	130	130	122	117	114	113	112	109	105	106	109	112	115	117
7 min	128	122	119	113	108	106	105	104	100	98	98	102	104	106	108
15 min	119	114	112	106	101	99	98	97	94	92	93	95	97	99	100
30 min	112	107	105	100	96	94	93	93	90	87	88	90	92	94	96
1 h	105	101	98	94	91	90	88	87	86	84	85	87	88	90	91
2 h	101	97	95	91	88	86	85	84	83	82	82	83	84	85	86
4 h	99	94	92	88	85	84	83	82	81	80	80	81	82	82	83
8 h	97	93	91	86	83	82	81	80	80	80	80	80	80	81	82
8-h Oct Bands of Noise, 80 dB(B)															
(a) Kryter	83		76	72	68		66		65		65				67
(b) Fr neg risk	82		77	72	70		68		66		64				62

NOTES: (1) The exposure levels for pure tones are from Kryter (ref 32, p 173)
(2) The 8-hour 80 dB(B) exposure contours for octave bands of noise, considered "safe, even for pure tones" have been added for comparison with Kryter's 8-hour pure tone levels.

TABLE 7 SUBJECTIVE ASSESSMENTS OF TRANSPORT VEHICLE PASSBY NOISE

TYPE OF VEHICLE	CORRESPONDING SOUND LEVELS IN dB(A) FOR SUBJECTIVE ASSESSMENT OF			
	QUIET	MODERATELY NOISY	NOISY	VERY NOISY
Petrol vehicle at 7,5 m	70 and less	71 to 79	80 to 88	89 and over
Diesel vehicle at 7,5 m	72 and less	73 to 80	81 to 87	88 and over
Motor cycle at 7,5 m	72 and less	73 to 82	83 to 91	92 and over
Aircraft overhead (from ground)	76 and less	77 to 93	94 to 107	108 and over
Tram at 7,5 m	82 and less	83 to 87	88 to 92	93 and over

NOTE: These data, for the passby noise of motor vehicles and aircraft heard and assessed individually, are from the Wilson 'Final Report on Noise' (ref 37, pp 178, 177, 182 and 200). The additional tram noise levels were included from a Melbourne assessment.

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TABLE 8 CRITERIA FOR NOISE WITHIN BUILDINGS: LIVING ROOMS AND BEDROOMS

TYPE OF AREA	MAXIMUM PERMISSIBLE L_{10} SOUND LEVELS IN dB(A)	
	Day	Night
Livingrooms and bedrooms in		
(a) Country areas	40	30
(b) Suburban areas away from main traffic routes	45	35
(c) Busy urban areas	50	35
Upper limit in buildings where speech intelligibility is required	55	

NOTE: These data are from the Wilson 'Final Report on Noise' (ref 37, paras 576, 577, p 134).

APPENDIX A Australian Standards for Architectural Acoustics

Group 1: Basic criteria for noise control in buildings

AS 2107-1987	Recommended design sound levels and reverberation times for building interiors
AS 2822-1985	Method of assessing and predicting speech privacy and speech intelligibility

Group 2: Methods of testing building materials for acoustic absorption or transmission loss

AS 1045-1988	Measurement of sound absorption in a reverberant room
AS 1191-1985	Method for laboratory measurement of airborne sound transmission loss of building partitions
AS 1276-1979	Methods for determination of sound transmission class and noise isolation class of building partitions
AS 1277-1983	Measurement procedures for ducted silencers
AS 1935-1976	Method for measurement of normal incidence sound absorption coefficient and specific normal acoustic impedance of acoustic materials by the tube method

Group 3: Methods of testing buildings in situ for acoustic absorption or transmission loss

AS 1807-	Clean rooms, work stations and safety cabinets: methods of test
1807.16-	Determination of sound levels in clean rooms
1807.20-	Determination of sound levels at installed workstations and safety cabinets
AS 2253-1979	Methods for field measurement of the reduction of airborne sound transmission in buildings
AS 2460-1981	Method for the measurement of reverberation time in enclosures
AS 2499-1981	Method for laboratory measurement of airborne sound attenuation of ceilings (two-room method)

Group 4: Standards concerned with the impact of environmental noise, in siting buildings, etc

AS 1055-	Description and measurement of environmental noise
1055.1-1989	General procedures
1055.2-1989	Application to specific situations
1055.3-1989	Acquisition of data pertinent to land use
AS 2021-1985	Aircraft noise intrusion: building siting and construction
AS 2363-1990	Assessment of noise from helicopter landing sites
AS 2436-1981	Guide to noise control on construction, maintenance and demolition sites
AS 2702-1984	Methods for the measurement of road traffic noise
AS 3671-1989	Road traffic noise intrusion: building siting and construction

APPENDIX B Australian Standards describing Methods of testing Equipment, Machinery, Plant and Vehicles for Noise Emission

AS 1081 Measurement of airborne noise emitted by rotating electrical machinery

1081.1-1990	Engineering method for free-field conditions over a reflective plane (from ISO 1680, 1681)
1081.2-1990	Survey method (from ISO 1680, 1682)

AS 1173.2-1986 Receivers for TV broadcast transmissions: electrical and acoustical measurements at audio frequencies

AS 1217 Determination of sound power levels of noise sources

1217.1-1985	Guidelines for the use of basic standards for the preparation of noise test codes
1217.2-1985	Precision methods for broad band sources in reverberation rooms
1217.3-1985	Precision methods for discrete frequency and narrow band sources in reverberation rooms
1217.4-1985	Engineering methods for special reverberation test rooms
1217.5-1985	Engineering methods for free-field conditions over a reflecting plane
1217.6-1985	Precision methods for anechoic and hemi-anechoic rooms
1217.7-1985	Survey method

AS 1359.0-1989 Rotating electrical machines: introduction and list of parts

1359.1-1987	Rotating electrical machines:definitions
1359.50-1983	Vibration limits (see also AS 2625.1 and AS 2625.2)
1359.51-1986	Noise level limits

AS 1861 Air-conditioning units: methods of assessing and rating performance

1861.1-1988	Refrigerated room air-conditioners

AS 1948-1987 Measurement of airborne noise on board vessels and offshore platforms

AS 1949-1988 Measurement of airborne noise emitted by vessels in waterways, ports and harbours

AS 2012 Measurement of airborne noise emitted by earth-moving machinery and agricultural tractors -- stationary test condition

2012.1-1990	Determination of compliance with limits for exterior noise
2012.2-1990	Operator's position

AS 2221 Methods for measurements of airborne souns emitted by compressor units including prime movers and by pneumatic tools and machines

2221.1-1979	Engineering method for measurement of airborne sound emitted by compressor/prime mover units intended for outdoor use
2221.2-1979	Engineering method for measurement of airborne sound emitted by pneumatic tools and machines

AS 2240-1979 Methods of measurement of the sound emitted by motor vehicles

AS 2254-1988 Recommended noise levels for various areas of occupancy in vessels and offshore platforms

AS 2374.6-1982 Power transformers: sound levels (based on IEC 551)

AS 2377-1980 Methods for the measurement of airborne sound from railbound vehicles

Appendix B (concl)

AS 2726-1984	Chainsaws -- safety requirements (incl noise and vibration)
AS 2991	Method for the determination of airborne noise emitted by household and similar electrical appliances
2991.1-1987	General requirements
2991.2-1991	Particular requirements for dish washers
AS 3534-1988	Method for measurement of airborne noise emitted by powered lawn mowers, edge and brush cutters, and string trimmers
AS 3713-1989	Industrial trucks -- noise measurement (see also AS 1763, AS 2359.1)
AS 3755-1990	Measurement of airborne noise emitted by computer and business equipment (ISO 7779-1988)
AS 3756-1990	Measurement of high frequency noise emitted by computer and business equipment (ISO 9295-1988)
AS 3757-1990	Declared noise emission values of computers and business equipment (ISO 9296-1988)
AS 3781-1990	Noise labelling of machinery and equipment (ISO 4871-1984)
AS 3782	Statistical methods for determining and verifying stated noise emission values of machinery and equipment
3782.1-1990	General considerations and definitions (ISO 7475-1:1985)
3782.2-1990	Methods for stated values for individual machines (ISO 7475-2:1985)
3782.3-1990	Simple (transition) method for stated values for batches of machines (ISO 7574-3:1985)
3782.4-1990	Methods for stated values for batches of machines (ISO 7574-4:1985)

and, in addition,

ISO 3981-1978	Procedure for describing aircraft noise heard on the ground

APPENDIX C Case history no. 1: Interference to speech intelligibility in a meeting hall

The following Table summarizes the original noise measurements

DETAILS OF NOISE, MEASUREMENT LOCATION	STATISTICS OF MEASURED SOUND LEVELS IN dB (note 1)					ESTIMATED LEVELS (note 2)	
	n	Mean	Std error	Max	Min	dB(A)	SIL4
PASSBY NOISE MEASURED AT KERB (7,5 m from 20 m wide roadway centre)							
- Background noise	6	67,7	2,704	76	62	60	52
- Groups of passing motorcars	6	80,7	2,290	86	74	75	66
- Individual large vehicles	3	98,0	3,512	105	94	88	79
- Trams on paved ballast track (at 25 to 35 km/h)	10	90,8	0,629	94	88	87	81
- Trams on concrete track (25-35 km/h)	16	95,1	1,167	105	90	91	85
PASSBY NOISE OUTSIDE HALL DOOR (15 m from roadway centre)							
- Trams on concrete track (25-35 km/h)	7	89,6	1,020	94	86	86	80
PASSBY NOISE INSIDE HALL (approx 18 m from roadway centre)							
- Background noise	-	54,0	-	-	-	54	46
- Groups of passing motorcars	-	-	-	74	70	64 to 68	55 to 59
- Individual large vehicles	2	77,5	1,500	79	76	72	63
- Trams on concrete track	5	72,2	1,210	76	69,5	69	63
- Est of trams on paved ballast track		68	-	-	-	65	59

NOTES: (1) Sound levels below 55 were measured in dB(A), between 55 and 85 in dB(B), and above 85 in dB(C).

(2) The corresponding estimated levels in dB(A) and speech interference levels (SIL4) have been added for comparison, and were derived according to the following conversion factors (from octave band analyses of similar noises):

(a) Background dB(C)-dB(A) = 15; dB(B)-dB(A)=8; dB(A)-SIL4=8
(b) Motor vehicles dB(C)-dB(A) = 10; dB(B)-dB(A)=6; dB(A)-SIL4=9
(c) Trams dB(C)-dB(A) = 4; dB(B)-dB(A)=3; dB(A)-SIL4=6

APPENDIX D Case history no. 1: Calculated variation of vehicle passby noise with time

As part of case history no. 1 (paras 56 to 58), a calculation of vehicle passby noise level decay from its maximum at the minimum distance to lower levels after various intervals of time was made in order to estimate the length of time by which any passby noise level might exceed some lower reference or tolerance noise level. For the purposes of this calculation the minimum source to measuring point distance was taken as 18m (as for the noise measurements made inside the meeting hall), average adopted vehicle speeds were 50 km/h (13,9 m/s) for cars and trucks, and 30 km/h (8,3 m/s) for trams, while the incremental decay of passby noise level with doubling of the distance was taken as 4,3 dB for motor vehicles (from Alfredson, 1974, ref 23) and 4,0 dB for trams. This increment can be of any value between 6 dB for point sources and 3 dB for long line sources. Since trams are slightly more of a line source than motor vehicles, an increment of 4,0 dB was adopted.

TABLE OF VEHICLE PASSBY NOISE DECAY AT 1 s INTERVALS

TIME INTERVAL IN s	DISTANCE IN m FROM SOURCE TO MEASURING POINT		PASSBY NOISE LEVEL DECAY IN dB FOR	
	Motor vehicles, 50 km/h	Trams 30 km/h	Motor vehicles	Trams
0	18,0	18,0	0 (max)	0 (max)
1	22,7	19,8	- 1,3	- 0,6
2	33,1	24,6	- 3,5	- 1,8
3	45,4	30,8	- 5,7	- 3,1
4	58,4	37,9	- 7,3	- 4,3
5	71,8	45,4	- 8,6	- 5,3
6	85,3	53,1	- 9,7	- 6,2
7	99,0	61,0	- 10,6	- 7,0
8	112,6	69,1	- 11,4	- 7,8
9		77,1		- 8,4
10		85,2		- 9,0
11		93,4		- 9,5
12		101,6		- 10,0
13		109,8		- 10,4
14		118,1		- 10,9
15		126,3		- 11,2
16		134,5		- 11,6
17		142,8		- 12,0
18		151,1		- 12,3

APPENDIX E Background noise levels in 'quieter' rooms proposed for audiometric testing (case history no. 5)

LOCATION, ROOM CONDITIONS	OCTAVE BAND LEVELS IN dB OF ROOM BACKGROUND NOISE AT FREQ (Hz)						
	125	250	500	1000	2000	4000	8000
AS 1269-1988 lowest maxima for audiometric testing	52	35	15	14	29	36	28
Room A: air-cond ON, ext noise ON	38	29	28	32	31	24	16
" " ON " " OFF (calc)	37	27	28	32	31	24	16
" " OFF " " ON	32	26	16	12	8	8	9
" " OFF " " OFF	27	18	13	10	8	8	7
Room B: general background in room	60	58	52	47	43	37	29
I/s audiometric booth, fans on (1)	41	41	22	20	14	15	16
" " " " " " (2)	51	38	19	16	11	11	12
" " " " " " (3)	45	32	17	13	10	11	12
" " " " " " (4)	34	27	15	12	13	15	16
Room C: general background in room	45	36	29	27	25	18	14

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APPENDIX F Case history no. 6: Dust collector fan noise in a factory building

FACTORY INTERIOR ENVIRONMENTAL NOISE	DUST COLLECTOR FAN NOISE OCTAVE BAND LEVELS IN dB AT FREQUENCIES (Hz)										OVERALL SOUND LEVELS		
	31,5	63	125	250	500	1k	2k	4k	8k	16k	dB(A)	SIL4	NR
Initial installation (w/o outlet muffler)	85	78	85	85	86	87	82	76	67	55	90	83	88
Modified with outlet muffler	77	81	80	76	77	76	71	66	57	54	80	72	76
The same after 6 mths	77	79	80	75	74	75	70	66	58	49	78	71	75
AS 1469 75 NR curve	106	95	87	82	78	75	73	71	69	-	82	74	75
70 NR curve	103	91	83	77	73	70	68	66	64	-	77	69	70

NOTE: From these tests it was concluded that a sound level of 70 NR, equivalent to 77 dB(A), represents the very maximum tolerable level for exposure to continuous background noise.

APPENDIX G Case history no. 9: Noise inside subway trains and trams at normal service speeds

METROPOLITAN AREA	OPERATING CONDITIONS	TYPICAL INTERIOR VEHICLE SOUND LEVELS IN dB(A)
Boston, USA	Subway trains: Harvard line 'Orange' line Quincy line Subway PCC trams	91-101 87-90 75-80 92-97
Brussels, Belg	Subway trams, 40 km/h	78-85
Chicago, USA	Rapid transit trains: subway ballast tracks " " " : concrete tracks, 50-80/110 km/h " " " : steel 'El' tracks	80-86 82-90/94-97 90-97
Frankfurt, Germ	U-trains	78-85
London	Subway 'tube' trains	80-88
Melbourne, Aust	City UG rail loop: smooth/rough wheels Ground level, suburban ballast track (average wheels) Reverb tunnel, conc track (average wheels)	78-82/85-88 78-82 90-94
Montreal, Canada	Subway rubber-tired Metro trains	78-86
Munich, Germ	Subway: S-bahn U-bahn	82-84 78-87
New York, USA	Subway trains: some on old (rigid) ballast tracks	92-102
Philadelphia, USA	Subway: older system Lindenwold line	90-97 80-85
San Francisco, USA	BART subway: 50-80 km/h/120-130 km/h	72-78/80-84
Sydney, Aust	City UG trains: resilient/rigid tracks	83-87/93-98
Toronto, Canada	Metro subways, 50-80 km/h	80-86

NOTE: All train and tram noise was measured inside vehicles, in subway or UG tunnels unless specifically stated otherwise (eg, Chicago rapid transit on 'El' (elevated) tracks), and at normal service speeds of 50-80 km/h unless otherwise specified. Measurements overseas were made in 1973.

APPENDIX H Case history no. 11: Octave band frequency spectra at the operator's ear position, of typically noisy equipment and machinery

NOISE FROM	OCTAVE BAND NOISE LEVELS IN dB AT OPERATOR'S EAR POSITION FOR FREQUENCY BANDS (Hz)										OVERALL SOUND LEVELS		
	31,5	63	125	250	500	1k	2k	4k	8k	16k	dB(B)	dB(A)	SIL4
Pneu paving breaker	85	94	93	99	106	106	107	105	108	98	113	113	106
Dressing grinding wheel	79	79	82	84	87	96	94	104	109	104	108	110	95
Circular saws:													
- lge, fixed (+amb)	78	85	86	95	88	98	100	97	92	80	104	105	96
(ambient)	78	85	83	87	78	73	70	66	61	-	88	82	72
- lge at 20 m(+amb)	70	76	76	76	75	85	78	80	76	66	88	87	80
(ambient)	70	71	73	72	69	65	59	55	46	-	75	70	62
- lge at 1m (-amb)	-	-	83	94	88	98	100	97	92	80	104	105	96
- lge at 20m (-amb)	-	74	73	74	74	85	78	80	76	66	88	87	79
- small, portable	64	70	72	72	82	78	82	89	97	96	96	98	83
Planer, idling	69	80	78	85	93	103	83	76	66	61	104	103	89
" , operating	72	82	78	86	93	95	91	83	72	62	98	98	90
Moulder, blunt blades	77	83	88	89	87	99	97	89	83	80	102	102	93
" , sharp blades	75	82	88	92	89	91	90	89	86	83	97	96	90
Truck diesel engine at smoke test speed	88	96	95	93	106	93	91	83	76	69	107	104	94
Foundry furnace burners:													
- lge, shielded	85	93	98	89	88	84	72	68	58	55	97	89	87
- small, unshielded	94	96	103	108	100	85	77	75	78	80	108	102	84
Pneu impact wrench	82	82	86	83	86	88	94	100	100	98	103	104	92
" " "	82	84	85	87	90	93	96	98	101	92	103	104	94
Kryter 8-h max, 90dB	106	99	92	88	84	82	81	81	83	79	96	90	82
Kryter 8-h max, 85dB	101	94	87	83	79	77	76	76	78	74	91	85	77
Fr 'neg risk' (orig)	100	90	85	80	78	76	74	72	70	68	88	82	75
Fr 'neg risk',80dB(B)	92	82	77	72	70	68	66	64	62	60	80	74	67

APPENDIX I Case history no. 7: Intrusive air-conditioner fan noise in a residential area

BAND CENTRE FREQUENCY (Hz)	FREQUENCY ANALYSES OF FAN NOISE: BAND LEVELS IN dB						
	METAL-BLADED FANS				PLASTICS-BLADED FANS		
	OCTAVE	⅓-OCTAVE	⅓-OCTAVE dB(A)	ADJUSTED ⅓-OCTAVE	OCTAVE	⅓-OCTAVE	⅓-OCTAVE dB(A)
20	-	22	-	-	-	47	-
25		30	-	-		46	1
31,5	50	40	1	1	51	47	8
40		49	14	14		46	11
50		55	25	25		45	15
63	69	69 T	43	49	50	46	20
80		54	32	32		46	24
100		57	38	38		57	38
125	66	65 T	49	54	62	58	42
160		53	40	40		57	44
200		53	42	42		55	44
250	57	52	43	43	60	56	47
315		51	44	44		55	48
400		49	44	44		53	48
500	54	48	45	45	58	50	47
630		49	47	47		49	47
800		47	46	46		47	46
1 000	51	45	45	45	52	47	47
1 250		45	46	46		45	46
1 600		42	43	43		44	45
2 000	46	41	42	42	47	42	43
2 500		38	39	39		40	41
3 150		38	39	39		38	39
4 000	40	35	36	36	41	36	37
5 000		32	32	32		32	32
6 300		31	31	31		31	31
8 000	34	29	28	28	35	30	29
10 000		26	24	24		28	26
12 500		23	19	19		27	23
16 000	25	20	13	13	29	25	18
20 000		15	6	6		19	10
Overall dB(A)	57	57	57	59	58	58	58

NOTE: 'T' in col 3 indicates a tonal component

APPENDIX J Case history no. 14: measurements of impulsive type noise

The investigations described here were begun as a result of Dr P V Brüel's comments made in his lecture on Sound Level Meters given in Melbourne on 92 Sep 16, in which he suggested that we need to measure maximum sound levels in dB_{pk} rather than the normal and averaged level in dB_{rms} if we are to satisfactorily correlate the exposure levels of intense, and particularly of impulsive type noise with human hearing damage risk.

Case history no. 14, briefly described here, concerns the measurements made recently of noise from a 2-stroke petrol-engined lawn mower and from a mechanical typewriter. The measuring instruments used comprised a Sound Level Meter (with F, S and I time-weightings, and capable of measuring sound levels in both dB_{rms} and dB_{pk}), Octave/one-third octave Filter Set, Printer, and Graphic Level Recorder. For all measurements, the instruments' calibration and batteries were checked and found satisfactory before and after each test. In each case, sound levels were measured at a distance from the noise source equivalent to that of the operator's ear position, at 1,6 m above ground with the mower, and at 250 to 300 mm with the typewriter. In addition, ambient noise levels were measured so that, if necessary, adjustments could be made to the mower or typewriter noise levels if ambient levels were insufficiently low.

Output from the octave and one-third octave analyses of the lawn mower noise (measured out of doors), and one-third octave analyses of the mechanical typewriter noise (measured indoors in a room of low reverberation time of around 0,35 s) provided the L_{eq} and L_{max} in dB_{rms}, and $L_{max.pk}$ in dB_{pk} for each frequency band time period, the durations at each frequency band being controlled by the programmed Sound Level Meter. For the typewriter noise, several additional sound level measurements were made with a graphic level recorder as a check on the frequency analyses.

The results of these tests are tabulated below. Their important features are that, for each measurement period, the L_{eq} value gives the averaged sound level in dB_{rms}, the L_{max} gives the maximum in dB_{rms}, and the $L_{max.pk}$ the corresponding maximum in dB_{pk}. Of these, $L_{max.pk} - L_{max}$ gives the Crest Factor, the amount by which, for a given waveform, dB_{pk} exceeds dB_{rms}, and a measure of the impulsiveness of the sound under test; $L_{max} - L_{eq}$ gives a measure of the variability of the sound level within each measurement period, and an indication of its intermittent characteristic; while $L_{max.pk} - L_{eq}$ indicates the amount by which $L_{max.pk}$ exceeds the normally measured averaged level (L_{eq}) in dB_{rms}, or by which the difference $L_{max.pk} - L_{eq}$ exceeds the Crest Factor, so including also a measure of the variability of the instantaneous level of the sound under test.

Examination of the tabulated results of these tests showed that the mower noise, with Crest Factors of 4 to 20 dB and $L_{max.pk} - L_{eq}$ of 5 to 21 dB, was neither highly impulsive, nor greatly intermittent (except at lower frequencies of 20, 31,5 and 80 Hz in the one-third octave analysis). By contrast, the typewriter noise, with Crest Factors of 5 to 27 dB and $L_{max.pk} - L_{eq}$ of 12 to 31 dB, was, as could be expected, both highly impulsive and intermittent, the impulsiveness being especially prominent at the higher frequencies, from 1000 Hz upwards.

This investigation has thus shown that, for the measurement of any sound levels, particularly of impulsive and intermittent sounds, and where hearing damage risk is a serious concern, the measurement (over some shorter or longer period of time) of a sound's L_{eq} and L_{max} in dB_{rms}, and $L_{max.pk}$ in dB_{pk} provides a good and useful measure of that sound's intensity, giving considerably more information than L_{eq} alone. This could well become and be made a standard procedure for the measurement of sound levels where there is any suspicion of hearing damage risk, with $L_{max.pk}$ given first priority in the assessment of the noise exposure level.

TABLE J-1 Octave band analysis of 2-stroke petrol-engined lawn mower noise

OCTAVE FREQUENCY (Hz)	LAWN MOWER NOISE (mean of 3 tests)					AMBIENT NOISE				
	dB_{rms}		dB_{pk}	Crest Factor	$L_{max.pk} - L_{eq}$	dB_{rms}		dB_{pk}	Crest Factor	$L_{max.pk} - L_{eq}$
	L_{eq}	L_{max}	$L_{max.pk}$			L_{eq}	L_{max}	$L_{max.pk}$		
31,5	78	80	87	7	9	54	59	66	7	12
63	88	89	95	6	7	55	58	66	8	11
125	86	87	98	11	12	46	48	58	10	12
250	90	91	103	12	13	40	42	52	10	12
500	86	86	98	12	12	37	38	49	11	12
1 000	82	83	96	13	14	38	39	52	13	14
2 000	83	84	98	14	15	35	38	50	12	15
4 000	76	77	94	17	18	31	33	45	12	14
8 000	70	72	91	19	21	23	27	41	14	18
16 000	62	63	83	20	21	13	14	27	13	14
dB	95	95	109	14	14	59	61	72	11	13
dB(B)	93	93	105	12	12	50	53	62	9	12
dB(A)	89	90	103	13	14	43	44	56	12	13

TABLE J-2 One-third octave analysis of 2-stroke petrol-engined lawn mower noise

$\frac{1}{3}$-OCT FREQUENCY (Hz)	LAWN MOWER NOISE (mean of 2 tests)					AMBIENT NOISE				
	dB_{rms}		dB_{pk}	Crest Factor	$L_{max.pk} - L_{eq}$	dB_{rms}		dB_{pk}	Crest Factor	$L_{max.pk} - L_{eq}$
	L_{eq}	L_{max}	$L_{max.pk}$			L_{eq}	L_{max}	$L_{max.pk}$		
20	62	69	75	6	13	54	59	65	4	9
25	64	68	74	6	10	51	59	64	5	13
31,5	63	69	75	6	12	51	62	68	6	17
40	73	75	81	6	8	56	62	68	6	12
50	87	88	92	4	5	55	60	68	8	13
63	78	79	88	9	10	50	54	62	8	12
80	75	79	85	6	10	48	52	59	7	11
100	83	84	91	7	8	56	60	67	7	11
125	80	82	90	8	10	47	50	59	9	12
160	83	85	93	8	10	47	49	58	9	11
200	84	85	94	9	10	44	50	59	9	15
250	87	88	98	10	11	42	44	55	11	13
315	81	83	93	10	12	40	42	50	8	10
400	79	81	91	10	12	38	40	50	10	12
500	82	83	93	10	11	38	40	50	10	12
630	82	83	94	11	12	38	40	50	10	12
800	79	80	90	10	11	40	41	54	13	14
1 000	77	78	88	10	11	40	40	52	12	12
1 250	78	79	91	12	13	39	41	52	11	13
1 600	80	80	92	12	12	39	41	54	13	15
2 000	79	80	92	12	13	32	32	45	13	13
2 500	74	75	87	12	13	28	29	42	13	14
3 150	73	74	89	15	16	32	36	51	15	19
4 000	72	73	89	16	17	23	25	37	12	14
5 000	68	69	85	16	17	23	28	43	15	20
6 300	67	68	86	18	21	25	29	42	13	17
8 000	65	67	86	19	21	15	18	29	11	14
10 000	62	64	83	19	21	12	13	24	11	12
12 500	58	59	78	19	20	9	10	21	11	12
16 000	55	56	73	17	18	6	7	17	10	11
20 000	48	49	64	15	16	6	7	16	9	10
dB	95	96	109	13	14	62	64	73	9	11

TABLE J-3 One-third octave analysis of mechanical typewriter noise

$\frac{1}{3}$-OCT FREQUENCY (Hz)	TYPEWRITER NOISE (mean of 2 tests)					AMBIENT ROOM NOISE				
	dB_{rms}		dB_{pk}	Crest Factor	$L_{max.pk} - L_{eq}$	dB_{rms}		dB_{pk}	Crest Factor	$L_{max.pk} - L_{eq}$
	L_{eq}	L_{max}	$L_{max.pk}$			L_{eq}	L_{max}	$L_{max.pk}$		
20	49	64	69	5	20	43	51	56	5	13
25	54	65	71	6	17	43	51	57	6	14
31,5	53	59	65	6	12	44	50	55	5	11
40	52	57	63	6	11	40	44	51	7	11
50	55	61	68	7	13	38	44	51	7	13
63	57	61	69	8	12	36	42	48	6	12
80	64	69	77	8	13	37	40	50	10	13
100	61	70	80	10	19	50	53	61	8	11
125	58	61	70	9	12	37	40	48	8	11
160	57	61	72	11	15	38	40	50	10	12
200	53	57	67	10	14	37	39	46	7	9
250	48	52	63	11	15	31	36	44	8	13
315	45	53	65	12	20	27	29	39	10	12
400	48	54	67	13	19	23	25	35	10	12
500	48	53	67	14	19	20	22	32	10	12
630	47	62	77	15	30	21	22	33	11	12
800	52	59	75	16	23	22	24	34	10	12
1 000	58	62	78	16	20	22	23	34	11	12
1 250	56	60	77	17	21	22	23	33	10	11
1 600	60	65	82	17	22	20	21	32	11	12
2 000	61	68	88	20	27	26	30	41	11	15
2 500	65	69	92	23	27	22	26	40	14	18
3 150	65	70	92	22	27	19	22	34	12	15
4 000	66	70	94	24	28	15	16	29	13	14
5 000	64	71	95	24	31	13	14	26	12	13
6 300	68	72	99	27	31	13	17	31	14	18
8 000	65	70	94	24	29	10	12	25	13	15
10 000	68	73	98	25	30	11	11	26	15	15
12 500	64	69	94	25	30	11	12	25	13	14
16 000	59	64	90	26	31	-	11	23	12	-
20 000	55	59	85	26	30	-	10	27	17	-
dB	74	80	105	25	31	52	54	63	9	11
dB(B)	74	79	102	23	28	46	49	57	8	11
dB(A)	75	80	104	24	29	36	39	49	10	13
Meas dB(A)	72	84	112	28	40	36	40	50	10	14
dB	-	-	-	-	-	55	58	67	9	12

EFFECT OF ATMOSPHERIC FLUCTUATIONS ON IMPULSE SOUND PROPAGATION

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Abstract

Outdoor sound transmissions often vary in time because of changes in the atmospheric conditions, especially in the presence of winds. While there are large scale effects such as wind and temperature gradients producing a shadow zone into which little sound propagates, there are also more localized fluctuations in levels caused by turbulence in the atmosphere. An understanding of these meteorological variations is useful for interpreting acoustical data. On the other hand, acoustic measurements can provide insight into the atmospheric mechanisms producing the fluctuations.

A seven microphone system, in conjunction with an impulse source, has been used to investigate how atmospheric fluctuations alter the pulse waveform. One microphone located close to the source was used as a reference to compensate for minor changes in level between shots. The other microphones were located on a circle, concentric with the axis of symmetry of the source, at distances out to 16m from the source. Measurements taken indoors in still air indicate that the waveforms closely agree from shot-to-shot and between microphones, although allowance must be made for ground reflections for the lower microphones. Outdoors, the range of waveforms recorded include changed peak-heights and wider or narrower peaks. Individual impulses have been categorized according to the change in waveform from the average value and correlations sought with meteorological data acquired while the propagation occurred. Analysis techniques and the implications of atmospheric fluctuations on propagating sound measurements will be outlined.

Introduction

Continuous wave propagation measurements have the characteristic that they represent an integrated effect over both space and time. Also, it is difficult to separate the effects of ground reflections from the behaviour of the direct signal. Generally the measurements involve averaging over at least a several minute data collection period[1,2], so any short term fluctuations are removed and it is hard to relate changes to anything other than the mean meteorological data. As an alternative, because of its short duration, impulse sounds[3] experience essentially a "snapshot" of the atmospheric conditions. In principle, changes to the waveform between successive impulses should relate to the changes in the meteorological conditions between shots. As it is impossible to determine fully these conditions along the propagation path, an estimate must still be made from a few sample measurements, however, time averaging effects are markedly reduced. As well as acoustic data, the impulse also carries a piece of meteorological data: the delay between creation and reception of the pulse is related to the mean sound speed, and therefore wind speed, along the propagation path. Thus a reasonable correlation is expected between pulse delay and wind speed and this offers a potential test of whether meaningful meteorological data have been acquired.

Experimental Arrangement

When a shot-shell primer is discharged down a 1m long tube to form an impulse source, it possesses conical symmetry about the tube axis[4]. Thus microphones arranged in a circle concentric with this axis, as indicated in Fig.1, show little variation in pulse waveform between positions for a given impulse. Furthermore, if shot-to-shot variations of the source are allowed for by adjusting peak heights with respect to those from a reference microphone kept close to the source, there is little shot dependence in the corrected pulse shapes recorded at larger distances. This is certainly true indoors, where conditions of minimal wind and temperature gradients exist. To demonstrate this, Fig.2(a) shows plus and minus a standard deviation from the average of 10 waveforms recorded indoors at a single microphone, located 4m from the source. A similar plot, showing the deviation from the mean of the 10 shots recorded at six microphones located on the cone, ie. a total of 60 waveforms, is given as Fig.2(b).

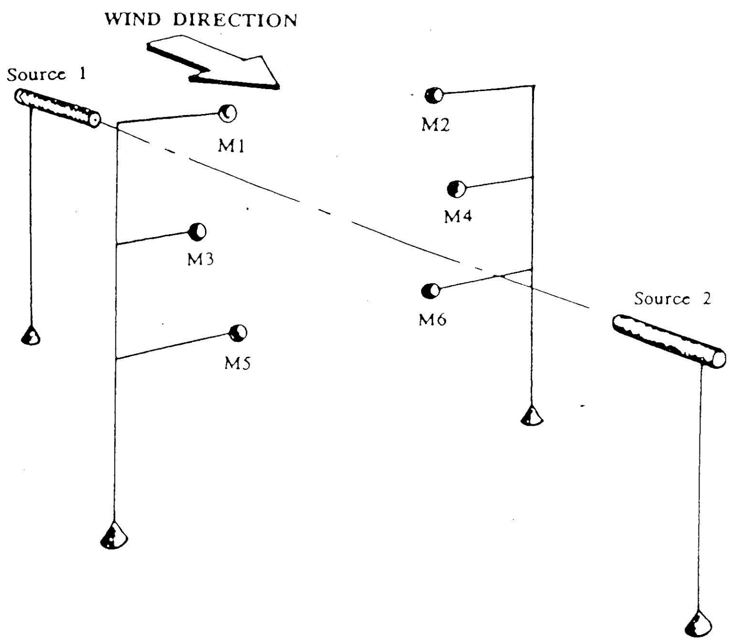

Fig.1: Measurement Geometry Showing Six Microphones Positioned on the Cone of Symmetry about the Source Axis.

The previous data were obtained by locating the microphones on a 1m diameter circle centered 2m above the ground and 4m along the source axis. When a similar geometry was used outside, with the wind blowing along the source-microphone axis, the waveform variations were significantly larger, as is apparent in Fig.2(c) and (d). Extending the previous geometry to an 8m and then a 16m source-receiver distance, with corresponding 2m and 4m diameter circles but retaining the axis 2m above the ground, produced much greater deviations. Examples are shown in Fig.2(e) and (f) for the 16m case, where waveforms for the two lower microphones have been excluded from the average as the ground reflected component is sufficiently merged with the direct to cause major alterations of the waveshape.

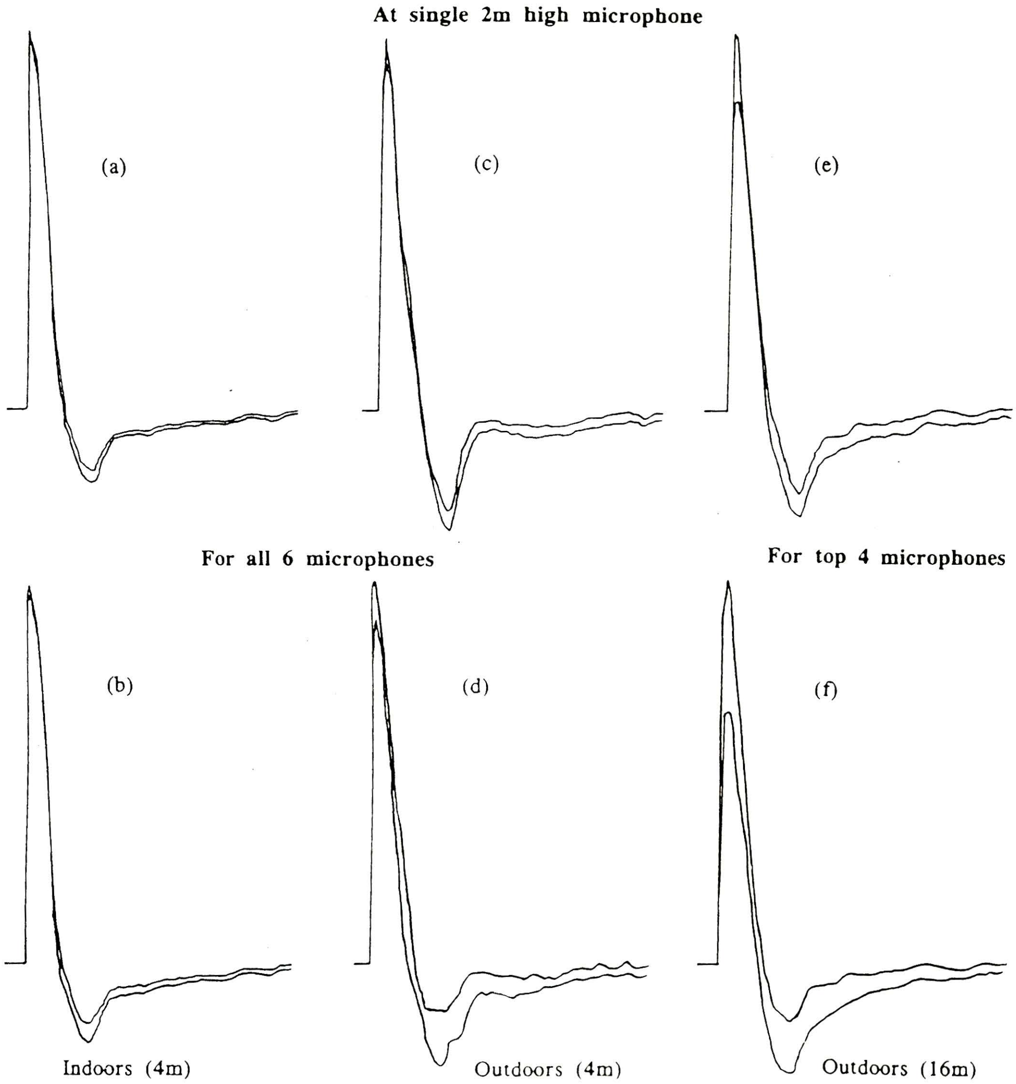

Fig.2: Repeatability of Pulse Shapes
(Contours represent ± a standard deviation around the average)

Wind and temperature data were collected at 1 second intervals from each of four Alnor thermo-anemometers located along the propagation path. Three were placed near the microphones, one at each of the three heights involved, while the fourth was positioned half-way along the propagation path at the source height. Readings were continuously stored in a computer, which was triggered by the impulse to output the mean of the three readings around the trigger point and also the average and standard deviation of ninety readings around the trigger instant. Thus both a short and longer term wind speed average were gathered in conjunction with each impulse. Unfortunately, wind direction was not automatically recorded, instead it was estimated from the direction of "flags" positioned around the measurement site. Care was taken to capture impulses when the wind direction was essentially along the propagation axis - with one set obtained when propagating alternatively up and down wind - which is why a second source is shown in Fig.1.

Results From Pulse Shape Considerations

Outdoors, at say 16m, the pulse waveforms displayed a range of shapes between shots and between similar microphone positions. When the differences between the individual and the average peak height, Δ-height, at a particular location are plotted and compared with the difference involving the total area of the waveform, Δ-area, as shown in Fig.3, it is apparent that the overall correlation is poor. An enhanced peak may be associated with a smaller than usual area or a larger one, so the waveforms were sub-divided into six broad types, as indicated in Table 1.

Fig.3: Δ-height and Δ-area of Consecutive Pulses

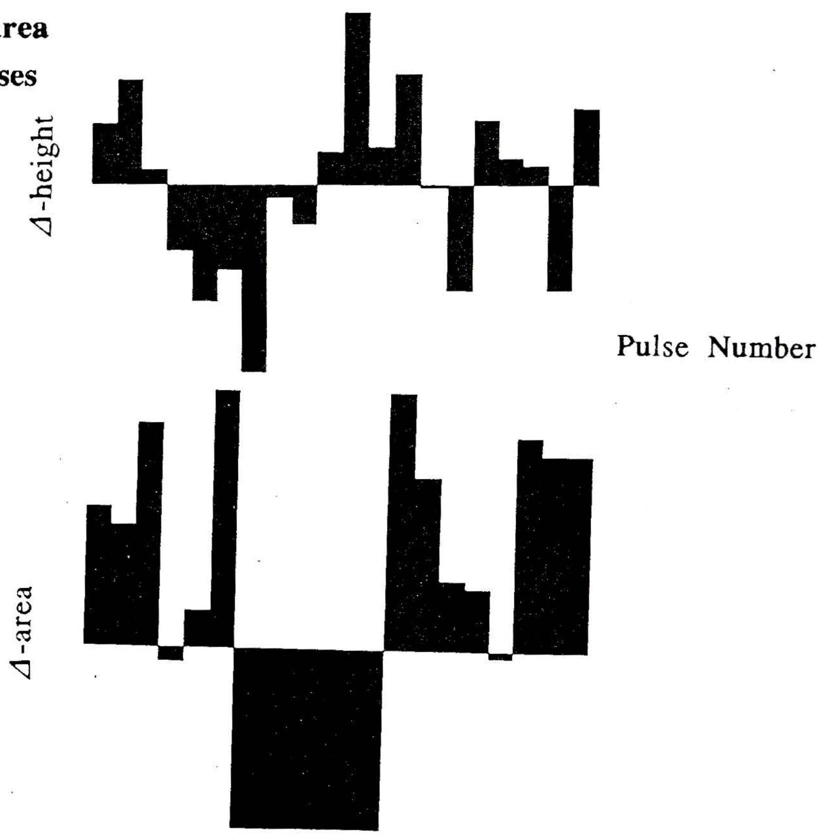

Table 1

Distribution of pulses at the three microphone positions, according to their height and area. Total of 264 pulses.

Δ-height	increase		equal		decrease	
Δ-area	+	−	+	−	+	−
Top	18	10	17	23	5	15
Middle	15	7	19	21	2	24
Lower	30	2	10	16	3	27

1) Analysis indicated that the effects observed at the left and right microphones at the same height were essentially identical. In what follows, no distinction will be made between results from such microphone pairs.

2) Comparison of the ensemble averaged waveforms indicate no significant difference in the peak value for up and down wind propagation out to 16m, with only minor differences in the shapes of the rarefactions.

3) From Table 1 it is apparent that a reduced peak height is rarely associated with an increased area, occurring only in 4% of cases, which is much lower than a random distribution would predict.

4) The predominant mechanism for pulse shape changes for the lower microphones is the reflected pulse combining with the direct, the degree of cancellation depending on the relative delay of the reflected compared with the direct pulse and the amount of inversion of the reflected component. Consequently, reduced height and area pulses are expected when inversion of the reflected pulse occurs while large pulse amplitudes and increased areas suggest little inversion. These cases occur with about equal probability. Pulses with a greater than average peak value and a decreased area only occurred in 3% of cases . This percentage increases with height.

5) At all measuring heights, an unchanged peak value is equally associated with smaller or larger areas. At the higher height there is a tendency for all categories to become equally populated.

Consideration of Meteorological Information

As a major objective of this research is to link meteorological fluctuations in the atmosphere with corresponding changes in the acoustic behaviour, it is important to establish that the meteorological data are meaningful. As indicated earlier, the delay experienced by an impulse is expected to correlate strongly with the wind speed.

A careful check was kept on the flight time of the impulses. As the reference microphone was only a metre from the source, the delay between its pulse and the arrival at the other microphones should give a direct indication of any fluctuation in the average wind speed experienced by the particular pulse. Although care was taken in the set-up to ensure that all microphones were located at the same distance from the source, an uncertainty of perhaps several centimetres remained in the 16m distance. Microphone supports were stayed by cords to restrict their movement as the wind gusted.

While there are differences between the mean delays at the three microphone heights, as shown in Table 2, these are within the uncertainties of locating the microphones. However, the associated uncertainties indicate that the spread in the delays increase significantly with height. In Table 2, $\bar{U}$ represents the mean value of the anemometer readings over the total time of some 45 x 90 = 4050 second recordings, while $\bar{u}$ is the average using the 45 x 3 = 135 second readings taken at the time of the pulses. These two averages closely agree, showing an increase in wind speed with height. The two anemometers at a 2m height gave essentially identical results for $\bar{U}$ and $\bar{u}$. If long term averages are at all meaningful, one might have expected the spread in delays at a given height, as measured by the quoted uncertainty, to be related to the corresponding RMS fluctuation parameter, however, this does not appear to be the case. The fact that $\bar{U}$ and $\bar{u}$ both increase with height, yet their RMS difference is essentially constant suggests that they are measuring the same quantity and that perhaps our sampling time of 1 second is too large.

Table 2
Comparison of Pulse Delay with Meteorological Data at the Microphone Heights

	Mean Delay	Av. Wind Speed 90 samples $\bar{U}$	Av. Wind Speed 3 samples $\bar{u}$	Fluctuation $(u - U)_{rms}$
Top	-0.30±0.13	3.8±0.6	3.7±0.3	0.64
Middle	-0.28±0.08	3.5±0.9	3.4±0.6	0.72
Bottom	-0.35±0.05	2.7±0.7	2.7±0.5	0.63

All delays in ms, all wind speeds in ms^{-1}.

If individual pulse delays are plotted against the corresponding short term wind speed, u, no significant correlation is obtained, Fig.4. The temperature measured at each location only changed by about 2^{o} during the measurement period, although the temperature distribution was not uniform. No trend was apparent when the delays were plotted against temperature, as evidenced by small and large delays occurring at the same temperature. This lack of correlation indicates that the meteorological parameters measured in the above experiments are inappropriate. This could be due to incorrect positioning of the anemometers, the need for more measuring points along the path or, very likely, the requirement of faster responding anemometers and more rapid sampling of the meteorological data, as one second is long compared to the 100ms flight time of the pulse. While the situation is not yet resolved, it does demonstrate the insensitivity of using mean parameters in such investigations.

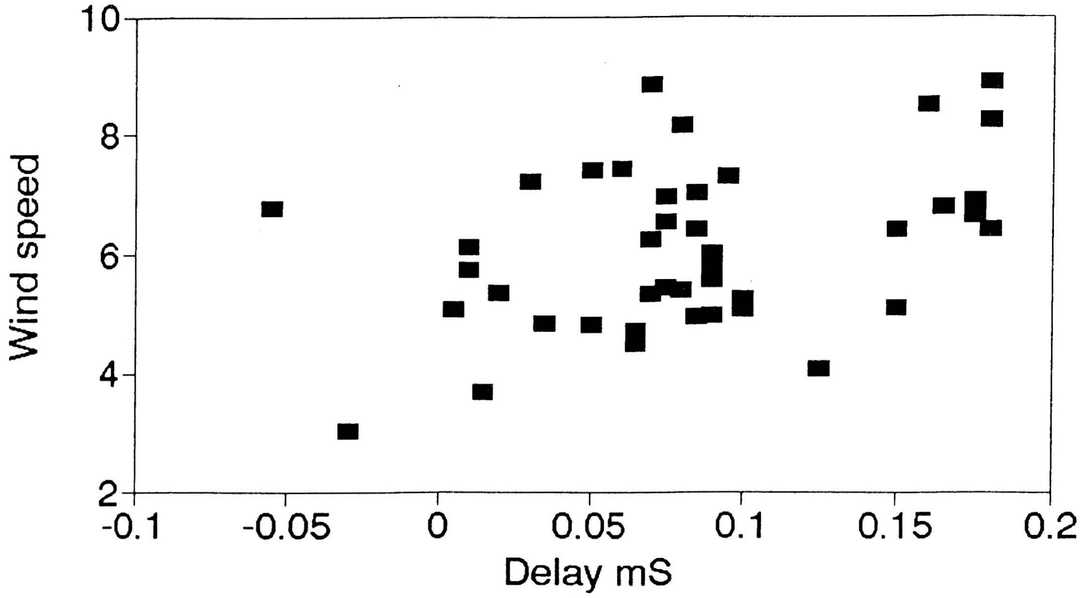

Fig.4: Test for Correlation between Wind and Pulse Delay

Energy of Impulse

A measure of the energy of a pulse can be obtained by summing the square of the instantaneous pressure occurring over the duration of the impulse. The energy ratio can then be calculated by dividing this energy measure for an individual pulse by the average value calculated for, say, an indoor pulse recorded at the same distance. This was how the data presented in Fig.5 was obtained.

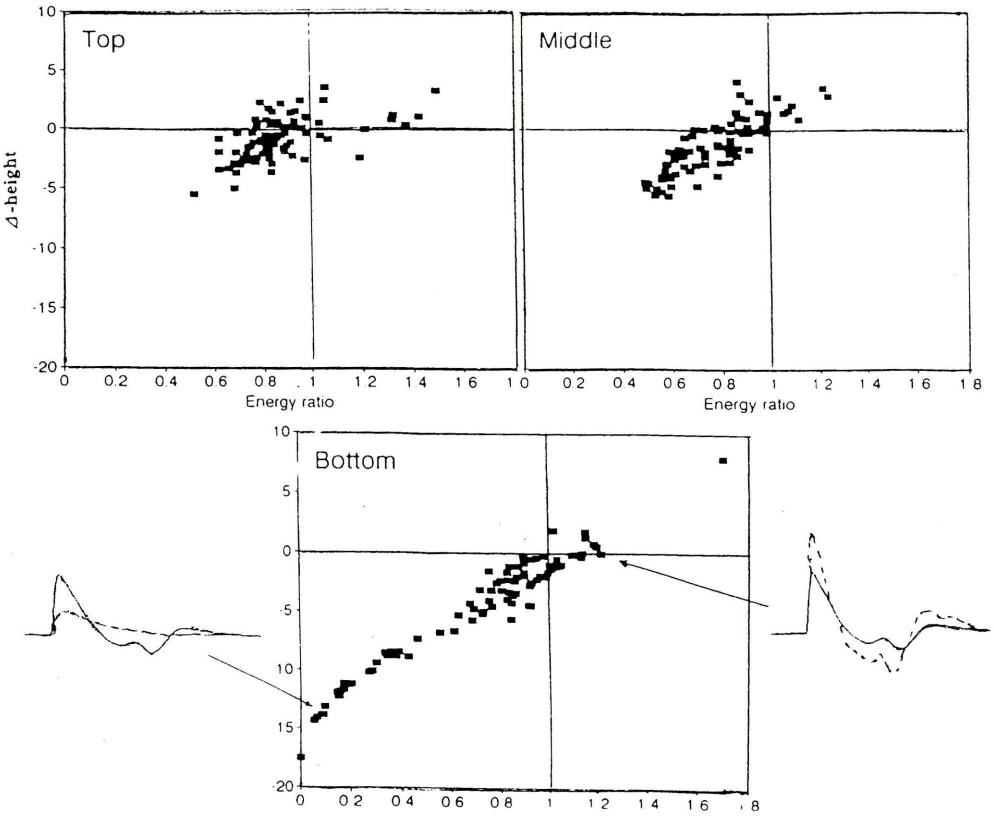

Fig.5: Energy Distribution Within the Pulses

Perhaps the most significant feature of the top two data sets is the presence of cases where the energy ratio exceeds one. When a similar plot is drawn for the 126 indoor measurements, the data groups tightly about one, with a maximum excursion to 1.1. Thus, approximately 5% of the outdoor measurements display an energy significantly greater than that expected on the basis of the indoor result. This increase in energy of the pulse suggests that a turbulent focusing mechanism is occurring in these cases, in contrast to the majority which loose energy through scattering.

The lower microphone case is dominated by the relative delay between the direct and reflected pulses. Waveforms for two extreme cases (dashed), compared with the average pulse shape (solid line), are also indicated. The behaviour of the larger Δ-height pulses is typical of the reflection being well delayed, allowing the full height of the direct to be observed. The average pulse has a significantly reduced height, due to partial attenuation by the reflection. When there is little delay between the components, as occurs for the very negative Δ-height cases, the rounded pulse shape typical of the ground wave results. Because of the summation of the direct and ground reflection, the lower microphone results can vary between zero and two without having to introduce turbulence to explain the results.

Frequency Content of Pulse

The above data have considered the pulses only in the time domain, however, it is interesting to see what information is available from the frequency domain. Figure 6(a) shows the difference in the relative amplitude of the frequency components in the pulses captured indoors while Fig.6(b) is the corresponding data for outdoor pulses recorded at the higher microphone locations. It is apparent that the atmospheric fluctuations cause a significant change to frequencies above several kilohertz, while little alteration occurs at lower frequencies.

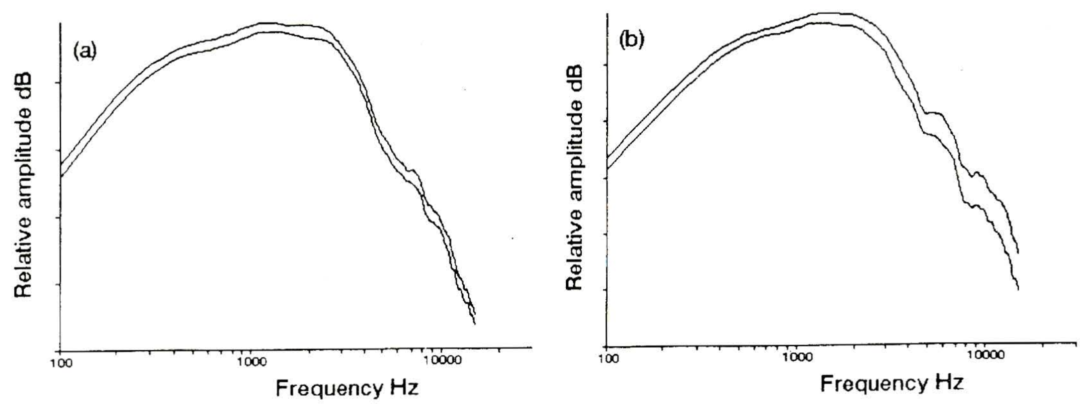

Fig.6: Variation amongst pulses (a) Indoors, (b) Outdoors

Conclusion

In the study reported here, the meteorological data clearly did not relate to the acoustic pulse delays: indicating that in-appropriate wind information was gathered. Impulse measurements offer the prospect of being able to correlate acoustic effects with meteorological data in a way not possible using continuous wave propagation. It is intended to pursue this approach until we have established a technique for gathering meaningful atmospheric information Once an appropriate measure of the wind properties is found, which correlates with the delays, it should be possible to proceed with greater confidence in examining how turbulence alters the pulse. A particularly intriguing aspect is how turbulence increases the energy content of some individual pulses.

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INSERTION LOSS OF ACOUSTIC BARRIERS

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Abstract

Acoustic barriers are a widespread way to shield populated areas from an unwanted noise, but what is the most effective design and how important is the surface treatment of the barrier?

This paper presents experimental results obtained using a short duration impulse sound as a source, over a range of barrier designs. The advantage of using an acoustic impulse is that it can be time isolated thus avoiding complications due to the ground or other reflections. The impulse used (produced by discharging a shot shell primer) has a broad frequency content ranging from 100Hz to 10kHz, allowing the investigation of this entire range at once.

It has been found experimentally that the insertion loss produced by two parallel barriers can exceed that of a single equivalent wide barrier. For example an increase in attenuation of approximately 3dB has been achieved using two parallel barriers 1.0m apart, as against a single 1.0m wide barrier. Also the effects arising from altering the impedance of one or more surfaces of the barrier has been investigated.

The experimental results are being used as a base to improve existing models for predicting barrier insertion loss, with an emphasis on including the effects of surface impedance and using multiple barriers.

Introduction

It has been found that the effective insertion loss of an acoustic barrier can be altered by applying small design changes in the construction of the barrier. Often it is impractical to examine these changes on a barrier installed on a full length of highway, so investigations into the effectiveness of different designs typically involve measurements taken using scale models, short lengths of barrier or by theoretical calculations.

Historically, theories have been based on geometrical diffraction. One of the first models was that derived by Maekawa[1] for a semi-infinite thin barrier. His design curve for the attenuation produced by a barrier can be represented analytically by

$$\Delta L = 20\log\left[\sqrt{2\pi N} / \tanh\sqrt{2\pi N}\right] + 5dB \quad (1)$$

providing the Fresnel number $N = 2\delta/\lambda$ exceeds -0.2 at the wavelength λ. Here δ is the difference in path length between sound passing over the barrier, d_o, and the direct distance from source to receiver, d_d : ie. $\delta = d_o - d_d$. This model can be adapted for a wide barrier by varying the height of the effective thin barrier, but the model is only approximate and does not make allowance for the nature of the barrier surfaces.

An exact theory for a hard wedge has been modified by L'Espérance et al[2] to include finite impedance sides of the wedge. However this is a very restricted barrier geometry. Other theories including a time domain approach by Medwin[3] which can, in principle, be applied to a variety of barrier shapes and an exact truncated wedge model derived by Tolstoy[4,5], but both these models assume an infinite barrier surface impedance.

A more general treatment is the boundary element method[6] which has the advantage that arbitrary barrier shapes and surface impedances can be accurately represented in the calculation. The principle of this theory is that an acoustic wavefront is moved step by step across the barrier and a modified wavefront calculated at each step, which then acts as the input for the following step. A major disadvantage is that long computing times and vast storage arrays are required.

Using an impulse technique[7] for determining the behaviour of various full size barriers, the validity of several of the above theories will be tested by comparison with experimental measurements.

Experimental Method

The acoustic impulses used to measure barrier attenuations are produced by discharging a shot-shell primer through a one metre length of 30mm diameter plastic tube. The impulse is typically 2-3ms in duration with a peak level of 120dB at 1m, originating from the mouth of the tube. Because of the conical symmetry of the sound source, the two microphone system described in Fig. 1 is used. The advantage of this technique is that it allows a comparison of the diffracted and direct waveforms of two, originally identical, impulses which have travelled the same distance, thus avoiding the need for corrections due to geometric spreading or atmospheric absorption. The barrier is rotated through 90° so that the diffracting edge is vertical to allow ground reflections to be eliminated by time isolation techniques. Also complications due to room and other reflections can be eliminated as the short duration impulse can be easily time isolated, as indicated in Fig. 2.

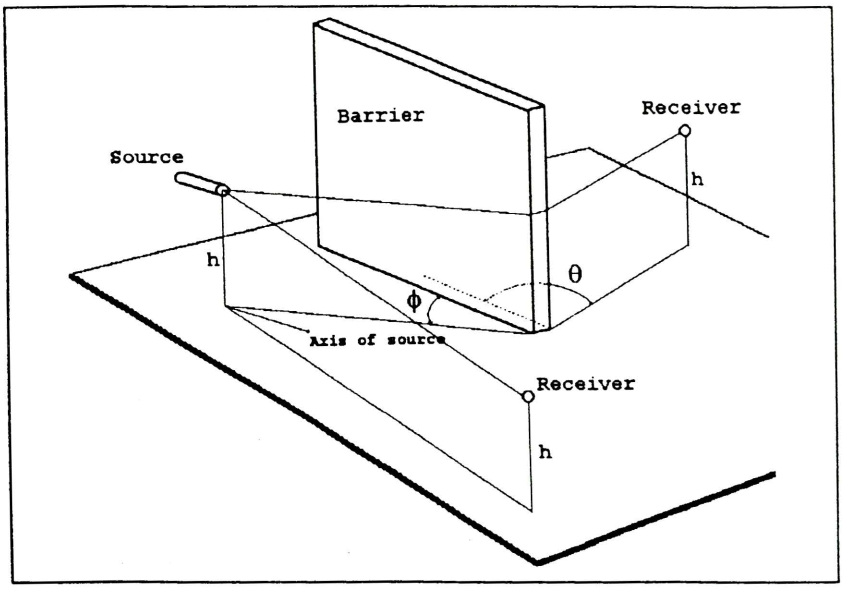

Figure 1: Geometry of measurement system used.

Previous work[7] indicated that fluctuations in the measured attenuation data which occurred at high frequencies were probably due to vibrations in the barrier itself. For the current measurements a very rigid barrier was constructed, using 18mm chipboard covering a pine backing frame, and this has significantly reduced such fluctuations.

Both the direct and diffracted impulses were recorded using B & K, type 2218 sound level meters with 1/4" microphones type 4135, their outputs being connected directly to a Data Precision DATA 6000 analyser with 16 bit analogue to digital conversion and storage. The impulses were then down loaded to a personal computer for ensemble averaging and later analysis. By doing a FFT on the appropriate waveforms and dividing the diffracted frequency components by the corresponding direct pulse ones the attenuation at the particular frequency can be converted to insertion loss by

$$\text{Insertion Loss} = 10\log\left(\frac{\text{diffracted}}{\text{direct}}\right) + 20\log\left(\frac{d_o}{d_d}\right) \qquad (2)$$

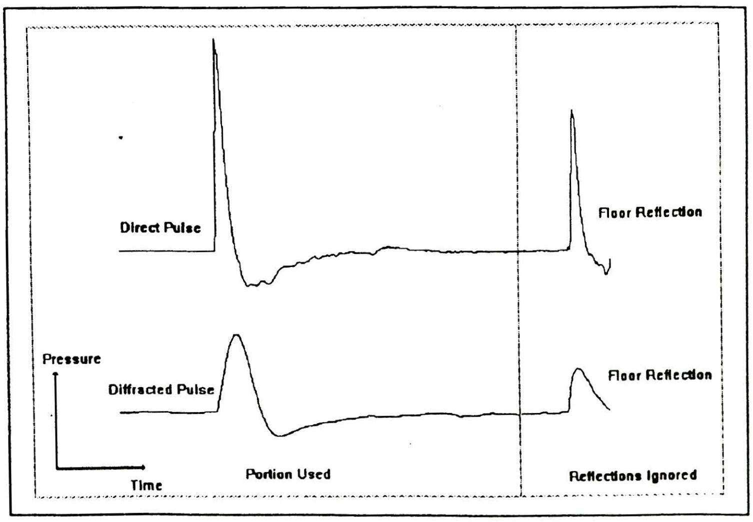

Figure 2: Example of how the direct and diffracted impulse are isolated from unwanted reflections.

Results

Measurements were taken over a rectangular barrier of width 0.1 and 1.0m, with source angle, ϕ, equal to 30°, 60°, 80°, and 90°. The receiver angle, θ, was set at an angle of 60° between the barrier and the ray from the top of the barrier to the receiver. Only one receiver angle has been used as it was found experimentally that the insertion loss for ϕ = 80°, θ = 60°, is the same for ϕ = 60°, θ = 80° for the symmetric rectangular barrier being considered. The surface impedance of the barrier walls was assumed to be infinite in model calculations, as the chipboard used in construction acts as a near perfect reflector.

In general it was found that for angles of ϕ around 60°, all the theoretical models considered produced a reasonable match with the experimental results for both the 0.1m and 1.0m wide barriers. However as ϕ approaches 90°, as shown in Fig. 3, Maekawa's approximation and especially Medwin's time domain calculation fail to match the experimental results. As expected, results calculated using the more sophisticated boundary element method give good agreement with experimental results for all source and receiver angles.

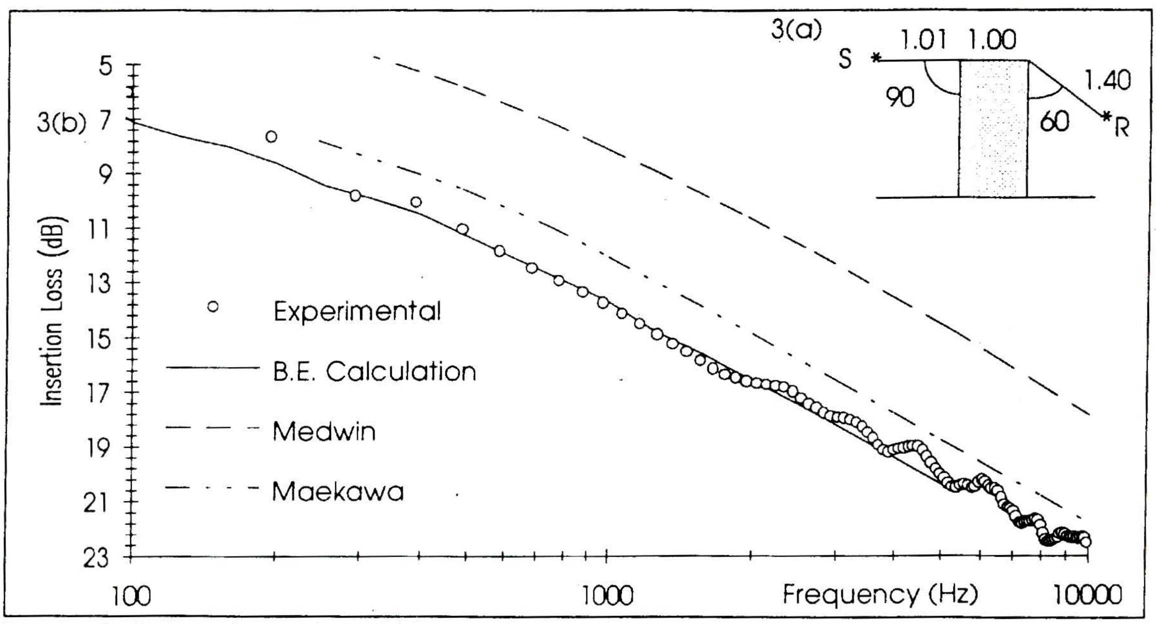

Figure 3: Comparison of experimental results with predictions from the boundary element theory, Medwin's theory and Maekawa's simple model for a wide barrier.

When the surfaces of the barrier are covered with an absorbent layer such as carpet with a measured flow resistivity of 350,000rayl/m, it was found experimentally that at frequencies above 2kHz an increase occurs in the insertion loss achieving an additional attenuation of 22dB at 5kHz. As shown in Fig 4(b), the side surfaces only contribute about 1dB of this. For a 0.1m barrier the additional attenuation is 6dB at 5kHz with the side surfaces contributing about 3dB of this. As one would expect, a finite impedance on the side surface has more of an effect as the source or receiver angles decrease. One interesting phenomena, found both experimentally and with the boundary element calculation, is that the insertion loss actually decreases at frequencies between 100Hz and 1.1kHz when one or more surfaces are covered with a finite impedance layer.

Measurements indicate that when the source geometry is equivalent to the receiver geometry the attenuation is the same, irrespective of the absorbing material being placed on just the source or receiver side of the barrier. This symmetry in the barrier geometry has also been found by L'Espérance et al[2].

Included in Fig. 4(c) are the experimental results produced for a wide barrier of width 1.0m with a 5cm layer of foam covering the top surface. As the foam has a much lower impedance (5000 rayl/m) the impulse will travel both through and around the foam causing an interference at appropriate wavelengths, producing the resonance pattern seen in the insertion loss graph.

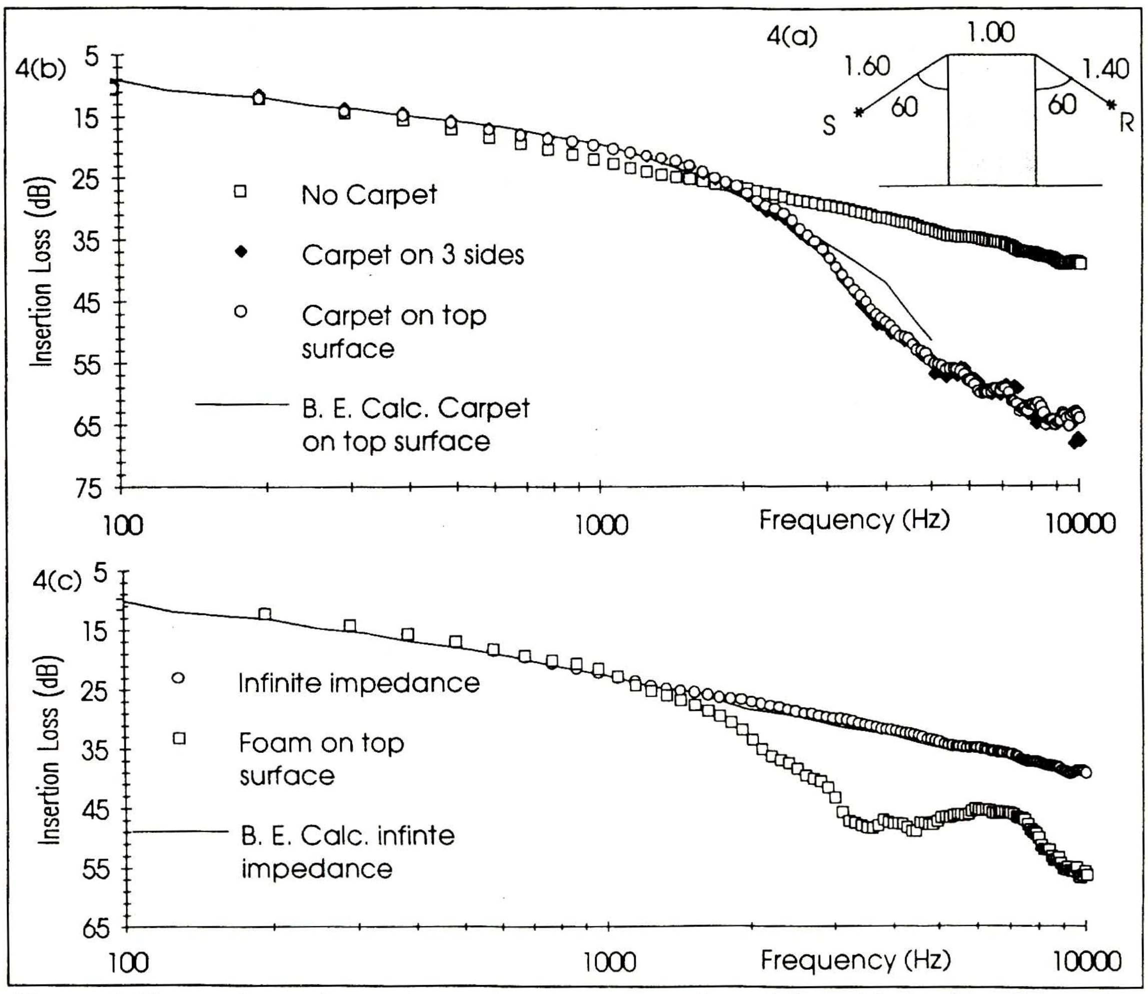

Figure 4: Comparison of experimental measurements for a 1.0m wide hard barrier and the same barrier with carpet or foam attached to one or more of the surfaces.

Perhaps more unexpectedly, when two parallel, 0.1m wide rectangular barriers were positioned 0.6m apart to form an effective 1.0m wide barrier but without a continuous top surface, the insertion loss was found to be greater than that of the hard surfaced 1.0m wide barrier. As shown in Fig. 5, the difference varies from 1dB at lower frequencies to 4 dB by 10kHz. This contrasts with carpet, which produces little alteration below 1kHz.

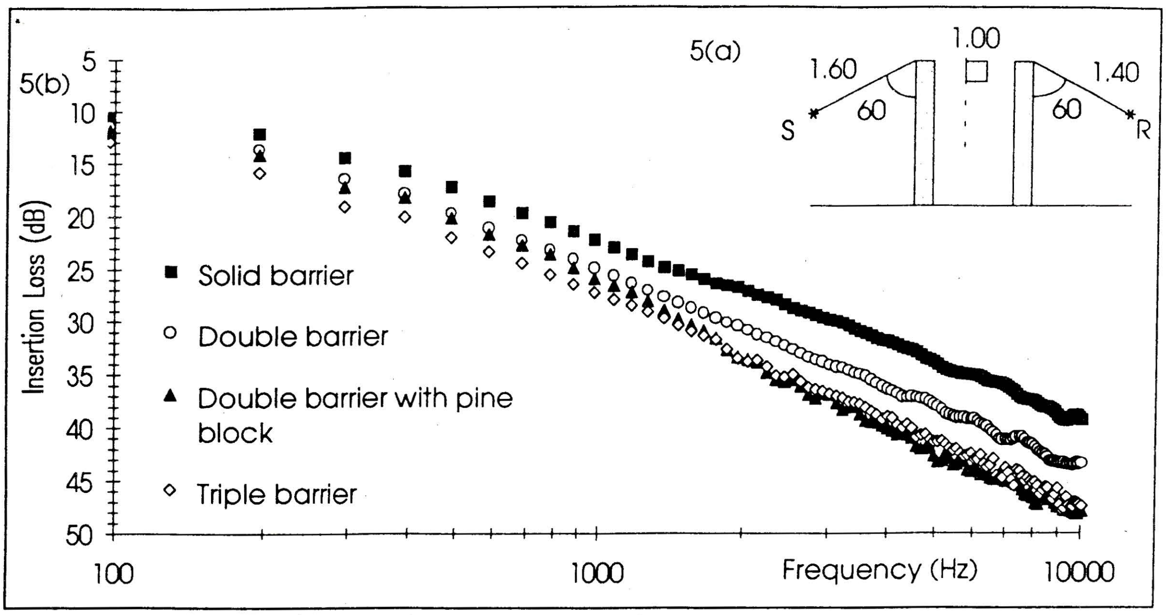

Figure 5: Experimental results for a multiple barrier system.

The formation of the double barrier increases the number of diffraction edges and produces a cavity, in which sound energy can be trapped. An intermediate block was added, with its upper surface in the plane of the barrier top. The block was fashioned from a 30mm wide by 88mm deep length of pine wood. This further enhanced the insertion loss at frequencies above 1kHz, which is perhaps unexpected, as the top surface area is effectively increased, approaching closer to that of the solid barrier.

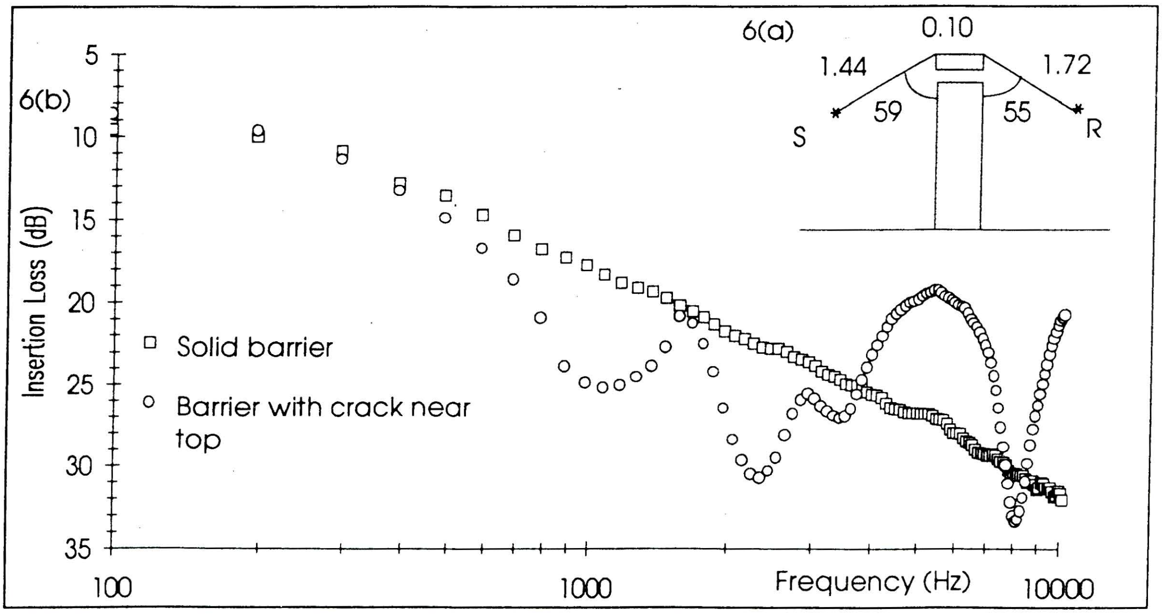

Figure 6: Experimental results for a 0.1m wide barrier and the same barrier with a 15mm crack 38mm from the top of the barrier.

The importance of the cavity itself was demonstrated by adding a chipboard sheet to the pine block, as indicated by the dashed line in Fig. 5(a). This had the effect of slightly reducing the insertion loss at the higher frequencies but improving it by several dB below 1kHz.

Fig. 6 shows the interference type pattern that occurs in the insertion loss when a crack is introduced near the top of a barrier[7]. Similar to the example involving a foam layer on the top of the barrier, Fig. 4(c), the interference pattern occurs due to part of the impulse energy travelling through the crack and some over the top causing cancelling and enhancement at appropriate frequencies.

Conclusion

Using acoustic impulses is a relatively simple technique for determining the insertion loss of a barrier over most of the audio range. In particular it is a convenient way of studying how changes to the surface impedance alters the barrier performance. While adding carpet to the top of a barrier may not be particularly relevant to the design of outdoor systems, it does indicate that the presence of grass covered berms (grass covered ground having a flow resistivity about that of carpet) are more effective than concrete ones. Further, such experiments act as a useful test for the theoretical models. For mid range angles of incidence and diffraction the simplistic Maekawa model is surprisingly effective, even for a 1m wide barrier, however, it can be in error by many dB at extreme geometries. Boundary element calculations are necessary for more complicated geometries, such as the multi-barrier system discussed here, and when the barrier has varied impedance surfaces. However, the computing time involved makes such investigations tedious so, in many ways, it is simpler to construct a barrier and test it directly using the diffraction of impulses.

Acknowledgments

The authors would like to thank Dr. S. Chandler-Wilde of the department of Civil Engineering, University of Bradford, for permission to use and adapt his boundary element program.

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The Attenuation of Impulse Propagation Peak Levels with Distance

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ABSTRACT

This paper presents the findings of an interim report on the study of peak levels propagated from an impulse noise source over long distances. The distances of measurement stations used for the study were 100m, 800m, 1.6km and 3.2km. Attenuation rates of the peak levels and a statistical analysis of the results are presented. Meteorological profiles of the prevailing atmospheric conditions were also gathered at the same time to examine the influence of wind strength and direction and the effects of the atmospheric temperature profile. This study is part of a much larger study of sound propagation through the atmosphere being conducted by the National Acoustic Laboratories and the Department of Defence.

INTRODUCTION

This paper represents part of an interim or progress report of an extensive amount of experimental data that has been gathered by the Prevention Services group at National Acoustic Laboratories. The main objectives of the project are to examine the effects of meteorological conditions on the propagation of sound over longer distances and to establish a stastical data base of noise level versus distance. This study has included both continuous and impulsive sound sources. Experimental work with impulsive sources commenced in 1989 and continued through until 1991, although to some extent more data is gathered from time to time if the opportunity presents itself. As will become apparent an enormous amount of data has been collected and the processing and interpretation of this data represents a mammouth task. As a consequence analysis is proceeding in stages and this paper presents the initial overall analysis of the impulse propagation data.

THE EXPERIMENTAL ARRANGEMENTS

The experimental setup was the same in each instance of the seven field trips from which data is taken for this paper. There was a source located at what we can term ground zero and out on a single radial from this source lay four manually operated recording stations. The direction of this radial was purely governed by the geography of the area and the ease of access that was required to access the measurement points. Usually they were, for this reason, placed along a single road or such that road access was available at each site. In this sense the radials along which the measurements were taken were not completely random. The terrain, while not being completely flat and free of bush was usually slightly undulating to flat and carried low scrub, grass and sometimes trees.

After much trial and error it was decided that the most effective impulse source would be a "plug" of high explosive, usually a 125gm stick of Tovex, Powergel or similar. The explosive was suspended by cord from a metal frame such that it was two metres off the ground, giving the closest approximation to an omnidirectional source.

The manually operated recording stations were situated at 100m, 800m, 1.6km and 3.2km. They consisted of computerised equipment that could not only record the maximum linear peak level (MAXP) of the impulse but also capture the resulting waveform for later analysis of frequency content, duration, etc. The only information that was required for this report was the MAXP. The aim is to analyse the other data at some future time. The co-ordination of all sites and the source was carried out through the use of portable radios.

At each of the manned measurement locations and in the vicinity of the source were portable ground meteorological stations and at each of these, ground meteorological conditions were recorded at half hourly intervals through out the measurement period. Also at a location not far from the propagation path was a meteorological team from the School of Artillery of the Australian Army. At specified intervals throughout the measurement period they would release a radiosonde which would send back information on the meteorological conditions prevailing. Data from the radiosonde then gave a complete meteorological profile of the atmosphere.

The duration of the measurement period was usually eight hours with the start and stop times being governed by whether interest was in a sunrise period or sunset period. For a sunrise period measurements were commenced one hour prior to sunrise, concluding eight hours later; for sunset periods, measurements were commenced such that the conclusion of the eight hour period would be one hour after sunset.
By convention the day is divided into three measurement periods; the morning, 00:00 to 10:00 hours; day time, 10:00 to 17:00 hours; and evening, 17:00 to 24:00 hours. These periods are adopted as they reflect the time periods that are frequently used when studying annoyance of noise from impulse sources such as rifle ranges. It can also take into account enhanced propagation conditions that can sometimes exist in the winter months when inversion conditions can be present in the atmosphere in the morning and early evening. A more detailed analysis of these propagating conditions will be undertaken in the future. In this analysis I have only considered the day as a single period from 00:00 to 24:00 hours.

Measurement sites were chosen such that they represented as many climactic zones in Australia as practically possible. Sites were at Innisfail, Queensland; Singleton and Holsworthy, NSW; Port Wakefield and Woomera, South Australia; and Bernacchi in the Central Highlands of Tasmania (twice).
A summary of the propagation directions is given in the following table:-

Summary of Propagation Directions	
Innisfail, Qld	232° (true)
Singleton, NSW	243° (true)
Holsworthy, NSW	222° (true)
Woomera, SA	184° (true)
Port Wakefield, SA	0° (true)
Bernacchi, Tas	143° (true)

As can be seen these directions are not completely random but were subject to the conditions discussed above.

RESULTS

Meteorological Data Summary

There were in the order of 300 radiosonde flights. The meteorological conditions necessary for this work are summarised as follows:-

a) Temperature: The ground temperatures measured were in the range -5°C to +35°C, naturally temperatures at altitude tended to be lower again.
b) Relative Humidity: The relative humiditiy fell in the range of around 20% to close to 100% as measurements were not taken when it was actually raining in order to protect the instrumentation.

c).Wind Direction: For this section of the experiment it was preferable that wind should come from any possible direction compared to the direction of propagation in order to give a true random sample. At this stage it is not possible to state definitively that all and any wind directions were involved with respect to the propagation directions, only that it is presumed that this is most probably the case.
d) Wind Speed: When the steady wind speed rose over 10m/s measuring ceased as it was found that there was too much wind noise generated near the microphones and this did not allow accurate measurement of the impulses. Hence, results for wind speeds greater than 10m/s are not included. Toward the end of the series of field studies several different forms of wind screens were tried with varying degrees of success. This is the topic of separate work.

Experimental Results
There were in the order of 2,500 impulses measured and recorded. The following table summarises all of the valid data for the maximum peak levels collected from all locations and for all day periods as mentioned above.

A SUMMARY OF MAXIMUM PEAK LEVEL DATA (00:00 to 24:00)

Measurement location	AVE MAXP (dB)	Maximum MAXP (dB)	Minimum MAXP (dB)	No of valid data points
100 metres	140.4	147.5	129.1	2097
800 metres	116.1	132.9	90.9	2102
1.6k metres	109.2	124.5	81.9	2088
3.2k metres	101.1	116.5	66.5	1882

DISCUSSION

The Relationship between MAXP and Distance
When the plot of the Average Maximum Peak Level, in dB, versus the logarithm of distance, in metres, is drawn (see above) it is found to be a good approximation to a straight line over the range of interest. On calculation the equation representing this line is found to be:-

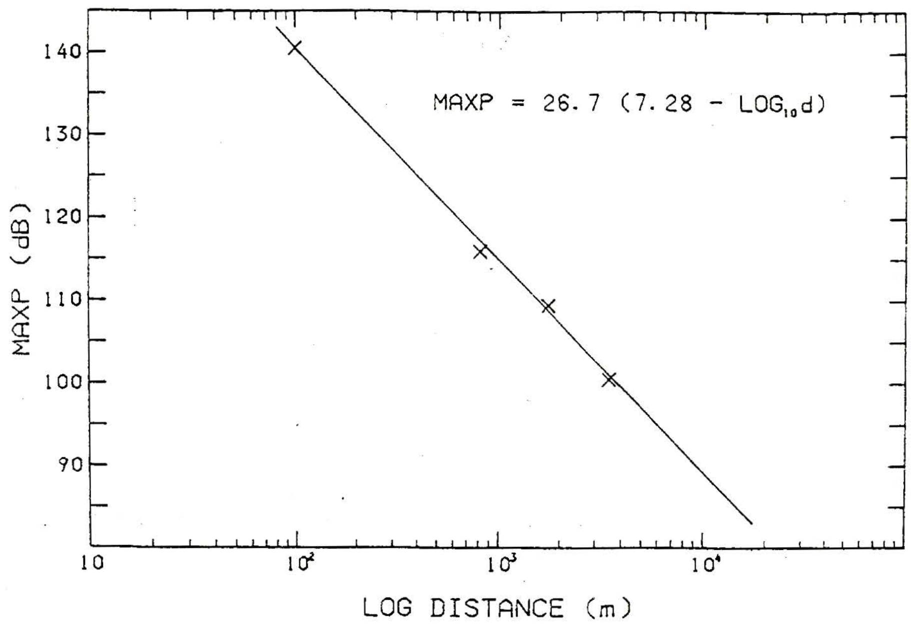
$$MAXP(dB)=26.7(7.28-\log d(m))$$

Given the sample size that was considered, this can be taken as a true indication of the relationship involved. The general equation becomes:-

$$MAXP(dB)=26.7(C-\log d(m))$$

where the constant C is dependent on the type of explosive source.

One important qualification on the use of this equation is that the type of source used should be of a similar nature to the one used for this experimental work. It is useful to say here that this equation is consistant with other studies conducted by NAL (see Peploe 1992).



Graph of Average MAXP (dB) versus Log Distance (m)

Spread of Results

One of the main results to note from the above table is the range of values of MAXP obtained at each measurement location. These can be summarised as follows:-

RANGE OF MAXP LEVELS	
Measurement distance (m)	Range of MAXP (dB)
100	18.4
800	42.0
1.6k	42.6
3.2k	50.0

The range of values indicated here graphically illustrates the wide variation in MAXP levels that can be experienced at a particular location. This is very important when the environmental effect of noise is being considered as predictions must take into account the worst case scenario.

The next table comperes the measured values to those that would be expected from,

a) a simple six dB per doubling of distance; and

b) the simple six dB per doubling of distance plus the maximum 3 dB per kilometer as proposed by Embleton (1982).

AVERAGE MAXIMUM PEAK LEVEL WITH RESPECT TO DISTANCE				
ASSUMED CONDITIONS	**DISTANCE (Metres)**			
	100*	800	1.6k	3.2k
Measured	140.4*	116.1	109.2	101.1
6dB/doubling of distance	140.4*	122.4	116.4	110.4
6db/doubling of distance plus 3dB/kilometer	140.4*	119.8	111.3	100.8

Note: * This is used as the reference value

As can be seen the 6 dB/doubling of distance plus 3 dB/kilometre comes the closest to the measured values.

When examining the environmental impact of a proposed development that would involve impulses of this nature the consideration of Accumulated Peak Levels (APLs) and hence the average MAXP is satisfactory but for the determination of annoyance the availability of percentile levels would be useful. This work is planned for the future.

CONCLUSION

From the above discussions there is now firm relationship between the average maximum peak level of an explosive impulse source and distance, up to a range of at least 3.2 kilometres. Possibly more important is the range of maximum peak levels that has been observed at these distances, up to 50 dB at 3.2 km.

While an extension of the MAXP versus distance relationship to longer distances would not seem unreasonable for predicting an *average* level, the omission of consideration of the possible wide range of levels could be disasterous.

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A FLUSH MOUNTED MICROPHONE TURBULENCE SCREEN

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Abstract

This paper describes the development of a flush mounted microphone turbulence screen for measuring the sound pressure at the wall of a power station chimney flue due to the sound waves propagating up the flue. The purpose of the turbulence screen was to reject the pressure fluctuations due to the turbulent gas flow in the flue. The turbulence screen was flush mounted to avoid generating self turbulence and noise. The turbulence screen was required to protect the microphone from the hot gases flowing in the chimney flue so that the microphone would not be used outside its rated temperature range.

The modal and flow velocity corrections in the international standard ISO 5136 (1990) "Acoustics - Determination of sound power radiated into a duct by fans - In-duct method" are designed for estimating the sound power propagating along the duct. Since the flush mounted turbulence screen was designed to measure the sound pressure at the wall of the flue, it was necessary to develop new modal and flow velocity corrections. The turbulence screen was calibrated and its directivity checked in an anechoic room. The turbulence rejection of the turbulence screen was measured at three different flow velocities.

1. Introduction

When measuring the noise emitted from brown coal fired power station chimney flues, it is desirable to measure the sound pressure level inside the chimney flue in order to avoid the large influence of the exterior climate on exterior propagation paths. The State Electricity Commission of Victoria (SECV) decided that it was necessary to measure the sound pressure level at the wall of the chimney flue using flush mounted devices because probes placed in the interior of the flue would disturb the exhaust gas flow and generate extra turbulence and noise. Thus they have specified maximum sound pressure levels on the inside of the chimney flue wall in their contracts.

The silencers at the Loy Yang A and Yallourn W power stations were retrofitted. SECV experience at these two power stations convinced them that the existing methods of measuring the sound pressure levels inside power station chimney flues were inadequate. This was because the magnitude of the turbulent pressure fluctuations could be comparable to the magnitude of the sound pressure fluctuations in the chimney flues which were fitted with silencers. In an effort to overcome this problem the SECV experimented with a prototype flush mounted microphone turbulence screen. They constructed a section of false wall which could be placed on the inside of the wall of a chimney flue through an existing inspection hatch. A conventional Brüel and Kjær microphone turbulence screen type UA 0436 was flush mounted in this section of false wall.

Unfortunately it proved impossible to rigidly mount this section of false wall to the inside of the chimney flue wall and it partially lifted off the flue wall in the presence of flow in the chimney flue.

Silencers were incorporated as part of the original design of the new Loy Yang B power station. There was also a clear need to develop a new system of measurement. A number of precision machined measurement ports were constructed in the walls of the chimney flues of the Loy Lang B power station. These were also part of the original design. Each of one of these measurement ports has a slit measuring 600 by 40 mm in which a flush mounted microphone turbulence screen can be rigidly mounted and four 24 mm diameter holes in which a flush mounted microphone can be mounted. To insure independence, the SECV and their contractor agreed that the flush mounted microphone turbulence screen and the equivalent of a flush mounted microphone should be developed by a third party with appropriate experience and ability. The CSIRO Division of Building, Construction and Engineering was contracted to carry out this development task. This paper describes the results of that contract.

CSIRO was required to develop a flush mounted microphone turbulence screen for use in measuring the acoustical pressure level at the wall of a 5.5 m diameter power station chimney flue at a temperature of 188 °C. The microphone turbulence screen was required to be flush mounted so that it did not generate any self noise or additional turbulence. It was also required to be provided with correction factors for the measurement of sound pressure level at the flue wall, rather than with corrections for the measurement of the propagating sound power in the flue as is the case in the international standard ISO 5136 (1990).

Another requirement was that the microphone must not be used outside its manufacturer's recommended temperature range. Brüel and Kjær (1982) state that their air condenser microphones can be used continuously up to a temperature of 150 °C. They also state that intermittent operation is possible up to 200 °C, but recommend that their microphones not be used above 150 °C.

Because the exhaust gases in the chimney stack are close to the dew point of their acidic vapour, and because the microphone turbulence screen is calibrated with the same temperature inside and outside, another requirement was that the microphone turbulence screen be thermally insulated. This would ensure that the screen's temperature would be close to that of the exhaust gases in the chimney flue, in order to prevent condensation or a change in the calibration of the microphone turbulence screen due to temperature differentials.

A further requirement was that the microphone turbulence screen be capable of measuring third octave band acoustical levels as low as 60 dB. An equivalent of a flush mounted microphone in which the actual microphone did not exceed the maximum temperature recommended by its manufacturer was also requested.

2. Design of the microphone turbulence screen

The initial idea was to use a piezoelectric pressure transducer which could be used up to temperatures of 250 °C. Unfortunately their stated background noise levels are of the order of 94 dB (1 x 10^{-5} atmospheres). This was confirmed by third octave band measurements on an actual piezoelectric transducer. The measurements were made in the hope that the third octave band noise levels would be significantly lower than the stated DC levels, but this proved not to be the case.

The use of a microphone probe tube to sample the sound pressure at one end of the turbulence screen was also considered. The idea was that the microphone probe tube would pass through the thermal insulation surrounding the chimney and thus keep the temperature of the microphone below 150 °C. However this idea was abandoned because of concerns about condensation obstructing the small diameter probe tube and the problem of calibrating the probe tube with a variable temperature gradient along it.

It was then decided to adopt the approach that has been used previously to make measurements in chimney flues, and which is used in the current Brüel and Kjær probe microphone. This consists of using a microphone or microphone probe tube in the wall of a tube which joins the chimney flue at right angles. The other end of the tube has an anechoic termination. The anechoic termination consists of a long length of coiled flexible tubing with the same internal cross sectional area as the rigid tube. Although the flexible tubing has only a small attenuation per metre, because of its long length and the fact that the reflected sound from its far end has to transverse the same long length of tubing before it reaches the rigid tubing, the long length of flexible tubing acts as a good anechoic termination. Because the microphone can be placed outside the thermal insulation which surrounds the chimney flue, it can be kept at a temperature much lower than the temperature of the flue. Use of a flexible goose neck to connect the microphone to its pre-amplifier will keep the microphone pre-amplifier at an even lower temperature.

The design of the microphone turbulence screen is based on Annex E of the international standard ISO 5136 (1990). However some deliberate changes were made. The length of the slit was increased from 400 mm to 500 mm so that the microphone turbulence screen would have the same directivity characteristics at 188 °C as a screen with a 400 mm slit at room temperature. Because the microphone turbulence screen had to be flush mounted a nominally square cross section tube was used instead of a cylindrical tube. The actual internal dimensions of the tube were 11.2 by 11.8 mm.

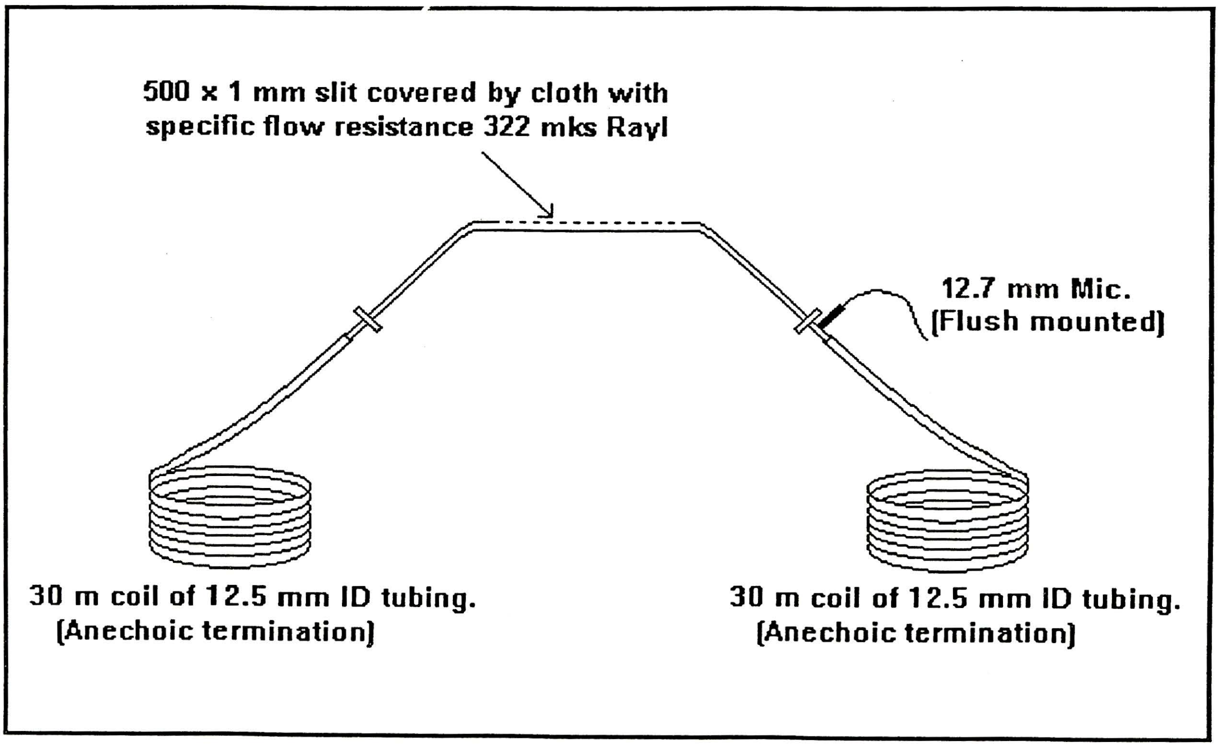

Figure 1. Schematic of microphone turbulence screen.

The microphone turbulence screen tube was extended beyond the slit length and bent in a large radius curve so that it extended outside the thermal insulation screen around the chimney flue. The microphone was flush mounted in the wall of the tube outside the thermal insulation screen. The tube was terminated at both ends with 30 m of coiled 12.5 mm internal diameter flexible tubing to provide an anechoic termination at both ends of the tube. The anechoic termination at the end without the microphone provides the opportunity to install a condensate trap or gas purge if found necessary. The layout of the microphone turbulence screen is sketched in Figure 1.

A slit width of 1 mm was retained and the slit was covered with two layers of a woven synthetic fabric. Each layer had a measured specific airflow resistance extrapolated to zero flow rate of 166 mks rayl. This synthetic fabric was chosen because it could withstand the temperature and the acidity. Seven different woven synthetic fabrics were tested for their specific airflow resistance. The values claimed by the manufacturer with a pressure drop of 196 Pa had almost no relationship to the measured values of specific airflow resistance extrapolated to zero flow rate. A fine stainless steel mesh was also tested, but its specific airflow resistance was far too low. Sintered metal was also considered, but there was concern about its resistance to acid attack.

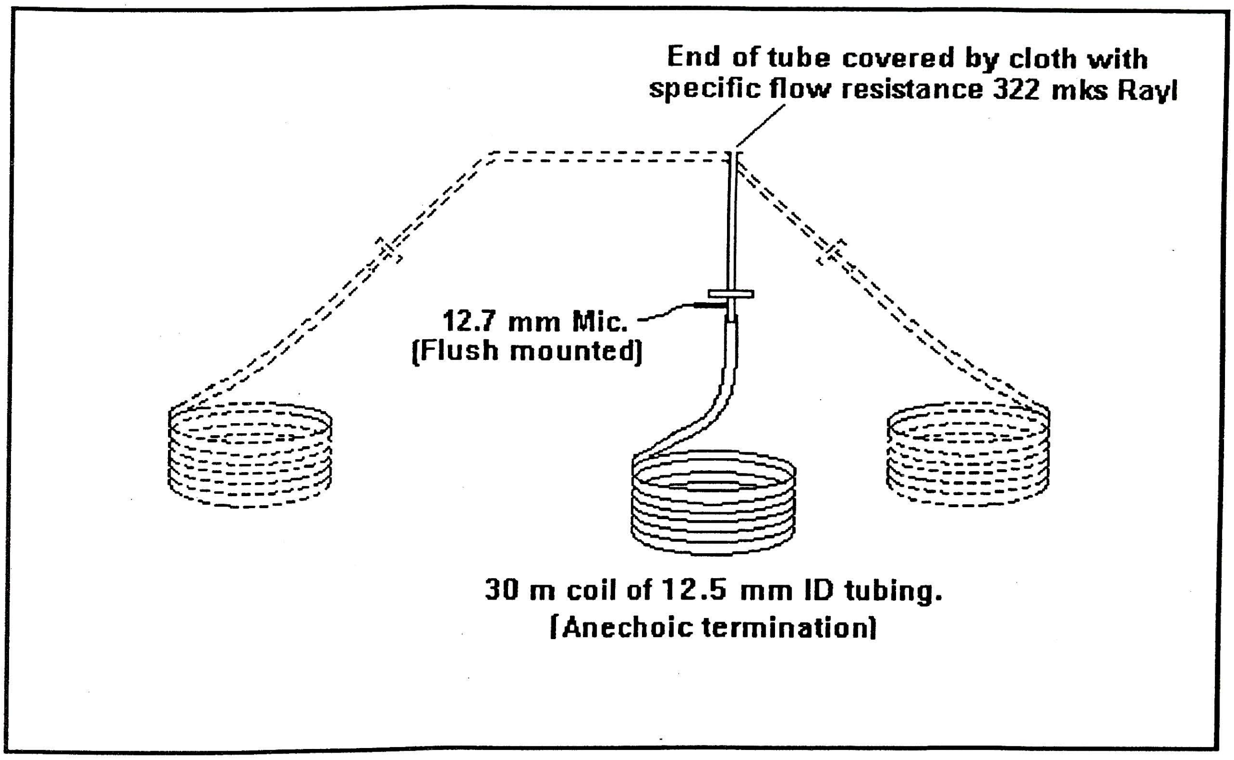

Figure 2. Schematic of perpendicular tube.

The two layers of fabric were sandwiched between the tube and the plate in which the tube was mounted. The slit was machined with the tube in the plate to ensure proper alignment of the two slits.

The microphone turbulence screen was originally tested without the anechoic terminations because this made the mounting of the microphone much easier. It was hoped that the attenuation due to the fabric in the slit would dampen any resonances due to the reflections at the ends of the tube. When tested with a fabric with a specific flow resistance of 2400 mks rayl, many pronounced resonances were observed in its Fast Fourier Transform (FFT) frequency response due to the low attenuation along the tube. When the specific flow resistance was changed to 332 mks rayl, some but not all of these resonances were damped out. The resonances which were damped out were obviously those involved with the length of the slit. It was then realised that the undamped resonances were due to reflections between the end of the slit where the impedance of the tube changes and the ends of the tube. To damp out these resonances, a flush mounted microphone and the anechoic terminations described above were employed. This damped out the remaining resonances and gave a locally smooth FFT and third octave frequency response. The damping out of the resonances ensures that the frequency response of the microphone turbulence screen will not depend on the temperature gradient along the two arms of the tube which extend through the thermal insulation around the chimney flue.

The frequency response of the microphone turbulence screen was also measured with only one anechoic termination which was placed at the end without the microphone. This configuration makes construction of the microphone mounting easier, but unfortunately some of the resonances returned due to reflections between the end of the slit and the microphone. In retrospect, the single anechoic termination probably would have been successful if placed at the microphone end of the tube. However there is little to be gained compared to the double anechoic termination design in terms of ease of construction.

The double anechoic termination design was also successful when used with fabrics of specific airflow resistance of 1170 and 493 mks rayl. However these fabrics were discarded because they did not meet the directivity requirements. The directivity problems were due to the fact that the directivity equation in the draft international standard ISO 5136 (1990) is a poor approximation to the directivity of microphone turbulence screens in the 30° to 45° angle of incidence range.

The equivalent flush mounted microphone comprised a length of the same tubing used in the microphone turbulence screen, mounted at right angles to the wall of the chimney flue with its inner end flush with the inner surface of the flue wall. This end was covered with two layers of the same fabric used in the microphone turbulence screen. These two layers were glued to this end of the tube to inhibit whistling due to the flow in the chimney flue. The microphone was flush mounted to the tube outside the thermal insulation around the chimney flue to keep the temperature of the microphone below 150 °C. The other end of the tube had an anechoic termination like those on the microphone turbulence screen ends. This equivalent flush mounted microphone was called the perpendicular tube. The layout of the perpendicular tube is sketched in Figure 2.

3. Frequency Response

The frequency response of the microphone turbulence screen was measured in an anechoic room at room temperature using third octave bands of random noise. A 300 mm diameter dual cone loudspeaker mounted in a baffle was placed 3600 mm from the centre of the microphone turbulence screen slit with the axis of the loudspeaker on the line which was an extension of the microphone turbulence screen slit. The loudspeaker was driven with pink noise which was passed through a third octave graphic equaliser set to boost the high and low frequency noise. The measured results were corrected for the effect of background noise which was minimal for the results reported here. A Brüel and Kjær half inch flat free field response air condenser microphone type 4133 without grid was used in the microphone turbulence screen. The response of the microphone turbulence screen was measured relative to the response of another 4133 microphone with a protecting grid positioned facing the microphone turbulence screen slit 6 mm above the centre of the slit. The relative 90° response of the two microphones with grids was measured by pointing the two microphones at each other with their grids 6 mm apart and their common axis at right angles to the line between them and the loudspeaker, and used to correct the measured results. The published Brüel and Kjær 90° free field response of a type 4133 microphone relative to the microphone's electrostatic actuator response was used to calculate the response of the microphone turbulence screen relative to the microphone's electrostatic actuator response.

The frequency response correction of the microphone turbulence screen is the negative of its frequency response. It is the amount in decibels that must be added to the output of the turbulence screen microphone to correct for the frequency response of the turbulence screen. The output of the microphone must also be corrected so that the microphone has a flat pressure or electrostatic actuator response. The frequency response correction of the microphone turbulence screen is shown in Figure 3 as a function of Helmholtz number. The Helmholtz number is the product of the wave number of the sound and the length of the turbulence screen slit. It is used to make the results independent of temperature. The theoretical frequency response correction for the microphone turbulence screen is 0.9 dB.

It should be noted that it would actually be more appropriate to use a flat pressure response microphone such as a Brüel and Kjær type 4134 in the microphone turbulence screen, perpendicular tube and as a flush mounted microphone. However the difference is only about 4 dB at 10 kHz. A 4133 was used in these measurements because the results in the Brüel and Kjær microphone turbulence screen data sheet are relative to a 4133 microphone. Also Australian sound level meters are equipped with flat 0° incidence free field response microphones.

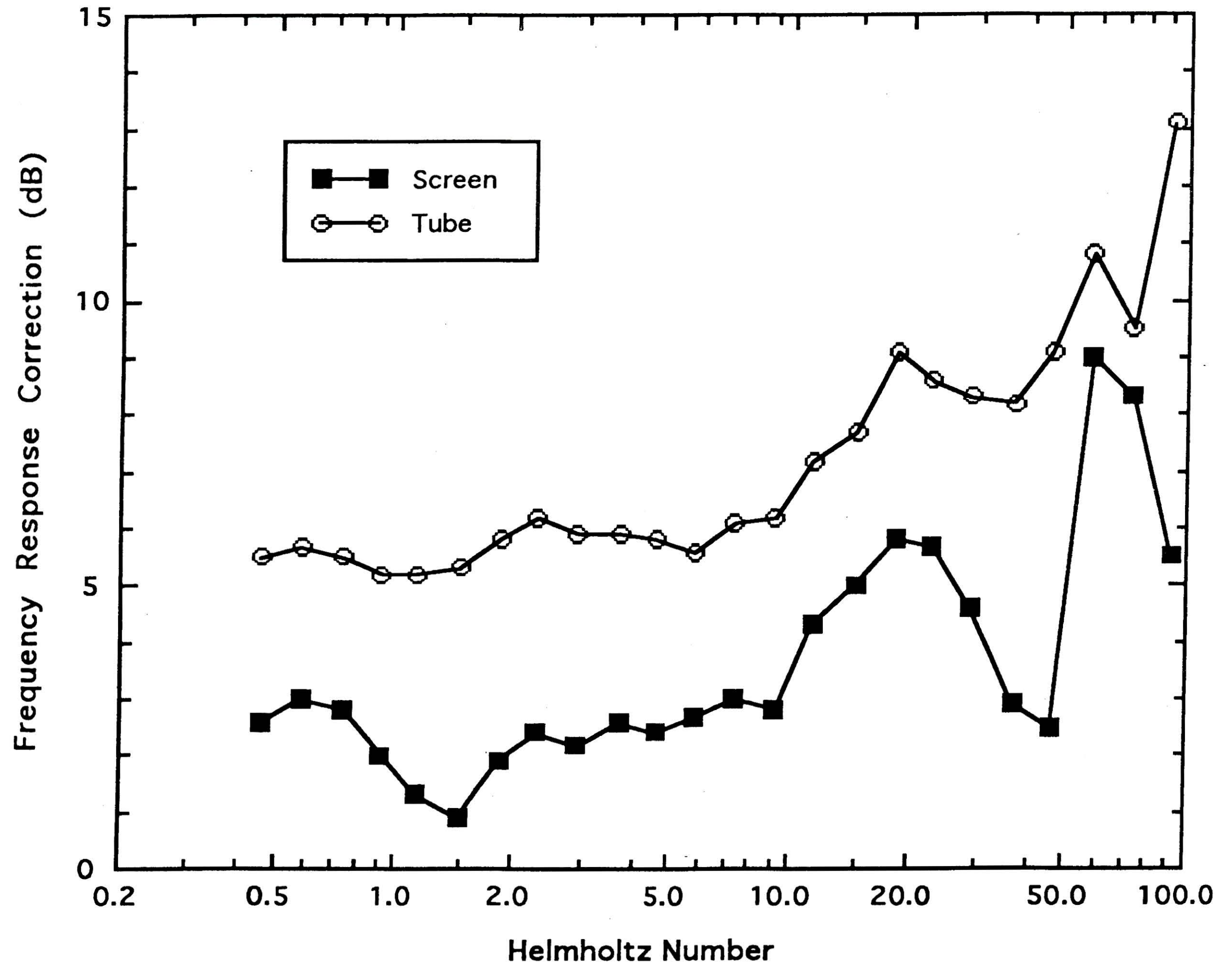

Figure 3. Frequency response corrections for microphone turbulence screen and perpendicular tube as a function of Helmholtz number.

The directivity of the microphone turbulence screen was measured from 0° to 75° in increments of 15°. The measurements were made using third octave bands of random noise from 50 Hz to 10 kHz inclusive. The turbulence screen easily satisfies the requirement that it be more omni-directional than the lower limiting curves given in Figure 2 of ISO 5136 (1990). However these curves as labelled do not agree with the equation on page 6 of ISO 5136 (1990) from which they are derived. The labels on Figure 2 of the International Standard have been drawn $\pi/18$ radians (10°) to the right of where they should actually be drawn.

When the directivity of the microphone turbulence screen was compared to the lower limiting equation on page 6 of the standard it did not satisfy the omni-directivity requirement. There are two reasons for this fact. The first is that the microphone turbulence screen was deliberately designed with a slit length of 500 mm rather than the 400 mm length recommended in the standard. This was done so that the microphone turbulence screen would have the same directivity at 188 °C as a turbulence screen with a length of 400 mm has at room temperature for the same frequency.

Secondly, the equation on page 6 of the international standard is based on an early empirical approximation of the directivity pattern of a microphone turbulence screen by Bolleter, Cohen and Wang (1973). They obtained their approximation by best fitting 'a simple empirical formula'. Their formula is a function of the Helmholtz number of the microphone turbulence screen and the cube of the angle of incidence. The equation on page 6 of the international standard removes the dependence on slit length by assuming a slit length of 400 mm, and expresses the wave number in terms of frequency in hertz by assuming a temperature of 20 °C.

These limitations can be overcome by multiplying the frequency used in the equation on page 6 of the international standard by either the ratio of the length of the microphone turbulence screen slit to the standard microphone turbulence screen length of 400 mm or by the ratio of the speed of sound at 188 °C to the speed of sound at the temperature at which the microphone turbulence screen directivity was measured. Both of these approaches lead to the same result because this is how the length of the microphone turbulence screen was selected. With this modification the omni-directional requirement was satisfied at all but three points. These three points have values of Helmholtz number which correspond to room temperature frequencies of 50 Hz, 200 Hz and 10 kHz. These frequencies are not the frequencies (1, 2, 4 and 8 kHz) at which the omni-directional requirement must be satisfied in Figure 2 of the international standard.

The frequency response correction of the perpendicular tube with an anechoic termination and a flush mounted microphone in the tube wall was measured at the same time as the frequency response correction of the microphone turbulence screen. The result is also shown in Figure 3. The theoretical frequency response correction of the perpendicular tube is 5.1 dB.

4. Higher order modal and flow velocity correction

The microphone turbulence screen is calibrated in an anechoic room with zero flow (Mach number M equals zero) and with angle of incidence equal to zero. For non zero Mach numbers and non zero angles of incidence a theoretical correction to the calibration must be calculated. If sound is incident from different directions at the same time, the theoretical correction must be averaged over the different angles of incidence with a weighting which is proportional to the sound energy incident from each direction. This approach assumes that sound incident from different directions is uncorrelated. Because of the large chimney flue diameter (5.5 m), back reflections can be ignored (see Appendix B of British Standard BS 4718 (1971)). This means that we only have to average over angles of incidence from 0° to 90°.

According to Morse and Ingard (1968), the cut on frequency of the first cross mode in a cylindrical duct is 0.5861 c/D where c is the speed of sound and D is the diameter of the duct. Thus for a 5.5 m diameter cylindrical chimney flue at 188 °C, the cut on frequency of the first cross mode is 46 Hz. This means that propagating cross modes will be present across the whole of the frequency range from 50 Hz to 10 kHz. Thus angles of incidence other than 0° have to be considered.

However we do not know the angular distribution of sound energy in the chimney flue. The obvious assumptions that might be made about the angular distribution of sound energy in the chimney flue are that every mode carries equal power down the duct, that every mode has equal energy density, that equal energy is incident from every angle or that equal energy is incident from every element of solid angle. This paper recommends the use of the equal modal energy density assumption, but also covers the other possibilities. The correction factor C is calculated by averaging the pressure-squared response $|p|^2$ of the microphone turbulence screen with the appropriate weighting factor over angles of incidence θ from 0° to 90° and dividing this into the product of the pressure-squared response for zero angle of incidence and zero flow and the average of the desired angular response with the appropriate weighting function. This gives

$$C = \frac{|p|^2_{M=\theta=0} \int_0^{\pi/2} f(\theta)\ w(\theta)\ d\theta}{\int_0^{\pi/2} w(\theta)\ |p|^2\ d\theta} . \qquad (1)$$

The function f(θ) is the ideal pressured squared response as a function of angle of incidence that is desired. For measurements of sound pressure squared f(θ) is equal to 1. For measurements of sound power propagating down a duct, f(θ) is equal to cos(θ), since the sound power is proportional to the projection of the duct cross sectional area onto a plane perpendicular to the direction of propagation of the sound. This projected area is proportional to cos(θ). For equal energy from every angle of incidence, w(θ) is constant. For equal energy from every element of solid angle, w(θ) is proportional to sin(θ). For every mode with equal energy density, w(θ) is proportional to the number of modes per unit angle of incidence.

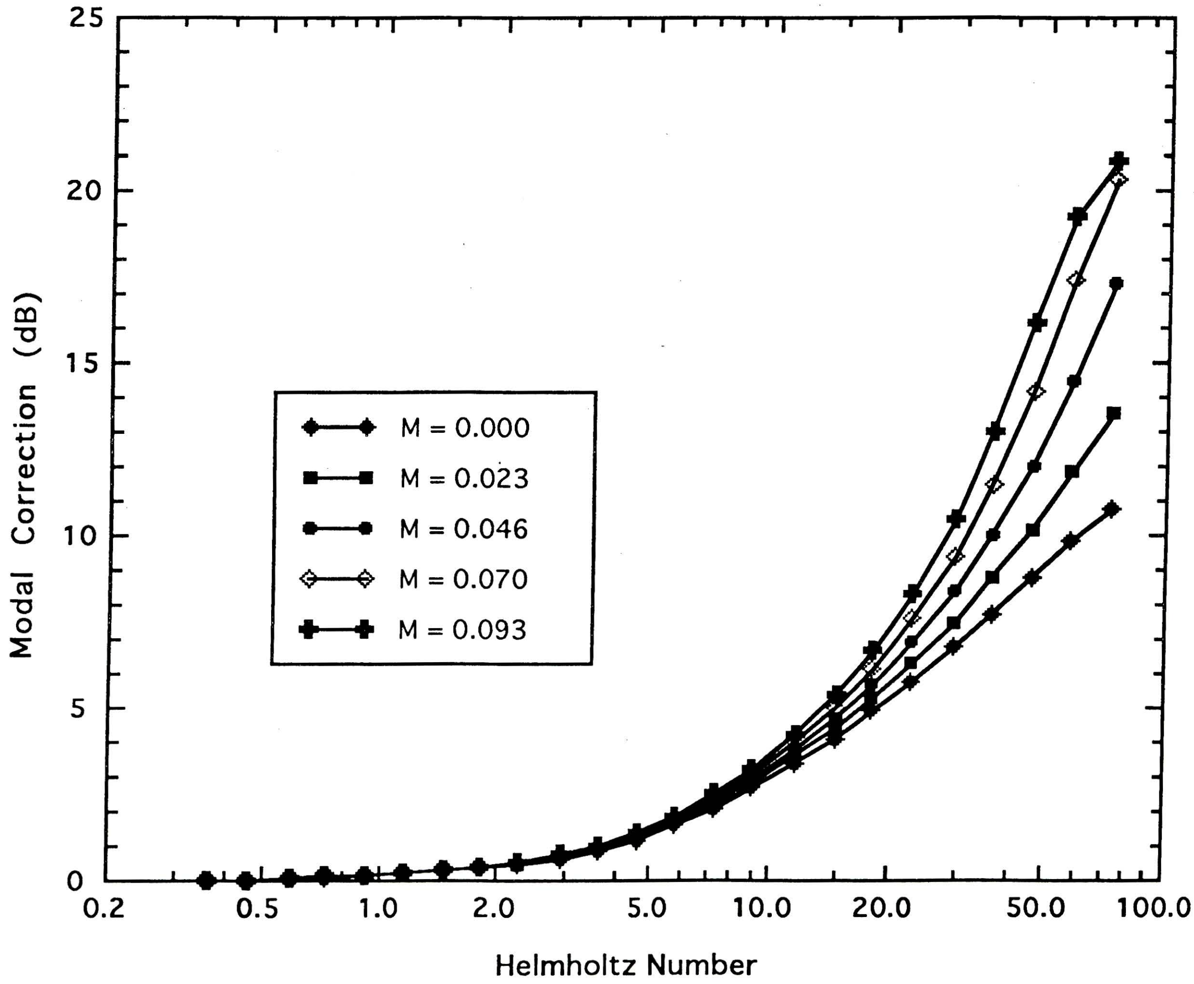

Figure 4. Combined cross mode and flow velocity correction for the case of equal modal energy density as a function of Helmholtz number and Mach number M.

The correction factor C is the factor by which values measured with the microphone turbulence screen are less than the desired values. Thus the values measured with the microphone turbulence screen must be multiplied by the correction factor. In practice the correction factor will be expressed in decibels and will be added to the sound pressure level.

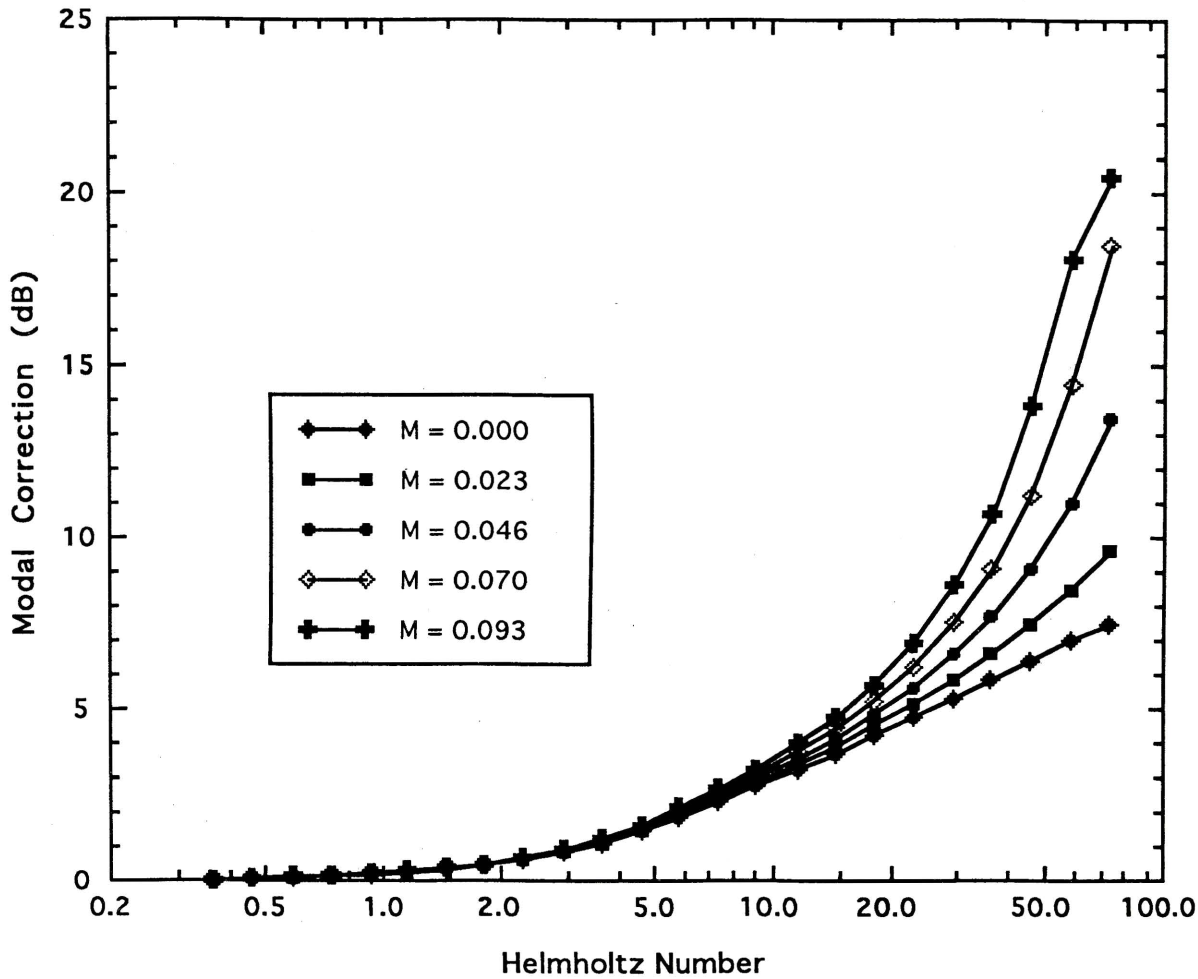

Figure 5. Combined cross mode and flow velocity correction for the case of equal energy incident from every angle of incidence as a function of Helmholtz number and Mach number M.

If the weighting function is relatively constant, the denominator of equation (1) is basically a measure of angular bandwidth. This means that directivity values which are more than 3 dB down will have little effect on the correction factor C. Thus for determining the correction factor C it does not matter greatly if our predictions or measurements of directivity are in error for those values which are more than 3 dB down.

For a temperature of 188 °C, a slit length of 500 mm, a slit width of 1 mm, a slit specific airflow resistance of 332 mks rayl, a tube depth of 11.2 mm, a tube width of 11.8 mm, a duct radius of 5500 mm and a pressure of 101.325 hPa, the correction factor C for measurements of sound pressure level has been calculated for the case of equal modal energy density and the case of equal energy per unit angle of incidence. The calculations were carried out for the third octave frequency bands from 50 Hz to 10 kHz inclusive and for flow velocities of 0, 10, 20, 30 and 40 m/s. The results are expressed in dB as a function of Helmholtz number and Mach number M. The equal modal energy density case is shown in Figure 4 and the equal energy per unit of angle of incidence case is shown in Figure 5.

The correction factor for sound power level measurements of sound propagating along the duct can be obtained by subtracting 2.0 dB from the values in Figure 5. For Figure 5 the adjustment for the lowest value of Helmholtz number is 1.4 dB, and increases to 1.8 dB for the highest value of Helmholtz number. If the propagating sound power level or the average sound pressure level across the duct is being estimated from measurements made at the wall of the duct, then 3 dB should also be subtracted from the values in Figures 4 and 5. This is to account for the fact that the incident and reflected waves are always in phase at the wall of the duct and thus add in the pressure domain rather than in the energy domain.

The values in Figures 4 and 5 are not strongly dependent on the value of the specific airflow resistance of the slit. If the specific airflow resistance of the slit is increased from 332 to 1170 mks rayl, the correction factor for the highest values of Helmholtz number and Mach number M is changed from 20.9 to 21.3 dB in Figure 4, and from 20.5 to 21.1 dB in Figure 5.

5. Turbulence rejection

The turbulence rejection of the microphone turbulence screen relative to a flush mounted Brüel and Kjær Type 4133 microphone was measured in a square cross section duct. The cross section of the duct measured 244 by 244 mm. The duct is described by Shepherd and La Fontaine (1986). Air is sucked through the duct, via an anechoic termination and silencer, by a remotely located fan. Measurements were made at average velocities across the duct cross sectional area of 11.6, 22.5 and 31.4 m/s. At the measurement temperature of 16.9 °C, these velocities correspond to Mach numbers of 0.034, 0.066 and 0.092. The turbulence was produced by the separation bubble from a sharp straight flanged entry to the duct. Although this produced much lower acoustic noise than other turbulence generators tested by Shepherd and La Fontaine, the acoustic noise was still a major problem when measuring the turbulence and limited the amount of turbulence noise rejection that could be measured.

The acoustic noise was removed using a cross spectrum technique. This method determines the acoustic noise by averaging a large number of cross spectra between the microphone turbulence screen and the flush mounted microphone. Because the turbulence detected by the two microphone systems is uncorrelated it is reduced by averaging, while the correlated acoustic noise is not affected by the averaging. The sensitivity of the cross spectrum to acoustic noise is the geometric mean of the sensitivities of the two microphone systems. Thus if $|H|^2$ is the square of the magnitude of the frequency response of the microphone turbulence screen relative to the flush mounted microphone, then the acoustic noise measured by the microphone turbulence screen is the magnitude of the average cross spectrum multiplied by $|H|$ and the acoustic noise measured by the flush mounted microphone is the magnitude of the average cross spectrum divided by $|H|$. H is the relative frequency response measured in the duct with noise coming from the area where the noise due to turbulence is generated. It was measured without flow using random noise from a loud speaker mounted on a baffle bolted across the entrance to the duct. It includes the effects of cross modes, end reflections and the variation of the relative frequency response with angle of incidence.

The turbulence measured by both microphone systems is calculated by subtracting the acoustic noise estimated above from the magnitude of the microphone system's auto spectrum. The difference in acoustic sensitivity between the two microphone systems must be taken into account by dividing the estimated turbulence measured by the microphone turbulence screen by the square of the magnitude of the frequency response $|H_f|^2$ of the microphone turbulence screen relative to the flush mounted microphone. $|H_f|^2$ is measured for zero degree angle of incidence relative to the axis of the microphone turbulence screen in an anechoic room. The anechoic room measurement is used since in actual use the microphone turbulence screen and the flush mounted microphone will have their acoustic measurements corrected using their frequency responses as measured in an anechoic room.

The turbulence rejection ratio t is the ratio of the turbulence measured by the microphone turbulence screen to the turbulence measured by the flush mounted microphone. It is given by

$$t = \frac{|G_{tt}|-|G_{tm}||H|}{|H_f|^2(|G_{mm}|-|G_{tm}|/|H|)}, \tag{2}$$

where G_{tt} is the auto spectrum of the microphone turbulence screen with flow, G_{mm} is the auto spectrum of the flush mounted microphone with flow and G_{tm} is the cross spectrum of the microphone turbulence screen and the flush mounted microphone with flow.

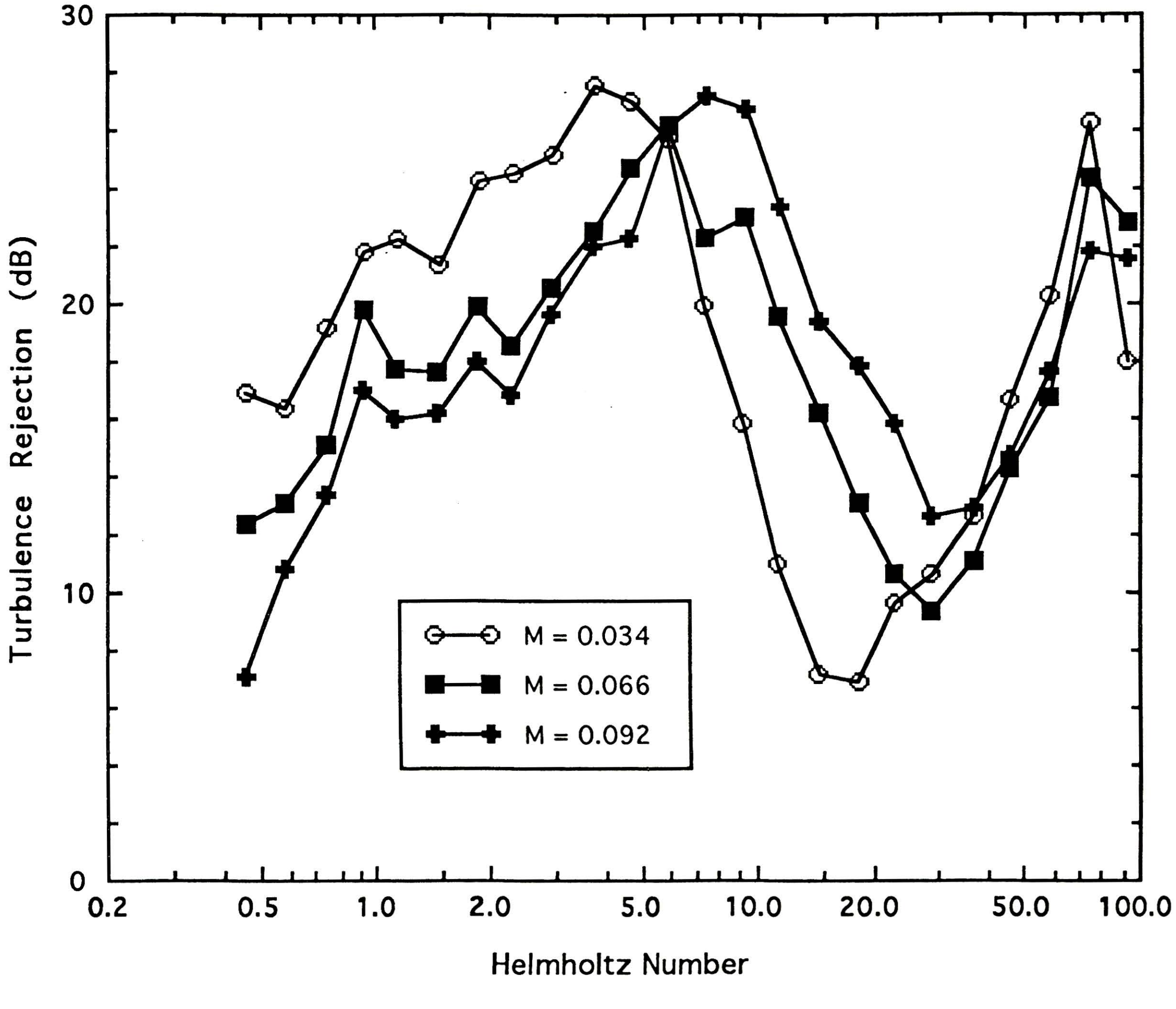

Figure 6. Turbulence rejection ratio for plane wave propagation as a function of Helmholtz number and Mach number M.

Because our equipment only allows measurement of the real part of the cross spectrum directly in third octaves, the measurements of turbulence rejection were made using Fast Fourier Transforms (FFT) and the results were combined into third octaves. The relative standard deviation of a single FFT line in the power domain when measuring random noise signals is equal to one. Averaging n FFT spectra reduces this relative standard deviation to $1/\sqrt{n}$. Thus even taking a relatively large number of averages will not completely eliminate the relative standard deviation of the turbulence and acoustic noise signals. Because the differences in equation (2) are often small compared to the quantities whose difference is being taken, the relative standard deviation of the differences are much larger than the relative standard deviations of the quantities whose difference is being taken.

This means that the uncertainty of the turbulence rejection ratio is often very large and because of statistical variation can sometimes assume physically unrealistic negative values which preclude the taking of logarithms to convert the ratio into decibels. The averaging of the turbulence rejection ratios for each of the FFT lines into third octave bands reduces this uncertainty but again does not completely remove it. The uncertainty is largest where the turbulence measured is very much less than the acoustic signal from the turbulence generator.

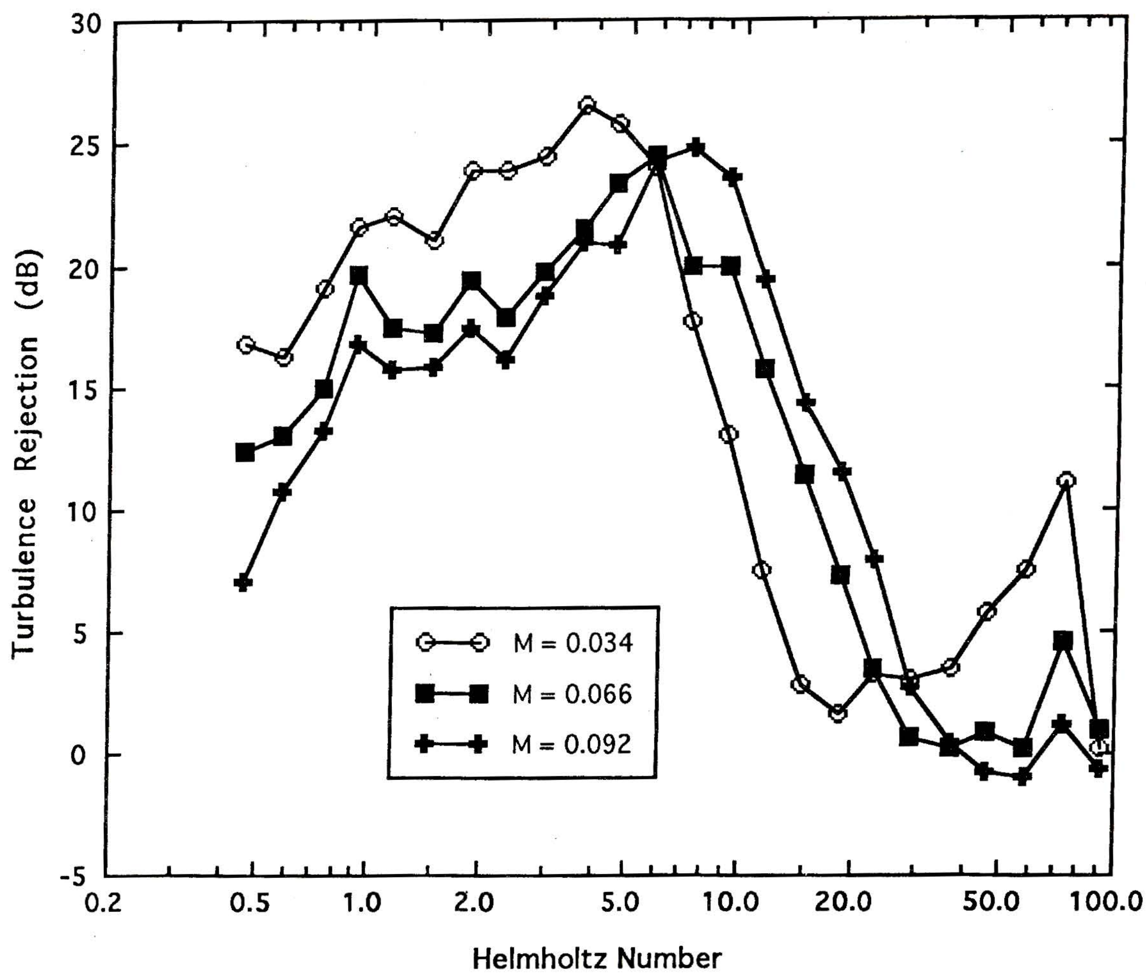

Figure 7. Turbulence rejection ratio for cross mode propagation as a function of Helmholtz number and Mach number M.

The means and the standard deviations of the turbulence rejection ratios in each third octave band are calculated and used to calculate the 95% confidence limits for the turbulence rejection ratios in each third octave band using Student's t distribution. The means and the 95% confidence limits are expressed in decibels after taking their inverses to give positive decibel values. This gives the turbulence rejection

$$T = -10\log_{10}(t). \qquad (3)$$

The values of turbulence rejection T for three different Mach numbers M were measured using an average of one hundred 2048 point FFTs which produced 768 useable FFT lines. To give a more equal distribution of the number of FFT lines in each third octave band, FFTs with maximum frequencies of 12 kHz, 3 kHz, 750 Hz and 187.5 Hz were used. These were combined into third octave bands from 50 Hz to 12 kHz inclusive. The centre frequencies of these third octave bands were then converted to Helmholtz numbers using the temperature of 16.9 °C at which the

Some Applications of Computers in Acoustic Measurement/Analysis

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ABSTRACT

The trend in the acoustic profession is towards greater number of measurements and more sophisticated analysis of data. This is true in both diagnostic measurement and site measurement.

Whilst the old adage "you get what you pay for!" is valid with current sound level meters, it is practical to link less expensive sound level meters to portable office computers to provide a powerful measurement instrument. The prerequisites are software to interrogate the sound level meter and software to run various applications on the data collected.

This paper illustrates some applications, within the building acoustic domain, of the combination of computers and sound measuring equipment.

INTRODUCTION

The trend in the acoustic profession is towards greater number of measurements and more sophisticated analysis of data. This is true in both diagnostic measurement and site measurement. Current technology allows greater detail to be extracted from field measurements.

This paper illustrates some applications, within the building acoustic domain, of the combination of computers and sound measuring equipment.

HISTORICAL VIEW

The acoustics profession has been rapidly evolving in its approach to the way it performs its tasks. We have seen, over the last two decades, a quantum leap forward in the array of techniques and equipment available to analyse tasks. The profession is becoming more scientific in its analysis techniques, both in measurement and calculations. This is partly due to research, education and the availability of more affordable and powerful instrumentation.

From the practitioners view point, current acoustic equipment, whilst still costly, is proving more flexible. Current equipment designs have been enhanced by the electronics industry's abilities to increase the power and performance of the current range of instrumentation. A redevelopment of design philosophies were required to switch from or selectively mate the best of the analog devices with the newer digital technology.

FIELD MEASUREMENTS

Gathering acoustic data out in the field has been a major focus with a range of current field use equipment that has the capabilities previously found only in laboratory based equipment. Often the new breed of equipment is more capable than the older laboratory equipment. The power of these new machines has caused the concept of field measurements to change.

The overall concept in any form of field measurement, is to be able to gather sufficient information in the most economical method possible. Field measurement can become a costly exercise when, due to incomplete data, or poor technique, additional measurements are required.

The cost of field measurement can be analysed in the following manner:

(i) Transportation, time, travel and accommodation;
(ii) Time for measurement;
(iii) Laboratory analysis;
(iv) Cognitive processing;

Cost reductions in the measurement procedure outlined above can be a major factor in the profitability of a project. The major concern in conducting field measurements is to gather sufficient pertinent data. In a problem solving situation, the acoustician requires to be in a position to interpret the data as it becomes available. In this way the acoustician can be assured that all necessary data has been collected.

The current octave band real time sound level meters when interfaced with a computer, offer significant diagnostic power for field measurements. In addition, they represent enormous time saving, in that all frequency data is measured simultaneously.

COMPUTER, SOUND LEVEL METER CONNECTION

Recently the School of Architecture at Curtin University purchased an octave band real time sound level meter for use in investigative projects.

The major influences in this decision can be listed:

(i) Real time analysis and display of octave data from 32 Hz up to 8 KHz, plus overall,
(ii) Memory capacity to measure and store 1500 complete spectras
(iii) Measurement modes including: Lp, Leq, LAE, LMAX, LN.
(iv) Serial Interface (RS232C)
(v) Precision Integrating, Type 1
(vi) Cost

This sound level meter has been interfaced with a computer and specific software has been written to interrogate the sound level meter, retrieve and store all relevant data. In addition computer applications have been written for specific measurement procedures.

Spatial Mapping

As a student project the spatial mapping of an auditorium and amphitheatre was carried out. The students were given brief instructions on the sound level meter operation and computer interface. With the auditorium evaluation, two students were able to complete the measurements in approximately 2 hours. Octave band sound pressure levels were recorded at 84 positions and stored in memory. At the completion of the measurement procedure, the data was down-loaded to a spreadsheet.

The purpose of the exercise was to consider the effect of acoustic reflectors located above the stage area. From the data collected, spatial contouring was used to present a visual map of the results. The students chose to present the data via Computer Aided Design software. In this case, they used EAGLE, though AUTOCAD has similar capabilities.

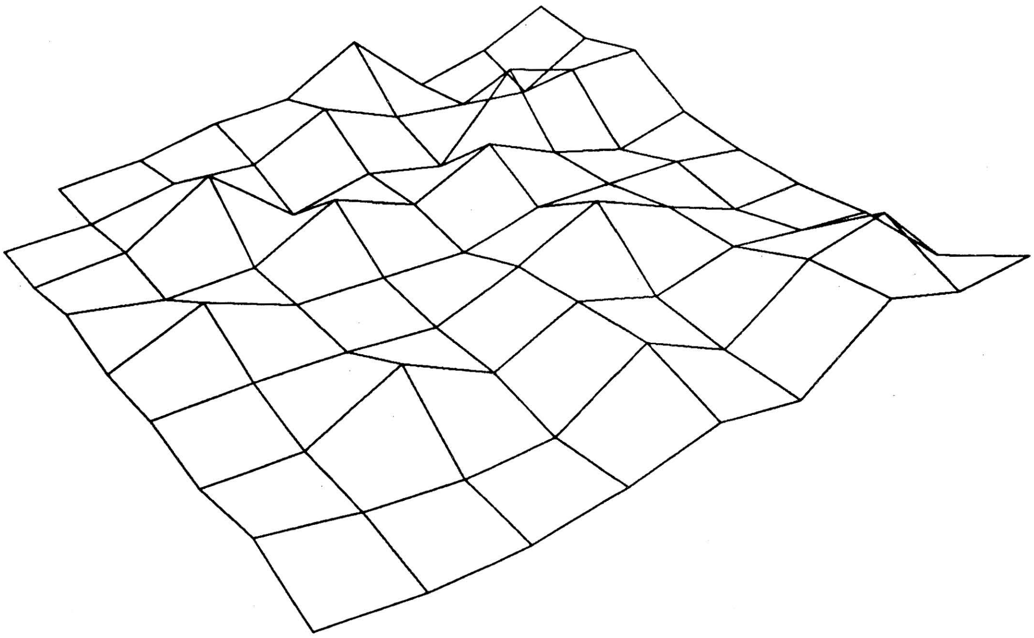

Figure 1 Spatial Mapping

Essentially the data required for these spatial maps, comprises the X and Y coordinates, ie the seat locations, plus the Z or height data. The appropriate frequency, or dB(A) data was used for the Z column.

Reverberation Time

With an octave band real time sound level meter, it is possible to conduct detailed reverberation analysis. The basic sound level meter offers the capabilities of measuring sounds at precise time intervals and then storing the data automatically into its memory. With time intervals selectable from 2 milliseconds to 10 seconds, this allows a maximum 'recording' time between 3 seconds and 4 hours. The instrument also features a trigger feature allowing impulse sounds to be used, in the form of clapper boards and pistols.

While decays may be viewed via the LCD screen of the sound level meter, increased flexibility can be achieved when the sound level meter is connected to a computer. Software has been developed enabling a IBM compatible computer to interrogate the sound level meter enabling semi-automatic reverberation times, across all frequencies to be made.

The software currently utilises a recorded ambient noise level to provide a minimum plus 10 dB cut-off for the calculations. A linear interpolation is used over the normal -5 dB down to the cut-off or base of the dynamic range as appropriate. An early decay time is also calculated in a similar manner.

Down-loading of 1500 full spectrums can take up to 6 minutes. Care needs to be taken to balance the quantity of data required to the time for down-loading. In other words, an RT of 1.2 seconds sampling at 2 milliseconds intervals requires approximately 400 spectra. An advantage of this instrumentation set-up is that a clapper board or pistol can be used as the source. As a by-product it is possible to record on disk, the rise time and decay times for each measurement.

This procedure provides the data in a format suitable for decision making at the time of the measurement.

Machinery Noise Database

This instrument configuration is being used in noise survey work where large numbers of machines are measured. The advantage presented are the sound levels can be down-loaded directly into a database.

Custom designing the interface software allowed:

(i) additional survey information to be recorded
(ii) measurements to be forced into dB(Lin) even if recorded in dB(A)
(iii) automatic up-loading to the database.
(iv) up-loading to the Hearing Protection Calculation program

This new procedure produced significant saving in time production of normal reports, and updating the database. They also improved the reliability of data by avoiding errors in data entry.

The cost benefits of such a database are self evident, especially in cases involving enquiries regarding equipment purchases, such as is product A quieter than product B, or please recommend the best three brands and type of a particular product. Table 1 below shows a sample listing:

Type Make	FREQUENCY								LEVEL (Leq.A)		PEAK (Lin)	
Model	63	125	250	500	1K	2K	4K	8K	Free	Working	Free	Working
ANGLE BOSCH	78	79	82	65	75	75	92	90	88	93	101	109
ANGLE BOSCH PWS115	78	82	88	61	72	71	88	89	87	93	99	108
ANGLE HITACHI METAL	72	81	87	63	65	64	95	90	87	97	99	103
ANGLE HITACHI	75	77	88	64	70	71	92	90	96	91	110	105
ANGLE HITACHI	69	85	86	54	58	61	89	92	91	94	101	115
ANGLE HITACHI 100C	82	89	90	61	65	70	97	97	88	102	100	118
ANGLE HITACHI G10SD	76	81	85	55	64	67	93	92	95	95	106	109
ANGLE HITACHI G10SD	74	77	83	66	65	65	92	93	98	95	109	108
ANGLE HITACHI G10SD	73	76	82	53	59	65	92	92	93	95	110	112
ANGLE HITACHI PDP-100C	85	86	91	63	72	70	97	107	94	107	108	115
ANGLE HITACHI PDP-100C	81	81	89	65	82	76	92	91	89	96	104	114
ANGLE HITACHI PDP-100C	71	73	82	62	74	66	95	86	88	93	101	105
ANGLE HITACHI PDP-100C	75	81	86	55	59	62	93	92	91	98	102	110
ANGLE HITACHI PDP-100C	74	85	89	54	62	63	93	92	86	98	99	110
ANGLE HITACHI PDP-100C	75	81	86	55	59	62	93	92	91	98	102	110
ANGLE HITACHI PDP-100C	84	90	95	68	77	80	93	96	93	100	105	114
ANGLE HITACHI PDP-100C	76	81	93	64	74	70	94	92	91	97	103	110
ANGLE MAKITA	77	83	95	62	71	73	99	96	89	100	92	113
ANGLE MAKITA 9501B	75	78	84	63	68	68	93	93	87	97	101	111
ANGLE MAKITA 9501B	81	86	89	61	72	71	88	86	88	91	100	105
ANGLE MAKITA 9501B	71	79	84	55	64	65	87	87	89	93	94	106
ANGLE MAKITA 9503B	73	82	91	60	66	67	99	98	92	101	105	115
ANGLE RYOBI	80	83	90	62	67	72	93	92	94	95	106	108
ANGLE RYOBI G1005	70	80	88	53	57	61	88	88	99	93	111	105
ANGLE RYOBI HG-100	70	80	85	56	66	67	91	90	92	96	104	107
ANGLE RYOBI SG-1000	82	89	98	65	69	72	96	93	89	99	104	102
ANGLE RYOBI SG-1000	83	85	86	68	83	87	89	92	89	97	101	111
ANGLE SKIL	75	82	84	61	65	75	92	94	87	95	99	110
ANGLE WOLF 4594	76	80	88	59	66	68	99	99	90	102	104	119
ANGLE WOLF 4594	72	75	81	58	62	66	93	95	89	97	102	110

Table 1 Machinery Noise Database

RASTI

Another example of computer interfacing is in conjunction with RASTI (Rapid Speech Transmission Index) measurement equipment.

The measurement procedure using the RASTI equipment has benefited with the interfacing of computers. The RASTI machine generates significant information which is readily down-loaded via the RS232C interface. The advantage of connecting a computer, is the data can be captured, then later processed to form the typical graphical output (see figure 2 below), as well the summary data (see Table 2 below).

SUMMARY OF RASTI MEASUREMENT DATA
JOONDALUP W.A.C.A.E. LECTURE THEATRE

Pos'n	Rasti	STI		Level (dB)		S/N Equivalent		EDT Equivalent	
No.		500Hz	2kHz	500Hz	2kHz	500Hz	2kHz	500Hz	2kHz
1	0.69	0.73	0.66	61.50	49.70	6.80	4.90	0.54	0.71
2	0.66	0.57	0.73	62.50	51.30	2.10	6.90	----	0.49
3	0.70	0.66	0.74	62.90	52.00	4.80	7.10	----	0.48
4	0.68	0.65	0.71	61.80	50.60	4.60	6.20	0.73	0.56
5	0.58	0.53	0.61	59.60	48.70	0.80	3.40	1.30	0.87
6	0.60	0.64	0.58	59.60	47.20	4.20	2.30	0.78	1.00
59	0.68	0.66	0.71	58.90	49.00	4.70	6.20	0.73	0.56
60	0.64	0.63	0.65	59.80	48.30	3.80	4.40	0.82	0.77
61	0.67	0.65	0.68	60.30	49.40	4.60	5.50	0.73	0.64
62	0.65	0.64	0.67	60.20	49.90	4.20	5.00	0.78	0.69
63	0.67	0.63	0.70	60.70	50.70	3.90	6.00	----	0.58

Average Rasti = 0.66
Standard Deviation = 0.03

QUALITATIVE ASSESSMENT

	Subjective Intelligibility
0 % of sample area	Excellent
96 % of sample area	Good
3 % of sample area	Fair
0 % of sample area	Poor
0 % of sample area	Bad

Table 2 RASTI Summary Data

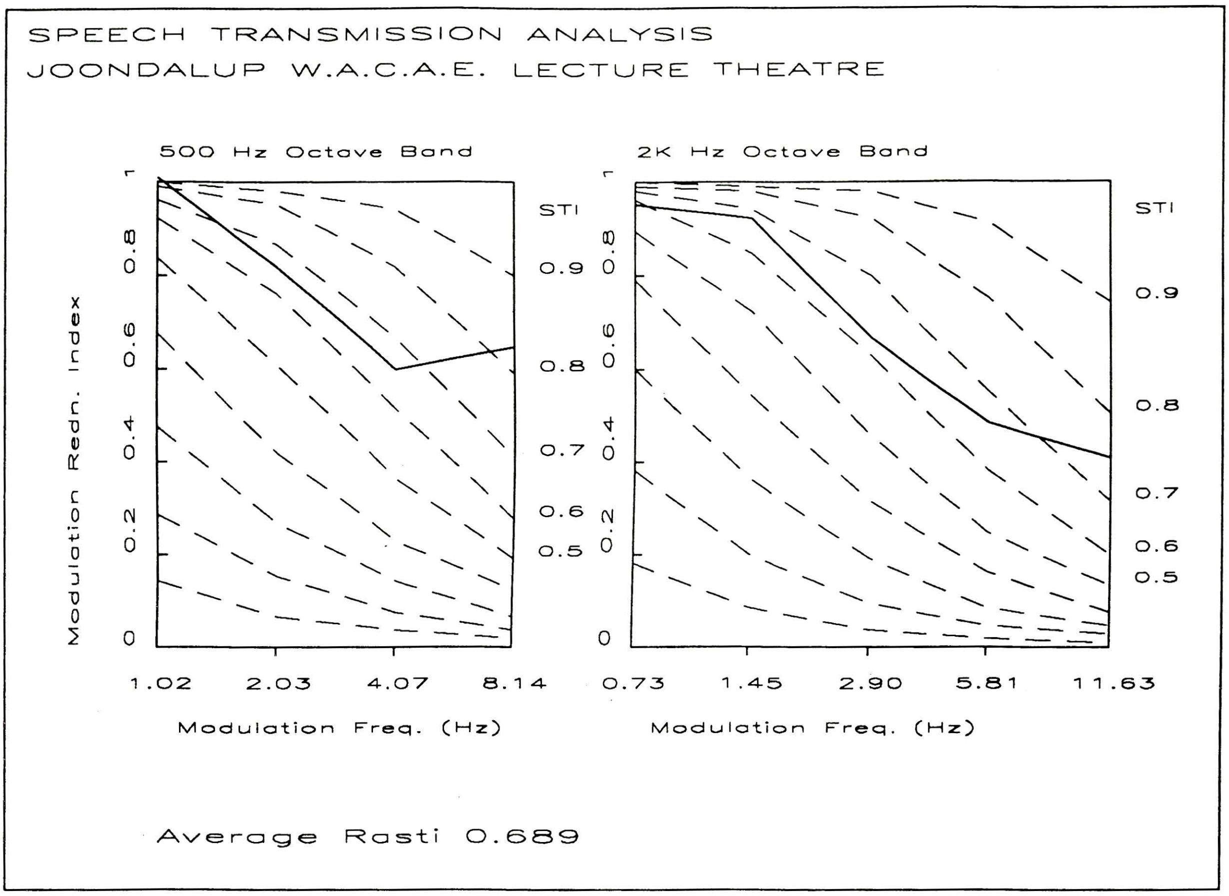

Figure 2 RASTI

TOO QUIET FOR THEIR OWN GOOD

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Principal, Geoffrey Barnes & Associates.

ABSTRACT.

The usual environmental noise problem for the acoustical engineer to solve, involves excessive noise generated by industry, entertainment or transportation in close proximity to residential properties.

However along with the increasing community awareness of the harmful effects of noise, has developed the not uncommon acoustic problem of the *'too quiet'* environment which brings about its own set of problems.

Three case histories are presented where a perceived annoying environmental noise problem was in fact a difficulty exacerbated by a low ambient sound level.

This paper seeks to establish that the human ear requires a reasonable degree of natural ambient sound for well being.

INTRODUCTION.

This paper addresses an acoustic difficulty which the author has encountered on increasing occasions in providing solutions to clients' noise problems. This difficulty may be due to an increasing community awareness of both the harmful physiological and psychological effects of noise.

The acoustic difficulty is that of the very low ambient sound level environment, which generally manifests itself following retirement to bed awaiting sleep.

The paper highlights three different situations where a very low background sound level has emphasised an annoyance which would be effectively *'absent'* in a *reasonable* background sound level environment.

In these cases, some understanding of the aural processes, and counselling skills were found an important adjunct to a working knowledge of acoustics, in presenting the client with an appropriate and acceptable solution.

DISCUSSION.

International and Australian Standards have been established for recommended ambient sound levels within areas of occupancy within buildings, which set guide-lines for an optimum range for background sound levels.

The Australian Standard AS 2107- 1987 sets out a range of recommended ambient noise levels in areas of occupancy within buildings. (1)

TABLE 1.

RECOMMENDED DESIGN SOUND LEVELS FOR DIFFERENT AREAS OF OCCUPANCY IN BUILDINGS.

Type of occupancy	*Recommended design sound level*	
	Satisfactory	*Maximum*
Private houses (rural & outer suburbs)		
Sleeping areas	***25 dB(A)***	***30 dB(A)***
Private houses (inner suburbs)		
Sleeping areas	***30 dB(A)***	***35 dB(A)***

Ambient sound levels in *excess* of the design sound levels as set out in ***Table 1*** may become increasingly unacceptable as the sound level increases, as shown in ***Table 2.*** (2)

TABLE 2.

ESTIMATED PUBLIC REACTION TO NOISE.

Amount in dBA by which Noise Level exceeds suggested noise criterion	*Strength of Public Reaction.*	*Expression of public reactions in a Residential Situation.*
0 - 5	***No***	***From no observed reaction to sporadic compaints.***
5 - 10	***Little***	***From sporadic complaints to widespread complaints.***
10 - 15	***Medium***	***From sporadic to widespread complaints to threats of community action.***
15 - 20	***Strong***	***From widespread complaints to threats of community action.***
20 - 25	***Very strong***	***From threats of community action to vigorous community action.***
25 & over.	***Extreme public reaction.***	***Immediate direct community action & personal action.***

From the author's experience, the potential for acoustic difficulties to arise ***also increases*** as the sound level ***decreases*** below the design levels presented in ***Table 1***, in the quiet environment generally conducive to sleep.

The strong psychological effects which can be generated at extremely low sound levels are well known, and have been used by security agencies as a *manipulative tool* for unco-operative detainees.

A much milder effect, but nevertheless one which stimulated significant annoyance can be experienced by the intrusive effects of unwanted sounds into the domestic sleeping environment, particularly in the absence of a reasonable background sound level.

Such an inappropriate acoustic environment may increase aural sensitivity causing sleep deprivation, and in more serious cases, irrational behaviour necessitating medication or professional counselling.

During the evening and into the night period, in most suburban environments, the ambient sound level falls. Traffic decreases in volume, neighbourhood activities cease, and activity noise within the home generally subsides as the occupants retire. The residual effective ambient sound level in the sleeping area is due to a range of both internal and external noises.

Internal sound sources during the quiet night period may include:

Electrical appliances, such as freezers and refrigerators, heating units, hot water services, clocks, and the sounds of other occupants.

External sounds which may vary during the night period may include the following:

Natural sounds such as rain, wind in trees, insects, croaking frogs and night birds. Background traffic noise and distant transportation noise, which is ever present although substantially diminished during the night period, with no local content in some suburban areas.

Typical intrusive sounds which have caused acoustic difficulties and become manifest as the ambient sound level has fallen below ***30 dB(A)***, include the following cases of which three are discussed in further detail.

Case examples of noise difficulties experienced in a low ambient sound level sleeping area environment.

Sound generated by squash balls at a squash centre.

Raised voices in a quality villa unit.

The sound of an unidentifiable persistent barking dog.

Next door to a night club closing at 3.00 am.

Extremely distant traffic noise.

The distant sound of entertainment music.

Wedding Reception Venue car-park noise.

The sound of one's own circulatory system.

SELECTED EXAMPLES.

1. RAISED VOICES IN A QUALITY VILLA UNIT.

Background.

In the multi-tenancy situation such as strata-titled quality villa units, it is quite likely that master bedrooms will have common walls.

Where this wall is of inferior construction, and/or conversational speech is generated at a higher level than normal, then the potential for an acoustic difficulty is hightened, allowing audibility to occur, particularly as the ambient sound level falls.

The client's acoustic problem was intrusion of voices into the bedroom through the common wall between the villa unit apartments, particularly during arguments which occurred regularly.

Raised voices were clearly audible, generally late at night in the adjacent bedroom when the ambient sound level was low.

A field transmission loss evaluation of this inter-connecting wall established that it performed poorly in relation to an optimum acoustic performance, with poorly sealed jointing allowing significant flanking sound. (3)

TABLE 2.

ACOUSTIC LIMITATION OF CLIENT'S INTER-TENANCY WALL.

	'A' weighting.	Octave band Centre Frequency (hz) 125	250	500	1k	2k	4k
Typical male voice in conversation at one metre. (dB)	*59*	*60*	*62*	*60*	*46*	*43*	*45*
Typical raised male voice at one metre. (dB)	*72*	*60*	*70*	*73*	*62*	*58*	*54*
Typical shouting male voice at one metre. (dB)	*78*	*55*	*74*	*78*	*71*	*65*	*59*
Noise Reduction through client's bedroom wall. (dB)		*38*	*39*	*47*	*52*	*53*	*62*
Resultant sound level of shouting, in client's bedroom. (dB)		*17*	*35*	*31*	*19*	*12*	-
Ambient sound level in client's bedroom. (dB)	*29*	*34*	*30*	*26*	*24*	*22*	*21*
Resultant excess (dB) providing audibility.		-	*5*	*5*	-	-	-

Acceptable Solution.

Having identified the major flanking paths for sound passing between the two bedrooms, the client was able to acoustically seal the joints eliminating the major flanking sound paths, thus optimising the acoustic performance of the brick wall.

The client then rearranged the bedroom, shifting the bed to the other side of the room, significantly increasing the distance from the inter-tenancy wall, along with *the hope that with the intensity of the neighbouring bedroom arguments, one party will soon quit,* restoring the tranquillity of the environment.

General Comments.

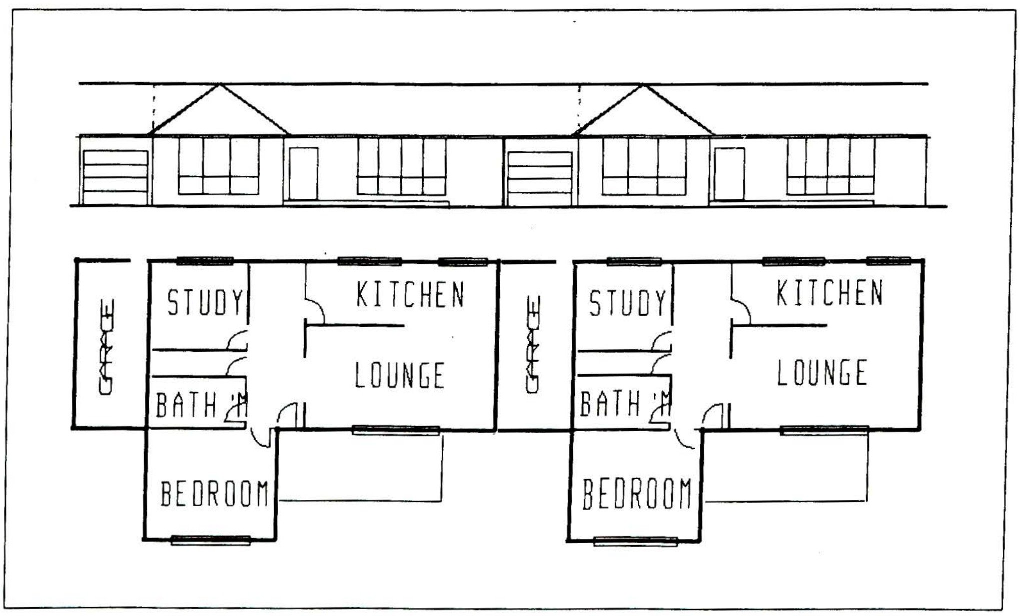
From experience, it appears that many suburban quality villa units are built as economically as possible, with minimal thought giving to adequate sound isolation between tenancies. The wall type may comply with building and planning regulations, but the construction is not adequate to provide optimum sound isolation. In addition, close proximity of bedroom windows may provide a significant flanking path for sound between adjacent bedrooms.

In addition to the above comments it should pointed out that with the present trend in construction of luxury multi-tenancy apartments in Melbourne, using single thickness light weight autoclaved concrete block work for inter-tenancy common walls, it is expected that these luxury apartments will provide inadequate sound isolation in terms of ensuring voice inaudibility between the apartments.

In past times the common wall between tenancies, as well as being of pressed clay solid brick construction, passed through the roof to form a parapet, and in doing so provided an optimum sound barrier between the two areas.

To minimise this acoustic isolation problem as experienced in this quality villa unit development, particularly in low ambient sound level environments, attention should be given to location of master bedrooms to provide no commonality of walls, with each other, and preferably with no other living area or utility area, as shown below.

FIGURE 1. QUALITY VILLA UNITS WITH ACOUSTICALLY ISOLATED BEDROOMS.



2. THE SOUND OF AN UNIDENTIFIABLE PERSISTENT BARKING DOG.

Background.

The client, who lives in an outer suburban area, where properties are divided into 2 hectare allotments was annoyed by the intermittent but persistent barking of a dog in the neighbourhood. Repeated night-time reconnoitres of the area had failed to identify the dog, with neighbours also being of no help in establishing its identity.

The background sound level within the bedroom was in the order of *22 dB(A)* with the windows closed. Bearing in mind the high sound level generated by a barking large dog, and the impulsive nature of barking which hightens annoyance (3) a distant persistent barking dog in this instance created a significant intrusive noise problem.

TABLE 3.

INTRUSIVE EFFECTS OF DISTANT BARKING DOG.

		Octave Band Centre Frequency (hz)					
	'A' weighting.	*125*	*250*	*500*	*1k*	*2k*	*4k*
Barking german shepherd dog, typical peak sound level at 1 metre. (dB)	*94*	*74*	*81*	*95*	*90*	*87*	*75*
Approximate resultant peak sound level at 500 metres from the dog, at residential facade. (dB)		*35*	*30*	*41*	*42*	*37*	*20*
Actual noise reduction through the front window of the residence. (dB)		*18*	*19*	*18*	*18*	*15*	*10*
Expected sound level peak from barking dog in bedroom. (dB)		*17*	*11*	*23*	*24*	*22*	*10*
Ambient sound level in bedroom. (dB)	*22*	*30*	*24*	*20*	*21*	*18*	*17*
Excess barking sound level in bedroom. (dB)		-	-	*3*	*3*	*4*	-

Acceptable solution

The client made use of an FM-tuner with a graphic equaliser, shaping the *random noise* between stations to provide a pleasant continuous background sound which was generated in the lounge of his home.

This sound filtered through to the bedroom effectively removing the annoyance of the barking by raising the level of internal noise within the bedroom. This effectively reduced the magnitude of sound level difference between the bark impulse peak and the ambient sound level within the bedroom.

General Comments.

Having failed to establish the identity of the barking dog, the client, had not been able to request the dog's owner to prevent the barking. Further, he was not able to involve the Municipality in taking action on his behalf, and in any case, other neighbours appeared not unduly upset with the barking.

Thus not being able to rectify the noise problem at the source, which is generally the most desirable point of acoustic treatment, carrying out the measure outlined above was found the most appropriate and cost effective.

It should also be borne in mind that impulsive noise may still be audible at a level below the actual measured background sound level, particularly when using a weighted sound level measurement.

3. THE SOUND OF ONE'S OWN CIRCULATORY SYSTEM.

Background.

This phenomenon has been experienced on several occasions, of which one is highlighted. The client was distressed and on medication when I was engaged to solve this acoustic problem, which apparently had been getting worse.

The engagement brief was to identify and control an environmental noise which had to that date eluded inspectors from various Statutory Authorities, as well as an 'acoustic expert'in identification of the noise source.

One inspector indicated that he could also hear the noise, although my client's spouse could not. The noise source was *'located'* in an industrial estate some 1km distance from the complainant's home, but when eliminated, the noise continued!

The background sound level in the sleeping area was below *20 dB(A)*, and the noise was explained to be like the sound of an idling diesel bus! *Vwoom.. Vwoom.. Vwoom..*

The sound being heard, I suspected, was my client's pulse, detected in the extremely quiet ambient environment, as had been observed by the author on previous occasions.

On diplomatically expressing my explanation of the noise source, the client pondered this and remarked simplistically:

> *`I can prove your hypothesis quite simply. If I place my fingers in my ears, according to your explanation, the sound I am hearing should remain. If it is from an external source then I would expect the level to diminish..... Your right, the sound is still there!'*

On further investigation it was discovered that the bedroom windows had been recently double glazed to escape from the annoying sound, and new heavy drapes had been fitted to the windows. In addition the client had organised for his spouse to sleep in a separate room to minimise his disturbance!

Acceptable solution.

The acoustic difficulty was easily rectified in this case by a series of measures.

1. The client invited his spouse back to bed! After consultation with his medical practitioner, ceased tranquilliser medication,

2. Removed the double glazing and opened the windows to let the environmental sounds filter into the bedroom,

3. Opened the bedroom door to allow residual ambient sound within the house, to pass into the bedroom, and was proposing to purchase a grandfather clock to further mask the problem sound.

General comments.

It is an established fact that at low ambient sound levels, body noises and also tinnitus (ringing in the ears which often accompanies a loss in hearing) become audible, and can become a severe annoyance. If these sounds are mistakenly thought to be of external nature, the annoyance and frustration is hightened.

When an acoustical consultant is confronted with a perceived annoyance with a sound which can be detected neither by the consultant's sensitive ears, nor sophisticated instrumentation, thought should be given to the possibility of an internal auditory stimulus.

At this point, where an acoustical solution may not be considered appropriate, the client should be encouraged, without causing undue alarm, to seek professional advice for what is likely a medical condition which may respond to suitable audiological or other treatment.

CONCLUSION.

The case histories presented all had a common basis. The ambient sound level in the sleeping area was too quiet for the occupant's comfort. In each case, raising the ambient sound level within the room would have lessened the acoustic problem, by masking the intrusive noise where elimination of the noise source was inappropriate.

It is desired by the author, that this paper will serve as a gentle reminder to acoustical engineers when offering solutions for noise problems associated with a low ambient sound level environment, that modification of this environment may be as important in providing an appropriate solution to the acoustic difficulty as identifying and quantifying the actual noise source.

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2. S.A.A. Doc 1707, ***DRAFT AUSTRALIAN STANDARD CODE OF RECOMMENDED PRACTICE FOR NOISE ASSESSMENT IN RESIDENTIAL AREAS 1971*** p 19.

3. Geoffrey A. Barnes, ***A CATALOGUE OF ACOUSTIC SHORTCOMINGS IN THE SCHOOL MUSIC DEPARTMENT***, AAS 1990 Conference proceedings, p 6.6

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A STUDY OF THE STRUCTURE-BORNE VIBRATIONS OF PLATE-BEAM SYSTEMS USING STATISTICAL ENERGY ANALYSIS

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ABSTRACT

Statistical Energy Analysis (SEA) provides a means of evaluating the average response of interconnecting elements based on energy flow relationships. This method is applicable to the analysis of complex mechanical and acoustical systems, especially at high frequencies where numerical methods are difficult to apply, mainly because of the large number of degrees of freedom involved and the high level of modelling detail required. This paper describes analytical procedures for calculating the coupling loss factors of plate-beam systems commonly used in naval ship constructions. The calculated coupling loss factors are then used to study the vibrational response of two plate-beam systems. The SEA predictions show good agreement with experimental results.

1. INTRODUCTION

Statistical Energy Analysis (SEA) is a technique for analysing the average vibrational response of interconnecting elements in a system and has been used successfully in a large number of engineering applications [1]. One of the major advantages of SEA is that it is capable of providing a fast estimation of the average response in different parts of a complex structure with a minimum of input data compared with other methods of analysis such as the Finite Element Method. The parameters required in SEA to describe the energy flow relationships between system elements include the modal density, the material loss and the coupling loss factors. For SEA to be used with a high degree of confidence, an accurate estimate of the above parameters is essential. The evaluation of material loss factors and modal densities of system elements has been considered by several authors [2 - 6]. Normally, the material loss factor may be measured experimentally or estimated from known material properties data. The modal density may be determined theoretically for simple homogeneous elements or measured experimentally for the case of complex elements. This paper describes analytical methods to determine the coupling loss factors of plate-beam junctions typical of those found in ship constructions based on wave transmission theory. The use of SEA to predict vibrational response is demonstrated by two examples of plate - beam systems. Results of SEA predictions and experimental measurements are presented and discussed.

2. COUPLING LOSS FACTOR

The coupling loss factors used in SEA define the amount of energy flow from one sub-system to the other. It can be shown [2] that for two coupled sub-systems, the power loss by sub-system 1 due to coupling to sub-system 2 is proportional to the energy of sub-system 1 and may be expressed in terms of the coupling loss factor as:

$$P_{12} = \omega\, \eta_{12} \langle E_1 \rangle, \tag{1}$$

where $\langle E_1 \rangle$ = time averaged energy of sub-system 1,
η_{12} = coupling loss factor between sub-systems 1 and 2,
ω = frequency.

Consider an SEA system that consists of two coupled plates as shown in Figure 1. If plate 1 carries a diffuse vibration field incident on the junction, the total power transmitted to plate 2 may be obtained by multiplying the power of plate 1 with the transmission efficiency and then average the results over the entire range of incident angles (see Cremer *et al.* [8],p 426):

$$\begin{aligned} P_{12} &= (c_{g1}\, L_c \langle E_1 \rangle / A_1\, 2\pi) \int_{-\pi/2}^{\pi/2} \tau(\alpha) \cos\alpha\, d\alpha\,, \\ &= (c_{g1}\, L_c \langle E_1 \rangle / A_1 \pi) \int_0^1 \tau(\alpha)\, d(\sin\alpha), \\ &= (c_{g1}\, L_c \langle E_1 \rangle / A_1\, \pi)\, \tau_m, \end{aligned} \tag{2}$$

where L_c = coupling line length,
c_{g1} = group velocity of plate 1,
$\tau(\alpha)$ = transmission efficiency at an incident angle α,
τ_m = mean transmission efficiency,
A_1 = area of plate 1.

From equations (1) and (2), the coupling loss factor between two coupled plates is given by:

$$\eta_{12} = (c_{g1}\, L_c / \omega\, \pi\, A_1)\, \tau_m. \tag{3}$$

The above analysis shows that the mean transmission efficiency τ_m may be used as a parameter for the calculation of coupling loss factor. In the following sections, the derivation of transmission efficiency for plate-beam systems will be discussed.

3. GOVERNING EQUATIONS FOR PLATE VIBRATIONS

Figure 2 shows a structural junction that consists of a number of plates coupled to a beam. The plates are assumed to be infinite along the y and x_1 to x_n directions while the beam is assumed to be infinite in length. The cross sectional dimensions of the beam are assumed to be small compared with the wave length so that the boundary conditions may be applied to the origin of the junction. Plate 1 is subjected to an oblique incident wave which can be either bending (B), longitudinal (L) or transverse shear (T). The incident wave is partially reflected and partially transmitted at the junction as bending, longitudinal and transverse shear waves as shown. The elastic deformations due to these reflected or transmitted waves in an arbitrary plate along a set of local co-ordinates x, y and z are defined as u, v and w respectively (see Figure 3). Using thin isotropic plate theory, the governing equations of motion for plate deformations may be derived (see Love [7] p 496).

The bending wave equation being:

$$\nabla^4 w + [12\rho(1-\mu^2)/Eh^2]\partial^2 w/\partial t^2 = 0, \tag{4}$$

and the in-plane wave equations are:

$$\partial^2 u/\partial x^2 + [(1-\mu)/2]\partial^2 u/\partial y^2 + [(1+\mu)/2]\partial^2 v/\partial x\partial y - [\rho(1-\mu^2)/E]\partial^2 v/\partial t^2 = 0, \tag{5}$$

$$\partial^2 v/\partial y^2 + [(1-\mu)/2]\partial^2 v/\partial x^2 + [(1+\mu)/2]\partial^2 u/\partial x\partial y - [\rho(1-\mu^2)/E]\partial^2 u/\partial t^2 = 0, \tag{6}$$

where $\nabla^4 = [\partial^2/\partial x^2 + \partial^2/\partial y^2]^2$,
ρ = material density,
μ = Poisson's ratio,
h = plate thickness,
E = Young's modulus.

The in-plane wave equations are functions of the plate deformations u and v. To obtain a solution for these equations, one can make use of the velocity potential ϕ and stream function ψ defined as follows (a detailed discussion on the use of velocity potential and stream function to analyse vibration waves is given by Cremer *et al.* [8], p 138):

$$u = -\partial\phi/\partial x - \partial\psi/\partial y, \tag{7}$$

$$v = -\partial\phi/\partial y + \partial\psi/\partial x. \tag{8}$$

Using equations (7) and (8), each of the in-plane wave equations is reduced to a function of one dependent variable (ϕ or ψ) only:

$$\nabla^2\phi - [\rho(1-\mu^2)/E]\ \partial^2\phi/\partial t^2 = 0, \tag{9}$$

$$\nabla^2\psi - [2\rho(1+\mu)/E]\, \partial^2\psi/\partial t^2 = 0. \tag{10}$$

The general solutions to equations (4), (9) and (10) may be expressed as:

$$w = W \exp(k_{Bx}x + k_{By}y + j\omega t), \tag{11}$$

$$\phi = \Phi \exp(k_{Lx}x + k_{Ly}y + j\omega t), \tag{12}$$

$$\psi = \Psi \exp(k_{Tx}x + k_{Ty}y + j\omega t). \tag{13}$$

The velocity potential ϕ is associated with longitudinal waves while the stream function ψ with transverse shear waves. For the solutions to be valid, the following conditions must be satisfied:

$$-(k_{Bx}^2 + k_{By}^2) = \pm [12\rho\omega^2(1-\mu^2)/E\, h^2]^{1/2} = \pm k_B^2, \tag{14}$$

$$-(k_{Lx}^2 + k_{Ly}^2) = [\rho\omega^2(1-\mu^2)/E] = k_L^2, \tag{15}$$

$$-(k_{Tx}^2 + k_{Ty}^2) = [2\rho\omega^2(1+\mu)/E] = k_T^2, \tag{16}$$

where k_B , k_L and k_T are the bending, longitudinal and transverse shear wave numbers respectively. For an incident bending wave travelling at an angle α to the normal, $k_{Bx} = j\, k_B \cos\alpha$ and $k_{By} = j\, k_B \sin\alpha$ (this is also true for an incident longitudinal or transverse shear wave except that in such cases the appropriate wave numbers must be used). Compatibility of trace velocity at the junction requires that the response of all plates along the y - direction to be the same. This implies that the y - component of wave numbers (ie, k_{By}, k_{Ly} and k_{Ty}) for the reflected and transmitted waves of all plates must be the same as that of the incident wave. The x - component of wave numbers may be determined from equations (14) - (16). For bending waves, equation (14) yields four roots, the negative imaginary root and the negative real root must be selected since they represent propagating and decaying waves respectively in the positive x - direction (ie, away from the junction). Similarly, the solutions for longitudinal and transverse shear waves must be negative imaginary. Equations (7), (8) and (11) - (13) give the elastic deformations of the plate u, v and w. These deformations are expressed in terms of four unknowns representing the complex wave amplitudes, namely, the amplitudes for longitudinal and transverse shear waves, as well as the propagating and decaying bending waves. For plate 1, the deformations must also include the component of incident wave which may be considered as having a unit amplitude. Hence, in a junction that consists of n plates, there are 4n unknowns to describe the wave motion. These unknowns may be solved by the appropriate boundary conditions.

4. BOUNDARY CONDITIONS

To consider the boundary conditions at a junction, it is convenient to introduce the suffix i to denote the plate number (i = 1, 2, 3.....n). The compatibility of plate motions requires that the component of displacements of all plates along a set of reference coordinates (eg, x_1, y and z_1) must be the same. In addition, the rotation about the y axis of all plates at the origin should be equal. The plate rotation is given by:

$$\theta_i = \partial w_i / \partial x_i. \tag{17}$$

The compatibility requirements between plate i and plate 1 lead to the following equation:

$$\begin{bmatrix} u_i \\ v_i \\ w_i \\ \theta_i \end{bmatrix}_{i=2}^{n} = \begin{bmatrix} \cos\beta_i , & 0 , & -\sin\beta_i , & 0 \\ 0 , & 1 , & 0 , & 0 \\ \sin\beta_i , & 0 , & \cos\beta_i , & 0 \\ 0 , & 0 , & 0 , & 1 \end{bmatrix} \begin{bmatrix} u_1 \\ v_1 \\ w_1 \\ \theta_1 \end{bmatrix} , \tag{18}$$

where β_i = angle between plate i and plate 1.

For thin isotropic plates, the forces and moment per unit length of plate (along the y - direction) may be expressed as:

$$F_{xi} = -[E_i h_i/(1-\mu_i^2)][\partial u_i/\partial x_i + \mu_i \partial v_i/\partial y], \tag{19}$$

$$F_{yi} = -[E_i h_i/2(1+\mu_i)][\partial u_i/\partial y + \partial v_i/\partial x_i], \tag{20}$$

$$F_{zi} = [E_i h_i^3/12(1-\mu_i^2)][\partial^3 w_i/\partial x_i^3 + (2-\mu_i)\partial^3 w_i/\partial x_i \partial y^2], \tag{21}$$

$$M_i = -[E_i h_i^3/12(1-\mu_i^2)][\partial^2 w_i/\partial x_i^2 + \mu_i \partial^2 w_i/\partial y^2]. \tag{22}$$

At the junction, the sum of the forces and moments due to all plates may be resolved along the beam co-ordinates x_b, y, z_b and expressed as the beam forces and moment (see Figure 3):

$$F_{xb} = F_{x1}\cos\beta_b - F_{z1}\sin\beta_b + \sum_{i=2}^{n} F_{xi}\cos(\beta_i-\beta_b) + \sum_{i=2}^{n} F_{zi}\sin(\beta_i-\beta_b), \tag{23}$$

$$F_{yb} = \sum_{i=1}^{n} F_{yi}, \tag{24}$$

$$F_{zb} = F_{x1}\sin\beta_b + F_{z1}\cos\beta_b - \sum_{i=2}^{n} F_{x1}\sin(\beta_i-\beta_b) + \sum_{i=2}^{n} F_{zi}\cos(\beta_i-\beta_b), \tag{25}$$

$$M_b = \sum_{i=1}^{n} M_i. \tag{26}$$

The motions about the beam centroid u_b, v_b, w_b and θ_b may be expressed in terms of the elastic deformations of plate 1:

$$u_b = u_1\cos\beta_b - w_1\sin\beta_b, \tag{27}$$

$$v_b = v_1, \tag{28}$$

$$w_b = u_1 \sin \beta_b + w_1 \cos \beta_b + \theta_1 S, \tag{29}$$

$$\theta_b = \theta_1, \tag{30}$$

where β_b = angle between the beam and plate 1,
S = distance between beam centroid and the junction.

The equilibrium of forces and moments at the junction must allow for the torsional, bending and inertia effects of the beam. Consider the balance of forces in the x_b direction, the beam force F_{xb} is augmented by the shear force as a result of beam bending in the x_b-y plane (a similar argument exists for forces in the z_b direction). Assuming that the beam centroid coincides with the shear centre and the beam axis x_b is a principal axis, the force balance equations are given by:

$$-F_{xb} - E_b I_{zb}\, \partial^4 u_b/\partial y^4 = m_b\, \partial^2 u_b/\partial t^2, \tag{31}$$

$$-F_{yb} = m_b\, \partial^2 v_b/\partial t^2, \tag{32}$$

$$-F_{zb} - E_b I_{xb}\, \partial^4 w_b/\partial y^4 = m_b\, \partial^2 w_b/\partial t^2, \tag{33}$$

where m_b = mass per unit length of beam,
E_b = Young's modulus of beam,
I_{xb}, I_{zb} = second moments of area about beam centroidal axes.

The variation in plate rotation θ_i along the y - axis causes the beam to twist and results in a torsional moment. Equilibrium of moments about the centroid leads to:

$$-M_b + F_{zb}\, S + T_b\, \partial^2\theta_b/\partial y^2 = J\, \partial^2\theta_b/\partial t^2, \tag{34}$$

where T_b = torsional stiffness of the beam,
J = polar moment of inertia per unit length about beam centroid.

Equations (18) and (31) - (34) provide the necessary boundary conditions for the solution of wave amplitudes. This set of 4n linear equations may be solved by a standard computer routine which handles complex coefficients. The above analysis can be easily modified to allow for other types of boundary conditions, eg, a plate-plate junction.

5. JUNCTIONS WITH THIN BEAMS

Some engineering structures (eg, ships, aircraft) quite often involve the use of thin beams to reinforce plate elements. A thin beam in the context of this paper implies that the beam thickness is of the same order as that of the plate element and is therefore subjected to bending and in-plane waves. A schematic diagram of the structure is shown in Figure 4. The analysis of this type of structures may be conducted by assuming that the thin beam behaves as a finite plate with waves travelling in both the positive and negative x - directions. Referring back to equations (14) - (16), the solutions to wave motions of the finite plate in this case must include the positive and negative roots. This results in eight unknown complex wave amplitudes (instead of four unknowns as in the case of an infinite plate). The additional four unknowns in a plate - thin beam junction represent waves in the finite plate that travel in the negative x - direction. Four additional boundary conditions are thus required to solve the wave motions at the junction. These boundary conditions may be obtained by considering the force and moment balance at the end of the finite plate. For example, in the structural junction shown in Figure 4, the forces and moment must vanish at the free end of the plate.

6. TRANSMISSION AND REFLECTION EFFICIENCIES

The above analysis solves the wave amplitudes of plate - beam junctions and leads to the calculation of wave power. The wave power per unit length of a junction may be expressed as the energy per unit area multiplied by the component of group velocity normal to the junction. Recall that the incident wave has a unit amplitude, the wave power due to an incident wave at an angle α is given by:

$$P_s = m_1 \omega^2 c_{g1} \cos \alpha, \qquad (35)$$

where m_1 = mass per unit area of plate 1.

Similarly, the transmitted and reflected wave power may be expressed as:

$$P_{Di} = m_i \, |\xi_{Di}|^2 \omega^2 c_{gi} \, (j \, k_{Dxi}/k_{Di}), \qquad (36)$$

where suffix D denotes the wave type (ie, bending, longitudinal or transverse shear) and ξ is the wave amplitude. For bending waves, the power is determined by the travelling waves only. The near field decaying waves carry no power. The complex number j appears in the equation since k_{Dxi} is negative imaginary, hence the value of power is a positive real number. Thus, the transmission efficiency is given by:

$$\tau_{Di} (\alpha) = P_{Di}/P_s, \qquad (37)$$

where i = 2....n.

For a diffuse incident vibration field, the mean transmission efficiency may be obtained by averaging the efficiencies over the entire range of incident angles as shown in equation (2) and Figure 1. The reflection efficiency of plate 1 may be obtained in a similar manner. Conservation of energy requires that the sum of all transmission and reflection efficiencies to be equal to one. In the next section, the above theory is applied to two plate - beam systems to calculate some of the parameters for SEA study.

7. SEA CASE STUDY

The SEA method involves modelling groups of similar resonant modes as individual elements in a given structure. Energy balance equations are then developed for all the elements based on the energy flow between different groups of resonant modes. The energy dissipated by an element is characterised by the material loss factor and those transferred to connecting elements by the coupling loss factor. Solution of the energy balance equations lead to the vibration energy of individual elements which can then be expressed in terms of the average response level.

7.1 Periodically stiffened steel panel

Figure 5 shows the test panel which consists of periodic T-sectioned stiffeners welded onto a flat plate. The SEA model was constructed by first calculating the coupling loss factors between plate elements by treating the stiffeners as beams under bending and torsional motions. A second set of calculations was performed by treating the web of the T - section as a finite plate and the flange as a beam attached to the web. Results from the calculation of transmission efficiencies suggest that the conversion of bending to in-plane vibration modes through the junction is negligible at the frequency range of interest (ie, 1000 - 5000 Hz). Hence only bending modes are considered in this study. The SEA model therefore consists of five elements each representing a group of resonant plate bending modes. For the smallest element, the number of resonant modes at the lowest frequency band (1000 Hz band centre frequency in the third octave band) is about four. Although the modal density is slightly on the low side for the lower frequency bands, it is believed that it will not significantly affect the accuracy of the present study. The material loss factor is evaluated by the steady state power injection method [3] and the value is found to be approximately 0.0015.

Tests were carried out with the panel suspended by cables and excited by an electromagnetic shaker through an impedance head. The force and acceleration measurements from the impedance head were processed to give the input power for the SEA model. Six accelerometers were randomly placed on each element in turn to measure the response. The accelerometer signals were processed to evaluate the mean square velocity of each element. Figure 6 shows the experimental set up.

Figures 7 shows the attenuation in mean square velocity between elements 3 and 5. It can be seen that the conventional beam theory significantly overestimates the attenuation, especially at high frequencies. Although the thin beam theory underestimates the attenuation, the discrepancy is only 2 - 3 dB and may be considered acceptable since it is within the normal range of experimental errors. The difference between the two models can be explained by the fact that the conventional beam theory models the system as a low pass filter (see Cremer *et al.* [8] p 442) and ignores the effects of beam bending and in-plane vibrations. Such effects are significant in this example since the thickness of web is of the same order as that of the plate.

7.2 Model of a ship structure

Figure 8 shows the basic model used in this example. The model has been studied by Nilsson [9] using the wave guide theory. In his analysis, Nilsson assumes that the ship frames act like rigid boundaries and the vibration power flow is considered to be mainly propagating in the plate elements through bending motions. Measurements on the vibration levels of the decks have also been carried out by Nilsson to verify his theory. A variety of noise suppression methods have been investigated by Nilsson including:

Model 1 Bare steel model as shown in Figure 9,
2 Damping layers on vertical elements between decks A and C,
3 Damping layers on decks B and C,
4 Superstructure connected to the main structure by frames only (ie, no plate connection between the main structure and superstructure).

The object of the present analysis was to apply the SEA method to the model ship structure and compare the results with those measured by Nilsson. In contrary to the wave guide theory, it was felt that the frames may contribute to a significant amount of vibration energy transmission. This is evident from the experimental results presented by Nilsson on model 4 which shows a significant level of vibrations in the superstructure (see Figures 11 and 12). On the basis of this consideration, it was decided to consider both the plate and frame elements in the SEA model. Only bending modes are modelled for plate elements while the frames were modelled as beams in bending and longitudinal modes of vibration. In calculating the coupling loss factors, it is assumed that the junctions are rigid. However, this condition is not completely fulfilled in Nilsson's model as some junctions are connected by brackets spot welded onto the plates. The loss factors of all plate elements were measured by Nilsson using the reverberation time method and such values are used in the present SEA model. In models 2 and 3, Nilsson reported some difficulties in measuring the effective loss factor of the damped plates. He observed that the damped plates tend to be driven by the adjoining undamped plates and thus affects the reverberation time. There are also some elements of uncertainty as to the effect of damping layer stiffness and other material properties on vibration transmission. As a result, models 2 and 4 were not considered in the present SEA study.

The attenuation of mean square velocity between deck A and decks B and D for models 1 and 4 is shown in Figures 9 - 12. In Figure 9, the SEA model underestimates the attenuation between decks A and B by about 2 dB. This small discrepancy may have been due to errors in the measurement of loss factors and the method of connection between elements as discussed. For model 4, the agreement between SEA and measured results is good. It should be noted that the wave guide model cannot be applied to model 4 since there is no plate connection between the main deck and the superstructure.

8. CONCLUSIONS

A method for evaluating the coupling loss factor of plate - beam junctions for SEA study has been developed. The method can be applied to junctions that consists of thin beams typical of naval ship constructions.

Results of SEA calculations and experimental measurements on the vibration levels of two plate - beam systems are presented. The results demonstrate that the SEA method can be used as a tool for the prediction of average system response.

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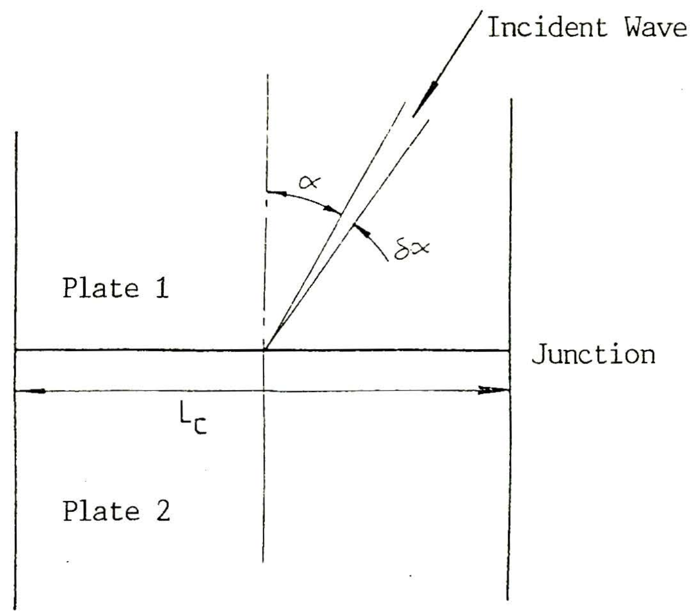

Fig.1 Interception of an Oblique Incident Wave

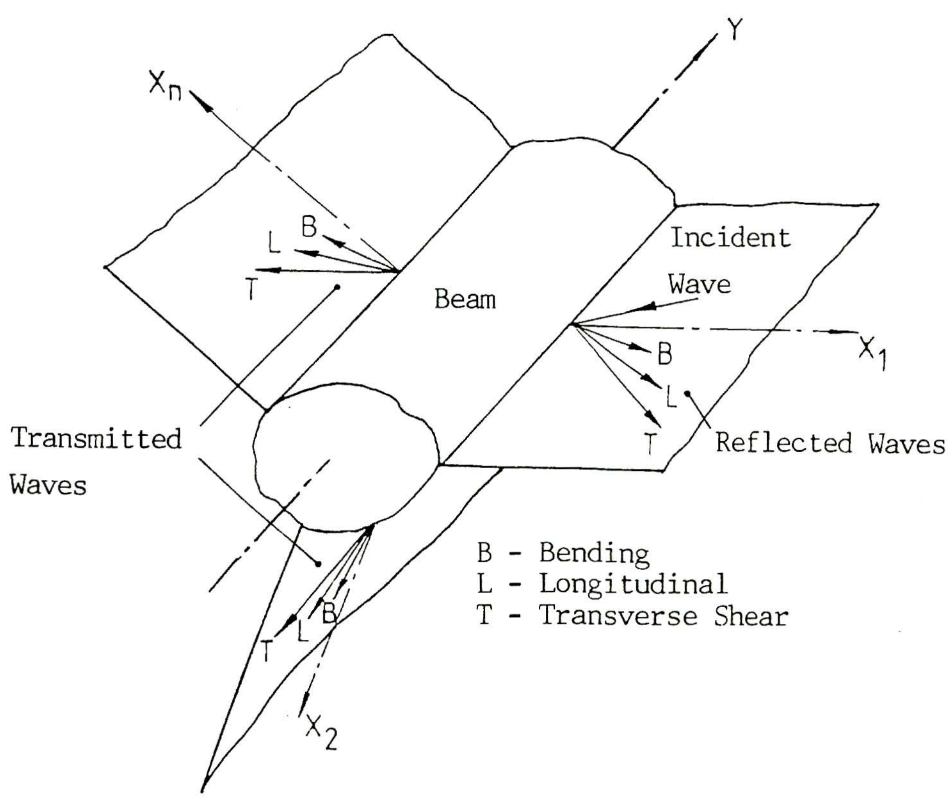

Fig.2 Schematic Diagram of a Plate-Beam Junction

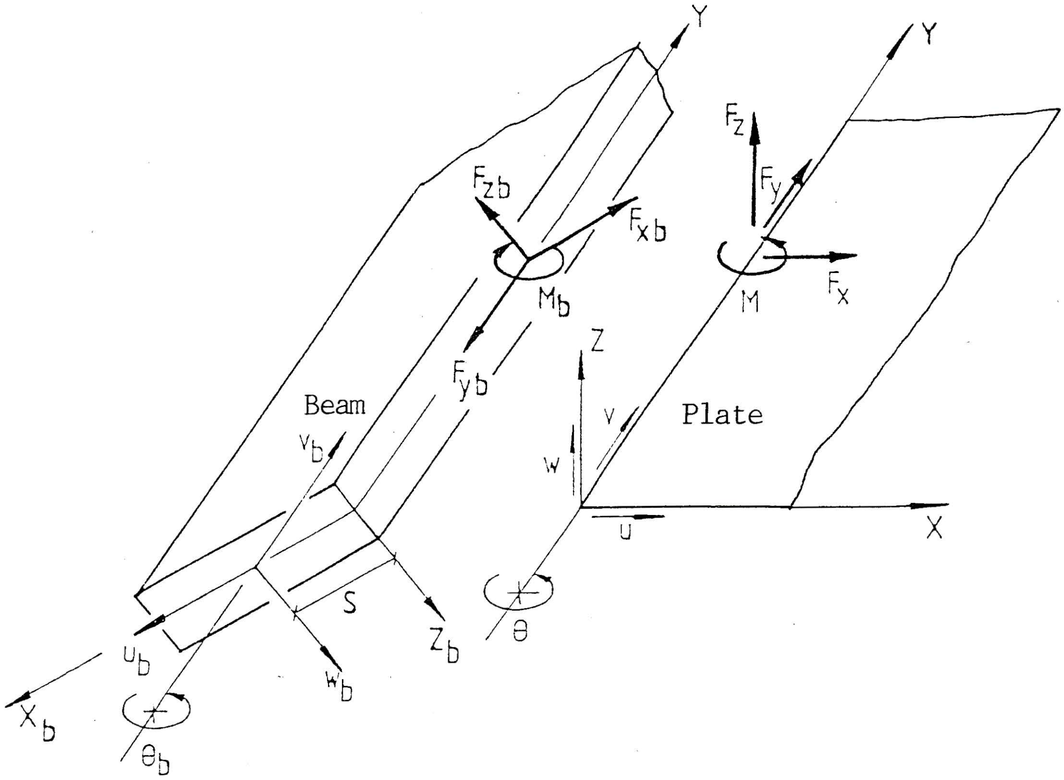

Fig.3 System of Co-ordinates

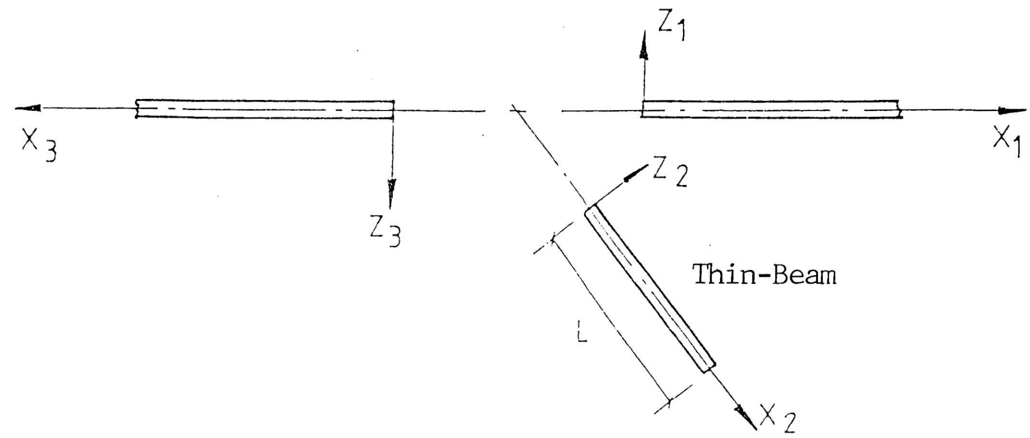

Fig.4 Plate-Thin Beam Junction

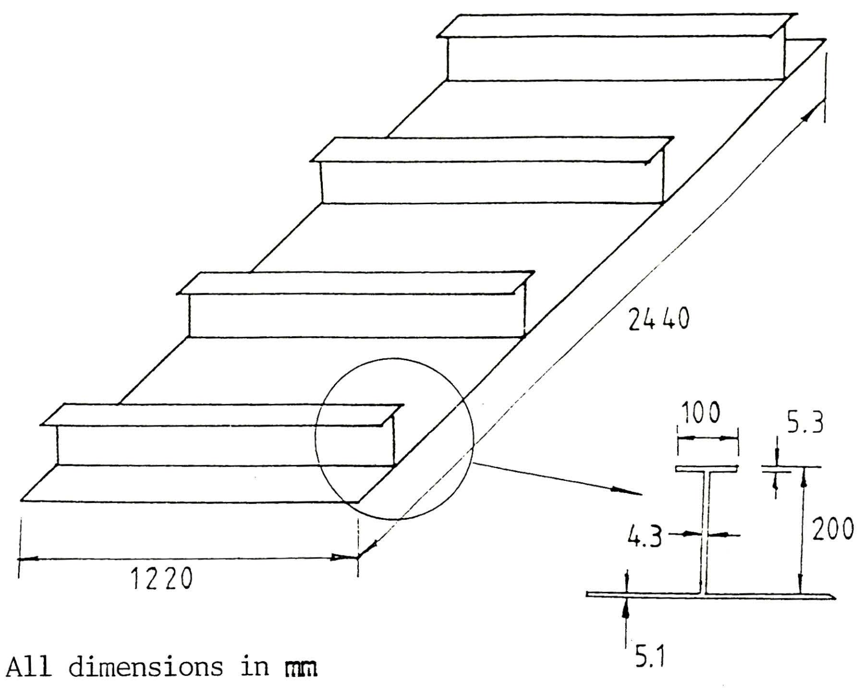

Fig.5 Periodically Stiffened Panel

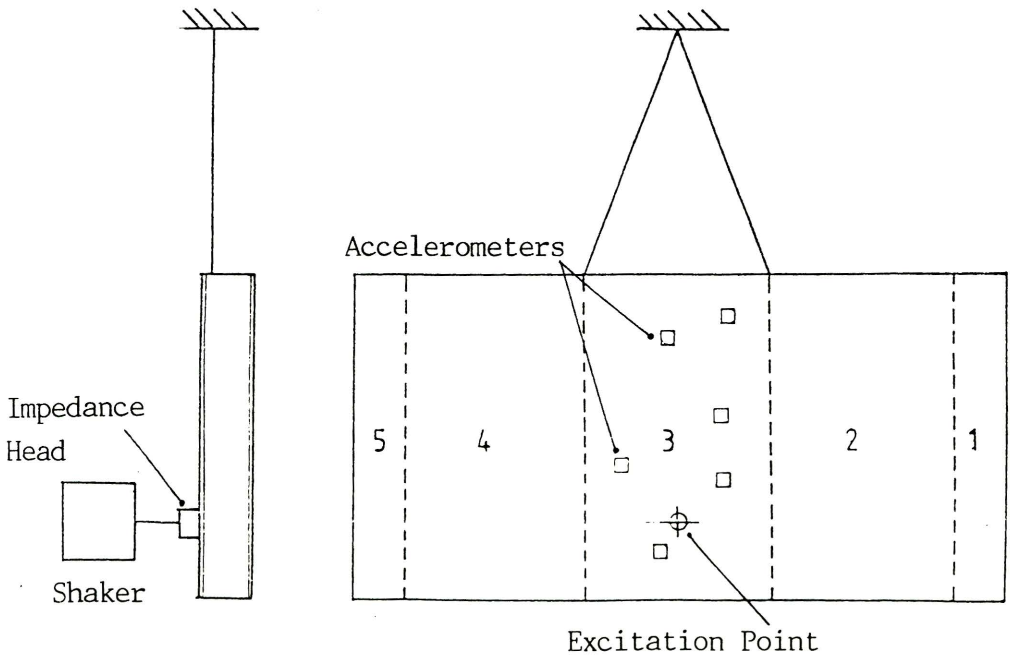

Fig.6 Experimental Set-up

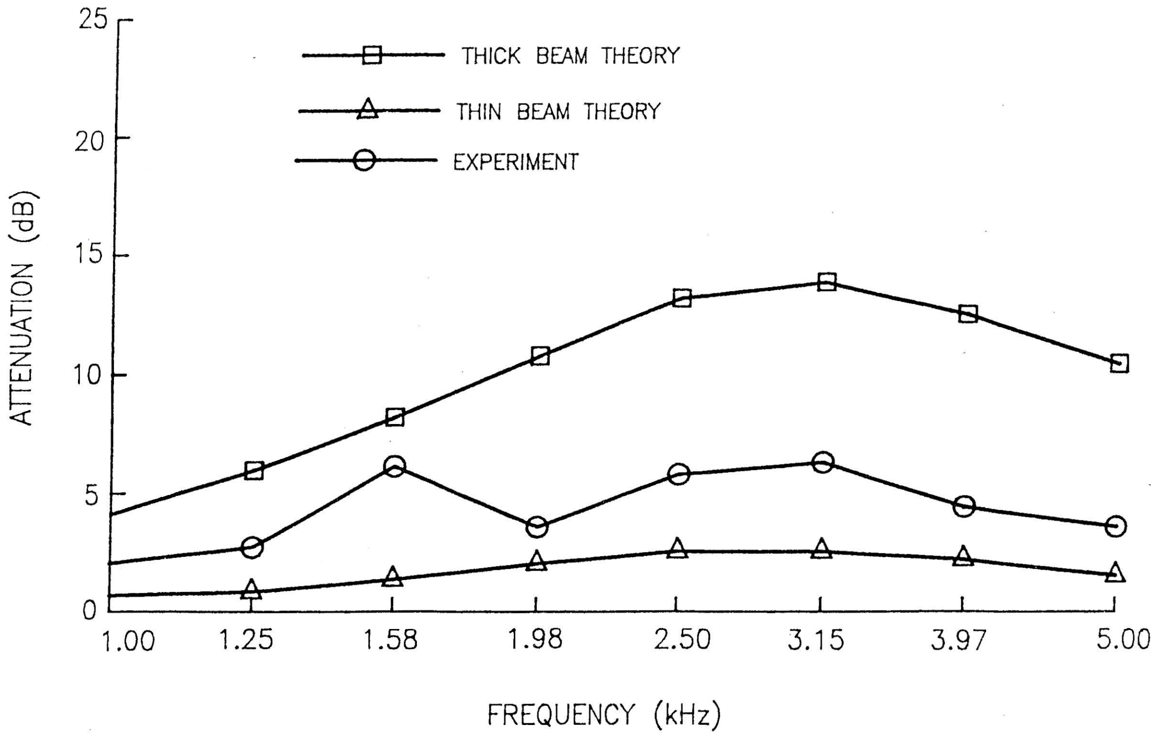

Fig.7 Attenuation Between Elements 3 and 5

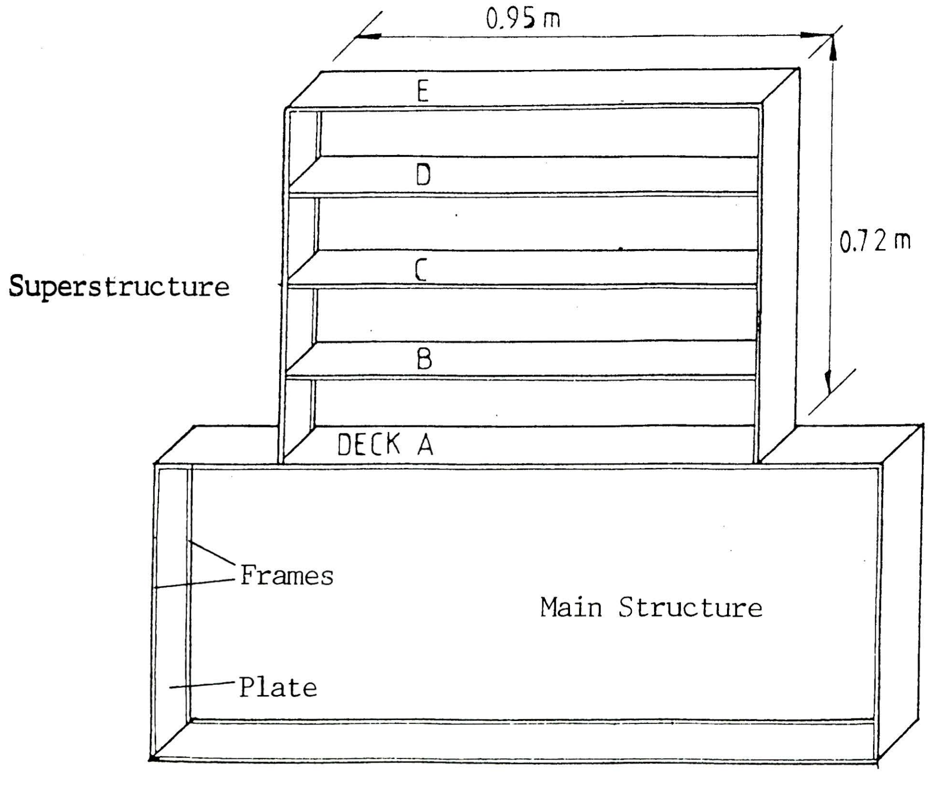

Fig.8 Model of a Ship Structure

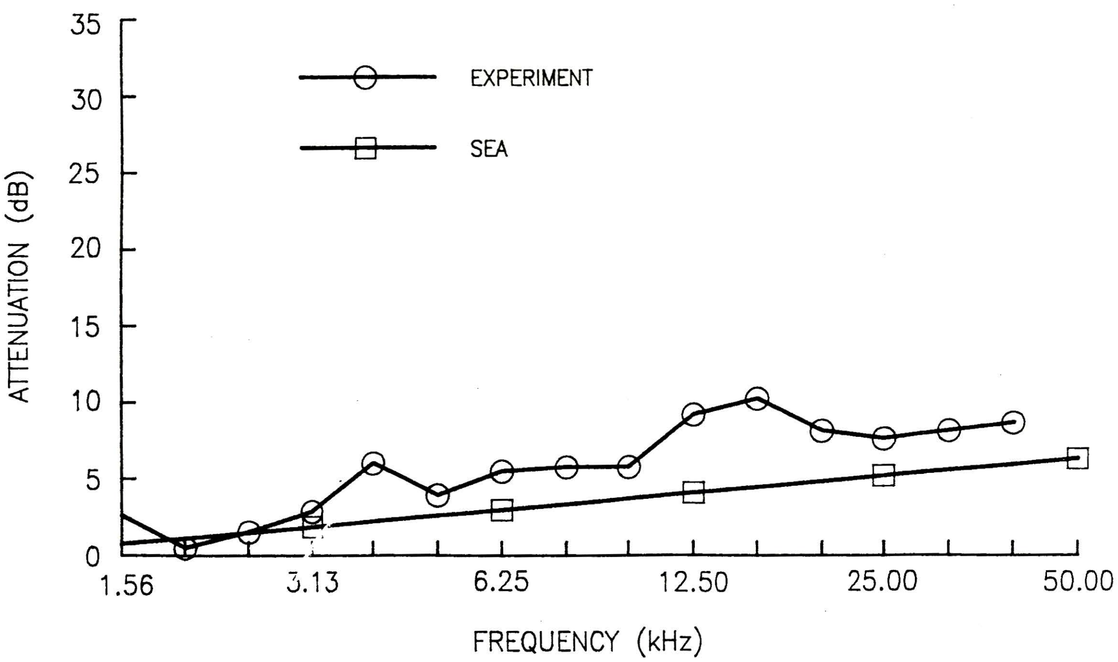

Fig.9 Attenuation Between Decks A and B of Model 1

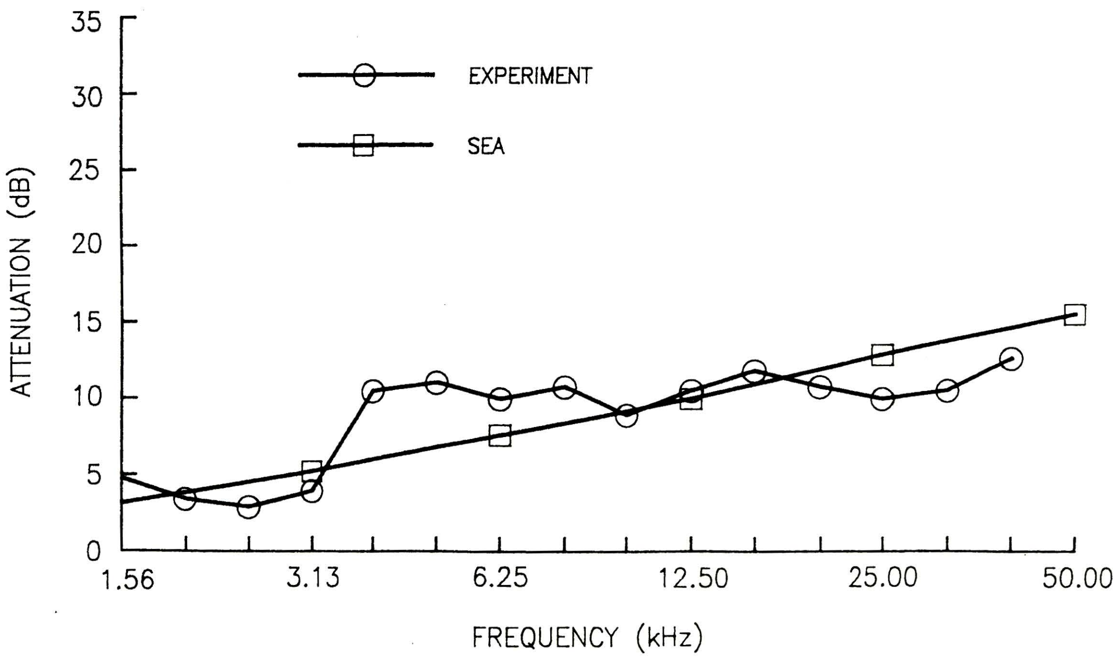

Fig.10 Attenuation Between Decks A and D of Model 1

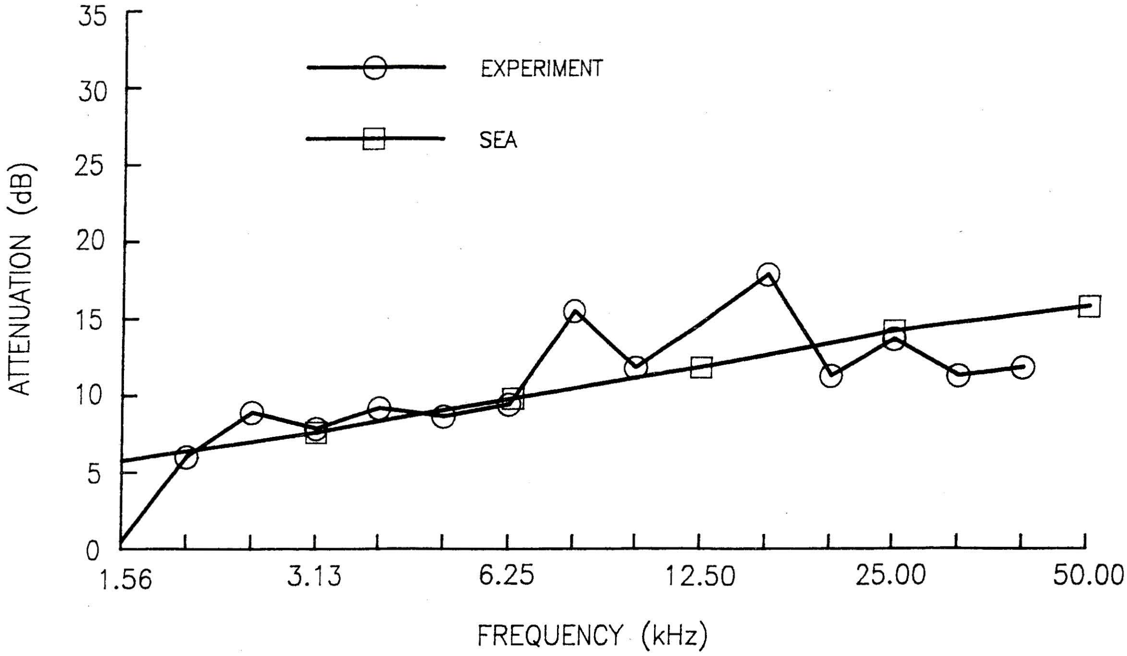

Fig.11 Attenuation Between Decks A and B of Model 4

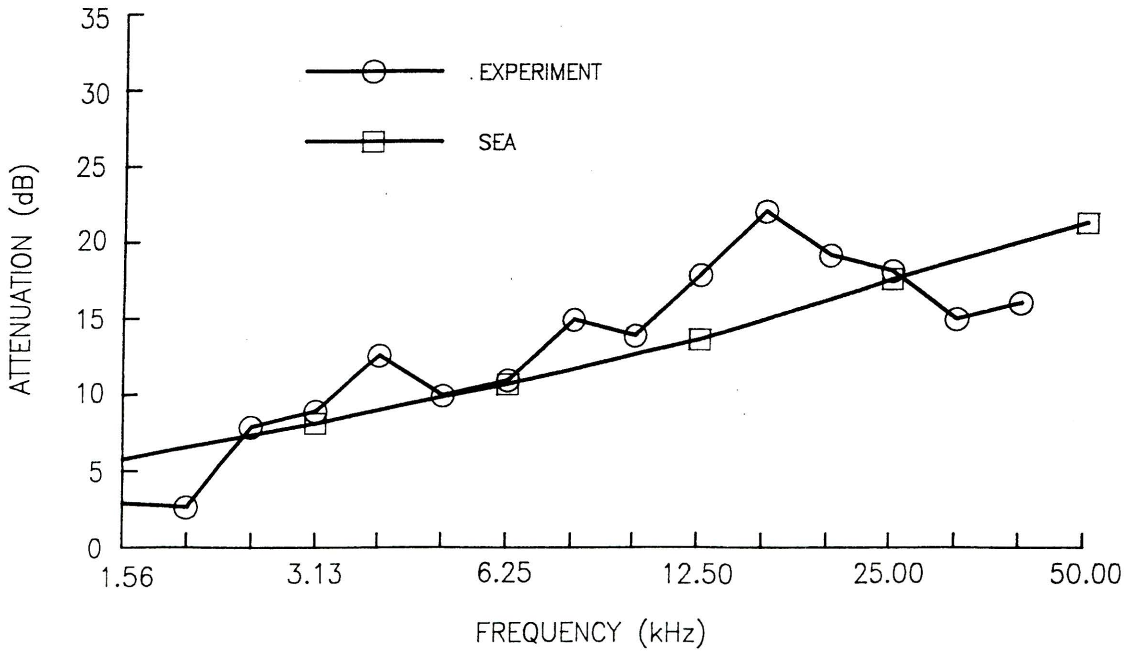

Fig.12 Attenuation Between Decks A and D of Model 4

MULTI-FREQUENCY OPERATION OF THE ACOUSTIC IMPEDANCE TUBE

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ABSTRACT

The acoustic impedance tube (or standing wave apparatus) has been used for many years as a means for measuring the absorption properties of acoustic materials. A disadvantage of the technique is that it is slow and laborious unless automated or expensive equipment is available.

The time required for a series of measurements could be reduced significantly if it were possible to operate at several frequencies simultaneously. Such an approach should be feasible given the capabilities of modern frequency analysis instrumentation.

A study is described in which the feasibility of this approach has been verified experimentally. A signal source capable of providing simultaneously up to seven single tone inputs in separate octave bands has been designed and built. The source has been used in an impedance tube and the absorption properties of various acoustic materials measured by both single frequency and multi-frequency excitation (with appropriate filtering).

It has been proved possible to test a sample over the full frequency range (in octave steps) with one traverse of the apparatus.

1. INTRODUCTION

The Acoustic Impedance Tube, also known as the Standing Wave Apparatus, provides a well-established technique for determining the absorption properties of acoustic materials [1,2]. The possibility of using this technique is recorded as having been noted in 1902 [3]. A commercial version has been available since 1955 [3] and a modernised version was marketed in 1991 [4].

The original idea and the first commercial version utilise testing at single frequencies in turn, with levels being measured by a travelling microphone. The technique can be made faster and less tedious by using mechanical microphone drive, computer control and digital data acquisition [5]. In the 1991 version, the two-microphone technique is used, giving extremely rapid testing over all frequencies at once. However, the apparatus is expensive and requires high precision equipment.

An alternative possibility, suggested by the availability of modern frequency analysis instrumentation, is the further development of the standing wave approach using excitation by several frequencies simultaneously. Such a technique has been implemented and has operated successfully.

2. BASIC APPARATUS

The starting point for this development was a conventional impedance tube with probe type microphone, as described previously [5]. The carriage carrying the microphone is driven by a stepper motor and pinion and rack. The motor control is program-generated by a personal computer. The microphone output is detected, filtered and converted to D.C. level by conventional acoustic equipment, then sampled by A/D conversion and stored, for each motor step, in the computer. The stored schedule of sound level v. distance is then analysed for absorption coefficient and acoustic impedance (real, imaginary, magnitude and phase).

The full traverse takes about two minutes and computed data is available immediately the traverse has concluded. This is considerably faster than the previous manual operation, which involved searches for maxima and minima, manual recording of level and distance, and subsequent calculations.

3. MULTI-FREQUENCY EXCITATION

The basic procedure, even with automatic operation, still required repetition at each test frequency. The question arises as to whether the apparatus will give satisfactory results if the excitation is composed of several frequencies together. This requires that the output filtering has sufficient discrimination to extract the standing wave data for each of the excitation frequencies without interactions producing invalid results.

A preliminary investigation was carried out using an existing multi-frequency supply - a unit used for a carried-based multiplexing system. The frequencies available for testing were 320, 560, 985, 1690, 2875 and 4860 Hz. Results over four materials at these six frequencies showed that good results can be obtained for each test frequency even though all six frequencies were applied together. Consequently a new signal generator was designed and built to supply signals at the more usual acoustic test frequencies.

The new generator has seven channels. each channel supplying a signal which can be adjusted over an octave range. The centre frequencies of the channels are 125, 250, 500, 1000, 2000, 4000 and 8000 Hz. The signal level from each channel is adjustable, and each channel can be switched on or off, i.e. the output can be from any one channel, or from any combination of any number of channels up to all seven channels on simultaneously.

TEST RESULTS

Results are presented for four test samples:

(1) Highly reflecting
(2) Polyurethane foam
(3) Fibreglass (low density)
(4) Perforated plasterboard with airgap.

Each sample was tested with single frequency excitation and with simultaneous excitation at five frequencies, 250, 500, 1000, 2000 and 4000 Hz. The frequencies 125 and 8000 Hz are outside the range of the impedance tube used.

Two data acquisition techniques have been used. A multi-frequency (FFT) analyser was used to obtain maximum and minimum sound pressure levels, at all test frequencies. However, the analyser could not identify position along the tube. The results presented here are taken from the second technique, which was simply to analyse one frequency at a time but with all frequencies excited. Thus full advantage of the multi-frequency principle has not been taken, but there was some merit in verifying the practicality of the idea before setting up a multi-frequency analysis system.

Figure 1 shows results for the highly reflective end. This figure and all the subsequent figures giving test results are presented in three parts:

(a) normal incidence sound absorption coefficient α_n
(b) specific normal acoustic resistance, R_S
(c) specific normal acoustic reactance, X_S

R_S and X_S are normalised with respect to characteristic impedance, ρc.

In each case the single frequency and multi-frequency results are presented together.

Test results for the three absorptive samples are presented in Figures 2, 3 and 4.

It can be seen from the results that there is good agreement and hence the principle of the technique is verified.

5. FURTHER DISCUSSION

The signal filtering has been performed by a Bruel and Kjaer Frequency Analyser Type 2107, set at frequency selectivity Max, i.e. the signal is attenuated 45 dB at one octave either side of the set frequency and 53 dB two octaves either side. The influence of a standing wave at one frequency on the results for an "adjacent" frequency depend on the relative positions of maxima and minima. Taking a worst case (i.e. a maximum at one frequency occurring at the position of the minimum for a frequency one octave different), equal maxima, and true level difference 40 dB, the indicated level difference would be approximately 39 dB and the calculated absorption coefficient 0.043 instead of the true 0.039. Such an influence is apparent in the results for the reflecting end, Figure 1. Thus results for highly reflective samples must be viewed with care. With this equipment, absorption coefficients less than 0.07 may not be valid.

More modern equipment has better resolution than was the case in this work, and hence the problems of the previous paragraph would not arise. It is the intention to aim for an analysis system which will permit simultaneous operation in one-third octave steps of frequency.

6. CONCLUSION

Multi-frequency excitation of the acoustic impedance tube gives a potential benefit in the sense of reducing the time necessary to conduct absorption measurements over a wide frequency range. The feasibility of multi-frequency excitation has been proved experimentally.

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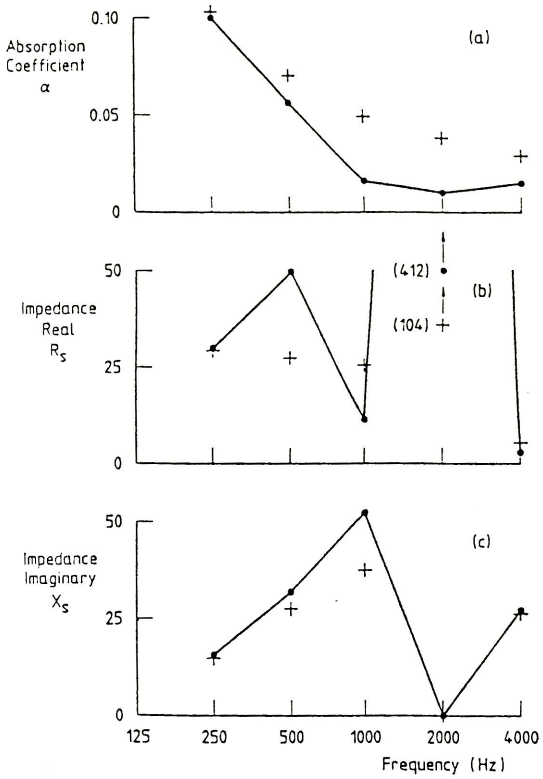

Figure 1. Reflecting end

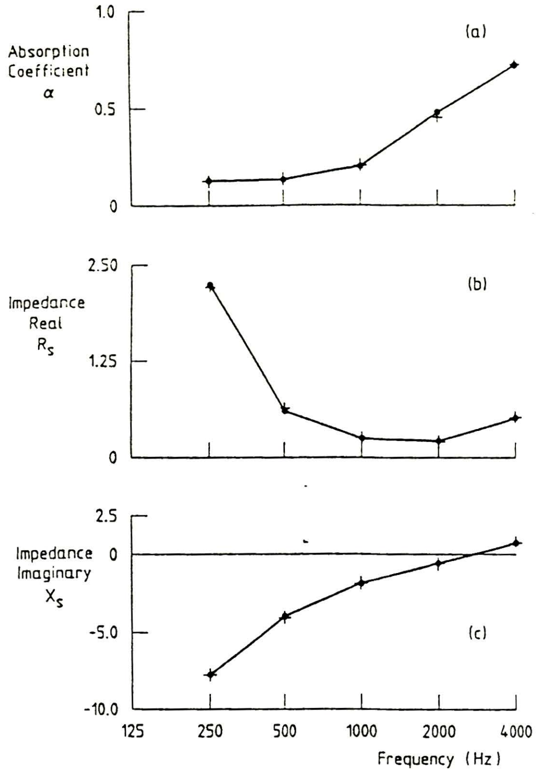

Figure 2. Polyurethane foam

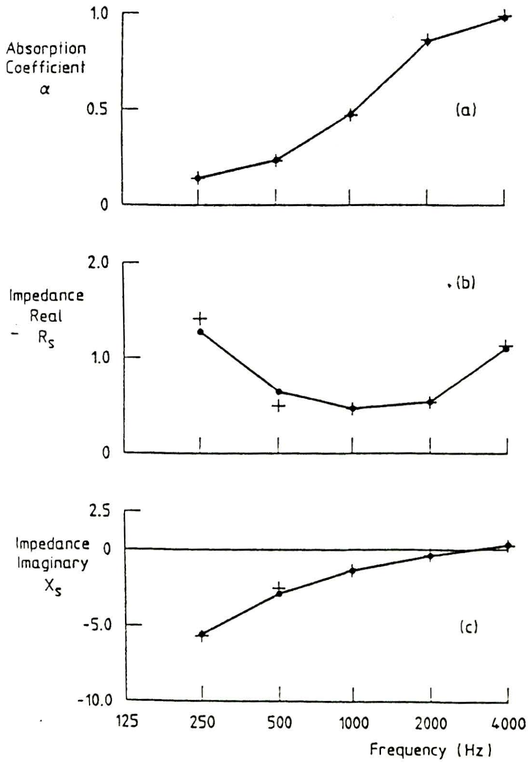

Figure 3. Fibreglass

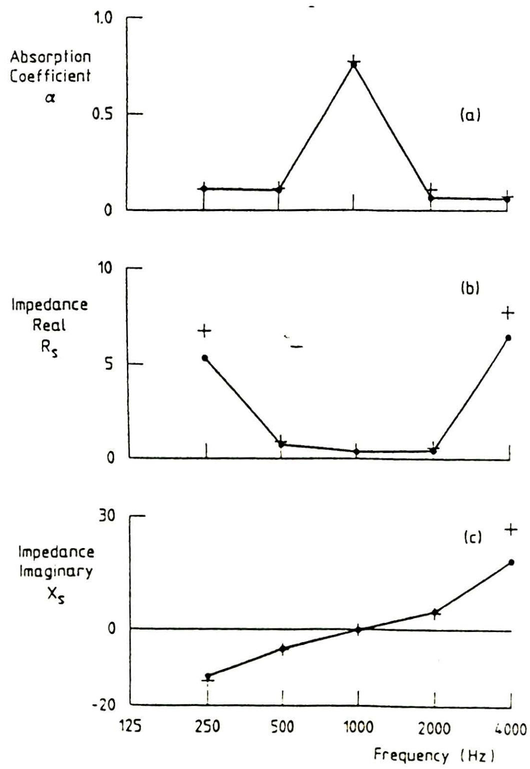

Figure 4. Perforated plasterboard

Figures 1-4. Absorption Properties

•—— Single frequency excitation

\+ Multi-frequency excitation

THE SOUND POWER LEVEL OF A SWITCHED RELUCTANCE MOTOR

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ABSTRACT

A switched reluctance (SWR) drive is basically an electronically controlled stepper motor with position feedback, specifically designed for variable speed drive operation. This type of drive offers the user a relatively simple motor configuration which leads to a significant reduction in manufacturing costs while the performance is comparable with conventional drives. The converter and control electronics associated with this type of drive is comparable with inverter technology in terms of drive complexity. In addition, the SWR drive is capable of high speed operation and the operating temperature of the motor is not constrained by the presence of permanent magnet (PM) as in the case with PM drives. Despite these advantages, switched reluctance drives generally tend to be noisier than conventional drives and thus are often precluded from widespread use. There is, therefore, a need to understand the noise generation mechanism of SWR so that effective engineering noise control strategy may be implemented. In this paper, the acoustic performance of a switched reluctance motor will be reported.

The sound power level of a switched reluctance motor under various operating load conditions has been measured by using the sound intensity technique. By using appropriate parameters, the sound power level results can be reduced to a single curve, thus implying a full description of the acoustic performance of this type of machines for all possible operating conditions can be obtained by simply testing the machine for a limited number of operating conditions.

1.0 INTRODUCTION

With the improvement in power electronics technology in the past decade, variable speed drives, such as inverter-driven induction motors or switched reluctance motors, are finding increasing usage in industrial and domestic applications because these drives are generally more efficient and offer potential savings in energy consumption. Variable speed drives are now being introduced to environment which may be subject to stringent acoustic regulations such as in office buildings where variable air flow airconditioning systems are used. It has been found that some of the variable speed drives produce unacceptably high acoustic noise levels and the noise spectra vary with the speed of the drives. In order that variable speed drives can be used widely in quiet environment, it is important to study the acoustic noise generation mechanisms in these drives so that appropriate noise control measures can be introduced at the design stage or retro-fit solutions can be implemented. Although there have been a few recent studies on acoustic noise generation in electrical machines (see, for example [1] & [2]), there is generally a lack of understanding in this area and the electrical design of these machines is usually more advanced than giving due considerations to acoustical performance.

The acoustic performance of a source (such as an electrical machine) is best specified by its sound power level which is basically a measure of the acoustic noise producing capacity of the source. Unlike the sound pressure level which is dependent on the acoustic environment and the distance from the source, the sound power level of a source is independent of these variables.

This paper is part of a major study aimed at reducing acoustic noise from electrical machines, in particular, a switched reluctance motor. In this paper, the principle of operation of a switched reluctance motor will be briefly described. The acoustic performance of a switched reluctance motor under various test load conditions will be quantified by its overall A-weighted sound power level and a semi-empirical technique in reducing the overall A-weighted sound power level data into a compact form using appropriate scaling parameters will be introduced.

2.0 OPERATION OF A SWITCHED RELUCTANCE MOTOR

A good description of the operation of a switched reluctance motor and its advantages and disadvantages has been given by Miller [3]. Figure 1 shows the schematic of a three phase, six stator pole/four rotor pole configuration (6/4). By exciting the windings of a pair of stator poles with a constant current, a torque will be produced due to the magnetic attraction so that the rotor will move from an 'unaligned' position until the rotor poles and the stator poles are aligned. Continuous motion will be produced by exciting the windings of each pair of stator poles in turn.

It is, therefore, not surprising that acoustic noise may be generated by the pulses exciting the resonant frequencies of the motor structure. Owing to the finite rise and fall times of the current pulse, a large negative torque will be produced if the excitation current is introduced at the unaligned position and turned off at the unaligned position. In order to reduce the negative torque produced and hence increase the average torque achieved, the excitation current is applied at an advanced angle θ_{ad} before the rotor poles reach the unaligned position and turned off at a delay angle θ_{de} before the rotor poles are aligned with the stator poles, as shown in Figure 2 which compares the torque produced with and without the use of delay and advance angles. Speed control of this type of motor is achieved by producing a square-wave pulse train with variable duty cycle (δ) to supply a voltage (proportional to δ) to the windings of the stator poles. It is, therefore, expected that both the electrical and acoustical performance will be dependent on θ_{ad}, θ_{de} and δ.

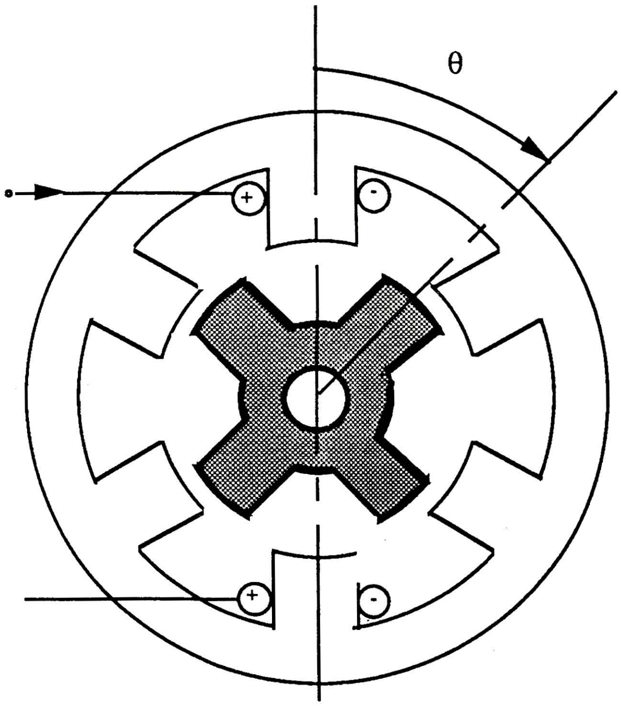

Figure 1 A switched reluctance motor with a 6/4 configuration.

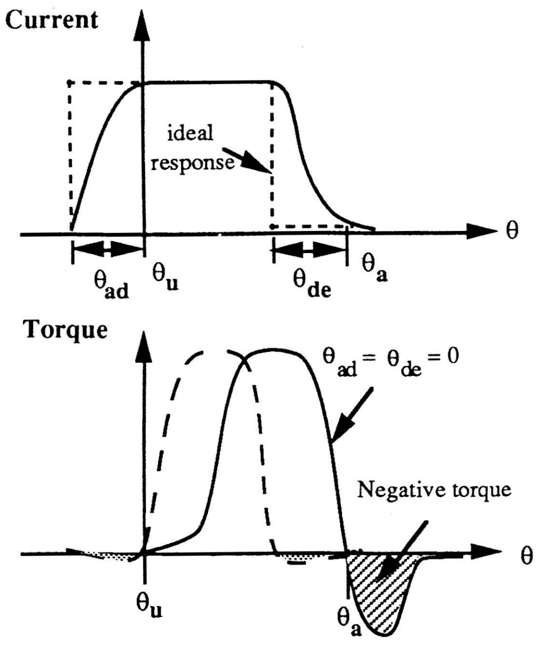

Figure 2 Comparisons of torque produced with and without the use of advance/delay angles.

3.0 EXPERIMENTAL SET-UP AND INSTRUMENTATION

Figure 3 shows the schematic set-up for measurements of the sound power level of a 600 W switched reluctance motor in an anechoic chamber with free-field dimensions of 3.5 m x 3.5 m x 3.5 m and a lower cut-off frequency of 150 Hz. The switched reluctance motor is connected to a dynamometer (which is a 2 kW DC generator) with a flexible coupling and both machines are supported by vibration mounts attached to a base plate.

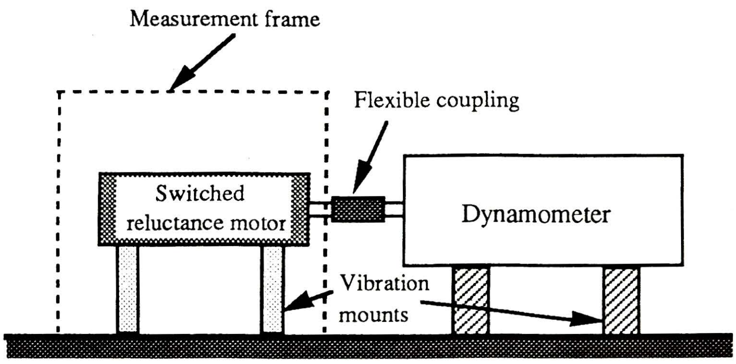

Figure 3 Schematic of experimental set-up.

In order to eliminate the contribution of the noise produced by the dynamometer to the measured sound power level of the switched reluctance motor, measurements have been made using the sound intensity technique. The theory of the sound intensity technique and its errors have been described in detail in the literature, see for example [4] and will not be repeated here. It is sufficed to say here that the main advantage of using the sound intensity technique in sound power measurements is its ability in suppressing the contribution of external background noise. As has been shown by a number of studies, for example [5], the external noise suppression capacity is determined by the dynamic capability L_D of the sound intensity measuring system and it is important to monitor the pressure-intensity index L_K during the measurements to ensure that L_K does not exceed L_D. Here L_K is defined as the difference between the measured sound pressure level L_p and the measured sound intensity level L_I.

In all the measurements described here, the sound intensity measuring system comprises a Bruel & Kjaer (hereinafter referred to as B&K) 2032 dual channel FFT analyzer and a sound intensity probe made up of a pair of B&K 4181 phase-matched 0.5 inch microphones mounted in a face-to-face configuration, separated at a distance of 12 mm. The valid frequency range covers the 1/3 octave frequency bands with centre frequencies from 125 to 5000 Hz. There has been considerable debate in the literature regarding the accuracy between measurements obtained at discrete points and those obtained by the scanning technique and the current International Standard for sound power measurements using the sound intensity technique is only for discrete point measurements [6]. However, our experience [5] indicates that as long as scanning is performed properly, the difference between the two techniques is insignificant. All measurements described here have been obtained by scanning the sound intensity probe over the five faces of a rectangular volume (with dimensions 0.27 m x 0.3 m x 0.3 m) which encloses the switched reluctance motor (Figure 3). The narrow band sound intensity data were processed and synthesized into 1/3 or 1/1 octave bands with a Hewlett-Packard series 300 microcomputer.

Although a range of rotational speeds, N (rpm) and duty cycle (δ from 0.3 to 0.7) was tested for the switched reluctance motor, only the results for a fixed advance angle ($\theta_{ad} = 21°$) and a fixed delay angle ($\theta_{de} = 6°$) are being reported here. The output power (shaft power),P, of the switched reluctance motor has been determined by measuring its rotational speed with a tachometer and its torque with a Staiser Monilo torque transducer.

4.0 RESULTS AND DISCUSSIONS

The shaft power (P) of the 600 W switched reluctance motor determined for various rotational speeds (N) and duty cycle (δ) is shown in Figure 4. It can be seen that for a given speed (N), the shaft power (P) increases with the duty cycle (δ) and for a given duty cycle (δ), there appears to exist a speed (N) at which maximum power (P) may be obtained.

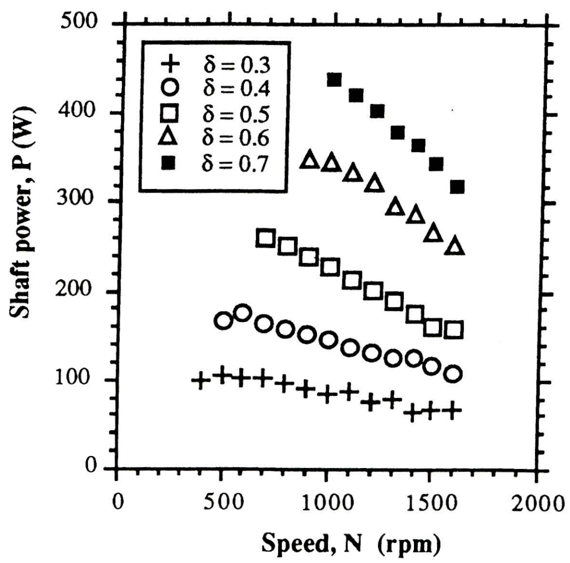

Figure 4 Variation of shaft power with speed for various duty cycles.

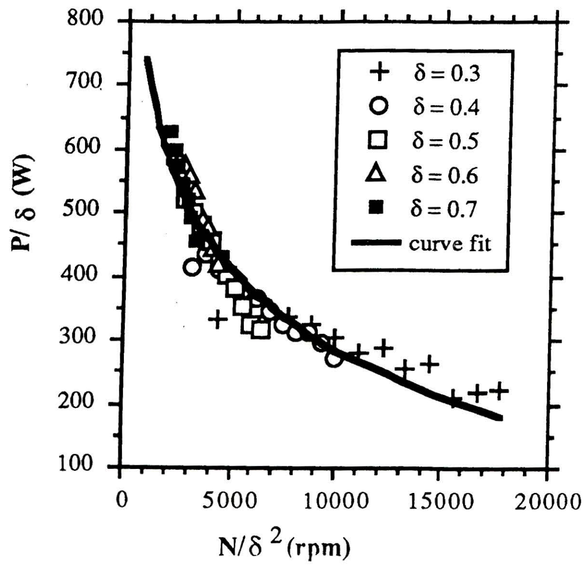

Figure 5 Variation of shaft power parameter with speed parameter.

Since the shaft power curves for various δ appear to be fairly similar and since for a given advance angle θ_{ad} and delay angle θ_{de}, P depends only on δ, it may be possible that with appropriate scaling of P and N using δ, the curves may be reduced to be geometrically similar. As displayed in Figure 5, with the use of a shaft power parameter, P/δ and a speed parameter N/δ^2, all the shaft power data for various δ collapse onto a single curve. The slight departure of some of the data from the curve is well within the limits of accuracies of the experimental measurements. The curve of best fit to the test data in Figure 5 is given by:

$$\frac{P}{\delta} = 1966 - 420 \log \frac{N}{\delta^2} \tag{1}$$

with a correlation coefficient of 0.95.

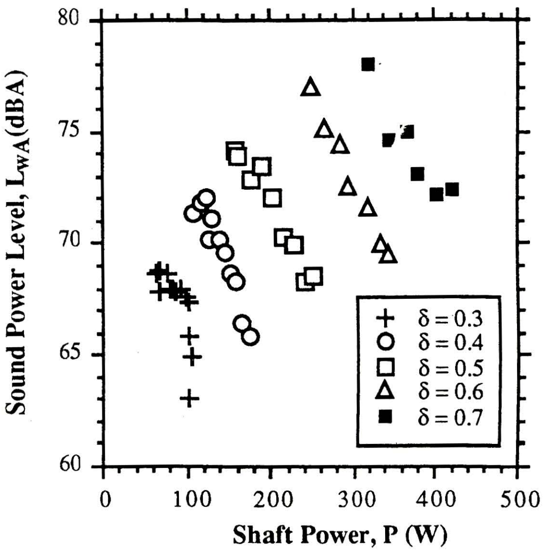

Figure 6 Variation of sound power level with shaft power for various duty cycles.

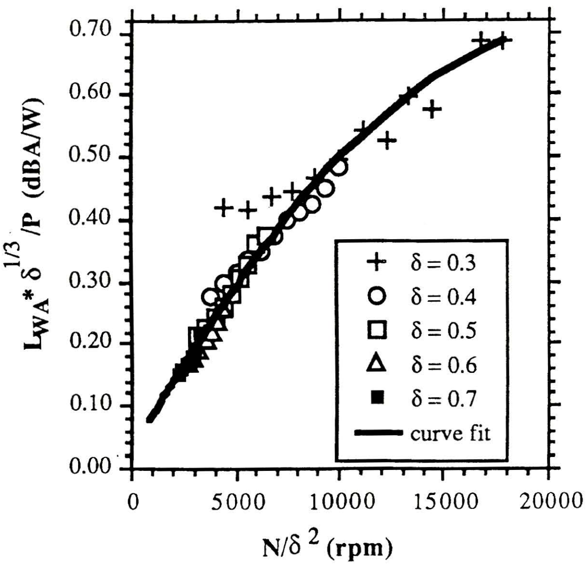

Figure 7 Variation of sound power parameter with speed parameter.

The variation of the overall A-weighted sound power level (L_{WA}) with the shaft power (P) for the same test conditions is shown in Figure 6. For a given δ, L_{WA} decreases with increase in P and for lower δ, there appears to be a shaft power at which L_{WA} is maximum. For a given P, L_{WA} increases with δ. As the sound power level curves show similar trends, it is expected from the above discussions on scaling for shaft power data that with appropriate scaling, the sound power data may be reduced to be geometrically similar. Since L_{WA} depends on P and δ, various combinations of L_{WA} , P and δ have been tried to formulate a sound power parameter to describe the acoustical performance of this switched reluctance motor as a function of the speed parameter N/δ^2. As shown in Figure 7, with the use of a sound power parameter $L_{WA}\delta^{1/3}/P$, the sound power data all collapse onto a single curve, thus supporting the postulation that these data can be scaled appropriately to yield geometric similarity. The curve of best fit to the test data in Figure 7 ,with a correlation coefficient of 0.97, is given by:

$$\frac{L_{WA}\delta^{1/3}}{P} = 0.032 + 6.08\text{x}10^{-5}\frac{N}{\delta^2} - 1.312\text{x}10^{-9}\left[\frac{N}{\delta^2}\right]^2 \qquad (2)$$

The successful collapse of the shaft power and sound power data with an appropriate choice of parameters indicate that ,at least for a given pair of advance and delay angles, the performance of a switched reluctance motor may be explicitly described by equations similar to (1) and (2). These results indicate that for switched reluctance motors, there will be a significant reduction in the number of motor tests that have to be conducted to produce performance data. Work is now being in progress to examine if the dependence of the performance of a switched reluctance motor on advance/delay angles can be described by modifying the shaft power parameter and sound power parameter with the introduction of an angle parameter. It is also important to examine through testing of motors with different power capacities whether, for the same motor configuration,control and mechanical structure, equations (1) and (2) are independent of the power output of the motor.

5.0 CONCLUSIONS

The basic principle of operation of a switched reluctance motor has been described. The overall A-weighted sound power level of a 600 W switched reluctance motor for various operating load conditions have been measured in an anechoic chamber by using the sound intensity technique in order to eliminate the noise contributed by the dynamometer. It has been shown from the results obtained that with an appropriate choice of relevant parameters (speed, shaft power and sound power parameter), the sound power data for various operating conditions can be collapsed onto a single curve and can be explicitly described by an equation. While it remains to be seen whether the same parameters can be used for switched reluctance motors with different power outputs, there is no reason why the same procedure cannot be used for reducing the data into geometrically similar form. The parameters not only allow a compact description of the performance (both mechanical and acoustical) of a switched reluctance motor but also they eliminate the need for extensive testings in order to produce adequate performance data.

ACKNOWLEDGEMENT

This project is being supported by the Australian Research Council. The experimental data have been collected by A. Dombek and N. Huynh. Discussions with J. Milthorpe and D. Pulle have been most stimulating.

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THE USE OF TREES FOR NOISE REDUCTION

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ABSTRACT

Many Defence bases are set in, or surrounded by, large areas of trees. Several tests have been conducted by the National Acoustic Laboratories to evaluate the effectiveness of trees in reducing environmental noise levels. Sound sources used for these tests include continuous noises and various types of impulsive noise.

Results of the tests are included and discussed.

1.0 INTRODUCTION

National Acoustic Laboratories (NAL) have been conducting research into propagation of sound outdoors since 1984. This has consisted of field trials at various locations around Australia using both continuous and impulsive noise sources and measuring the characteristics of these sounds at distances up to 3.2 kilometres. The research has been sponsored by the Department of Defence because of concern about the possible effect of its noisier activities on the community.

Recently these studies have included measurements of the effects of trees on the propagation of sound. To date four field trials have been conducted:

* At Greenbank, south of Brisbane in Queensland, 570 gramme slabs of TNT were detonated at the edge of a stretch of trees and peak levels and waveforms measured both through and alongside the trees.

* At Williamtown, north of Sydney in NSW, continuous sounds from horn loudspeakers were measured at various distances through trees and at the same distances alongside the trees.

* Also at Williamtown, peak levels and waveforms were recorded of both detonators and Powergel explosive along these propagation paths.

* Using blanks from a rifle, peak levels were measured at Kingswood near Sydney for propagation through trees and at similar distances over open ground.

Typically, the vegetation through which these tests were conducted consisted of dry sclerophyll with angophora, eucalypt and casuarina trees.

2.0 MEASUREMENTS

2.1 GREENBANK

An area was selected with a clearing approximately 100 metres wide and 500 metres long with an access road cutting through the trees at one end. At the other end there was a small clearing in the trees off the main clearing. Explosive noise sources were placed in this small clearing and identical sound measuring equipment were located at the other end of the clearing near the road and also on the road behind the trees.

With this set up, noise measurements could be taken of the explosions for the sound travelling along two paths, one through the clearing in the trees and one through the trees, with about 20 metres of clearing between the sound source and the tree line and 10 metres at the other end between the microphone and the trees. In both cases, the source to microphone distance was almost 480 metres.

2.2 WILLIAMTOWN - CONTINUOUS

Tests were conducted at a location which had a depth of trees of over 750 metres with a wide road/service corridor running alongside. Tracks lead into the trees from this corridor at several points allowing access for measurements.

A high power horn loudspeaker was mounted so as to propagate sound through the trees. An identical unit was used to propagate sound in a parallel path along the corridor. The speakers were alternately driven so that levels could be measured in the trees and at the same distances in the open. Each speaker was driven from a 200 watt amplifier using a swept tone signal from 2000Hz to 200Hz.

Sound levels were monitored 10 metres forward of each loudspeaker.

Tape recordings were taken along the road at distances of 190 metres, 570 metres and 750 metres from the sound source in the open and recordings were taken in the trees at similar distances from the speaker located in the trees.

2.3 WILLIAMTOWN - IMPULSIVE

The same propagation paths were used for the impulsive noise tests. Sticks of Powergel and electric detonators were used as impulsive sound sources. The typical firing sequence would be to fire a detonator in the open, then in the trees, Powergel in the open and then in the trees, after which new charges would be set and the sequence repeated.

Peak levels and waveforms were measured at distances of 190 metres, 570 metres and 750 metres, both in the open and in the trees.

2.4 KINGSWOOD

Using an L1A1 SLR rifle firing blanks, as a noise source, peak level readings were taken at distances of 100 metres, 200 metres and 400 metres along the small arms range. The noise source was then moved to the edge of an area of trees at the end of the range and peak levels were measured in the trees at these same distances and for the same direction of propagation.

3.0 RESULTS

3.1 GREENBANK

570 gramme TNT slabs measured at 480 metres

Linear Peak Level dB		
Open	**Trees**	**Open - Trees**
134.8	131.8	3.0
136.2	133.0	3.2
137.5	135.0	2.5
139.3	135.3	4.0
133.5	131.5	2.0
138.0	135.3	2.7
137.3	136.9	0.4
137.5	134.4	3.1
139.7	134.8	4.9
Mean		2.9
Standard Deviation		1.2
No. of Samples		9
Maximum		4.9
Minimum		0.4

3.2 WILLIAMTOWN - CONTINUOUS

Level Differences - Continuous Noise Propagation through Trees

	Measurement Distance from Source (m)		
	190	**570**	**750**
	Open Level minus Tree Level (dBA)		
	7.2	10.4	8.2
	4.8	12.6	14.0
	2.8	12.6	14.3
	6.8	13.7	10.6
	4.9		17.3
	0.3		13.6
	1.4		14.3
	4.5		18.1
	6.7		16.8
			18.5
Mean	4.4	12.3	14.6
Standard Deviation	2.3	1.2	3.1
No. of Samples	9	4	10
Maximum	7.2	13.7	18.5
Minimum	0.3	10.4	8.2

3.3 WILLIAMTOWN - IMPULSIVE

Excess Attenuation due to Trees, using Detonators

Statistic	Peak Level Difference (Open - Trees) - dB 190 metres	570 metres	750 metres
Mean	6.9	10.8	>7.3*
Standard Deviation	3.9	6.1	5.8
No. of Samples	30	31	33
Maximum Value	13.8	19.7	>15.6
Minimum Value	-0.4	-3.6	-6.8

Excess Attenuation due to Trees, using Powergel

Statistic	Peak Level Difference (Open - Trees) - dB 190 metres	570 metres	750 metres
Mean	3.9	10.4	8.2
Standard Deviation	1.2	2.3	2.7
No. of Samples	29	35	34
Maximum Value	7.8	14.3	13.2
Minimum Value	1.0	0.8	2.0

* When measuring through the trees, some levels were at, or below, ambient noise levels.

3.4 KINGSWOOD

Rifle Blanks Measured Through Open

Statistic	Peak Levels - dB: 100 metres Linear	100 metres A-Weight	200 metres Linear	200 metres A-Weight	400 metres Linear	400 metres A-Weight
Mean	120.6	121.4	114.8	112.7	98.3	92.8
Standard Deviation	3.1	0.8	1.3	1.4	2.5	2.4
No. of Samples	10	10	10	10	10	10
Maximum	125.0	122.5	116.9	115.2	102.0	97.5
Minimum	115.6	120.0	113.0	110.0	94.1	88.7

Rifle Blanks Measured Through Trees

Statistic	Peak Levels - dB: 100 metres Linear	100 metres A-Weight	200 metres Linear	200 metres A-Weight	400 metres Linear	400 metres A-Weight
Mean	106.3	108.0	92.6	88.2	86.8	81.0
Standard Deviation	3.4	2.1	0.8	1.5	1.2	1.3
No. of Samples	8	8	9	10	10	10
Maximum	112.6	111.0	93.7	91.0	90.2	83.5
Minimum	102.0	104.9	91.3	86.0	85.7	78.8

Excess Attenuation due to Trees

Peak Level Difference (Open - Trees) - dB					
100 metres		200 metres		400 metres	
Linear	A-Weight	Linear	A-Weight	Linear	A-Weight
14.3	13.4	22.2	24.4	11.5	11.8

4.0 DISCUSSION

Trees can provide useful reductions for both continuous and impulsive noises where there is a good depth of trees.

This depth will depend on the type and density of trees, but for those tested the depth should be at least 100 metres.

In general, impulse noises of short duration (rifle blanks, detonators) will be attenuated more than those of longer duration (TNT, Powergel).

Attenuation appears to be non-linear with distance and frequency. For instance, at Williamtown, the A-weighted attenuation rate decreased with distance and, using detonators, the attenuation at 750 metres was not much greater than at 190 metres. Also at Williamtown, with a continuous noise source, although the attenuation rate generally decreased with decreasing frequency, in the range 200 to 300 Hertz the attenuation rate increased and did so rapidly at the 190 metre position.

Some of these non-linear effects may be explained by variations in the type and density of growth along the propagation path and also the contribution of flanking paths around and over the trees. Where the attenuation increases rapidly over a narrow frequency range, as with the continuous noise tests at Williamtown, then this may be related to the average spacing of the trees.

5.0 CONCLUSIONS

Although many factors affect the sound attenuation performance of trees, useful reductions of both continuous and impulsive noises are provided by depths of 100 metres and more.

In general, the greater the depth of trees or the higher the frequency of the continuous sound or the shorter the duration of the impulse then the greater the attenuation.

Buffer zones around noisy activities, such as firing ranges and engine run-up areas, will be far more effective if heavily timbered. Clearing these zones of trees for farming, mining and other development will greatly reduce their effectiveness.

IMPULSE MEASUREMENTS OF ACOUSTIC IMPEDANCE USING A MICROPHONE ON THE SURFACE

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Abstract

Many acoustic problems, including noise propagation studies, require a knowledge of the impedance of surfaces. While impedance tube techniques are appropriate for uniform materials, they are limited for situations involving multi-impedance paths, or where dynamic conditions exist, such as when rain-water is altering the properties of soil.

One established impulse technique uses two microphones to compare the direct and reflected waveforms measured about a metre above the surface to determine it's average impedance. A recently developed technique uses a single microphone located on the surface to capture the resultant waveform from an impulse source also located on the surface. Because the direct and reflected signals have merged, and there is a ground wave component, an iterative analysis is required to deduce the impedance from the ground wave. Apart from being a simple set-up, this method permits the direct measurement of the effective impedance experienced by sound propagating over composite surfaces. By contrast, when the impulse source is located, say, 1m above the microphone on the surface, the resulting sound field is influenced by a restricted area around the receiver producing localised impedance data, which is especially useful for investigating the layering properties of wet soils.

Impedance measurements obtained over layered foam, carpet, and different types of soil will be presented, and a comparison of the above techniques will be made, demonstrating their advantages and limitations.

Introduction

Calculating the interaction of sound with a boundary surface requires a knowledge of the characteristic or normalised impedance of that surface[1,2]. Experimental methods to determine these parameters include using a standing wave tube[3] and the reflection of either continuous[4] or impulse sounds[5,6]. An impulse method has the advantage of obtaining impedance data over a wide frequency spectrum simultaneously, and, under suitable geometric conditions, providing the ability to time isolate the required signal from unwanted reflections. In this paper three impulse methods are compared; one which is well established in the literature[6] uses a two microphone technique while the other two require only one microphone and use a simplified geometry. Providing the latter techniques are reliable, they offer the advantage of convenience when taking measurements. Results over different surfaces using the three methods are compared and the limitations of the individual techniques are discussed.

Principle of measuring ground impedance.

Above flat ground, the sound pressure due to a point source above the surface consists of two components, the direct wave from the source and the wave reflected by the ground, as shown in Fig.(1a). The principle of measuring impedance by the impulse technique is to capture a pulse which has been modified in amplitude and shape by reflection from the impedance surface. A direct or free field pulse, that is, one which travels the same distance to the microphone without interacting with any surface, is also required. By dividing the frequency components of the reflected pulse by the corresponding direct ones a quantity Q is calculated, at each frequency, which theoretically is given by

$$Q = R_p + (1 - R_p)\, F(w) \tag{1}$$

where both the plane wave reflection co-efficient, R_p, and the boundary loss factor, $F(w)$, depend on the normalised surface impedance, Z, given by

$$Z = \frac{1+R_p}{(1-R_p)\sin\psi}, \tag{2}$$

where ψ is the angle the incident sound ray makes to the surface. $F(w)$ is a relatively complicated function which depends on the geometry of the system as well as the impedance. When ψ is greater than 30^o, $F(w)$ becomes sufficiently small that the approximation $Q = R_p$ holds, so that Z can be readily determined for either the geometry of Fig.1(a) or 1(b). As ψ approaches 0^o, $F(w)$ becomes a dominant term. In the geometry of Fig.1(c), where $\psi = 0^o$, Eq.(2) indicates that $R_p = -1$ and Eq.(1) now simplifies to $Q=-1+2F(w)$. If $R_p = -1$ then the incident pulse is exactly inverted and since it arrives simultaneously with the direct, in the geometry of Fig.1(c), they both cancel leaving only the $2F(w)$ term, which is often called the ground wave. As indicated by the typical pulse waveforms shown in Fig.1, the ground wave pulse is much more rounded than those recorded in the other methods and the reflected pulse may become inverted when ψ is sufficiently small.

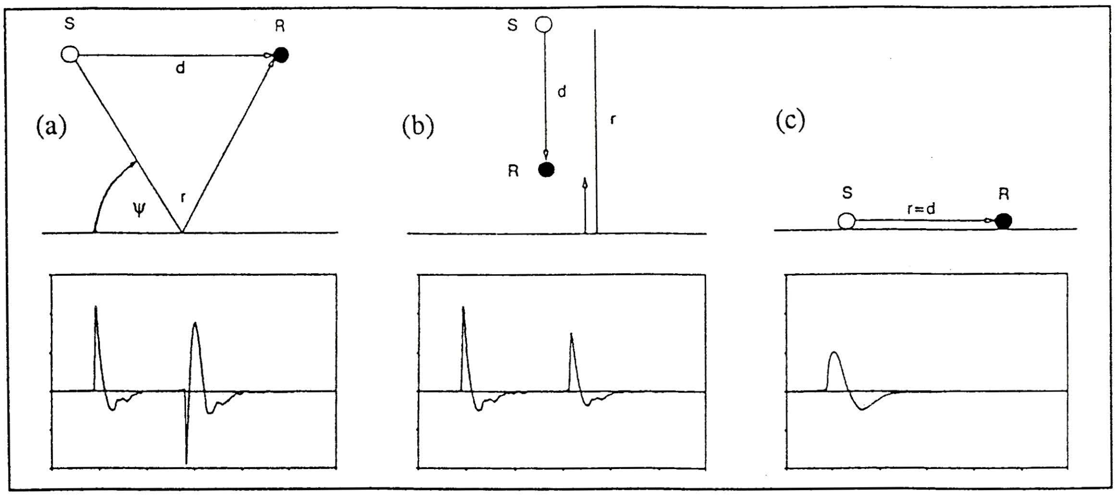

Fig.1: Three methods used to determine impedance and typical pulse shapes.

Measurement Techniques

The data was processed in a PC, by ensemble averaging at least 10 direct and 10 reflected waveforms before performing fast fourier transforms to obtain the frequency data required to determine Q. B & K 1/4 inch microphones and type 2218 sound level meters connected into a Data 6000 waveform analyser were used to record the signals. In most of this work shot shell primers were discharged to produce the impulse signal, however, in some measurements using method 3, a speaker driven by a waveform synthesiser was used to produce an impulse with a reproducible amplitude and frequency content. The reduced signal level from the speaker was acceptable for this method as the sound only had to travel one metre, whereas method 1 typically requires a path distance of 4 m to obtain the necessary time separation between the direct and reflected wave recorded at the microphone.

A geometry similar to that shown in Fig.1(a) was used for the two microphone technique[6] where one microphone records the reflected pulse and a second microphone was located a distance r away from the source such that it recorded the pulse travelling directly from the source. This avoided having to allow for geometric spreading of the signal.

Method 2 involves using the geometry in Fig.1(b) with the one microphone measuring the direct pulse and then some time later, depending on the distance of the microphone from the ground, recording the reflected pulse. If the time separation is small, so the reflected pulse is recorded on top of the direct pulse, the direct pulse must be determined in an independent measurement and then appropriately subtracted from the initial recording. Since there is a difference in the distance travelled between the direct and reflected pulses, the direct pulse must be rescaled according to the inverse square law.

The third method involves placing both the microphone and the source on the ground, thus both the direct and the reflected pulses arrive at the same time, Fig.1(c). This means that a direct pulse must always be recorded separately to be used in the calculation of Z. Because this method involves finding an appropriate value of Z to fit the measured F(w), an iteration technique must be used, whereas the other two methods involve a simple calculation to find Z from Eq.(2).

Experimental results

Carpet is a useful test situation as it is easy to obtain a large uniform surface area, in this case laid over a hair felt backing. Method 2 results are the average of seven independent sets of data taken at microphone heights varying between 0.8m and 1.1m with the source at 1.5m or 1.8m. All the individual data sets agreed to within the size of the data points used on the graph. Both the sets of data used to form the average for the method 3 results were taken with a source receiver distance of 1m. As shown in Fig.2. all three methods give good agreement above 2kHz, with some divergence at lower frequencies.

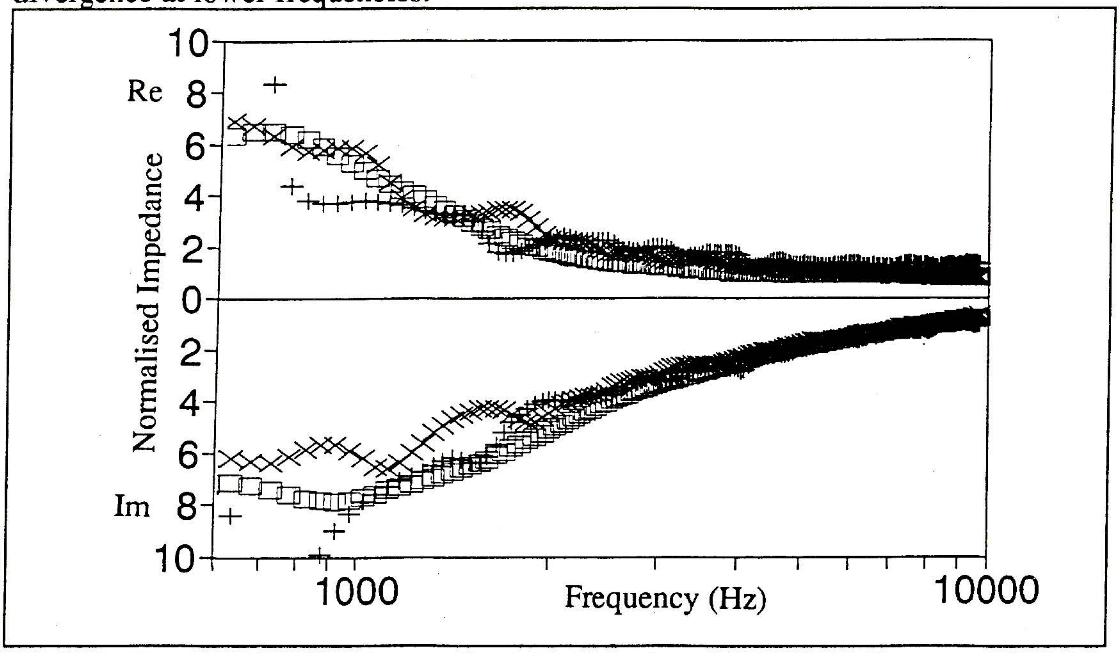

Fig.2: The complex impedance, Real (Re) and Imaginary (Im), of carpet using, □ method 1, × method 2, + method 3.

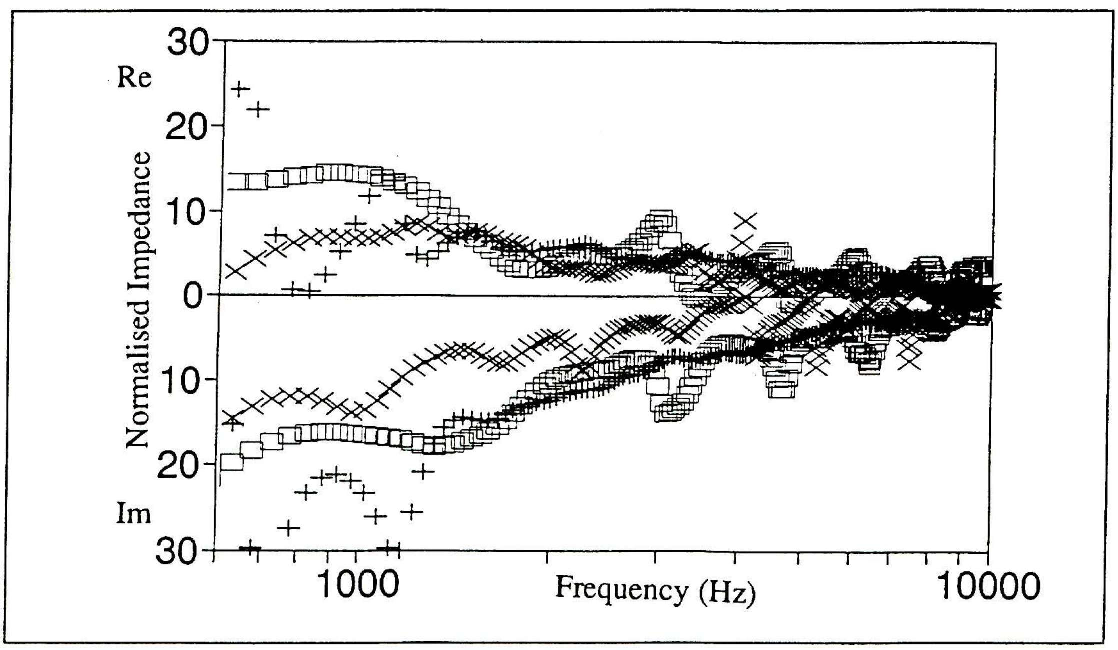

Fig.3: The impedance of wet grassland using, □ method 1, × method 2, + method 3.

An example of the impedance of grassland determined by the three techniques is presented in Fig.3. The measurements were taken over a very wet sports field covered sparsely with grass mown to about 2cm. Method 1 data are an average of two impedance measurements taken over the same area but with the point of reflection moved 10cm, while the method 2 results were obtained at the mid-point of the reflections from the previous experiment. An average of two impedance data sets obtained by method 3, using a source to receiver distance of 1m is also shown. The most notable feature of the data is the lack of "resonances" in the method 3 results. These resonances are believed to occur because of layering in the wet soil[7]. The frequencies at which these resonances occur are known to be sensitive to the nature of the soil around the point of reflection. It appears that method 3 does not detect this layering in the same way or simply averages out such effects along the 1m path. This averaging effect may also occur in the data of Fig.4, where method 3 was used, at varying distances between source and microphone, over much drier grassland. While the three sets of data in Fig.4(a) are in good agreement, the smoothest curve is the 3m data set while the most fluctuations occur in the 1m result. The average of these three data sets is in agreement with method 1 data obtained over the same area, as shown in Fig.4(b).

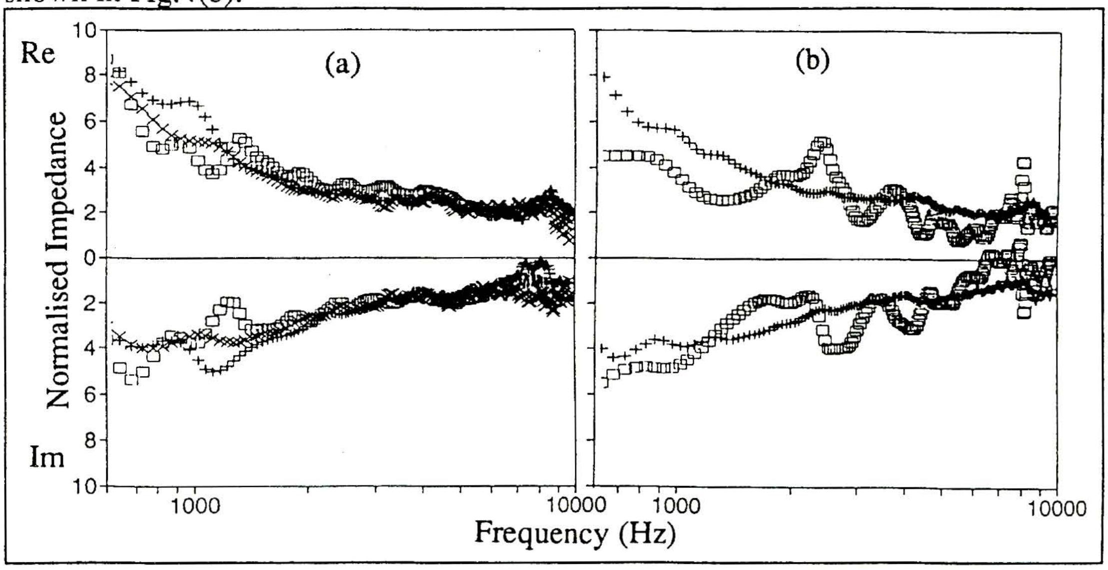

Fig.4: (a) The impedance of dry grassland using method 3 over □ 1m, + 2m, × 3m.
(b) method 1 □ compared to the average of method 3 +.

Results from a thick layer of grass, which formed a very porous mat about 5cm deep over the soil, are shown in Fig.5. There is good agreement between the results of method 1 and 2, except when the microphone was placed on the ground in method 2. These results appear to agree more favourably with data obtained using method 3. At this stage we have not been able to resolve why such differences occur, although it appears to be due to the extreme layering.

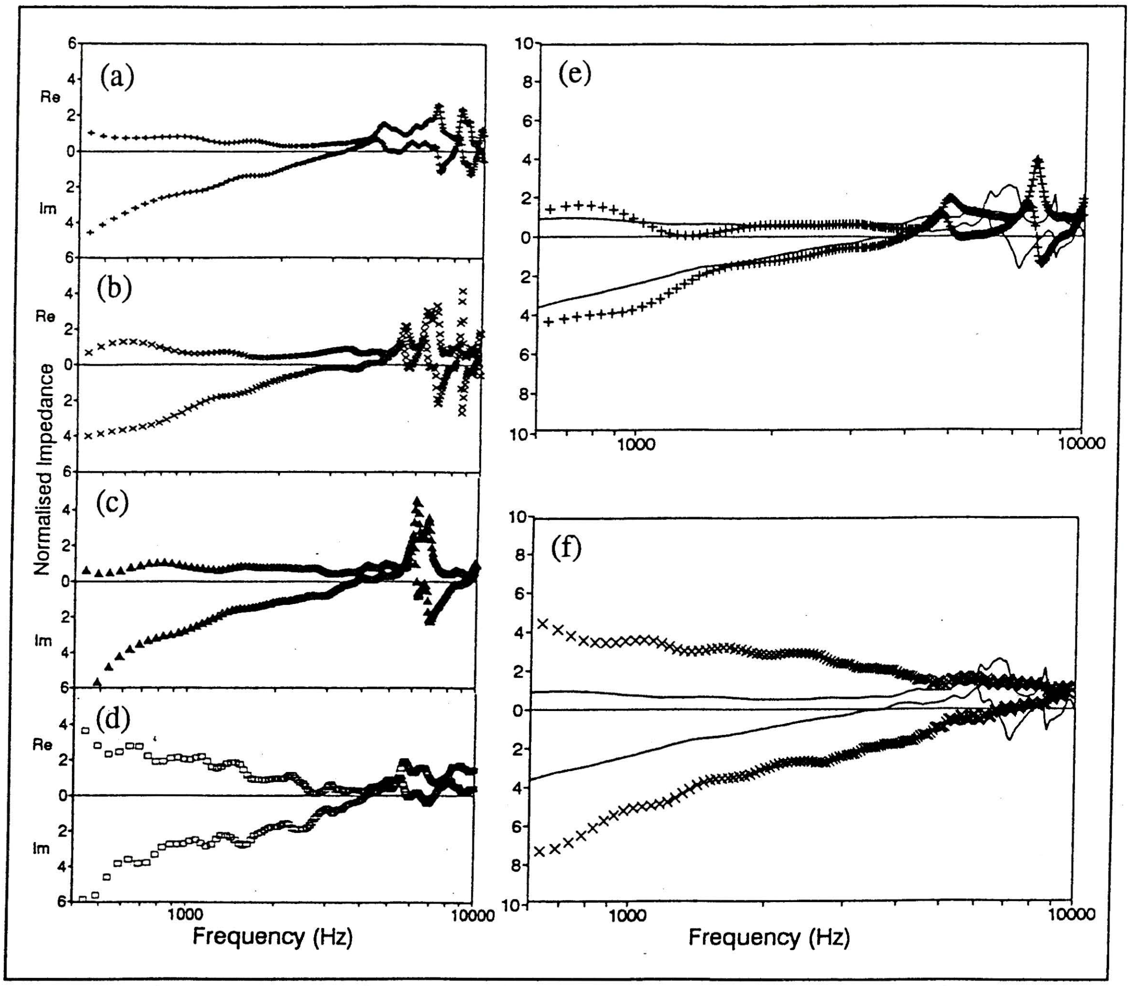

Fig.5: The impedance of thick grass: (a), (b), (c), (d) method 2 with receiver heights of 0.5m, 0.25m, 0.05m, 0.0m respectively; (e) — trend line of (a), (b), and (c), + method 1; (f) — trend line, × method 3.

A layer of foam is a more complicated system than those considered above, as the surface is not locally reacting but one of extended reaction - this means that the pressure incident at one point influences the motion elsewhere on the surface. Under this condition, Eq.(2) no longer applies and the effective impedance will be angle dependent. To illustrate this, Fig.6 shows measurements at various values of ψ, obtained over a 5cm layer of foam on a hard surface. The resonances, caused by interference of sound reflected from the upper and lower interfaces, occur at different frequencies which can be calculated theoretically at higher angles, such as 90^o and 60^o, as demonstrated in Fig.6(e) for the latter case. At lower angles the complication of theoretically including extended reaction in the ground wave term has not been resolved. Fig.6(f) compares a measurement taken at 60^o with a set where the microphone was placed on the surface such that the ray from the source to the microphone was at 60^o to the surface. Unlike earlier method 2 measurements, where the microphone on the surface gave inconsistent results, these are in excellent agreement.

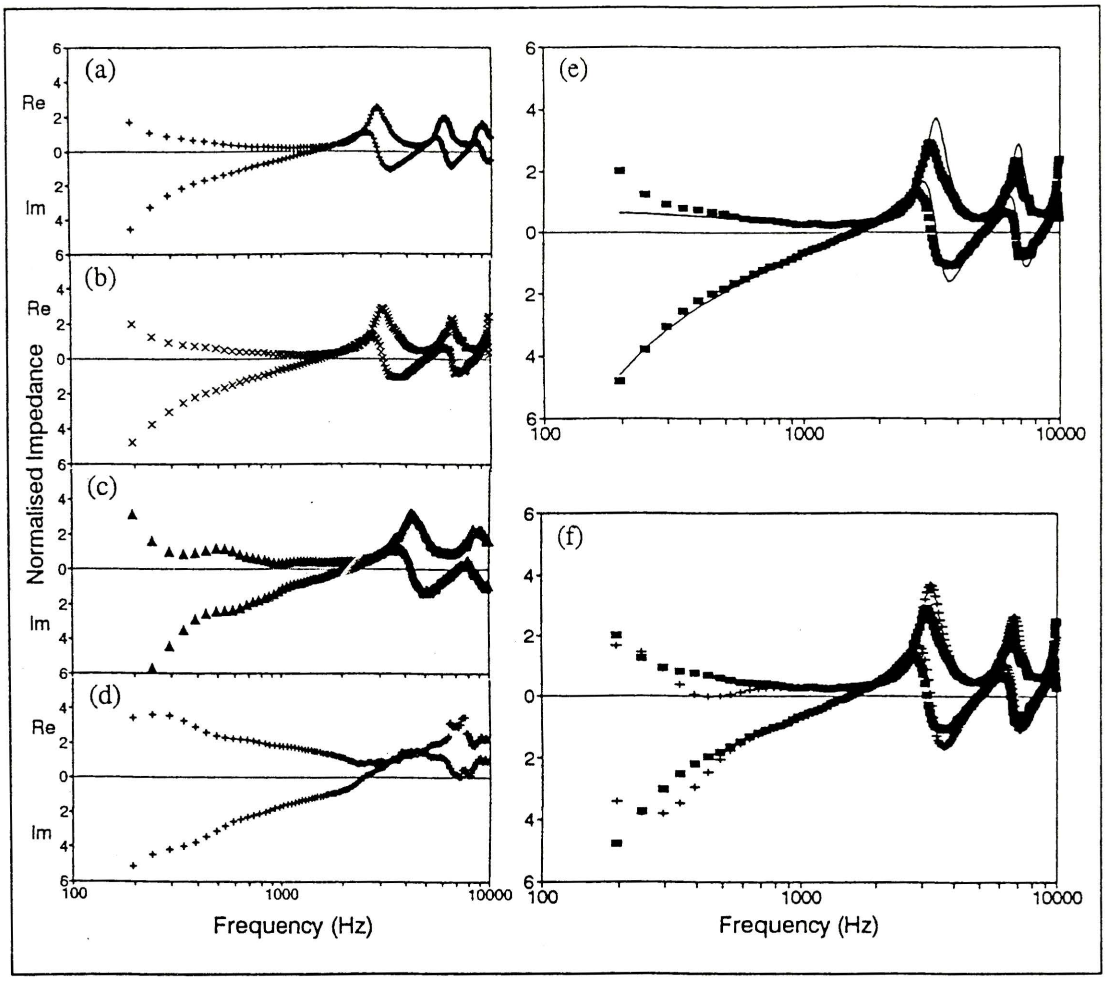

Fig.6: The impedance of foam: method 1, (a) $\psi = 90^{o}$; (b) $\psi = 60^{o}$; (c) $\psi = 30^{o}$; (d) method 3, $\psi = 0^{o}$; (e) - theory, ■ method 1, $\psi = 60^{o}$; (f) ■ method 1, + method 2, $\psi = 60^{o}$;.

Conclusion

Results for all three methods are generally consistent over ground and over thin layered surfaces, such as carpet. At this stage it appears necessary to take care when interpreting method 2 data taken when the microphone is placed on the surface. Only data taken over the extended reaction surface of foam seem to be consistent with other techniques. Method 3 did not agree with the other data over a thick grass layer and in a number of cases seems to diverge at frequencies up to 1kHz. Further, this method appears to give a more averaged impedance than the other techniques. A major advantage of method 2 is that it can be applied to smaller areas than method 1, which is, to a lesser extent, also true for method 3.

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MICROPHONE PLACEMENT, WIND NOISE and WINDSCREENS

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When measuring impulsive noise in an outdoor environment, air movement across the microphone diaphragm generates "wind noise". This wind noise component may be of sufficient amplitude as to dominate the measurement process and result in meaningless data.

The effect of the microphone height above the ground is discussed with respect to both the amount of wind noise and the waveform shape. Microphone windscreens constructed from "shadecloth" are discussed and a windscreen design, suitable for a low mounting height microphone, is also presented.

Microphone Placement, Wind Noise and Windscreens

1. Introduction

Various Environmental Agencies throughout Australia specify that annoyance caused by impulsive noise is to be assessed by the measurement of the linear (broadband) peak level of the impulse. The measurement of impulses in an outdoor setting is regularly interfered with by the presence of wind noise components in the measured signal. For example, in wind speeds of 9 m/sec (Beaufort scale 4, "moderate breeze") broadband (2 Hz to 70 kHz) linear peak levels of nearly 130 dB SPL are experienced when using a 12 mm diameter microphone without a windscreen.

Wind noise is low frequency dominated. Cooper [Reference 1] and Daigle *et al* [Reference 2] report measurements of wind turbulence using a hot wire anemometer which indicate that the amplitude of frequency components reduces at a 6 dB/octave rate with rising frequency.

Reference 3 describes the mechanism involved in the generation of wind noise and the use of shadecloth as a windscreen material.

2. Wind speed versus height.

In the lower level of the earth's atmosphere, the interaction of the atmosphere and the earth's surface creates a "friction layer" which reduces the wind speed as height is decreased. Figure 1, below, shows an example of a graph of wind speed versus height.

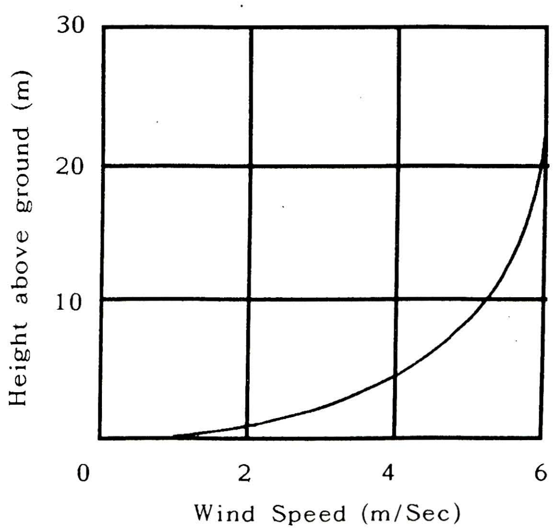

Figure 1. Wind speed versus height.

It can be seen that the wind speed decreases rapidly as the ground surface is approached. From measurements conducted by Thornthwaite [Reference 4] the reduction in wind speed is of the form :

$$u = u_1 z^a$$

where u = wind speed (m/sec)
u_1 = wind speed at 1 m height
z = height of interest
and a = factor which varies as to height, temperature surface roughness and time of day.

The expected result of reducing the microphone height from say 1.5 m to 10 cm is that the wind speed should reduce by approximately 50 % and thus a decrease in wind noise of approximately 15 dB should also be expected.

Another way of looking at this proposal to lower the microphone height is as follows : Geiger [Reference 5] cites measurements taken by Hellmann of wind speed checks taken with cup anemometers over a period of several months at heights of 5, 25, 50, 100 and 200 cm above the ground. Figure 2 shows one result of these measurements. The Figure expresses in percentage terms the frequency of "hours of calm" in the lowest 2 metres of the atmosphere.

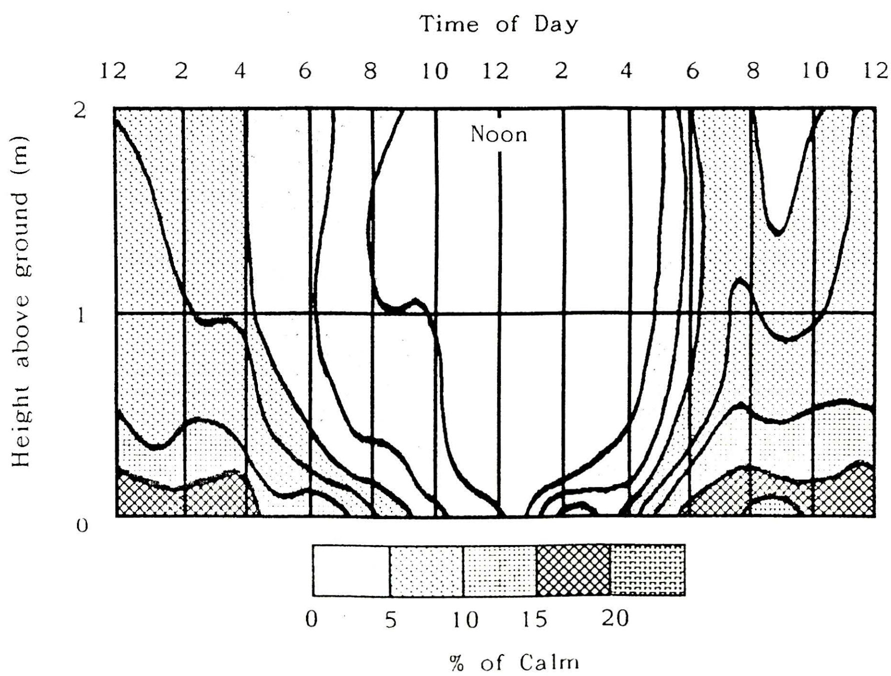

Figure 2.. Frequency of "hours of calm" in the lowest 2 m of the atmosphere.

It can be seen from Figure 2 that for a microphone height of 1.2 to 1.5 metres the frequency of "hours of calm" between 0800 and 1700 hours is substantially 0%. Thus at this microphone height one can always expect to have wind present during normal working hours. However, if the microphone height is reduced to 10 cm then a significant proportion of time will be spent in calm conditions.

3. Acoustic effects versus height.

In order to examine the effects, if any, of reducing the microphone height from 1.2 m to 10 cm, many measurements were conducted using the equipment set-up detailed in Figure 3.

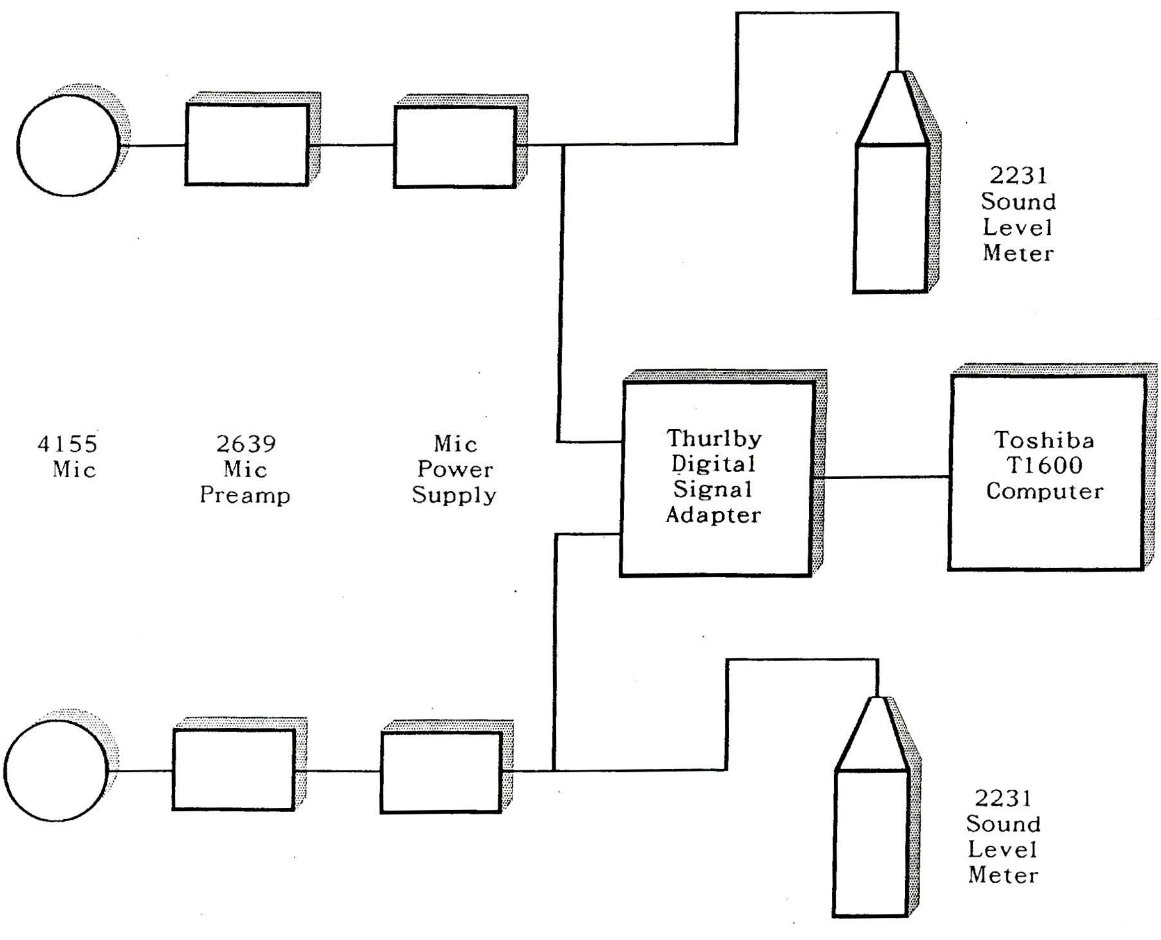

Figure 3. Equipment used for acoustic measurements.

Simultaneous measurements were conducted with one microphone mounted at the standard height of 1.2 to 1.5 metres above ground whilst a second microphone was located in close proximity at 10 cm above the ground. The acoustic sources were all impulsive and were recorded in conditions as detailed below in Table 1.

TABLE 1.

Details of the measurement conditions.

Location	Source	Ground Conditions	Meteorological Conditions	Distance from Source
RAAF Williamtown	No. 8 electrical detonator plus 120 g stick of Powergell "Magnum" 3151 explosive.	Sandy soil, covered with mown grass.	24°C, 61% R.H., no wind, medium height broken cloud.	180, 570 and 750 metres
Holsworthy Army Range	400 g Dynagex explosive, 1.25 kg TNT, or, 1.25 kg ANFO explosive.	Clay soil, small shrubs to 0.5 m height. Measurement location in open area.	18°C, 40% R.H., wind 1-2 m/S across propagation path.	3.2 km
RAAF Orchard Hills	Small arms weapons. Muzzle, and shock waves.	Long grass, scattered 5 m high trees.	16°C, 60% R.H., wind 0-2 m/S in direction of propagation.	400 m
Puckapunyal Army Range	80 mm Mortars, 105 mm and 155 mm Artillery. Muzzle, shock wave and impact.	Sparse grass, small stones together with small trees. Measurement location elevated.	26°C, 60% R.H., wind 5 m/S in various directions relative to the propagation path.	1.3 to 10.2 km

Several examples of waveforms recorded at some of these sites are presented below. The waveforms were recorded using the set-up detailed in Figure 3 and are presented here in a manner that enables a close comparison of the waveform shapes.

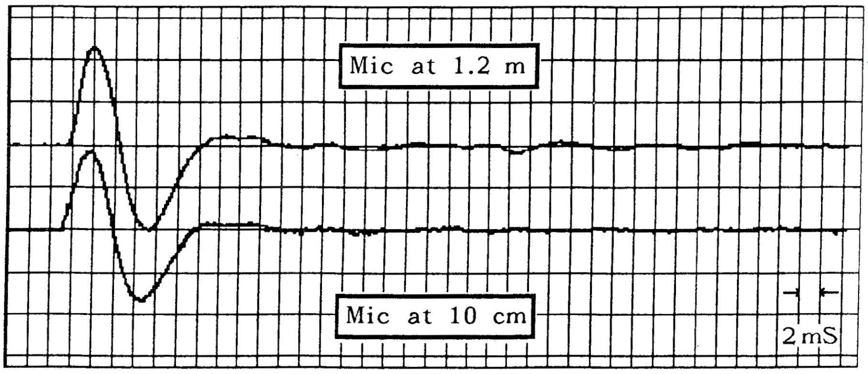

Figure 4. Waveforms recorded at RAAF Williamtown, 180 m from the source, at microphone heights as indicated. Waveform sample rate 50 kH.

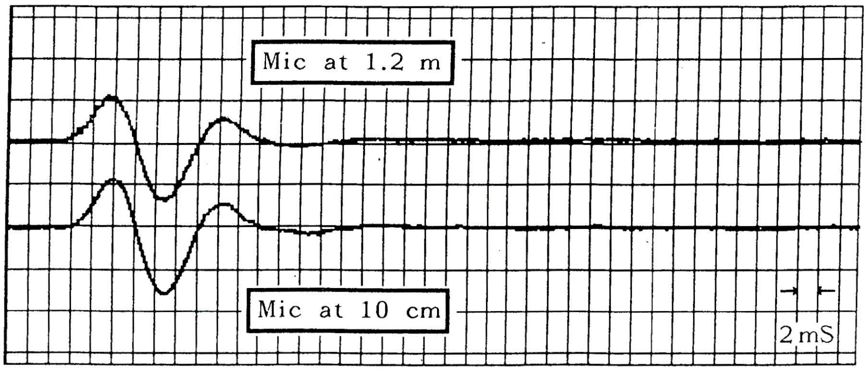

Figure 5. Waveforms recorded at RAAF Williamtown, 570 m from the source, at microphone heights as indicated. Waveform sample rate 50 kHz.

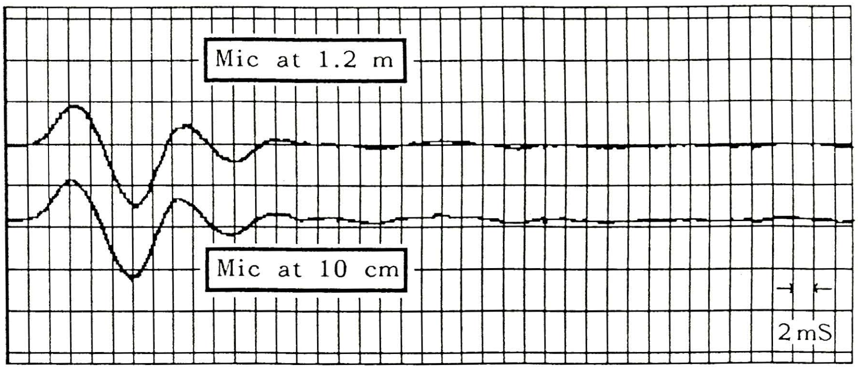

Figure 6. Waveforms recorded at RAAF Williamtown, 750 m from the source, at microphone heights as indicated. Waveform sample rate 50 kHz.

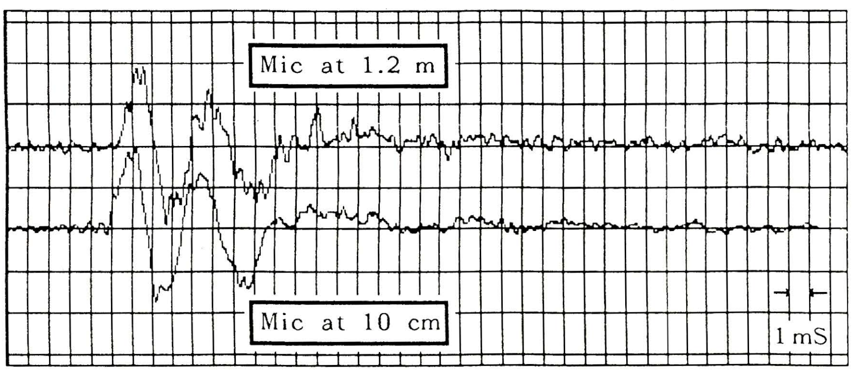

Figure 7. Waveforms recorded at RAAF Orchard Hills, 400 m from the source, at microphone heights as indicated. Waveform sample rate 100 kHz.

It can be seen from examination of Figures 4 to 7 that for all practical purposes there is no difference in the peak level recorded by each microphone.

Linear peak sound pressure level measurements were also conducted at Puckapunyal over a period of two days of 80 mm mortars and 105 mm and 155 mm artillery. For the 80 mm mortars, measurements were of the impact detonation, whilst for the artillery weapons the muzzle and projectile impact were recorded. Commencing as early as 0624 hours and finishing as late as 1813 hours (a total measurement time over the two days of > 17 hours) the results obtained from two microphones mounted at heights of 1.2 m and 30 cm indicated that the mean difference between the two microphones was 0.05 dB over a total of 178 events.

4. Windscreen design for low mounting height microphones.

Reference 3 describes the characteristics of "shadecloth" when used for the construction of microphone windscreens. Briefly, shadecloth is a readily available knitted material, with low stretch characteristics, good resistance to tearing, is UV stabilised and strong. The grade used is classed as 70% shade. Tests conducted in an anechoic chamber reveal that for frequencies as high as 20 kHz the shadecloth is acoustically transparent.

Several designs for windscreen have been investigated at NAL but for reasons of ease of installation by one person, and for the reasons detailed above, the preferred design is a hemisphere as shown below. Such a design should be constructed with a diameter of approximately 2 metres so that significant wind noise reduction is obtained.

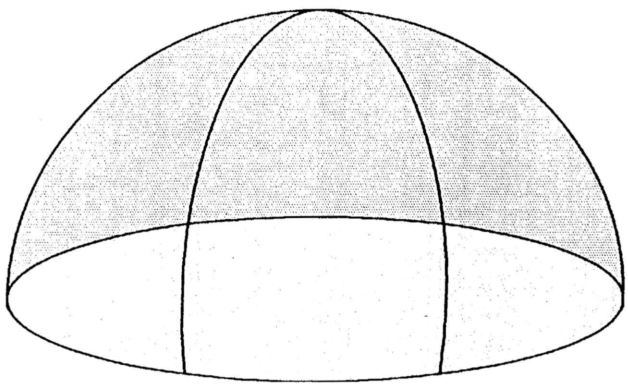

Figure 8. Hemispherical windscreen design.

5. Conclusion.

The problems encountered when measuring low level linear peak sound pressure levels in an outdoor environment are well known. One significant problem is the interference with the desired signal by wind noise components in the microphone output. It has been shown that by mounting the measuring microphone close to the ground it will be subjected to less wind. Mounting the microphone in a suitable windscreen will further reduce the wind noise components that can be present in the microphone output.

At the source to receiver distances usually involved (> 100 m) when environmental measurements are being conducted it has been shown that there is little or no effect from lowering the measurement microphone to a mounting height of 10 cm above the ground.

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Confidentiality/Crosstalk

Richard Latimer

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The paper will be supported by specific test data slides of installation and comments from the installers and final users of the building.

The paper will be presented in conjunction with Nylex, the Australian agents for the material.

Copies of this paper are obtainable from the author at the following address:

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FORMAL CALL FOR ABSTRACTS

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Wednesday 25 November to Friday 27 November 1992

AUSTRALIAN ACOUSTICAL SOCIETY

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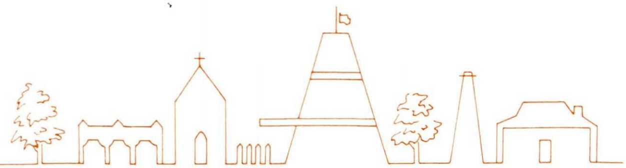

25 — 27 November 1992

PROGRAMME

Wednesday 25 November

5.00 p.m.		Evening Registration & Dinner

Thursday 26 November

9.00 a.m. –	9.30 a.m.	Opening
9.30 a.m. –	12.30 p.m.	Morning Sessions
12.30 p.m. –	2.00 p.m.	Lunch
2.00 p.m. –	5.00 p.m.	Afternoon Sessions
5.15 p.m. –	5.45 p.m.	Annual General Meeting
7.00 a.m. –	11.00 p.m.	Conference Dinner

Friday 27 November

9.00 a.m. –	12.30 p.m.	Morning Sessions
12.30 p.m. –	12.45 p.m.	Closure
12.45 p.m. –	2.00 p.m.	Lunch
	2.30 p.m.	Buses Depart

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Formal Papers will be 20 minutes long with 10 minutes for questions.

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Papers should directly seek to identify an acoustic problem, show what methods were used to solve it, and the subsequent results.

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