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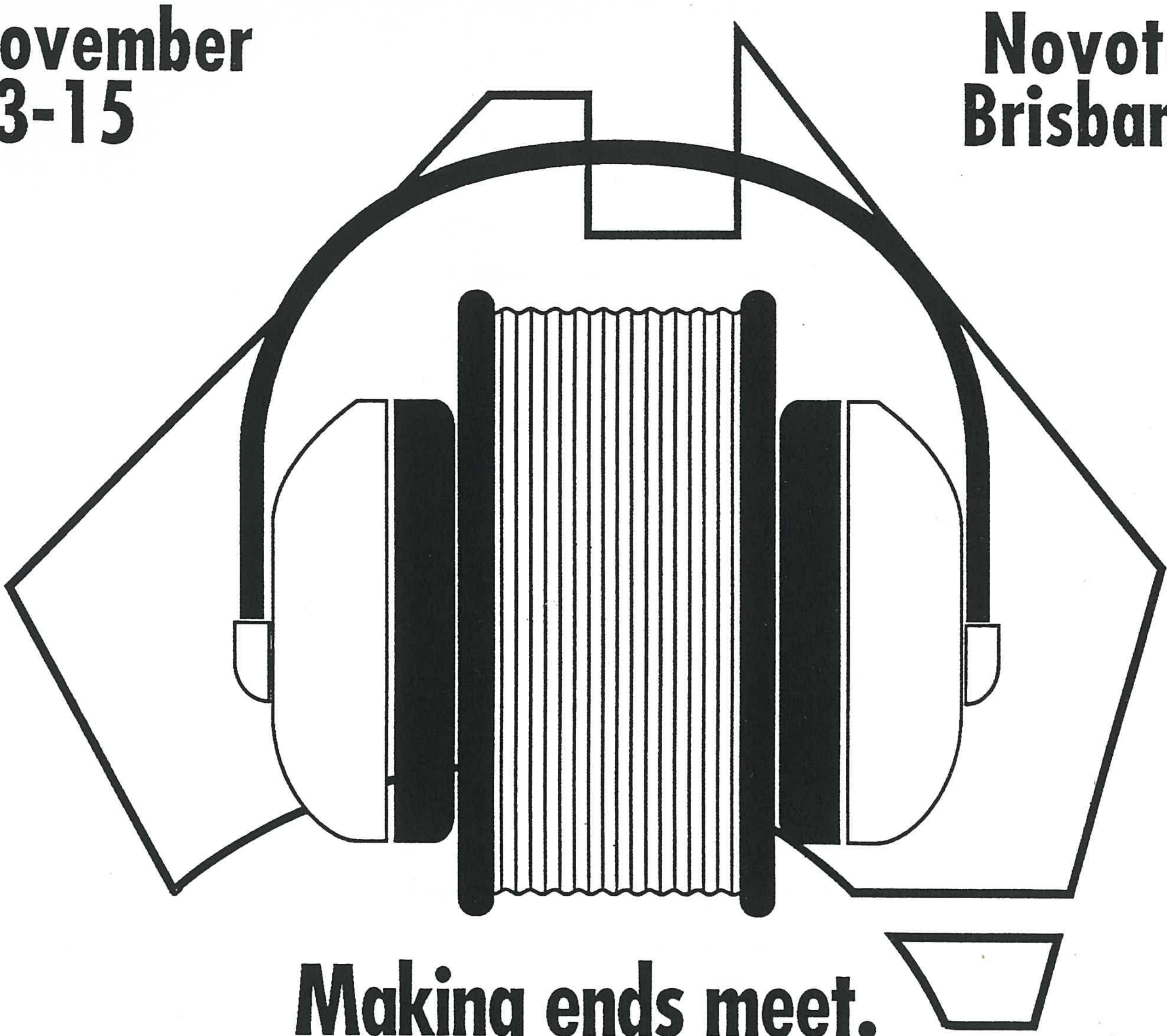

Making ends meet. Innovation and legislation.

TECHNICAL PAPERS

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MAKING ENDS MEET: FROM 10% TO 90%
THE CONTROL OF FLUCTUATING NOISE

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SUMMARY

A continuing difficulty for noise regulating bodies is the problem of deciding whether or not a fluctuating noise is acceptable. The measurement, assessment and control (by regulation) present difficulties, and are all the subject of continuing debate and new proposals. In this paper several of the historical treatments are collected, and several of the current guidelines and techniques presented and reviewed.

INTRODUCTION

Many methods have been used or proposed for assessing the effect of noise on a community and for setting criteria for acceptability of noise sources in the community. For many years, the concept of "nuisance" at common law was a basis. Written laws and regulations developed, but in general terms which prohibited the making of noise that caused undue stress or annoyance. The uncertainties which arose in assessment and the increasing intrusion of noise has led to the setting of numerical criteria in terms of measurable quantities.

For a noise which is steady and has no special distinguishing features the A-weighted sound level is almost universally used for assessment. Corrections may be made for tonal or impulsive characteristics. If the noise is not steady and continuous then difficulties arise - the problems of intermittent and/or fluctuating noise. Many methods for dealing with duration and fluctuation have been proposed, recommended, used and (for some) abandoned. This address is a journey amongst the past and the present of fluctuating noise assessment.

THE WILSON REPORT: 1963

Was this the first attempt? The question is posed because the writer has not made an exhaustive study of the subject and by no means is this presentation the definitive work.

The Wilson Report [1] was published in 1963. It included a proposed assessment procedure for intermittent noise.

> "In some cases the noise from the process is not constant, but significantly louder noises occur at intervals, say for less than half the time. When these louder noises occur during the day the following allowances may also be made to determine an intermittent limiting level:
>
> | | |
> |---|---|
> | Noise occurring approx. 15 mins. per hour | add 5dB(A) |
> | Noise occurring approx. 5 mins. per hour | add 10dB(A) |
> | Noise occurring approx. 1 min. per hour | add 15dB(A) |
> | Noise occurring approx. 1 min. per half day | add 20dB(A)" |

Thus the report addressed the intermittency case but not the fluctuation case. Note that the corrections are made to determine the limiting level. Some might regard as liberal the allowance of 20 dB(A) for a short exposure. For comparison, and anticipating later development, the corrections based on equal energy for the times quoted are respectively 6, 11, 18 and 24 dB(A).

BRITISH STANDARD 4142:1967

Was this the first Standard? The question is posed for the same reason as before.

British Standard 4142, "Method of rating industrial noise affecting mixed residential and industrial areas", was issued (possibly for the first time) in 1967. It was

> "intended to meet the need for rating various noises of industrial origin with respect to their effects on persons living in the vicinity."

The Standard specified

> "a method of measuring, at the outside of a building, the level of noise being emitted from industrial premises; the application of corrections according to the character and duration of the noise; and comparison of the corrected noise level with a criterion which takes account of various environmental factors, or with the background level, to determine whether the noise is likely to give rise to complaints."

Intermittency and duration were considered. For noise that is not continuous, the typical on-time duration and the sum of the on-times expressed as a percentage of the total time considered enabled a correction to be determined according to Figure 1 for day or evening periods or Figure 2 for night-time. The corrections are applied to the measured level, i.e., they are negative, rather than the positive corrections of the Wilson

Report applied to the limiting level. Again, anticipating, the corrections of Figures 1 and 2 are almost all less than the corrections based on equal energy, and are therefore less favourable to the noise-maker.

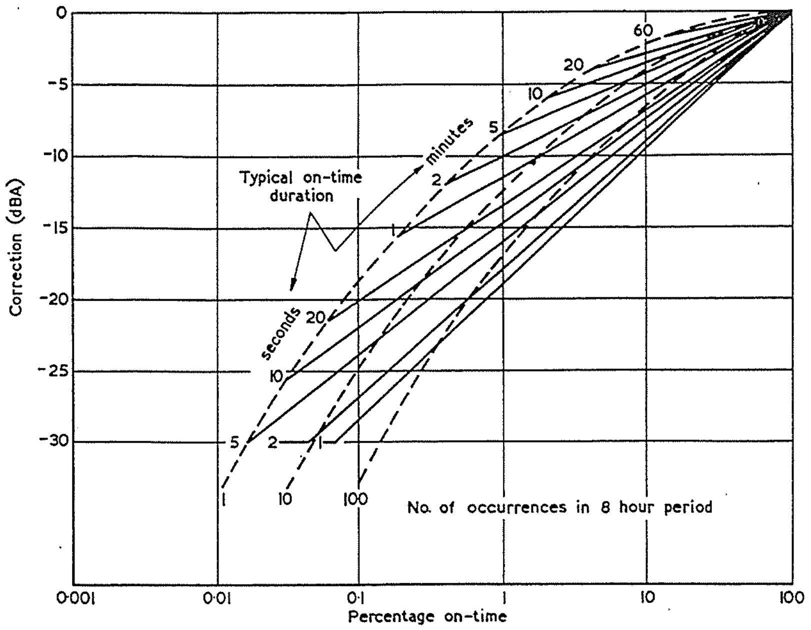

Figure 1. Intermittency and duration correction for other than night-time

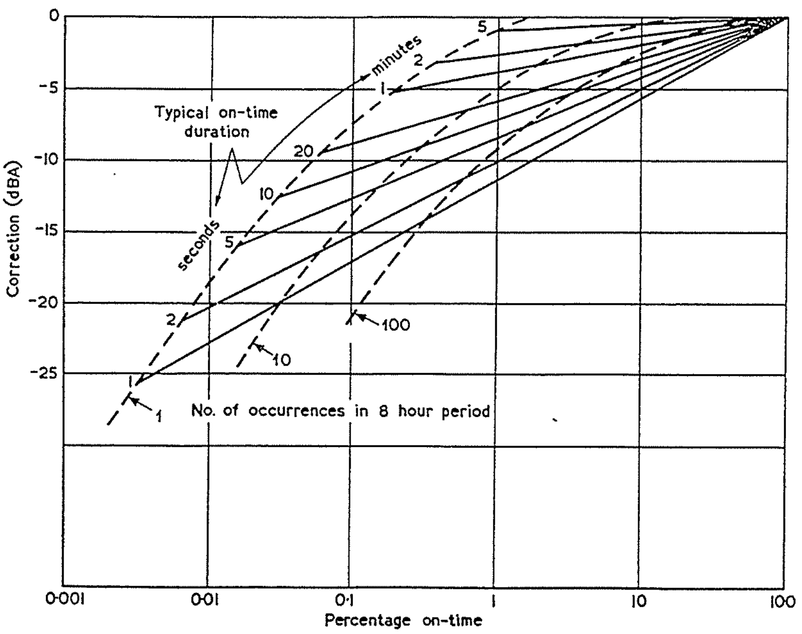

Figure 2. Intermittency and duration correction for night-time

INTERNATIONAL STANDARD R1996:1971

Was this the first appearance of L_{eq}? ISO Recommendation R1996 first appeared in 1971. It included separate provisions for duration (intermittency) and fluctuation. The corrections to be applied for duration are shown in Table 1. The percentage on-times are chosen to "split" the correction steps, that is, on an equal energy basis, 56% corresponds to a correction of 2.5 dB, 18% corresponds to 7.5 dB and so on. It can be noted that this specification allows no correction for noise that is on for about half the time. In other words, there is just as much annoyance from a sound which comes on and off as for one which is on all the time. Some would argue that the on-off situation is more annoying than continuously on, especially if the onset is irregular and without warning (e.g., cycling of refrigeration plant).

TABLE 1. DURATION CORRECTIONS ISO R1996

Duration of noise as a percentage of relevant time period	Correction dB(A)
Between:	
100 and 56	0
56 and 18	-5
18 and 6	-10
6 and 1.8	-15
1.8 and 0.6	-20
0.6 and 0.2	-25
Less than 0.2	-30

(+30dB for 10sec/1 hour = +1dB)

ISO R1996 also stated that if the noise varied with time in a more complicated manner than is appropriate for the use of the duration correction table, the equivalent sound level L_{Aeq} should be obtained. The L_{eq} is based on energy and therefore has the advantage of giving emphasis to high levels, making it effective as acriterion. However, it has (or had) two limitations as a criterion. The first is that it allows high levels of noise for very short periods. This difficulty is avoided by setting an absolute permissible level (or maximum limit to the correction) and by choosing a suitable (not too long) relevant time period. The second limitation is that it requires appropriate measuring instrumentation and procedures, of a statistical or integrating kind. In 1971, many L_{eq} results were obtained by non-integrating sound level meters, level recorders, and calculations using charts and tables. Now, the integrating sound level meter is commonplace. Another question - what was the link between the rise of L_{eq} and the development of the integrating meter?

The Victorian Environmental Protection Agency in 1974 in a draft policy [4] adopted duration corrections based on ISO. The relevant time period was set at one hour, and correction limits were set for noises of short duration at 30 dB for day and 15 dB for night.

The corrections discussed so far are shown on Figure 3.

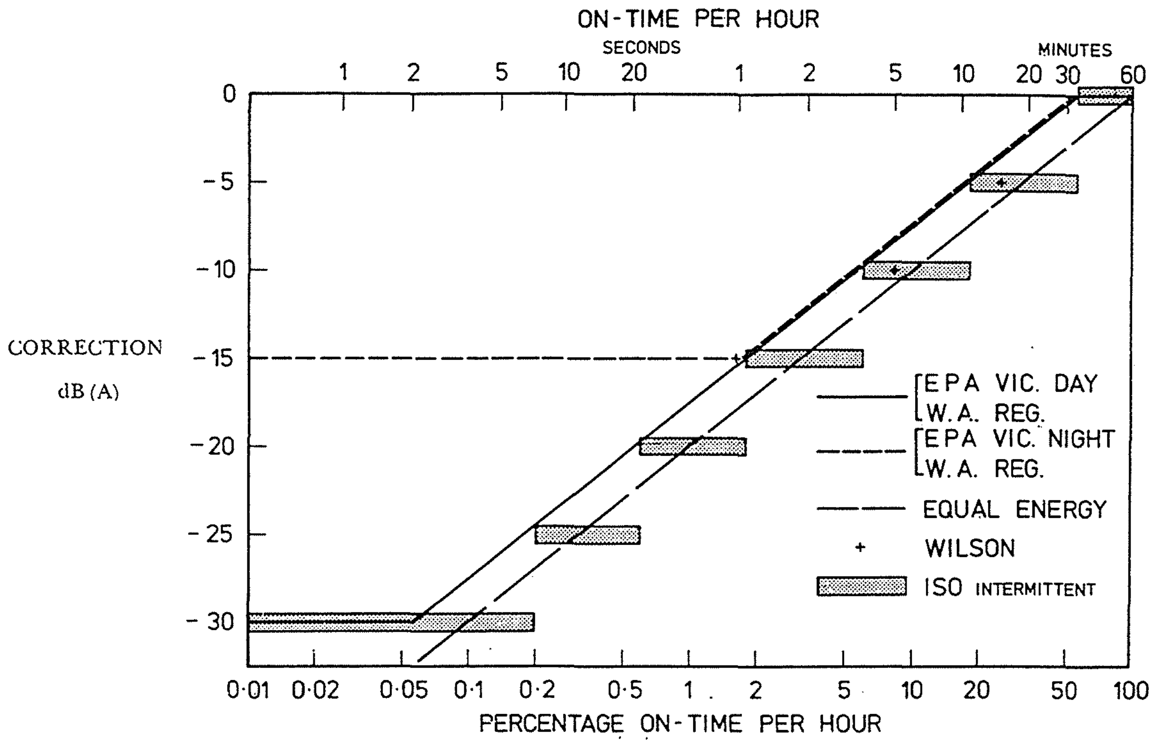

Figure 3. Duration allowances

AUSTRALIAN STANDARD 1055:1973 - 1997

AS1055 was first issued in 1973 and followed very closely BS4142. The same charts were used (Figures 1 and 2) and a procedure for assessment specified, which gave a statement of the likelihood and severity of complaints. Incidentally, the writer is not aware of the origin of the correction charts then in use. Over several revisions AS1055 has changed its handling of fluctuating noise. The current situation (1989) is use of L_{Amax} and L_{Abg}. It is noted that L_{A10} is commonly taken as an approximation of L_{Amax} and L_{A90} for L_{Abg}. Assessment is made in terms of the difference between L_{Amax} (with corrections for tonality and impulsiveness if appropriate) and L_{Abg} (or a reference level).

AS1055 is under review and a revised version may be issued soon (in 1997?). The revision shifts the emphasis from L_{Aeq} to $L_{A\%}$.

> "For fluctuating, impulsive or other non-steady sound the measurements shall be .... presented in a statistical format."

Assessment tables are expressed in terms of comparisons between $L_{A\%}$ (adjusted if appropriate) and L_{A90bg}. The intrusive noise may be expressed at the 0.1, 1 or 10 percent exceedance levels, and various corresponding permissible margins and numbers of occurrences are specified.

Two interesting points arise from this revision. The first is the shift from L_{Amax} to $L_{A\%}$, i.e., a move to a precisely defined quantity. The second is a hint at consideration of a non-steady background noise. There is a statement that

> "The determination of relevant noise emission and extent of noise exceedance for the steady sound source ..... require the following parameters to be determined and compared:
>
> (i) The difference between $L_{A10,T}$ (noise source operating), and $L_{A10,T}$ (noise source inoperative).
>
> (ii) The difference between $L_{A90,T}$ (noise source operating), and $L_{A90,T}$ (noise source inoperative)."

The item (i) appears not to be addressed further, nor is there any similar paragraph for non-steady sound. The question of fluctuating background noise will be addressed in the Comment section below.

ISO 1996:1982-1987

At this stage it is either a comfort or a worry to quote from the ISO introduction.

> "Extensive research concerning the way in which human beings are affected by noise from a single kind of source such as rail or road vehicles, aircraft or industrial plants, has led to a variety of measures for assessment of different kinds of noise, many of which are in common use. Conversion from one measure to another is often beset with serious uncertainty."
>
> "The methods and procedures described in this International Standard are intended to be applicable to sounds from all sources, individually and in combination, which contribute to the total noise at a site. At the present stage of technology this requirement seems to be best met by adopting the equivalent continuous A-weighted sound pressure level as a basic quantity. Results shall always be expressed in terms of this quantity even if supplemented by corrections or other descriptors that, in certain cases, may be deemed appropriate."

In common with other Standards, ISO has moved away from prescription to techniques of measurement and presentation.

> "The aim of the ISO1996 series is to provide authorities with material for the description of noise in community environments."
>
> "This part ..... lays down guidelines for the specification of noise limits and describes methods for the acquisition of data that enable specific noise situations to be checked for compliance with specified noise limits."
>
> "This part ..... does not specify noise limits."
>
> "It is assumed that noise limits are established by local authorities."

Another question now arises. Is AS moving back to statements of criteria?

ISO clearly specifies L_{Aeq} and also requires, when assessing compliance limits, that a number of measurements be made and consideration given to their average and statistical distribution.

AUSTRALIAN POLICIES AND REGULATIONS

All Australian States and the Australian Capital Territory have Environmental Protection Policies for noise and appropriate regulations. There is general commonality in principle. If fluctuations are less than a specified range (typically ± 3 dB), the noise is classified as steady. Premises are typically classified as noise-sensitive, commercial or industrial. Day, evening and night periods are specified. Margins above L_{Abg} are specified (typically 5 dB, but 10 dB for daytime, Victoria, 1989), and base levels are specified. In two states, zoning is applied. Most if not all States have noise limits for specified items or activities, e.g., lawnmowers, shooting ranges.

There are also many differences from State to State. In this section some of the approaches are mentioned, but the information given is based on a limited availability of documents, and might or might not be up-to-date and a correct interpretation. The point to be made, however, is the variability of approach.

In Queensland, the sound levels for non-steady sound are to be measured as the adjusted average maximum sound level, i.e., following AS1055:1989. The general criterion is that the sound is reasonable if the level does not exceed specified values, depending on time and place. For steady sound, the background level (noise source operating) is compared with the background level (noise source not operating). One of the criteria for excessive noise is based on L_{A1}.

In Victoria, the measure is L_{Aeq} over a period of at least 30 minutes, and the background is the average of L_{A90} levels for each hour in the relevant period. There are adjustments for duration and intermittency. The duration adjustment is on the equal energy basis, intermittency on number of events involving rapid increases and level change.

South Australia appears to allow measures of L_{Amax} or L_{Aeq}, and also uses an $L_{Aeq,adj}$. In Western Australia there is included circumstances in which L_{A5} is used. There are yet more sets of adjustments for intermittent noise (Table 2) and duration.

TABLE 2. INTERMITTENCY ADJUSTMENTS

Cumulative period for which noise measured is present in any hour	Adjustment dB(A)
More than 15 minutes	$\pm$ 0
Exceeding 5 minutes but not exceeding 15 minutes	- 5
Exceeding 1 minute but not exceeding 5 minutes	- 10
Not exceeding 1 minute	- 15

A special case relating to fluctuating noise arises in two States and concerns music noise from public premises. The Queensland policy, which follows a Victorian Order of 1989, is as follows:

"The quantitative criteria for entertainment noise from an indoor venue is that the noise is reasonable -

(a) if, at any time during the day or evening, the adjusted average maximum sound level measured during the use of the music system does not exceed the background level by more than 5dB(A) measured outside the most exposed boundary of any affected dwelling; or

(b) if, at night, the noise level from the music when measured as the average maximum sound level, L_{OCT10}, in any full octave band with centre frequencies from 31.5Hz to 4kHz, does not exceed by more than 8dB the background level, L_{OCT90}, in one or more octave bands when measured outside the most exposed boundary of any affected dwelling.

The two cases mentioned are the only ones known to the writer in which an environmental noise policy specifies octave band measures. Similarly the origin of the specification is not known.. However, it does give rise to further discussion and will be taken up in the next section.

COMMENT

There are a great many measures used for the description of noise, and many take account of fluctuation - L_{eq}, $L_{\%}$, noise pollution level, community noise exposure level, traffic noise index. Many are suitable as descriptors of an environment, but less suitable for control by incorporation into regulations. The two most satisfactory for regulatory purposes are L_{eq} and $L_{\%}$, and almost all specifications use one or both.

Nevertheless, both have shortcomings. L_{eq} allows high levels for short durations and must be supplemented by overall limits. $L_{\%}$ requires multiple specification to be fully effective. For example, if a noise level must not be exceeded for 10% of the time, then it can be exceeded by a large margin for 9% of the time. Clearly an unsatisfactory situation which would be prevented by an L_{eq} specification. Thus, even with these measures, uncertainty remains.

Consideration of the problems of fluctuating noise brings out a further serious complication. In almost all discussions it is implicitly assumed that the background noise is reasonably steady. This is not necessarily so. Take the rules relating to music noise from public premises. Often this will occur in a residential area where the major background noise is a moderate flow of traffic. The background noise is itself fluctuating. Consider the following argument, in which additional subscripts m for music and bg for background are applied. For the evening case, L_{A10m} is compared with L_{A90bg}, and the difference must be not greater than 5 dB. But L_{A10bg} may be more than 5 dB greater than L_{A90bg}. It happens - data of this nature is available.

Similarly for the night requirement. $L_{OCT10bg}$ may be more than 8 dB greater than $L_{OCT90bg}$. Subtracting L_{A90bg} (for one type of noise) from L_{A10m} (another type of noise) is somewhat akin to subtracting apples from oranges.

In fact, this special case applies equally throughout the whole concept of comparing L_{Amax} or L_{A10} or L_{Aeq} from a source with L_{Abg} or L_{A90} for a background. It is all very well when the background is steady, but if there is significant fluctuation in the background, the background itself might exceed the margins set, and there is a strong case for the view that such criteria are inappropriate.

This is a major matter of principle which has widespread ramifications.

CONCLUSION

The ISO words "beset with serious uncertainty" are well-chosen in the context of handling the problem of regulatory control of fluctuating noise. The title of this paper is intended to convey the theme that L_{10} and L_{90} (or L_1 and L_{99}) are ends which do not meet, but the task of the regulation writer is to somehow bring them together in a meaningful specification.

The further theme of this conference is Innovation and Legislation. An address on fluctuating noise is appropriate - there is a need for Innovation in the Legislation. May I hope that this address will promote debate and progress.

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APPLICATIONS OF ACTIVE NOISE ATTENUATION TECHNIQUE TO ENGINE EXHAUST NOISE CONTROL - A REVIEW

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ABSTRACT

This paper introduces the basic principles of active noise control technique applied to engine exhaust noise control system. The authors reviewed the work done by others over the last decade in this challenging research area. The active noise control technique can be effectively applied to attenuate low frequency duct noise, car and aircraft cabin noise and structure vibration. However, many problems still remain in the field of active control of engine exhaust noise. This paper discusses the achievements and problems in this area together with the current progress of the authors.

INTRODUCTION

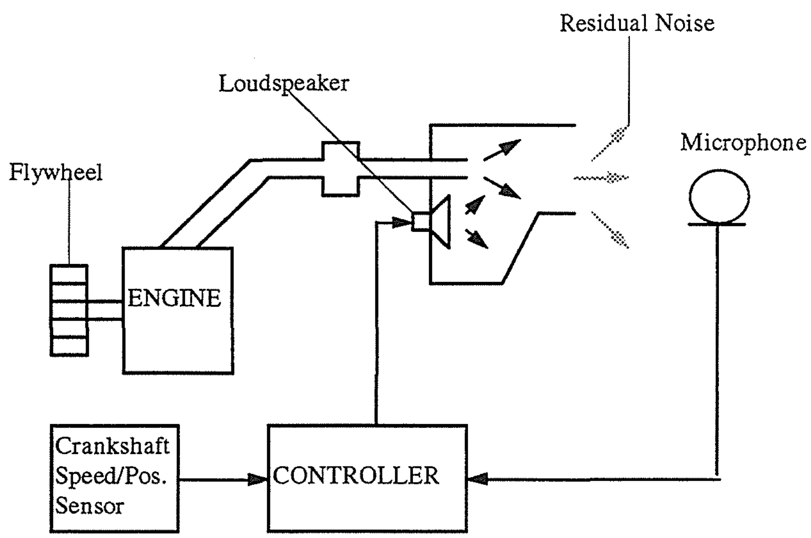
The traditional method of attenuating the engine exhaust noise is by using passive mufflers. Passive mufflers have three types: absorptive, dispersive and reactive. Low restriction passive elements including expansion chambers, pipe resonators, volume resonators or lined ducts are used in these conventional mufflers (Denenberg, 1992). They are very effective in reducing high frequency engine exhaust noise. However, they are expensive and have little effect on the low frequency tonal noise where a significant portion of engine exhaust noise is concentrated, unless the mufflers are made impractically large and heavy. Active noise control technique, well developed in the last two decades, is particularly useful at low frequency noise control. Considerable work has been carried out to apply the active noise control technique to reduce duct noise (Shepherd et al., 1984), car and aircraft cabin noise (Elliott and Nelson, 1990), fan noise (Eriksson et al., 1988), structural vibration (Fuller et al., 1995) and engine exhaust noise attenuation (Arnold et al., 1991).

PRINCIPLES OF ACTIVE NOISE CONTROL

The active noise control technique is not a new idea. Lueg filed for a US Patent early in 1934 and was granted a US Patent in 1936 (Lueg, 1936). In his Patent, Lueg attempted to engineer the principle of wave superposition so that the destructive interference of sound waves could be used to eliminate noise (Warnaka, 1982). Lueg's pioneering patent introduced the concept of active attenuation of sound by using an artificial acoustic sound mixed with the unwanted sound. If the second sound is the same power and intensity as the first sound but 180° out of phase, the first sound will be cancelled.

To turn this laboratory curiosity into a useful noise control technique required an understanding of both the acoustics and electrical control technology appropriate to the particular problem under consideration, namely: the time delay and the phase correction characteristics. Unfortunately, the electronic technology was not sufficiently advanced to meet the control requirements of the active noise control system until the mid-1970's, when a huge step was taken to use adaptive filters to generate the second sound. Another breakthrough in the mid-seventies by Fuller and von Floton was the recognition that many noise sources, particularly those produced by artificial machines exhibit periodic or tonal noise. This tonal noise allowed for a more effective solution, since each repetition was similar to the last, and the predicability enabled the production of an accurate anti-noise signal (Fuller and Von Flotow, 1995).

Today, rapid growth in digital computer technology and improvement in control transducers has propelled great developments in active noise control. Two kinds of active noise control approaches are currently being used: digital feedforward and synchronous feedback. The former approach is to obtain a prior knowledge of the noise by using an upstream microphone. This method is mainly used in controlling low frequency noise in ducts, propeller noise inside aircraft and low frequency engine noise inside cars. In the latter method, a tachometer signal is used to provide information on the rate of the noise. It is mostly used in the attenuation of low frequency engine exhaust noise as shown in Figure 1.



Figurer 1: Engine Exhaust Noise Active Control System General Arrangement

ENGINE EXHAUST NOISE ATTENUATION

Engine exhaust noise control has been and continues to be a challenging research area. Traditionally, the engine exhaust noise is attenuated by passive mufflers. However, they are ineffective to low frequency exhaust noise where a significant portion of engine exhaust noise is concentrated. The primary engine exhaust noise is a mixture of low frequency, harmonically-related tones. As engine exhaust noise has a large energy component below 500 Hz it is an ideal candidate for active noise control. However, the active noise control technique has not been applied to attenuate the engine exhaust noise until the last decade because of the high pressure and high temperature of the exhaust gas. In recent years, some companies and institutions have been undertaking research on the engine exhaust noise active control, such as: CSX Transportation Inc., USA; NCT (Noise Cancellation Technologies, Inc. USA); Detroit Diesel Corp., USA; University of Michigan, USA; and PMA (Production Engineering, Machine Design and Automation, Katholieke Universiteit Leuven, Belgium). According to published papers listed in 'References', two types of engine exhaust noise active control systems are being tried to replace the traditional mufflers in the laboratory, namely: the actively tuned passive system; and engine exhaust noise active control system (EENACS).

ENGINE EXHAUST NOISE ACTIVE CONTROL SYSTEM

Of all applications of active noise control, the electronic active muffler is eagerly sought and an active area of research. A literature review into active control of engine exhaust noise has concluded that only two methods have been employed, namely: microphone feedforward and synchronous feedback.

The feedforward method uses a microphone to detect the primary noise carried in the exhaust gas. The microphone signal is processed by a digital control device which produces a mirror image of the noise waveform. The secondary sound emitted by the loudspeakers, causes destructive interference as the primary noise emerges from the exhaust stack. According to Ffowcs Williams (Eghtesadi et al.), the performance of this system is about 10 dB reduction at the gas turbine's rumbling peak across the lowest octave.

Feedback is the most popular method used in EENACS. Generally the system has four main parts: an active muffler, which is a loudspeaker enclosure fitted around the tailpipe; a digital-signal-based control to process the noise signal; an error microphone to sample residual noise; and a passive muffler designed to cancel the high frequency noise. In the feedforward method, the noise was detected by a microphone installed at the upstream of the loudspeaker, while in the feedback system, the microphone was installed downstream to detect the noise error signals. The feedback control system is depicted in Figure 1.

Early in 1984, Japanese researchers Kosaka et al. carried out a simulation study on the active noise attenuation of low frequency noise of engine exhaust (Kosaka et al., 1984). In this work, they simulated the active control system for reducing engine exhaust noise. A wave generator with two inputs was used to generate the signals driving the loudspeaker. One of the inputs was the waveform from the microphone, while the other was the synchronous signal obtained from sound wave generator. These two analogue signals were converted to digital signals by an A/D converter and then fed into a micro-computer Z-80A. After processing by the micro-computer they were passed through a D/A converter into an amplifier and into a loudspeaker to produce a sound wave for attenuation of the noise which was artificially generated for simulating the engine exhaust noise. The system reported could reduce the sound up to 15 dB within one second after starting operation, finally achieving over 20 dB of attenuation.

From 1989 to 1990, Arnold et al. (Arnold et al., 1991) carried out a research program to demonstrate an active muffler system on a heavy duty diesel engine. A Detroit Diesel Corporation 6V-92TA industrial diesel engine used to power a generator set was chosen for their study. This system reported 5 dB to 13 dB (idling condition) reduction over a conventional baffled passive muffler and the system has a significant reduction in engine fuel consumption due to a lower exhaust flow restriction. The arrangement of the system is similar to that shown in Figure 1.

ACTIVE TUNED PASSIVE MUFFLER

This is a very interesting idea based on the principles of active control technique and the characteristics of the internal combustion engine. EENACS has some limitations and disadvantages, such as the need for sensors (microphones), complex electronic hardware and efficient low frequency sound sources. The actively tuned passive muffler (Lamancusa, 1987) is a simple alternative to active cancellation, which combines some of the desirable attributes of a passive muffler and eliminates the need for cancelling sources. In practicality, this system would actively tune the silencer so that its maximum attenuation always coincided with a pure tone component of the noise regardless of engine speed. The schematic view of this system is shown in Figure 2. According to Lamancusa's experimental results, this tuned muffler can provide at least 30 dB of transmission lost of a single pure tone at the resonant frequency of the muffler. The range of volumes tested is sufficient to cover a range of frequencies of 100 to 170 Hz. Lower frequencies can be obtained by further increasing the volume. Insertion loss of 10 dB or more was obtained over a range of approximately 100 Hz for all volumes tested.

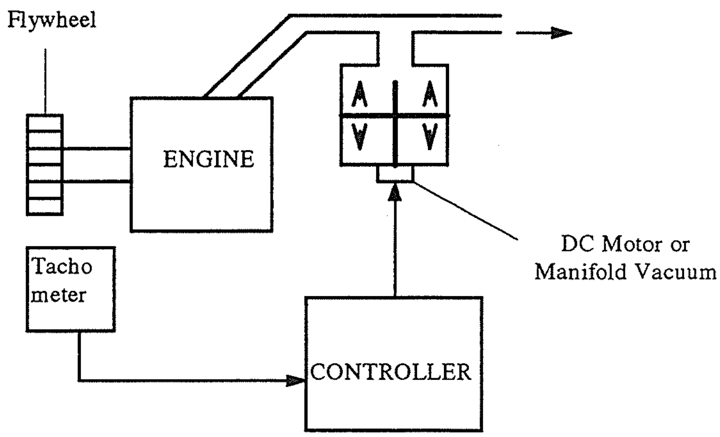


Figure 2: Actively Tuned Passive Muffler

DISCUSSION AND FURTHER STUDIES

As can be seen from the above, the noise active control technique has the potential to attenuate engine exhaust noise, reduce engine exhaust back pressure, improve engine power output and fuel efficiency. In addition, the active noise control system has a smaller packaging. The disadvantages of this system are that the electrical control system is complicated and produces high level low frequency sound within a small enclosure, which is a challenge. The actively tuned passive muffler does not need complex electronic hardware. However, to cancel the lower frequency (below 100 Hz) noise, a resonator of impractically large volume may be required.

Although many companies and institutions are undertaking research on the engine exhaust noise control, there are still some remaining problems. First of all is the high temperature working environment. In the research of DDC (Arnold et al., 1991), the actuators (loudspeakers) were put in the exhaust pipe which was 7.6 meters. At the exit end of this pipe, the exhaust temperature was much lower than the real engine exhaust temperature such as that of cars and buses. Therefore, the actuators (loudspeakers) used in this research may have some difficulties in practical engine exhaust noise active control applications, where the exhaust pipes are much shorter. According to O'Connor, the best active mufflers operating temperature is approximately -40 to 150°C at present (O'Connor, 1994).

The second complicated factor in the active control system is that the exhaust noise sound pressure level and its power spectrum are changing with the engine speed and load. According to the published papers, most of the work on active engine exhaust control systems was similar to that of DDC. The engines used were operated in a stationary condition (used as an auxiliary electrical power generator). The exhaust noise generated by this kind of engines is easier to control than that generated by the engines at transient operating condition, such as those in automobiles, trucks, construction equipment, as well as garden machines.

Currently, the authors are undertaking a research program on the active control of engine exhaust noise. A single, 4-stroke, gasoline, spark ignition engine has been chosen. This engine can be used to power air compressors, pumps, pressure washers, construction equipment, garden vacs and turf equipment. The research program aims to develop the understanding of the control mechanisms and to address problems limiting the success of active control of engine exhaust noise. The initial work of the program was reported in the accompanying paper (Hong et al., 1996).

CONCLUSIONS

Active noise control technique can be used effectively in controlling low frequency duct noise, car and aircraft cabin noise and structure vibration and noise. The application of active noise control technique to engine exhaust noise control has the potential to attenuate engine exhaust noise, reduce fuel consumption and reduce packaging. However, to develop effective, reliable and economical engine exhaust noise control system or actuators used in attenuating the noise generated by variable engine speed and load exhaust remains a challenge to the application of active noise control technique, namely: the high temperature working environment, variable spectrum and sound pressure level of engine exhaust noise and variable engine combustion conditions.

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DYNAMIC DESIGN OF A VIBRATION ISOLATOR TEST FACILITY

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ABSTRACT

Vibration isolators are an important element in the reduction of the structure borne noise transmission. In determining the frequency dependant dynamic properties of the isolators it is important to have a set of descriptors that is independent of the testing arrangement, and the four-pole parameters provide one such set. It is also important to be able to measure the properties with the isolator pre-loaded to the levels experienced in normal operation, as the properties are both frequency and pre-load dependant. A test facility has been designed to measure the four-pole parameters of vibration isolators with pre-loads of up to 30 kN and over the frequency range from 10 to 2000 Hz. In undertaking the experimental measurements the force and velocity above and below the isolator need to be determined, and it is vital that the dynamics of the testing machine structure do not affect the results. This paper describes the modelling of the modal behaviour of the elements of the test rig and the use of harmonic response analysis modelling to ensure there is no effect on the experimental measurements from the structural modes of the test rig. The results show that while the structural framework and other elements of the test rig have natural modes within the frequency range of interest, careful design can ensure that these do not affect the results.

1. FOUR-POLE PARAMETERS

In the design, development and selection of vibration isolators it is important to have a description of the isolators' behaviour that is independent of the testing arrangement and, for their actual use, the vibration source and the installation foundation system. The four-pole parameters provide one such description (Molloy [5], Snowdon [6], Snowdon [7] and Verheij [8]) and relate the force F_1 and velocity V_1 at the isolator input to the force F_2 and velocity V_2 at the isolator output:

$$\begin{bmatrix} F_1 \\ V_1 \end{bmatrix} = \begin{bmatrix} A & B \\ C & D \end{bmatrix} \begin{bmatrix} F_2 \\ V_2 \end{bmatrix} \tag{1}$$

where A, B, C, and D are the four-pole parameters, and are complex, time invariant functions of ω.

Application of Maxwell's law of reciprocal deflections to the isolator leads to the relationship:

$$AD - BC = 1 \tag{2}$$

A symmetric isolator is one that behaves the same if the input and output ports are interchanged. For this case the additional relation is applicable:

$$A = D \tag{3}$$

From equations (2) and (3) it is evident that for a symmetric isolator, only two independent four-pole parameters need to be measured in order to completely characterise it. At lower frequencies an isolator may be assumed to be a massless spring of dynamic stiffness k. This assumption yields $A = D = 1$, $B = 0$ and $C = j\omega/k$, where $j = \sqrt{-1}$ and ω is the circular frequency.

From equation (1) two particular cases can be derived. Case 1 is for the output free, $F_2 = 0$, which yields $F_1 = BV_2$ and $V_1 = DV_2$. Case 2 is for the output blocked, $V_2 = 0$, which yields $F_1 = AF_2$ and $V_1 = CF_2$. While the first case is experimentally convenient it does not allow the determination of the isolator properties under pre-load, and therefore the properties measured in this way will not be representative of those for the installed isolator.

Verheij [8] developed a method for determining the blocked transfer function $F_2/j\omega V_1$. Dickens and Norwood [4] developed a system that followed Verheij's basic method but determined the four-pole parameters. Improvements to the measurement technique correcting for the small but finite velocity of the blocking mass, and measuring the input force directly were suggested by Dickens and Norwood, [2, 3]. It was decided to implement these measurement improvements in an upgraded test rig. Additional improvements would also include increasing the upper frequency limit to 2 kHz and the dynamic force capacity to 5.3 kN.

2. TEST RIG

The developed test rig is shown schematically in Figure 1. The isolator under test is mounted between two large masses, static pre-load is applied by air-bags top and bottom and the dynamic load by an electro-dynamic shaker. The rig has two supporting frames, the upper frame that supports the shaker and the lower frame that provides the reaction forces for the upper pre-loading air-bags. The lower pre-loading air-bags sit on a base plate mounted on top of a seismic mass.

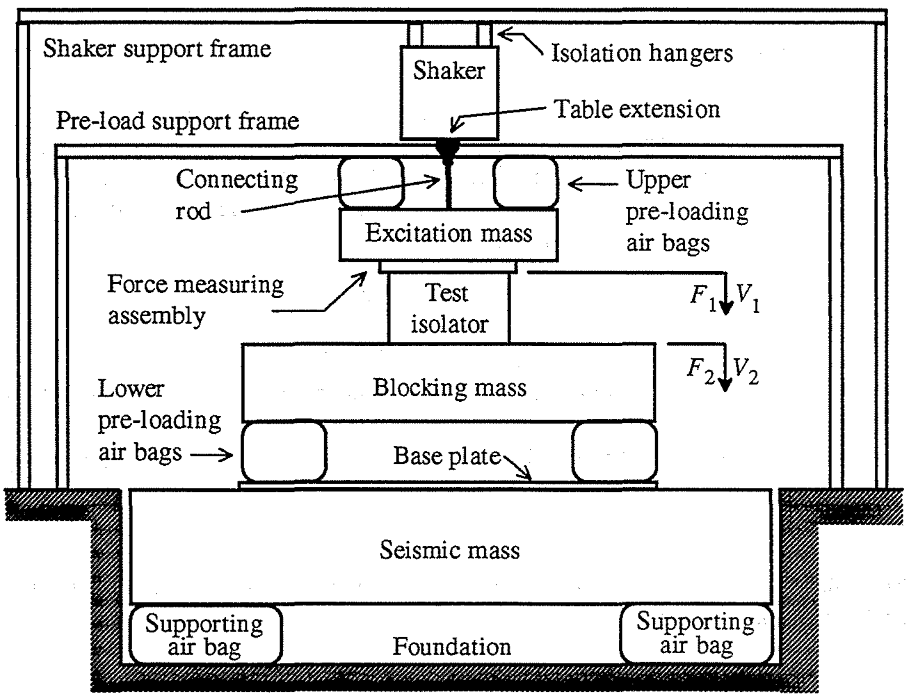

Figure 1
Vibration Isolator Test Rig

Two frames are used to reduce coupling between the pre-loading structure and the shaker. The shaker is decoupled from its supporting frame by four isolation hangers and drives the excitation mass through a single centrally located connecting rod. The seismic mass is a block of reinforced concrete of dimensions 3m x 3m x 1m, supported on four air-bags connected to air reservoirs and having a mounted natural frequency of approximately 1.2 Hz. The use of the seismic mass decouples the blocking mass from the laboratory floor, reducing the input of extraneous forces and transmissions from the two supporting frames.

Air-bags are used to provide the pre-load as the static force can be easily adjusted, while at the same time giving a degree of isolation between the masses and the supporting structure. The dynamic force between the excitation mass and the isolator is measured directly by an assembly of force transducers. The dynamic force between the isolator and the blocking mass is inferred from the acceleration of the blocking mass. The motions of the excitation and blocking masses are measured directly using accelerometers.

The rig was required to be able to test isolators with dynamic stifffnesses in the range from 1×10^5 to 2×10^7 N/m, with pre-loads adjustable over the range from 1.5 to 30 kN and over a frequency range from 10 to 2000 Hz. In designing the test rig it was important that the dynamic response of the test rig did not affect the measurement of the isolator's four-pole parameters. This implies that where possible the components of the test rig should not have structural modes within or near the frequency band of interest, and where this is not possible the structural response of the rig should not affect the results.

3. DYNAMIC ANALYSIS OF THE TEST RIG

The analysis can be divided into three separate areas. Firstly modal analyses were conducted to determine the natural frequencies and mode shapes of the test rig components and sub-structures. Where possible components were designed not to have structural modes within or near the test frequency band; these included the excitation mass, the blocking mass, the table extension for the shaker, the force measuring assemblies and the base plate for the seismic mass.

For components that could not be designed to satisfy the modal frequency criterion, an harmonic response analysis was performed. Each analysis was chosen to determine the affect of the component modal behaviour on the measurements. Items in this group included the supporting frames and the seismic mass. The latter was set in the laboratory foundation and not able to be altered.

Finally a harmonic response analysis for the entire assembly was performed and compared with the response for an idealised spring/mass system. This was done to ascertain if there were any effects from the modal behaviour of the individual rig components on the assembly as a whole.

3.1 Modal Analyses

In this series of analyses the excitation mass, blocking mass, shaker table extension and air-bag base plate were modelled to determine their natural frequencies and mode shapes to ensure there were no modes in the frequency range of interest. Each item was modelled using eight-noded brick elements, and where necessary, sections in contact with the element under study were included in the model to ensure appropriate boundary conditions. A subspace iteration procedure was used to solve for the modal frequencies and shapes. The modal analysis for each component is detailed below.

(a) Blocking mass: The model included the supporting air-bags and the isolator. Two extreme case were considered, the stiffest isolator under the maximum pre-load and the softest isolator under the minimum pre-load. The air-bags and the isolator were modelled as springs spread over the contact areas. The optimum design selected for the blocking mass was a steel cylinder of diameter 480 mm and height 398 mm, which gave a first modal frequency of 4.00 kHz for the (1,0) mode. The (1,0) mode has one nodal diameter and zero nodal circles. To apply the two mass method described by Dickens and Norwood [2] for testing non-symmetric isolators, two blocking masses of different masses are required. A second blocking mass made from an aluminium alloy was therefore analysed to ensure its modal properties were adequate. It had the same height but a smaller diameter compared to the first mass, and slightly higher predicted modal frequencies.

(b) Excitation mass: This model included the pre-loading air-bags and the isolator. As for the blocking mass the air-bags and isolator were modelled as spring elements, and the two extreme cases described above were analysed. Similarly to the blocking mass, the optimum shape selected was a cylinder; in this case it had a diameter of 360 mm and a height of 355 mm, with the first natural frequency of 4.58 kHz. The first mode was torsional, and the second mode was the (1,0) mode at 5.66 kHz. To accommodate low pre-loading forces down to 1.5 kN, a second lighter excitation mass was required. This second mass had the same dimensions as the first but was made from an aluminium alloy, and gave marginally higher predicted modal frequencies.

(c) Table extension: In order to limit the length of the stinger required it was necessary to provide an extension to the shaker table. This would be bolted to the shaker table in six locations and was required to have a minimum height of 175 mm. The extension was made of aluminium to minimise its mass. The optimum design selected was a base cylinder of diameter 125 mm and height 23 mm leading into a 117 mm high conical frustum and tapering to a 20 mm diameter cylinder at the top. This design had an overall height of 175 mm and a first mode at 4.26 kHz.

(d) Base plate: A base plate attached to the seismic mass was to be used to locate the six lower pre-loading air-bags under the blocking mass. The diameter of the plate was set at 480 mm diameter, which was the minimum required to support the air-bags. The plate was initially modelled as being attached to the seismic mass by 15 bolts on three concentric circles. The optimum solution for this arrangement was a steel plate with thickness of 45 mm giving a first mode at 3.57 kHz. One of the problems to be resolved with this model was that the mode shape would be rectified. The plate could deflect away from the mass only. Unfortunately this one-way motion was not able to be modelled, so the mode shapes were calculated as for normal bi-directional deflection. It was reasoned that the uni-directional motion constraint would increase the modal frequency so the results calculated would be conservative.

A second model using adhesive to attach the plate to the seismic mass was made. This consisted of a plate 40 mm thick and three locating bolts, with a continuous layer of epoxy adhesive between the plate and the mass. The adhesive layer was modelled as a continuous compliant layer and the first structural frequency was 11.86 kHz. This plate and attachment method was selected in the design of the test rig.

3.2 Harmonic Response Analyses

In this series of analyses the two support frames and the seismic mass were modelled using harmonic response analyses to ensure that the modes present within the frequency range did not affect the results. The two frames were modelled using three-dimensional beam elements, and the seismic mass used eight-noded brick elements. Initially a reduced modal analysis was performed using Guyan reduction to reduce the total matrix size. Then an harmonic response analysis was performed by modal superposition, with appropriate forcing functions and damping applied. The forcing function modelled the force input from the shaker, and the harmonic response solutions were clustered at the modes to give better resolution of the expected peak responses. Each analysis was carried out for several different damping cases to assess the importance of the damping to the peak responses.

(a) Shaker support frame: The model of the frame included the columns, cross beams and braces for the frame itself and the isolation hangers on which the shaker was supported. The shaker was modelled as a lumped mass and the input force was applied to it. The frame had a large number of modes in the frequency range up to 2 kHz. The response analysis was used to determine the level of force transmitted to the floor by the frame, and hence the amount of feedback to the air-bags supporting the blocking mass through the seismic mass. The columns were considered to be fixed at their bases and the reactions forces calculated. The transfer function between the mounting points for the columns on the floor and the top of the seismic block was measured using an instrumented hammer and an accelerometer. This transfer function was then combined with the predicted forces to give displacement levels on the top of the seismic block.

The displacements predicted by this method were at least 120 dB less than those predicted on the seismic block via the direct path through the isolator and the blocking mass over the frequency range of interest. Therefore it was concluded that feedback via the shaker supporting frame was not a problem.

(b) Pre-load support frame: The pre-load support frame consisted of a pair of portal frames cross connected by an "H" structure which carried the pre-loading air-bags. The model included the upper and lower pre-loading air-bags, the excitation mass, isolator, blocking mass, seismic mass and supporting air-bags. The air-bags and isolator were modelled as spring elements and the masses were modelled as rigid mass elements. The system was excited by applying an harmonic force to the excitation mass, as the shaker would do. The analysis was intended to predict the errors caused by the feedback of the reaction forces at the base of the supporting frame columns via the laboratory floor and the seismic mass' base.

The transfer function between the laboratory floor and the top of the seismic mass was measured experimentally as described in the previous section. Over the frequency range of interest it was found that the predicted displacement peaks due to the feedback path were at least 90 dB below the amplitude of those due to the forward path via the blocking mass. This shows that the pre-load support frame is more important then the shaker support frame as a potential error path. However the levels predicted are so far below the displacement levels via the forward path that errors due to feedback through the frame will not be significant.

(c) Seismic mass: A modal analysis of the seismic mass showed that it had a considerable number of modes below 2 kHz, so that the important consideration would be that these modes did not affect the results. From the previous analyses of the supporting frames which used measured transfer functions, it was clear that the modal behaviour of the seismic mass due to the reaction forces was not an important consideration. However, it is also possible for the seismic mass to be excited by the forward path forces through the pre-loading air-bags under the blocking mass, and these motions could affect the results. Therefore a model of the seismic mass, excitation mass, base plate, blocking mass, adhesive layer, locating bolts, air-bags and isolator was made. The seismic mass, base plate and blocking mass were modelled using eight-noded brick elements and the excitation mass was modelled as a lumped mass, while the air-bags and the isolator were modelled using spring elements.

An idealised seismic mass system was assembled with the seismic mass modelled as a rigid mass element and its supporting air-bags combined together. In addition, the bottom of the adhesive layer in contact with the seismic mass was constrained to move rigidly in sympathy with the seismic mass. The other elements in the model remained the same as above.

The results for the displacements on the blocking mass and the contact forces between the blocking mass and the isolator for the actual system were compared for the two models. The comparison of these corresponding displacements and forces showed that they differed from each other by less than 0.03 and 0.01 %, respectively. This indicates that the modal behaviour of the seismic mass has an insignificant effect on both the predicted displacements and forces.

4. MODELLING AND ANALYSIS FOR COMPLETE SYSTEM

In order to determine the system frequencies and establish whether the modes of the system would affect the measurements, the system was modelled as a series of rigid masses and springs, as shown in Figure 2. The model does not include the supporting frames, so that the stiffness elements attached to the two frames are assumed fixed at these points. The isolation hangers are represented by the springs k_1 and k_2, and the mass m_1. The springs k_1 and k_2 respectively represent the stiffnesses of the multi-layered rubber isolation elements and the steel springs, and the mass m_1 represents the mass of the hangers including one half of the masses of the rubber and steel spring elements. The shaker complete with its table extension is represented by the springs k_3, k_4 and k_5, and the masses m_2, m_3, m_4 and m_5. The masses m_2, m_3, m_4 and m_5 correspond respectively to the trunnion, body, table with the attached extension, and driving magnetic coil element. The springs k_3, k_4 and k_5 respectively represent the stiffnesses of the toroidal elastomeric isolators between the trunnion and the body, the suspension attaching the table to the body, and the linkage between the driving magnetic coil element and the table. The force produced by the shaker is generated between the body and the driving magnetic coil element, and is shown as F_{in}.

The shaker drives the excitation mass through a connecting rod. The connecting rod and the upper pre-loading air-bags are modelled respectively by the springs k_6 and k_7. The mass m_6 represents the excitation mass including the top mass of the force measuring assembly, the masses of the end supports for the pre-loading air-bags, and one half of the masses of the pre-loading air-bags. The stiffness of the force measuring assembly is modelled by the spring k_8.

The sample vibration isolator is represented by the mass m_7 and the spring k_9. The isolator is considered to comprise an upper plate, elastomer and a lower plate. The mass m_7 corresponds to the mass of the upper plate with one half of the elastomer's mass plus the bottom mass of the force measuring assembly, while the stiffness of the elastomer is modelled by the spring k_9.

The mass m_8 represents the blocking mass including the mass of the lower plate of the isolator, one half of the elastomer's mass, the masses of the end supports for the lower pre-loading air-bags and one half of the masses of the supporting air-bags for the blocking mass. The combined stiffness of the air-bags that support the blocking mass are represented by the spring k_{10}. The mass m_9 corresponds to the seismic mass including the mass of the base plate, the masses of the end supports for the supporting air-bags for the blocking and seismic masses, and half of the masses of the supporting air-bags for the blocking and seismic masses. The spring k_{11} represents the stiffness of the air-bags that support the seismic mass.

There are two limiting cases that need to be considered. The softest case, comprising the softest expected isolator under minimum pre-load with the lighter excitation mass; and the stiffest, comprising the stiffest expected isolator under maximum pre-load with the heavier excitation mass. As

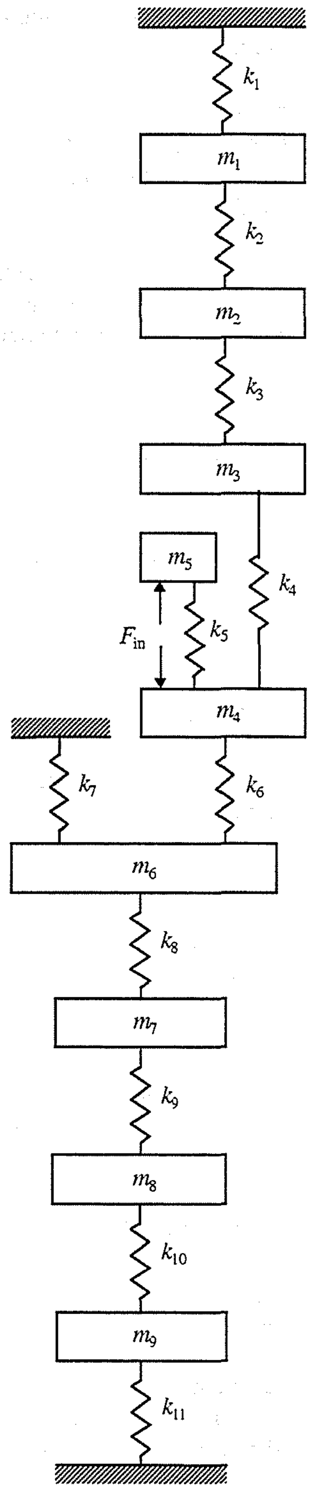

Figure 2
Complete system model

modelled the system has nine degrees of freedom and hence nine natural frequencies. The equations of motion for the system were solved to give the eigen-vectors and eigen-values using the numeric computation MATLAB software for the two cases outlined above. While it is not possible to know the masses of all isolator plates and elastomers expected to be tested *a priori*, representative minimum and maximum total isolator masses from isolators already tested were used, being 0.5 and 17.8 kg respectively. Masses and stiffnesses for the two cases investigated are given in Tables 1 and 2. The nine natural frequencies f_n obtained from the solution of each case are given in Table 3, where f_n corresponds to the nth mode.

Modes 4, 5, 6, 7 and 8 fall near or within the planed frequency range of measurement, which is from 10 Hz to 2 kHz. The fourth mode has the table, coil, excitation mass and the top plate of the isolator in-phase with each other, but out-of-phase with the isolator's bottom plate and blocking mass. The stiffness of the isolator has a major influence on the modal frequency f_4. The hanger exhibits the dominant motion in the fifth mode, and is out-of-phase with the trunnion. The frequency f_5 is determined by the stiffnesses of the rubber and steel spring elements, and thus is the same for both cases. In the sixth mode, the table and coil are in-phase with each other and out-of-phase with the excitation mass. The frequency f_6 is predominantly determined by the stiffness k_6 of the connecting rod. In the seventh mode, the shaker's trunnion is out-of-phase with its body and the frequency f_7 critically depends on the stiffness of the toroidal elastomeric isolators, and remains the same for both cases. The table and coil are out-of-phase with each other in the eighth mode, and the frequency f_8 is critically dependant upon the stiffness k_5, and so is equal for both cases.

Table 1
Masses

Mass (kg)	m_1	m_2	m_3	m_4	m_5	m_6	m_7	m_8	m_9
Softest case	16.6	41.1	607	4.34	6.38	105	1.65	572	22200
Stiffest case	16.6	41.1	607	4.34	6.38	291	10.3	576	22200

Table 2
Stiffnesses

Stiffness (N/m)	k_1	k_2	k_3	k_4	k_5	k_6	k_7	k_8	k_9	k_{10}	k_{11}
Softest case	1.57×10^{7}	1.63×10^{5}	2.65×10^{9}	1.37×10^{5}	5.49×10^{8}	3.57×10^{7}	2.62×10^{4}	4.00×10^{9}	1.00×10^{5}	2.31×10^{5}	2.87×10^{6}
Stiffest case	1.57×10^{7}	1.63×10^{5}	2.65×10^{9}	1.37×10^{5}	5.49×10^{8}	3.57×10^{7}	4.37×10^{5}	4.00×10^{9}	2.00×10^{7}	7.80×10^{5}	2.87×10^{6}

Table 3
Natural frequencies

Frequency (Hz)	f_1	f_2	f_3	f_4	f_5	f_6	f_7	f_8	f_9
Softest case	1.80	2.73	3.75	7.87	156	302	1320	2350	7900
Stiffest case	1.89	3.32	7.04	70.4	156	301	1320	2350	3290

From the work by Dickens and Norwood [3] and since the direct force at the input of the isolator is being measured, these modes 4, 5, 6, 7 and 8 will not adversely affect the test measurements. The sixth and eighth modes dominate the behaviour of the excitation mass, which shows maximum acceleration levels at the frequencies f_6 and f_8. When testing at or near these frequencies, care must be exercised to prevent excessive acceleration and force amplitudes of the table and coil. The upper frequency limit of the measurements is determined by the modal behaviour of the assembly of force transducers, represented by the ninth mode. In the ninth mode, the excitation mass and the top mass of the assembly are in-phase with each other but out-of-phase with the bottom mass of the assembly and the top plate of the isolator. The force transducer assembly measures the direct forces satisfactorily until it exhibits modal behaviour, and so the upper limit is determined by the frequency f_9, which predominantly depends upon the axial stiffness of the force transducers and the mass of the isolator's top end plate and elastomer. Therefore the upper frequency limits of the measurements for the softest and stiffest cases are 3.29 and 7.90 kHz, respectively.

Dickens and Norwood [3] derived an expression for the lower frequency limit of the measurements using direct forces. This limit is governed by the square root of the sum of the stiffnesses of the isolator and the lower pre-loading air-bags divided by the mass of the blocking mass together with the isolator's mass contribution. For the softest and stiffest cases they equate to 3.83 and 30.2 Hz, respectively.

Allowing factors of approximately two for the lower, and one half for the upper frequency limits gives practical testing frequency ranges for the softest and stiffest cases of 10 Hz to 1.6 kHz, and 60 Hz to 3.9 kHz, respectively. Because the modal behaviour of the individual components of the system has been designed for a maximum frequency of 2 kHz, the resulting system frequency ranges for the softest and stiffest cases are 10 Hz to 1.6 kHz, and 60 Hz to 2 kHz, respectively.

5. CONCLUSIONS

A test facility has been designed to measure the four-pole parameters of vibration isolators with pre-loads of up to 30 kN. The structure and construction of the test rig do not influence the results and do not limit the frequency range over which the isolators can be tested. The rig was designed to test up to 2 kHz and there are no influences from the structure in that frequency range. The lower frequency is governed by the stiffnesses of the isolator and the lower pre-loading air-bags, and the masses of the blocking mass and the isolator's lower plate, and not the structure of the test rig. The only components of the test facility with modal behaviour in the frequency range of interest are the two frames and the seismic mass. The feedback from these frames through the ground and the seismic mass is at least 90 dB lower than the direct path excitation levels and therefore the modal behaviour of the frames will not influence the test results. The effect of the modal behaviour of the seismic mass on the measured isolator's velocities and forces is less than 0.03 and 0.01 % respectively, and so is negligible.

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EXAMPLES OF NUMERICAL MODELLING USAGE TO SOLVE ACOUSTIC PROBLEMS

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ABSTRACT

Various numerical modelling techniques are presented to illustrate how acoustic problems can be solved during the design process. These techniques are presently becoming more relevant with the onset of more stringent sound emission regulations and higher customer expectation levels. These methods enable a detailed investigation into the physical phenomena prior to experimental implementation and thus present a considerable shortening of the overall design cycle. Examples presented include auditorium design, vehicle interior noise, fuel tank design and sound radiation from a ship.

1. INTRODUCTION

Computer simulation and numerical analysis of acoustic phenomena are fast becoming an integral part of the design cycle. In contrast to the traditional design approach in which prototyping plays a central role, the concurrent engineering design cycle aims to predict the physical behaviour of a prototype prior to its manufacture - thus for example the structural, thermal, aerodynamic and acoustic properties are optimised together in parallel. This integrated approach yields a higher quality product combined with a shorter time to market and at a lower cost.

Tougher government legislation demands improved sound quality and noise levels in our technology based environment. One technique for analysing these problems is based on ray tracing techniques and geometric acoustics. The versatility of this approach makes it extremely useful for a wide range of applications including industrial noise control, room acoustics and environmental acoustics. Typical applications for industrial noise control include the determination of noisy areas through noise ratings and evaluation of modifications such as changing absorber linings, reorganising machinery lay-out and modifying hall geometry. Room acoustics can be assessed using quality parameters such as early decay time, definition and clarity index allowing for optimal positioning of sound amplification and amplifier systems. The environmental impact of noise from factories, highways etc.... can be assessed and the subsequent resolution of the problem by optimisation of screens and barriers (position, length, height and effectiveness).

General acoustic and elasto-acoustic problems are being modelled more and more using finite element and boundary element techniques. Often the acoustic and structural behaviour of a mechanical system are strongly inter-related as shown in figure 1. Thus, vibration of a structural component induces pressure oscillations in the surrounding acoustic medium and similarly pressure waves impinging on the component will induce vibrations in the structure.

Usually it is adequate to consider only one of these two interactions - either the pressure distribution on a structure is calculated due to an acoustic source or the sound pressure level distribution is calculated in the vicinity of a vibrating structure. This class of problem is known as an **uncoupled problem** and is summarised in figure 2.

However, in many cases where the acoustic medium is dense or the structure is particularly flexible it is not possible to ignore the mutual interaction. These problems are known as **coupled problems** where the structural behaviour is influenced by the presence of the acoustic medium which in turn influences the generated pressure field. The two problems are solved together by taking coupling constraints into account which are mathematical representations of the physical conditions of:

- fluid loading of the acoustic medium on the structure.
- structural vibration inducing pressure waves in the fluid.

Four examples are presented to illustrate practical implementation of these computer modelling techniques. Auditorium design - ray tracing technique; 3-D Car Cavity Design - Boundary Element Analysis; Fuel Tank - Finite Element Analysis; and Acoustic Radiation of a Ship Hull - Boundary/Finite Element Analysis.

2. AUDITORIUM DESIGN.

It is a well known fact that fan-shaped auditoriums are attractive for combining large audiences with good site-lines but are not attractive acoustically due to the lack of early reflections in the centre of the main floor. One method for overcoming this problem is to incorporate saw tooth shaped walls which increases the lateral efficiency of hte auditorium. The lateral efficiency is an important acoustical quality parameter measuring the amount of energy arriving laterally at the receiver.

In 1994 an International Round Robin on Room Acoustical Computer Simulations was organised by Michael Vorlander. Acoustical parameters in a speech auditorium were measured experimentally and compared to those calculated using the geometrical acoustics package RAYNOISE Rev 2.1A. Experimental measurements were carried out by several different teams (Lundeby et al 1995).

The speech auditorium has a volume of about 1800m^3, a floor area of approximately 22 x 14 m^2 and a seating area inclined by 12°. Simulations and measurements are performed in one octave band (1kHz) with two source positions on the stage and with five receiver positions. The geometry model is shown in figure 3 and consists of eight different material linings with absorption coefficients derived from measurements in a reverberation room. A statistical reverberation time calculation was performed in order to validate the overall accuracy of these coefficients, yielding the results shown in table 1. The chosen absorption coefficients can therefore be considered to be sufficiently accurate compared to an experimentally determined space averaged result of $T_{30,\,meas.} = 1.16 \pm 0.10$ sec.

Various acoustical quality parameters were calculated and measured experimentally. However, this paper will concentrate on lateral efficiency results. Calculations take into account statistical tail correction, the number of rays per source was 20, the order of reflection was 30 and the calculation took 64 sec per source on a HP 9000/735. Figure 4 shows an auditorium plot of lateral efficiency with source 1 and figure 5 displays measured vs calculate values for the 10 source receiver combinations together with absolute differences.

3. 3D CAR CAVITY DESIGN

The modelling of elasto-acoustic problems is of primary interest in the study of vibration induced noise in passenger compartments. Internal sound pressure levels are directly related to the vibration of the boundary panels of the compartment which in turn are subject to excitation from the road and powertrain. Boundary Element Analysis is a useful tool to analyse the structural performance of the passenger compartment panels with a view to minimising sound pressure levels at a point near the drivers ear.

To demonstrate some features of the boundary element technique a three-dimensional model of a VOLVO 480 is presented (courtesy of Volvo, Sweden). The model, including internal front seats, is shown in figure 6 and comprises 786 nodes and 782 elements. The aim of the analysis is to show the effect of the finite impedance of the vibrating firewall on the sound pressure level. The pressure is calculated for different values of admittance of the vibrating panels from zero admittance (rigid panel) through a low admittance (0.001 m^3/Ns) to a higher admittance (0.002 m^3/Ns). The problem geometry was solved using the direct variational approach. It is evident from figure 7 that the Sound Pressure Level at the drivers ear decreases as the admittance increases.

Contribution analysis is a useful tool whereby the contribution of each panel to sound pressure level at a given point can be assessed. The boundary element mesh has been divided into six sets of elements representing different panels or surfaces of the model: front seats, roof, front part of the back seats, back part of the back seats and back window. An admittance boundary condition of 0.0021 m^3/Ns is defined on the 5 selected sets of elements and a unit velocity boundary condition is assigned on the firewall. All remaining components are assumed to be rigid.

The contribution of each panel is represented in figure 8 with the largest contribution coming from the front seats and the roof panel.

4. FUEL TANK

Fully coupled internal acoustic problems can also be solved using the finite element method. Typically the problem is formulated from two separate meshes - one representing the fluid pressure degrees of freedom and one representing the structural displacement degrees of freedom.

It is possible to specify an identical fluid structural interface mesh which requires duplicate grids of the structure and fluid elements. It is also possible to define a non-identical fluid structure interface whereby the user supplies an offset value to decide whether a point belonging to the structure lies on the surface of a fluid element. The fluid is modelled from elements which can assume the properties of irrotational, compressible fluids that are governed by the three dimensional wave equation.

A demonstration of the coupled fluid-structure interaction capabilities of MSC/NASTRAN was performed for the analysis of a large fuel tank. The aim of this analysis was to establish how the modal characteristics of the tank changed for full and empty conditions. The full tank problem was complicated by the fact that the deformed shape of the tank due to fluid loading had to be calculated initially. This was necessary as "bowed" panels have a different inherent stiffness than "flat" panels. The static deformation field is used to modify node locations prior to the modal extraction.

The analysis shows that the first natural frequency of the system drops from a value of approximately 20 Hz to 10 Hz for empty and full conditions respectively. The structural deformation and associated pressure field for coupled mode 1 is shown in figure 9.

5. ACOUSTIC RADIATION OF A SHIP HULL

This example presents a technique whereby vibrational modes of a structure can be used to compute a radiated acoustic field. This method is a combination of finite element analysis - to calculate structural modes (or indeed harmonic surface displacements) and boundary element analysis - to predict the acoustic field.

The object of this study is a semi-submerged ship hull comprising of 287 nodes and 490 shell elements as shown in figure 10. The acoustic boundary element mesh includes only the nodes and elements directly in contact with the water. The structural model yields a first bending mode at 3.66 Hz which is then transformed into normal velocity boundary conditions. The direct/collocation boundary element approach is used to calculate the acoustic field which is presented in figure 11. It should be noted that the boundary element model contains the definition of a free surface to model the water level.

6. CONCLUSION

Finite element, boundary element and ray tracing methods are now well established tools for solving acoustic problems. The frequency and scope of application of these techniques is increasing rapidly as acoustic analysis becomes an integral part of the modern day design process. The generalised use of simulation techniques provides engineers with tools for improving their design under tighter schedules and higher quality standards.

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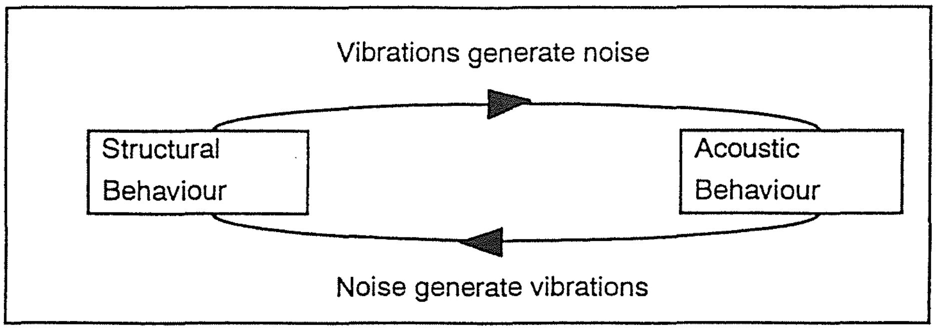

Figure 1 - Interaction between acoustic and structural behaviour

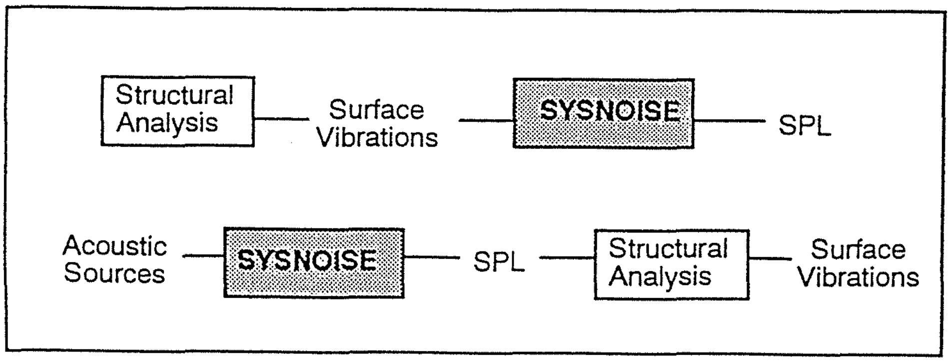

Figure 2 - Uncoupled Problems

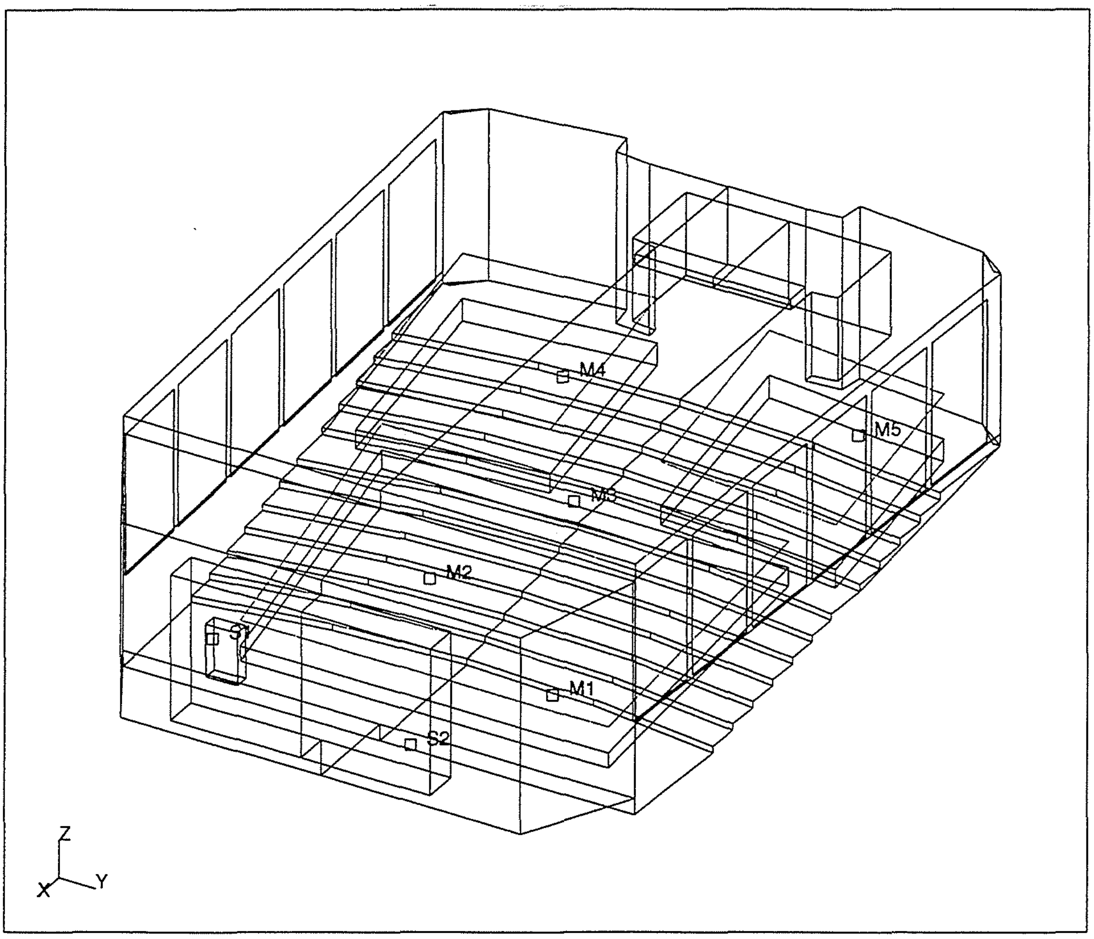

Figure 3 - Geometry model of the auditorium

REVERBERATION

Total Area (m2) : 1297.2
Volume (m3) : 1894.7

Mean Free Path (m) : 5.8
Relative variance : .64

Number	Mat_id	Name	Area(m2)	A-weight	S-weight
1	1	Hard walls	291.4	.22	.22
2	2	Ceiling-plasterb	302.3	.23	.27
3	3	Woodenlin	42.4	.03	.04
4	4	Floor	149.4	.12	.11
5	5	Stairs	175.8	.14	.05
6	6	Seating Area	169.4	.13	.13
7	7	Windows	117.1	.09	.14
8	8	Absorber panels	49.2	.04	.04

	63	125	250	500	1K	2K	4K	8K Hz
mean absorption :	.199	.199	.199	.199	.199	.199	.199	.199
Sabine area (m2):	273.	273.	273.	273.	273.	273.	273.	273.
RT(Sabine) :	1.18	1.18	1.18	1.18	1.18	1.18	1.18	1.18
RT(Eyring) :	1.06	1.06	1.06	1.06	1.06	1.06	1.06	1.06
RT(statis) :	1.13	1.13	1.13	1.13	1.13	1.13	1.13	1.13

Table 1 - Results from Statistical Reverberation Time Calculation

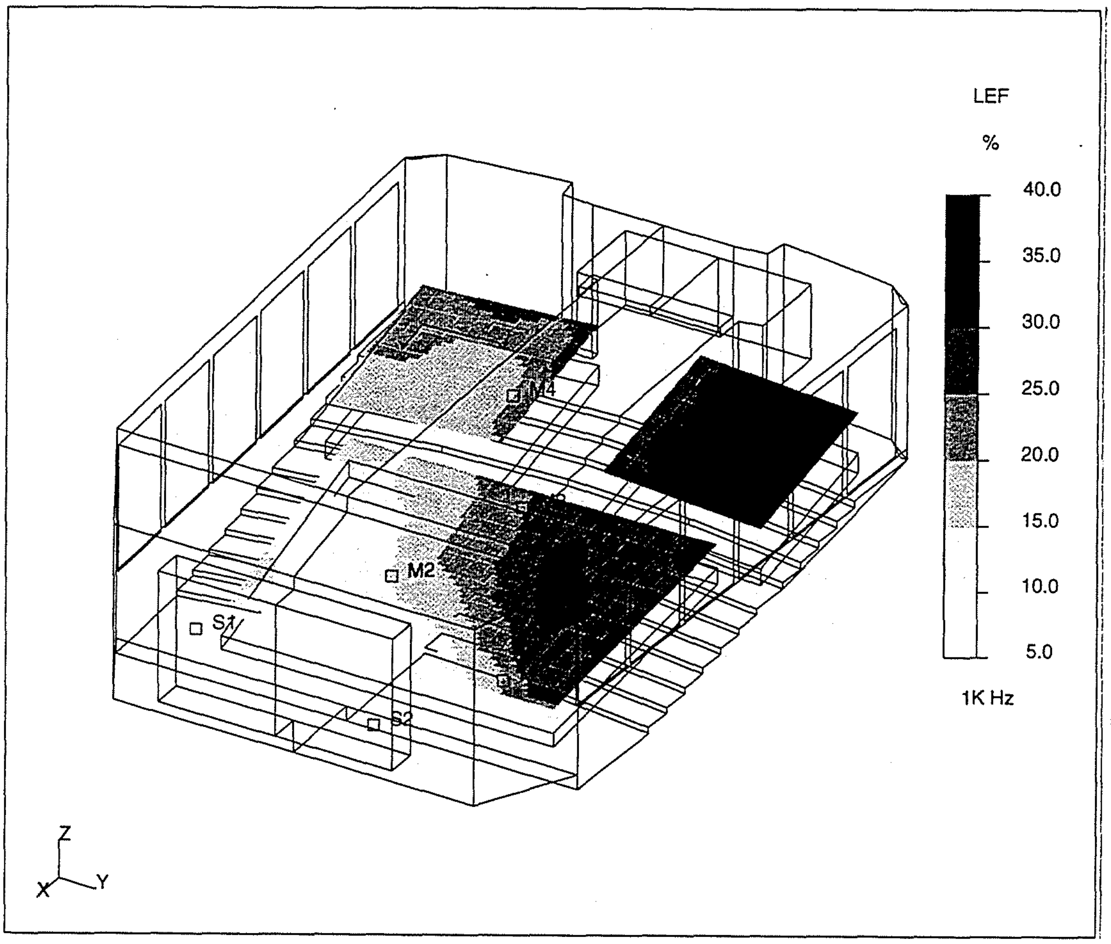

Figure 4 - Lateral Efficiency with source 1

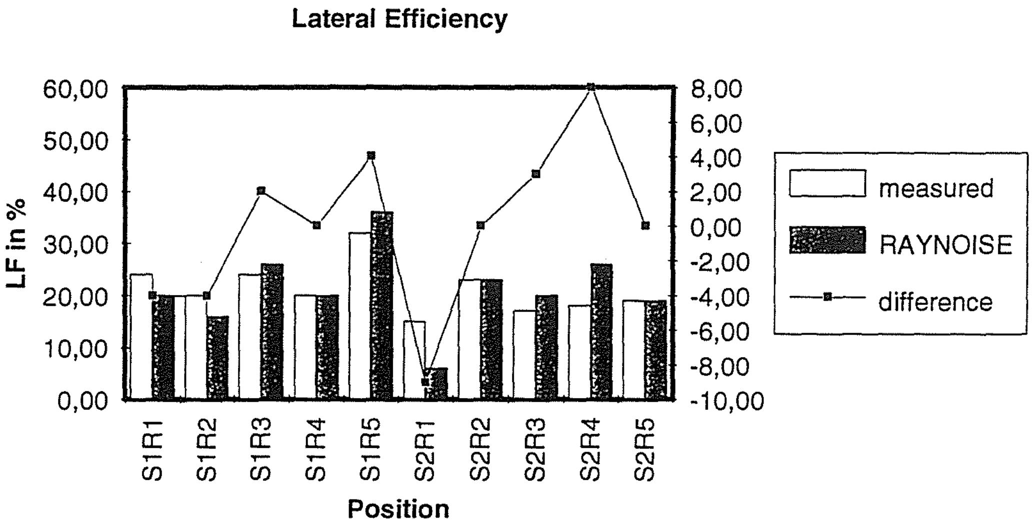

Figure 5 - Lateral Efficiency: Measurement vs Calculation

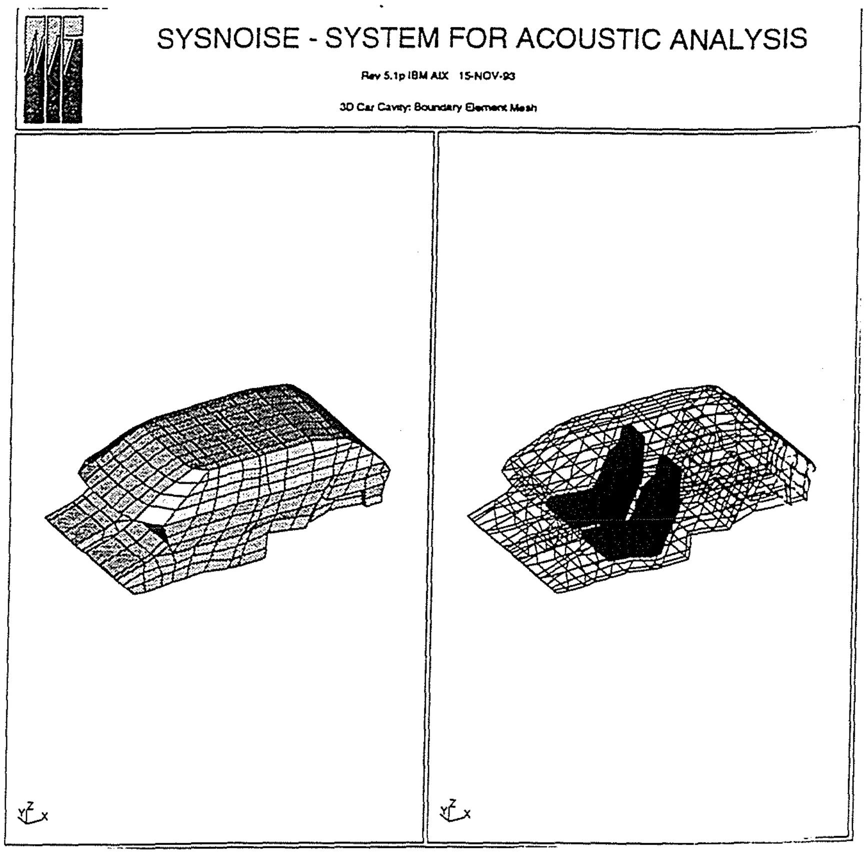

Figure 6 - 3D Car Cavity Boundary Element Mesh: 786 Nodes and 782 Elements

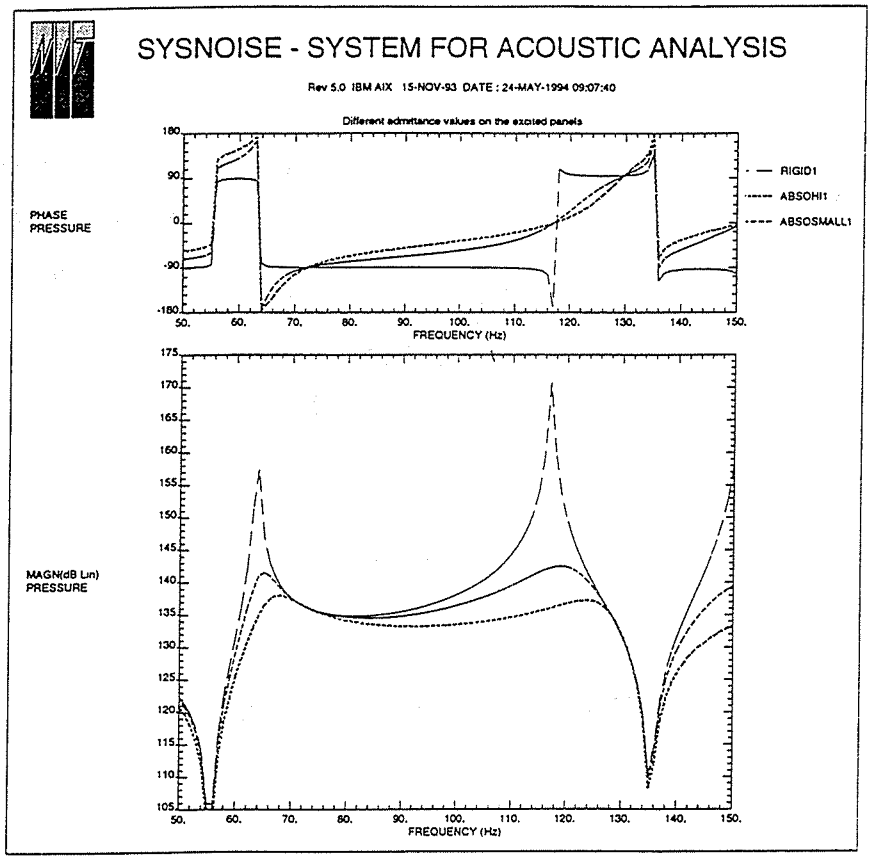

Figure 7 - Sound Pressure level at drivers ear for different values of the admittance of the excited panel

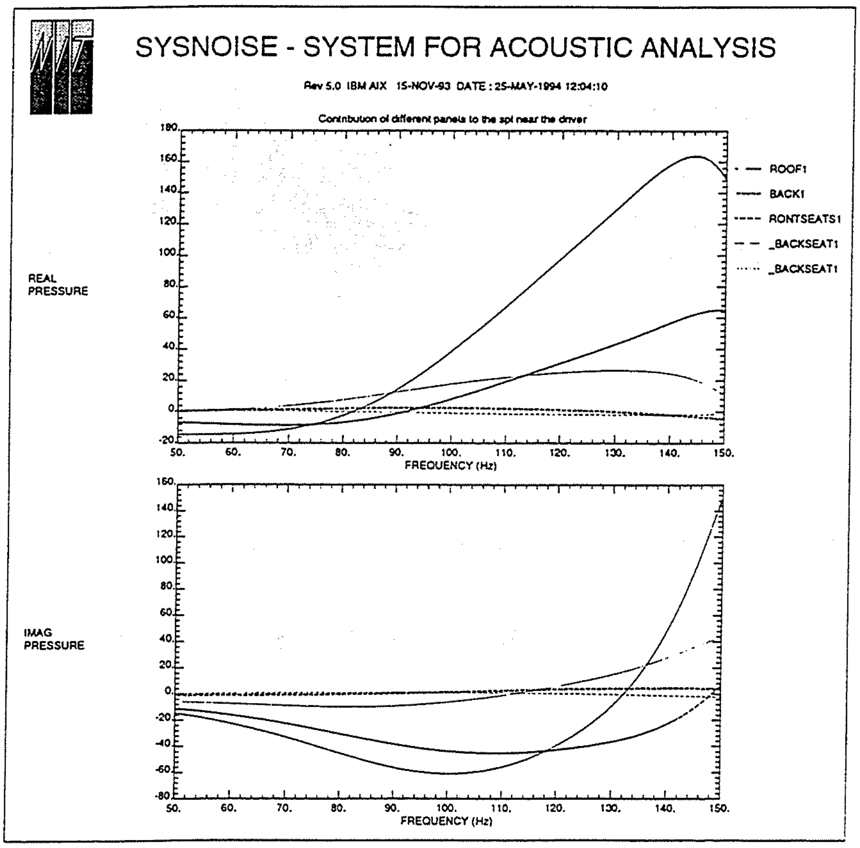

Figure 8 - Contirbution of the 5 defined panels

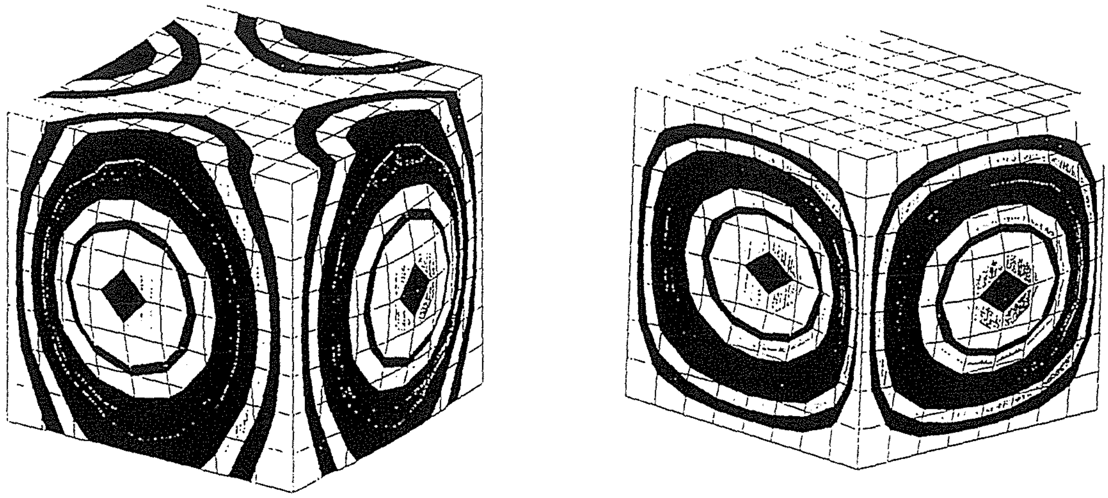

Figure 9 - Coupled Mode 1 of Fuel Tank. Left: Pressure Plot, Right: Structural Deformation Plot.

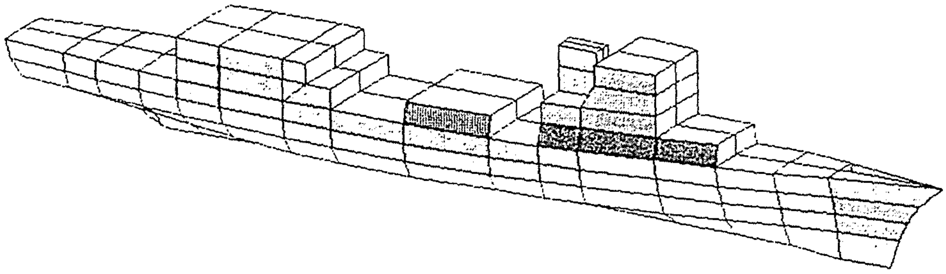

Figure 10 - Finite Element Mesh of Ship

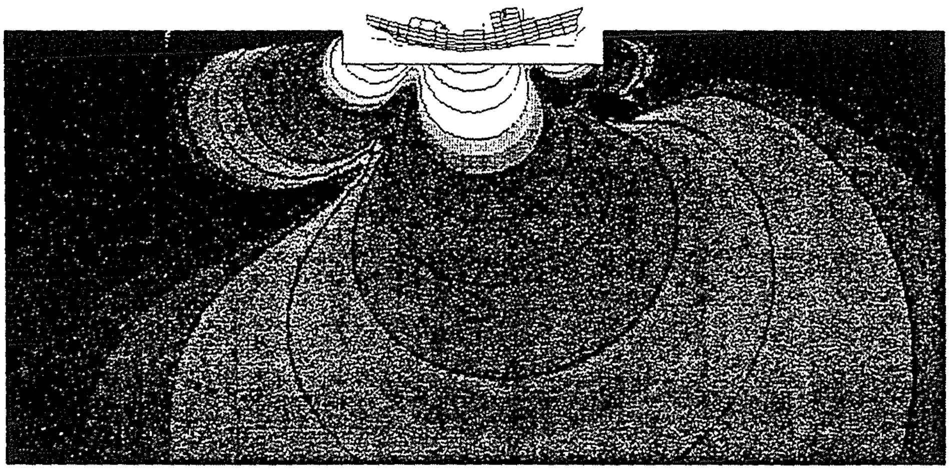

Figure 11 - Pressure Contour of Acoustic Radiation Pattern due to First Mode Vibration (3.66 Hz)

A MODEL OF ULTRASONIC ECHOLOCATION

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ABSTRACT

For several years, researchers have been working on the development of ultrasonic sensing systems for mobile robots. Their efforts have been met with limited success. Some systems take a long time to produce low resolution maps of the environment. Other systems can recognize a small set of known objects very quickly. This paper will discuss research aimed at the development of object recognition methods which offer an improvement over these techniques (in a limited domain) and consequently may serve to complement them.

INTRODUCTION

The most popular approaches for ultrasonic object recognition for mobile robotics are based on the arc model and impulse response model, although other approaches abound (McKerrow, et al.). The arc model (McKerrow & Hallam) involves taking several range readings to an object as the sensor moves past it. One draws arcs with a locus at the position of the transmitter and a radius of curvature equal to the range reading obtained. By drawing lines that intersect the tangents to these arcs we obtain an estimate of the object shape. The problem with this method is the poor resolution of the images that result, even the simplest objects such as a circular bin will end up reconstructed as a polygon. The other drawback is the speed, or lack thereof, of the method since the sensor has to move around the object before any image can be built up.

An alternative is the impulse response method (Lhemery A.) whereby a broadband ultrasonic pulse irradiates the object and fourier analysis is performed on the received echo. Since different objects will absorb different frequencies by different amounts, the fourier components of the returned echo will differ from one object to another. These echoes can then be used to form a database of recognizable signals. Although a much faster method than the former, it can obviously only be used for object recognition with those objects that it has previously encountered and hence has limited applicability.

The aim of the research reported here is to take a step back and examine the physics of scattering in an attempt to find the basic geometric characteristics that produce the echoes observed, with the hope that an inverse model can then be created enabling determination of the physical characteristics of the object from a knowledge of the echo. It is envisaged that such a model will enable the development of sensing systems that complement current ultrasonic sensing technology.

FORWARD MODELING

A number of different models are available for calculating the returned echo from an object of arbitrary shape irradiated by an acoustic wave. They fall roughly into two categories which we could call analytical models and numerical models. The former essentially provide us with an equation that relates the geometry of the object to the echo received. Because the echo and the geometry are linked via the equation, it has the potential to be mathematically inverted which would enable us to determine object geometry from the echo. In practice though, for many of these models a wide range of objects can give rise to the same echo, the many-solutions problem, making
inversion useless.

Numerical models, on the other hand, approach the problem by calculating the entire ultrasonic wave field at small discrete time steps from the moment of initial transmission until the moment the echo is received. Because the entire field is recalculated for each time interval, accurate echoes can be calculated for even relatively complex shapes. The drawback of these models is the time required for the echo calculation, from minutes to hours with current processing speeds, but more importantly they lack inversion potential because mathematical inversion only provides the wavefield for the previous time step and at this stage you've already encountered the many-solutions problem. Consequently, analytical models offer the best hope for our work, the model of choice being the work of A. Freedman in the late 50's.

Freedman's model (Freedman A.) is based upon a number of assumptions, the most important are that the scattering body is convex, rigid and in the far field with respect to the transmitter. The central equation of the model, for the case of coincident transmitter and receiver, is given by:

$$E_g = j\frac{VPH}{r_m^{\ 2}\lambda}\exp\left(j(t\omega - 2kr_g)\right)\sum_{n=0}^{\infty}\frac{D(A,g,n)}{(j2k)^n} \tag{1}$$

where, E_g is the voltage at the receiver due to an echo originating at range r_g,

V is the voltage applied to the transmitter during transmission,

P & H are sensitivity factors for the transmitter & receiver respectively,

r_m is the mean range to the object,

λ, ω and k are the wavelength, angular frequency and wave number of the acoustic pulse,

$A(r_g)$ is the cross sectional area subtended at the receiver by all those parts of the scattering body within a range r_g,

$$D(A,g,n) = \frac{d^n A}{dr^n}(r_{g-}) - \frac{d^n A}{dr^n}(r_{g+}) \tag{2}$$

and is a measure of the discontinuity in the nth order rate of change of $A(r_g)$.

The implications of the discontinuity Equation 2 are best illustrated by example. At an arbitrary range r_g on a smoothly varying surface of the scatterer, the nth order rates of change at a point preceeding r_g by an infinitesimally small amount and at another point located beyond r_g by an infinitessimal amount will be the same. Hence, Equation 2 is zero, thus there is no discontinuity and so no echo is generated. At those points at range r_g where there is a discontinuity in the value of one of the nth order derivatives , Equation 2 will have a non-zero value and hence contribute to the echo. Thus discontinuities in cross sectional area or any higher order rates of change of cross-sectional area are responsible for the formation of echoes.

Referring to Figure 1.0 as an example, the cross sectional area of the sphere (as "seen" by a transducer located to the far left of the sphere) is not discontinuous at any point. However, at the front (left) face the first and second order rates of change are discontinuous and so each will produce an echo component. These are superimposed to form the echo from the front face. Similarly, at the equator of the sphere the second order rate of change is discontinuous and hence an echo will originate from there too.

One of the assumptions of the model is that the surrounding medium is a non-attenuating fluid, such as water. For our purposes we need a model that works in air, hence we need to add an attenuation factor of the form exp($-2r_g\,\alpha$) to the above equation, where α is the attenuation constant. Currently, work is being carried out to verify the validity of this model in air because Freedman developed and tested the model for water.

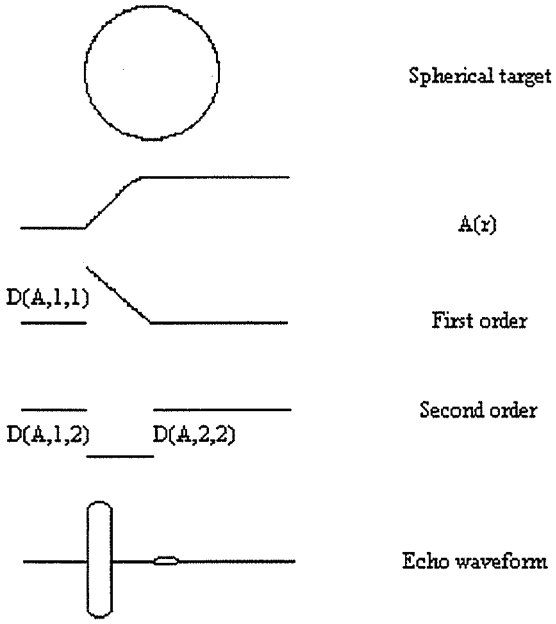

Figure 1.0 Echo reflected off a sphere.

INVERSE MODELING

Once the forward model has been validated, our aim is to invert it. Although the equation is complex (ie. real & imaginary components), we can only measure the real components, leaving us with,

$$\mathrm{E}_g = \Re(E_g) = \frac{VPH}{r_m^{\,2}\lambda}\exp\left(-2r_g\,\alpha\right)\left[D(A,g,0)\sin\theta + D(A,g,1)\frac{\cos\theta}{2k} - D(A,g,2)\frac{\sin\theta}{4k^2}\right] \quad (3)$$

where $\theta = t\,\omega - 2kr_g$ (4)

We only include components up to the 2nd order discontinuity because higher components are negligible in amplitude and hence unobservable (the second order discontinuity is already 10^4 times smaller than the zeroth for a 10 cm diameter sphere). This can be further simplified,

$$\mathrm{E} = a.D(0) + b.D(1) + c.D(2) \qquad [\text{ where } D(n) = D(A, g, n)\ ] \quad (5)$$

At this stage our equation would suffer the many-solutions problem if inverted, but we circumvent this in the following way. Clearly a, b and c are determinable functions of frequency, time, distance, humidity and temperature. By measuring time, distance, humidity and temperature and holding them constant, we can generate simultaneous equations by varying frequency. We only need three simultaneous equations (ie. three frequencies) in order to generate the 3x3 matrix needed for the inversion, the inverted matrix having the following form,

$$\begin{bmatrix} D(0) \\ D(1) \\ D(2) \end{bmatrix} = \begin{bmatrix} a(f_1) & b(f_1) & c(f_1) \\ a(f_2) & b(f_2) & c(f_2) \\ a(f_3) & b(f_3) & c(f_3) \end{bmatrix}^{-1} \begin{bmatrix} \mathrm{E}(f_1) \\ \mathrm{E}(f_2) \\ \mathrm{E}(f_3) \end{bmatrix} \qquad (6)$$

Thus once a, b and c are calculated or measured for each of the three frequencies, the matrix can be inverted and then used along with the returned echo amplitudes measured for each frequency to determine the three discontinuities.

SCATTERER GEOMETRY RECONSTRUCTION

Once D(0), D(1) & D(2) are obtained for each range along the scatterer for which there is an echo, we can build up a picture of the scatterer in the following way. For the front face of the scatterer at a range r_o, we have the following equations,

$$D_{ro}(0) = A(r_{0-}) - A(r_{0+}) \qquad (7)$$

$$D_{ro}(1) = \frac{dA}{dr}(r_{0-}) - \frac{dA}{dr}(r_{0+}) \qquad (8)$$

$$D_{ro}(2) = \frac{d^2 A}{dr^2}(r_{0-}) - \frac{d^2 A}{dr^2}(r_{0+}) \qquad (9)$$

but of course $\frac{d^n A}{dr^n}(r_{0-})$ are all zero, since r_{o-} is the range just prior to the front face where there is no variation in cross sectional area. Hence,

$$\text{Area at } r_o = -D_{ro}(0) \tag{10}$$

$$\frac{dA}{dr} \text{ at } r_o = -D_{ro}(1) \tag{11}$$

and $$\frac{d^2 A}{dr^2} \text{ at } r_o = -D_{ro}(2) \tag{12}$$

Now, assuming that the second order derivatives remains constant over the range r_{o+} to r_{m-}, where r_m is the range at which the next echo originates, then we can apply the following relations at each successive point within this range,

$$A_{x+1} = A_x + \frac{dA_x}{dr}(r_{x+1} - r_x) \tag{13}$$

$$\frac{dA_{x+1}}{dr} = \frac{dA_x}{dr} + \frac{d^2 A_x}{dr^2}(r_{x+1} - r_x) \tag{14}$$

Once $A(r_{m-})$ and $\frac{dA}{dr}(r_{m-})$ have been calculated via the above method and assuming $\frac{d^2 A}{dr^2}(r_{m-}) = \frac{d^2 A}{dr^2}(r_{0+})$, we can calculate the area and higher order rates of change at r_m using the following relations,

$$\text{Area at } r_m = A(r_{m-}) - D_{rm}(0) \tag{15}$$

$$\frac{dA}{dr} \text{ at } r_m = \frac{dA}{dr}(r_{m-}) - D_{rm}(1) \tag{16}$$

and $$\frac{d^2 A}{dr^2} \text{ at } r_m = \frac{d^2 A}{dr^2}(r_{m-}) - D_{rm}(2) \tag{17}$$

This process is continued for all ranges at which echoes originate and thereby the shape of the scatterer is reconstructed if it is symmetric, or a symmetric equivalent if it is non-symmetric. Either way, the shape information could be used to discriminate between a set of known objects.

EXPERIMENTAL WORK

Our setup consists of a software controlled chirp generator card (which enables the generation of an arbitrary wave shape) whose output is amplified then directed to a Polaroid ultrasonic transducer. The echo is received via an ultrasonic microphone, the signal amplified and channeled into a 12 bit analogue to digital converter and then sampled at a maximum rate of 1 MHz by a chirp capture card.

The main issue of concern currently is that of reducing experimental error. The electronic noise has been minimized with the electrostatic shielding of wires and the synchronization of the transmit and receive cycles, but errors remain a problem. Specifically, they can be localized to three sources, the transducers response, the long term drift in atmospheric conditions and short term fluctuations due to air turbulence and measurement system vibration. The transducers output for a fixed amplitude waveform can decrease in amplitude by as much as 20% over a seven hour period. This is believed to be due to the mylar film of the transducer stretching under the tension of the constant 300 volt bias, the result being a loss in rigidity. We can compensate for this problem by either modeling the amplitude drift (assuming the drift is repeatable), recalibrating at regular intervals, or removing the bias between measurements.

The drift in environmental conditions (such as temperature, humidity and pressure) over a number of hours are of concern because they affect the attenuation constant in air, a figure which is vital in our modeling. We had hoped to be able to continuously monitor these conditions and use them to calculate the attenuation constant in real time using the series of equations specified by the American national standard method for air attenuation calculation (Acoustical Society of America). Unfortunately, the model they employ has an error of 10% in our frequency range (approx. 50 kHz), making it unreliable. Thus it seems that an experimental determination of air attenuation at regular intervals will be necessary. How these will affect the end product of this research, namely mobile robot sensing, is that the robot every once in a while may need to recalibrate it's system by sensing a known object.

The problem of short term fluctuations in air (from moving objects, drafts, etc.) and vibrations in the measurement system (from doors opening, people walking, etc.) can cause errors of up to 10%. The solution is to either shield the setup in a box to minimize turbulence effects and support the setup on shock absorbent material or simply to average over multiple readings to improve the signal to noise ratio. For the experimental work, both methods are adequate although for the final working setup the latter is the only practical solution.

The verification of the forward model, specifically Equation 3, will be done by testing the model on a number of objects (namely spheres and cones of varying dimensions) over a range of frequencies and distances. The measured amplitude at a discrete point in time along the echo from each of the objects will be compared with the amplitude values calculated with Equation 3 for that time. The quantities used in the calculation are derived from both known (e.g. time, frequency, range, etc.) and experimentally determined (e.g. atten. constant and VPH factor) values. In this way the range of conditions under which the forward model is valid will be established.

Similarly, the inverse model will be tested over a range of distances for different objects. The functions $a(f)$, $b(f)$ and $c(f)$, from the matrix in Equation 6, will each be calculated for a particular point in time at three different frequencies. Again, the quantities used in the calculations will be derived from both known and experimentally determined values. The amplitudes of the echoes for this point in time for each of the three frequencies will then be measured and Equation 6 evaluated. Comparisons of the calculated discontinuities with the measured ones will enable the determination of the range of conditions under which the inverse model is valid.

CONCLUSION

At this point, we have achieved better than 10% accuracy with the forward model. To improve this accuracy, we have spent considerable time identifying sources of error so that they can be eliminated or compensated for in future tests. This will either improve the accuracy or indicate the need for changes to the model. Ultimately, we foresee that the techniques illustrated above will lead to the development of a real-time object reconstruction system that will complement current systems by specializing in the reconstruction of convex scatterers or their modeling with equivalent symmetrical objects. Major applications of such a system include map-building by mobile robots and object recognition conveyor belts.

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REDUCTION OF SOUND TRANSMISSION THROUGH A WINDOW USING ACTIVE ABSORBERS.

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Abstract: The usefulness of maximally absorbing sources in preventing sound power from being transmitted through a square aperture in an infinitesimally thin infinitely large plane wall is considered. A coupled boundary element technique is used to calculate the acoustic propagation through the aperture. Two types of maximally absorbing sources are considered: a monopole; and a monopole and dipole combination. Results of numerical calculations show that the monopole and dipole combination source is capable of providing significant attenuation of sound power transmission through the aperture, whereas the monopole source tends to increase the sound power transmission for large wavelengths.

1. Introduction

In the active control of noise the most common way to control the noise is to minimize some cost function of the acoustic field over some region of space, e.g. minimize the sum of the square of the pressures at a number of points in a region. Now, whilst this type of control may successfully reduce the noise over the region of space considered, the effect elsewhere may be uncertain. It is possible that reducing the noise in a small region will result in an increase of acoustic power output from the sound sources, implying that the noise levels will be greater in other regions. This point leads to another way to control noise, and that is the control of the sound power from the noise sources.

2. Sound Absorption Using Monopole Sources.

Assume that we have an array of single frequency monopole (i.e. point) sound sources which have complex volume velocities given by the vector $\mathbf{q}$. Some of these sources are `primary sources' ¾ those we have no control over, and some are `secondary sources' ¾ those which we can control. The complex volume velocities of the primary and secondary sources are given by $\mathbf{q}_s$ and $\mathbf{q}_p$ respectively (note that $\mathbf{q}=[\mathbf{q}_p\ \mathbf{q}_s]^T$, where ^{T} denotes matrix transpose). At each of these sources there is a sound pressure, these complex pressures are denoted by the vector $\mathbf{p}$. The time averaged (i.e. averaged over one period) power output of one single-frequency monopole is $(1/2)\mathrm{Re}\{p^*q\}$ [Nelson & Elliott, Sec. 9.11], where q is the complex volume velocity of the source, p is the complex pressure at the point source, * denotes complex conjugation and Re{ } is the real part of a complex number. Hence, the total time averaged sound power emitted from the array of sound sources is $W=(1/2)\mathrm{Re}\{\mathbf{p}^H\mathbf{q}\}$, where ^{H} denotes the conjugate transpose.

We can express the acoustic pressure at a point due to a monopole of volume velocity q by $p=zq$, where z is the acoustic transfer impedance from the monopole to the measuring point. We can therefore write that

$$\mathbf{p} = \begin{bmatrix} \mathbf{Z}_{pp} & \mathbf{Z}_{ps} \\ \mathbf{Z}_{sp} & \mathbf{Z}_{ss} \end{bmatrix},$$

where the submatrix $\mathbf{Z}_{pp}$ is the transfer impedances of the primary sources to the primary source positions, $\mathbf{Z}_{ps}$ is the transfer impedances of the secondary sources to the primary source positions, etc. After much manipulation and assuming $\mathbf{Z}=\mathbf{Z}^T$ (i.e. acoustic reciprocity holds), the acoustic power flow out of all of the sources, W, can be written as

$$W = (1/2)(\mathbf{q}_s^H\mathrm{Re}\{\mathbf{Z}_{ss}\}\mathbf{q}_s + \mathbf{q}_s^H\mathrm{Re}\{\mathbf{Z}_{ps}\}\mathbf{q}_p + \mathbf{q}_p^H\mathrm{Re}\{\mathbf{Z}_{ps}^T\}\mathbf{q}_s + \mathbf{q}_p^H\mathrm{Re}\{\mathbf{Z}_{pp}\}\mathbf{q}_p) \quad .$$

The minimum value of W occurs when [Elliott & al.]

$$\mathbf{q}_s = -\mathrm{Re}\{\mathbf{Z}_{ss}\}^{-1}\mathrm{Re}\{\mathbf{Z}_{ps}\}\mathbf{q}_p \quad .$$

Observe that in order to minimize W it is necessary to know the volume velocities of and transfer impedances to the primary sources.

If $\mathbf{Z}_{ps}$ and $\mathbf{q}_p$ are unobtainable a possible course of action is to maximize the power absorbed into the secondary sources. Even though this may result in an increased power output from all of the sound sources, it is possible that there will be reduced noise levels in a region close to the secondary sound sources. The maximum of the sound power input to the secondary sources is achieved when

$$\mathbf{q}_s = -(1/2)\mathrm{Re}\{\mathbf{Z}_{ss}\}^{-1}\mathbf{p}_n \ ,$$

where $\mathbf{p}_n$ is the primary source pressures (also called the incident field) at the secondary source positions, or, equivalently, when

$$\mathbf{q}_s = -(\mathbf{Z}_{ss}^*)^{-1}\mathbf{p}_s \ ,$$

where $\mathbf{p}_s$ is the total pressures at the secondary source positions.

From the above equation it is apparent that the secondary sources maximize the power absorption by operating as a collective group. It is quite likely that in order to achieve this maximum power absorption some of the secondary sources will be sources of sound power rather than absorbers of sound power. Instead of maximizing the power absorption of the secondary sources a collective group, one can opt to maximize the power absorption of each secondary source individually. This occurs when $q=-p/z^*$ for every secondary source, where p is the sound pressure at a secondary source and z is the complex input impedance of the same source. Although the total power absorption in this case will be less than or equal to the maximum power absorption of the secondary sources operating as a collective group, we have the obvious advantage that it will not be necessary for the secondary sources to communicate information amongst each other.

It can be shown that the absorption of a maximally absorbing monopole in a free field situation with a plane incident field [Nelson & Elliott Sec.9.12] is equivalent to that of perfect absorbing area of size (l^2/4p). One is tempted, therefore, to imagine that an array of such sources spaced (l/2Öp) apart would act as an acoustic screen. One is further tempted to wonder whether placing a monopole in an open window of area less than (l^2/4p) would prevent significant amounts of sound power from being transmitted through the window. It is this last question which this paper attempts to address.

3. **Calculating the pressure field near a window.**

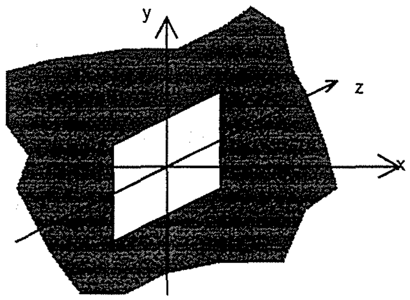

Figure 1: Window in an infinite plane wall.

In order to predict what will happen with monopole sound absorbers near a window we first have to calculate the sound propagation through a window. For the purposes of making the problem relatively easy to solve it is assumed that the `window' is a square hole in an infinitesimally thin, perfectly rigid wall of infinite extent. The cartesian co-ordinates are arbitrarily set such that the wall lies on the y-z plane at x=0 (See Figure 1).

If all the sound sources are monopoles then the acoustic pressure field is given by

$$\sum_l p_l(\mathbf{r}),$$

where $p_l(\mathbf{r})$ is the field generated by source l . $p_l(\mathbf{r})$ satisfies

$$\nabla^2 p_l(\mathbf{r}) + k^2 p_l(\mathbf{r}) = -j\,\omega\rho q_l\,\delta(\mathbf{r}_l - \mathbf{r})$$

in the volume considered as well as the boundary conditions on the surface of the volume, where q_l is the volume velocity of the monopole, $\mathbf{r}_l$ is the location of the monopole, j^2=-1 , w is the angular frequency, r is the air density and d is the dirac delta function. Using Green's theorem which relates the field in a volume to the field on the surface of the same volume, and by using the Green's function for a wall with no hole (represented by $h(\mathbf{r}½\mathbf{r}_0)$) we get,

$$p_l(\mathbf{r}) = -j\,\omega\rho q_l h(\mathbf{r}|\mathbf{r}_l) + \int_{S_1} h(\mathbf{r}|\mathbf{r}_0)\frac{\partial p_l(\mathbf{r}_0)}{\partial x}dS_0,$$

when **r** is on the same side of the wall as the monopole ($\mathbf{r}_l$) , where S_1 is the square hole area. We also get,

$$p_l(\mathbf{r}) = -\int_{S_1} h(\mathbf{r}|\mathbf{r}_0)\frac{\partial p_l(\mathbf{r}_0)}{\partial x}dS_0,$$

when **r** is on the other side of the wall to the monopole ($\mathbf{r}_l$) .

To solve these equations we need to find the gradient of p_l in the direction of x over all of the hole, S_1 . This is achieved by ensuring p_l and its gradient given by the above two equations are equal over the surface S_1 [Morse & Ingard Sec. 10.4]. Doing so yields the equation

$$\int_{S_1} h(\mathbf{r}|\mathbf{r}_0)\frac{\partial p_l(\mathbf{r}_0)}{\partial x}dS_0 = -\frac{1}{2}j\,\omega\rho q_l h(\mathbf{r}|\mathbf{r}_l),$$

which is an implicit expression for the gradient of p_l over S_1 . We can find an approximate explicit expression for the gradient of p_l over S_1 by breaking S_1 into small elements and assuming the gradient of p_l to be constant over an element. Having done this we are now in a position to calculate p_l for monopole l , hence find the transfer impedance, maximize the sound power absorbed by the monopoles and ultimately find the total acoustic pressure field generated by such a group of monopoles.

4. **Calculating the sound power flow through a window.**

In order to assess the performance of sound absorbing monopoles a plane incident wave was imagined to insonify one side of the wall and the sound power which flowed through the window to the other side of the wall was calculated.

Single Monopole:

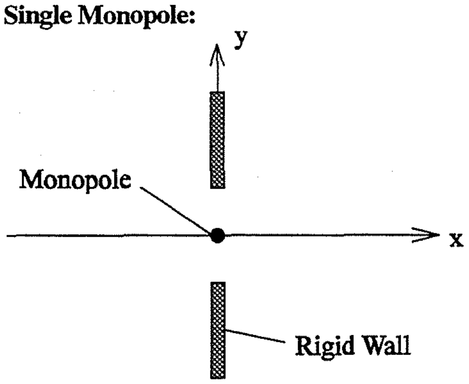

Figure 2:Side on view showing the position of the monopole in the window.

In the first instance the case where a single monopole was theoretically placed in the centre of the hole was examined, i.e. at $(x,y,z) = (0,0,0)$ (See Figure 2). Figure 3 shows plots of the calculated power flow out of the hole with and without absorption, as well as the power absorbed into the monopole. Note that when the window has an area less than (l^2/4p) , i.e. when the wavelength is greater than 2Öp, the power output from the window increases with increasing wavelength ¾ a somewhat surprising result. Analytical analysis shows that the power absorbed by the monopole is

$$-\frac{1}{2}\mathrm{Re}\{p^* q\} = \frac{\pi}{2k^2 c\rho}|p_n|^2,$$

i.e. the power absorbed is independent of the hole size. This is because, since the wall is infinitely thin and the monopole is on the same plane as the wall, there is no sound reflected back onto the monopole. It also can be shown that as the wavelength approaches infinity the power flow into the hole (i.e. power flow through the hole plus power flow into the monopole) when the monopole is maximally absorbing approaches

$$\frac{3\pi}{2k^2 c\rho}|p_n|^2.$$

Therefore, the ratio of sound power emitted from the hole to the sound power absorbed by the monopole approaches 1/3 as the wavelength approaches infinity.

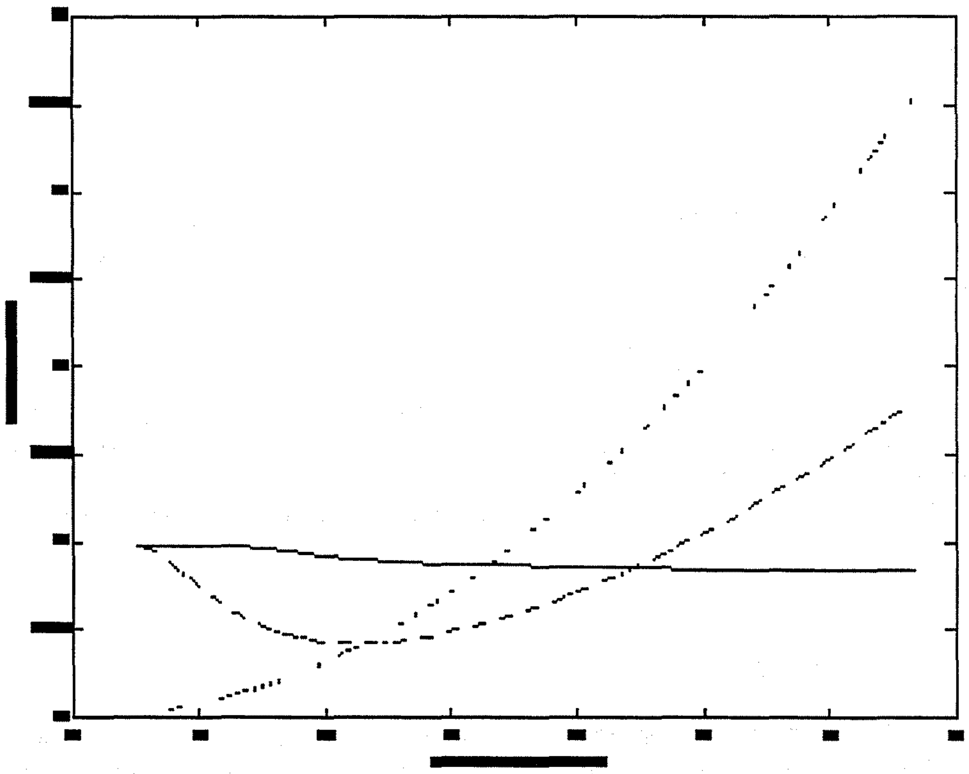

Figure 3: Plot of sound power transmitted through window in the case of no control and when there is one maximally absorbing monopole (dot-dash). Also, plot of the sound power absorbed by the monopole (dot-dot).

It is interesting to compare this result to that for the case of one dimensional sound propagation, for example, in a duct, where the effect of a one dimensional maximally absorbing monopole is considered. It can be shown [Nelson & Elliott, §5.5] that, in such a case, 1/4 of the incident sound energy is transmitted beyond the monopole and 1/2 is absorbed by the monopole (implying, of course, that 1/4 of the sound energy is reflected).

A dipole/monopole combination (two closely spaced monopoles):

It is well known in the analysis of one dimensional active noise control, such as ducts or tubes, that two closely spaced monopoles (or, equivalently, a dipole and monopole combination) have the ability to absorb all the incident sound energy — there is no sound transmission nor any sound reflection.

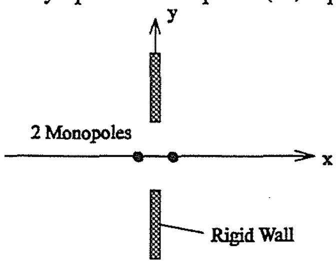

Figure 4: Side on view showing the position of the two closely spaced monopoles in the window.

With this in mind let us consider the case where we have two closely spaced monopoles, arranged so that the line joining the two monopoles is perpendicular to the surface of the window, and so that the monopoles are in the centre of the window (see Figure 4). The monopoles were placed at (-l/40,0,0) and at (l/40,0,0), their volume velocities independently set so that the two monopoles maximally absorbed sound power, and the sound power flow through the aperture calculated. The monopole separations are wavelength dependent since numerical instabilities occur in the calculation of the optimum monopole volume velocities if the separation is too small compared to a wavelength.

Figure 5 shows plots of uncontrolled sound power flow through aperture, sound power flow into the x>0 side of the aperture when the two monopoles are maximally absorbing, and the sound power absorbed by the two monopoles, for a plane incident wave propagating in the positive x direction with unit intensity.

These results show that this monopole/dipole active absorber arrangement can reduce the sound power transmission significantly especially for large wavelengths.

5. Conclusion

Two types of active absorbers have been compared for their ability to reduce sound transmission through a square aperture in an infinitesimally thin infinitely large rigid wall. The types of active absorber tested were a maximally absorbing monopole and a maximally absorbing dipole and monopole combination. The latter combination was found to provide significant attenuation of the sound power transmitted through the aperture for wavelengths large compared to the dimension of the aperture, viz. greater than 10dB for wavelengths greater than three times the length of the aperture side. However, the monopole active absorber appeared to be capable of focusing the sound power through the aperture for large wavelengths, resulting in an increase in sound power transmission even though most of the sound power was being absorbed by the active absorber.

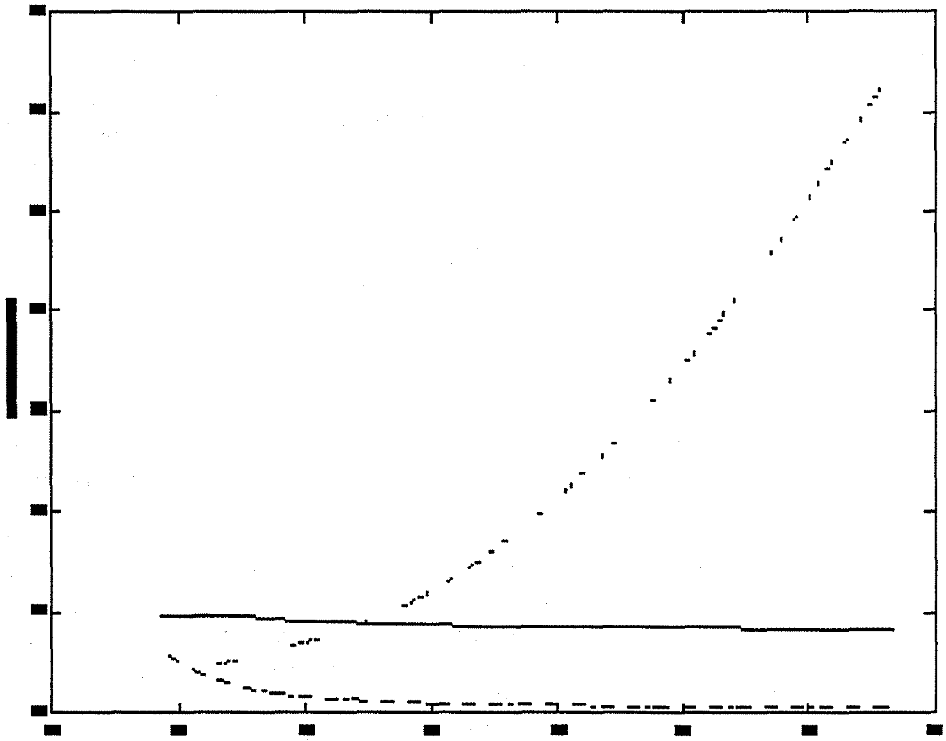

Figure 5: Plots of uncontrolled power flow, controlled power flow (dash-dot), power absorbed (dot-dot), for a maximally absorbing monopole and dipole combination at the centre of a square aperture of dimension 1m x 1m.

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A SOUND INTENSITY BASED INVESTIGATION OF THE EFFECT OF STIFFENERS ON THE SOUND TRANSMISSION LOSS OF A SINGLE LEAF MDF PARTITION

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ABSTRACT

As part of a study into the sound transmission properties of partitions constructed from medium density fibreboard (MDF), an investigation into the fractions of the incident sound energy transmitted through the stiffened and unstiffened portions of a $10m^2$, single leaf, 18mm thick, MDF partition was undertaken by making sound intensity measurements at 81 points over the surface of the stiffened partition. In addition sound intensity measurements were made at identical points over an unstiffened partition. An indication of the spatial variability of the radiated sound intensity induced by the stiffeners then could be obtained. The validity of the sound intensity measurements was assessed by using them to determine the transmission losses of the stiffened and unstiffened partitions and these sound transmission losses were compared with those derived by the conventional two room measurement method. It was found that the lightweight channel stiffeners of the type widely used with MDF partitions produce no significant effect on the sound transmission loss with a single leaf construction.

INTRODUCTION

Medium density fibreboard (MDF) is made of a material with a density of approximately $730kg/m^3$. Because it can be manufactured in thicknesses up to 37 mm, the resultant product can have a high mass per unit area and it also has a high strength. These attributes make MDF well suited to the construction of building elements such as partitions in which high sound transmission loss is sought.

Usually partitions incorporate stiffeners such as wood or metal studs and this is so with MDF partitions. The effect of stiffeners on the sound transmission loss of partitions has been of interest for some time, particularly with regard to their use in cavity walls. The paper of (Davy, 1990) is an example which demonstrates this interest. Although it would be expected that light channel stiffeners of the type commonly used with MDF partitions would not have a significant effect on the sound transmission loss of a single leaf partition, and some data supports this, it was decided, as part of a larger study of the sound transmission losses of various MDF partition constructions, to look in some detail at the effects of light channel stiffeners on the sound transmission loss of a single leaf, 18mm thick, MDF partition. The sound intensity technique provides a means of measuring the sound intensity at a point and by measuring the sound intensity at 36 points directly opposite the stiffeners and at 45 points between the stiffeners, information about the relative sound powers radiated from the stiffened and unstiffened portions of the partitions can be obtained. Further, useful comparisons can be made by comparing statistics of the sound intensities measured at the 36 'on-stiffener' points and at the 45 'off-stiffener' points on the stiffened partition with each other and with those at the corresponding points on the unstiffened partition and this was done. These statistics include the means and normalised standard deviations of the intensity levels. The point intensities when integrated over the partition surface enable the sound power radiated from the partition and hence its sound transmission loss to be found and as a means of verification, this sound transmission loss can be compared with that obtained by use of the conventional two room method. It should be noted that other investigators such as (Lai et al, 1993) have conducted detailed studies relating to the use of sound intensity measurements to determine sound transmission loss.

TEST FACILITIES

The unstiffened and stiffened partitions were mounted in turn in the $10m^2$ opening (2718mm wide and 3714mm high) between the $185m^3$ and $200m^3$ reverberation rooms of the School of Mechanical and Manufacturing Engineering at The University of New South Wales. The smaller room is used as the source room and the larger room is used as the receiver room. The partition was located axially in the opening so that the receiver room face of the specimen was 200mm from the plane of the receiver room wall. The sound transmission losses of these partitions were measured in the conventional way according to Australian Standard 1191, Method for Laboratory Measurement of Airborne Sound Transmission Loss of Building Partitions, 1985.

When the sound intensity measurements were made at the 81 points 200mm from the receiver room side of the partition, the large access door of the receiver room was left fully open and several polyester batts were placed in the receiver room to reduce the reactivity of the sound field at the measurement points. This was done to allow the sound intensity measurements to be made as accurately as possible. The measurement plane was in the plane of the receiver room wall.

TEST PARTITIONS

The $10m^2$ test partitions were made of 2715mm long and 1800mm wide sheets of 18mm thick MDF of nominal density 730 kg/m^3. The sheets were joined to each other by routing grooves in the long edges of the sheets and fitting plastic flooring tongues into these grooves to join the sheets together. A 92mm $\times$ 34mm channel formed of 0.8mm thick galvanised steel was bolted to the heavy steel edge plates at the periphery of the $10m^2$ opening. The MDF sheets were screwed on to one of the 34mm long outstanding legs of these peripheral channels. In the case of the stiffened partition the 18mm thick MDF sheets were also screwed to one of the 34mm long outstanding legs of four 65mm $\times$ 34mm channels formed of 0.8mm thick galvanised steel which were in turn attached using clips to the upper and the lower peripheral channels so that they spanned the 3714mm height of the $10m^2$ opening and were nominally 600mm apart as used in partition construction. Caulking compound was used to ensure that any gaps between the MDF sheets and the edge plates of the opening were sealed. Features of the stiffened partition along with the locations of the points at which the intensity measurements were made are shown in Figure 1. The unstiffened partition installation was identical except that the four 65mm $\times$ 34mm channels were not present.

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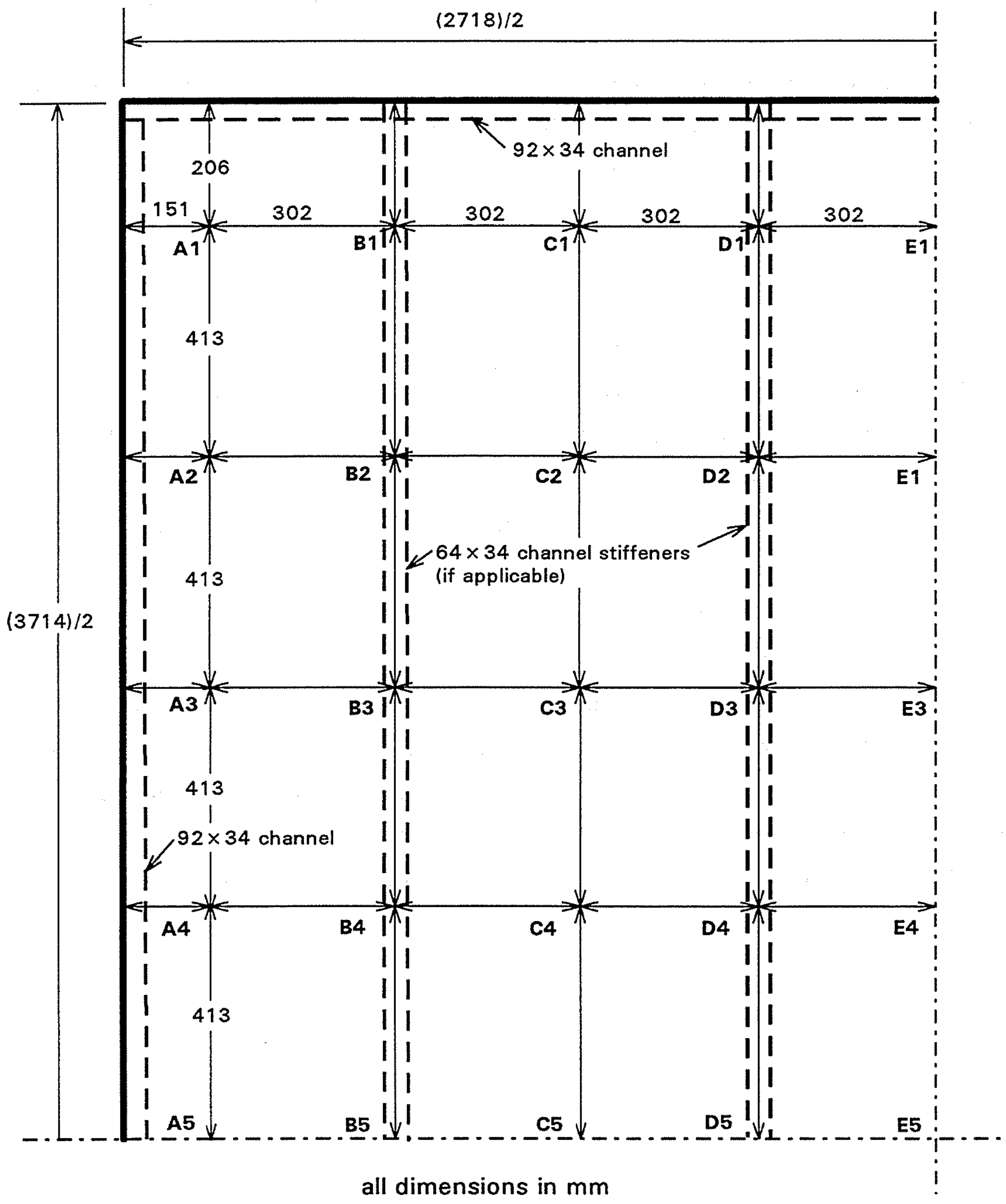
Figure 1 One Quarter of Test Partition Showing Stiffener Positions (if applicable) and Intensity Measurement Locations (A1 to I9).



10 m^2 times the average intensity. The sound transmission loss for the partition then could be determined from the incident and radiated powers and compared with the corresponding result obtained by the standard two room method. The point measurement technique enabled the radiated sound power to be evaluated in accordance with the requirements described in ISO 9614-1 Determination of Sound Power Levels of Noise Sources Using Sound Intensity. The validity of the intensity measurements for calculating the sound power is determined, according to this standard, by evaluating a number of field indicators from the individual intensity measurements and this was done. Results for all one third octave bands including and above 200 Hz were valid for even precision grades of accuracy. The normalised standard deviations for the sets of 'on-stiffener' and 'off-stiffener' intensity data were calculated in each one third octave band to give an indication of the spatial variability of the intensity at the 'on-stiffener' and 'off-stiffener' measurement locations. This quantity is in fact the F_1 field indicator used in ISO9614-1 to assess the temporal variation in measured intensity.

UNSTIFFENED PARTITION RESULTS

Figure 3 gives the sound transmission loss curves of the unstiffened partition derived by the various methods. The results obtained from the two room method are plotted along with the results obtained by the intensity method. Also shown on Figure 3 are the theoretically derived sound transmission losses. The theoretical results are derived from the computational procedure developed by (Byrne, 1989) which is based on the assumption that the panel is of infinite extent and is subject to a diffuse field in the solid cone with a cone angle of 84°. The 18mm thick MDF partition was assumed to be made of a material with a density of 730 kg/m^3, a Young's Modulus of 3 Gpw, a Poisson's Ratio of 0.3 and a loss factor of 0.015. It can be seen that there is good agreement between the three transmission loss curves.

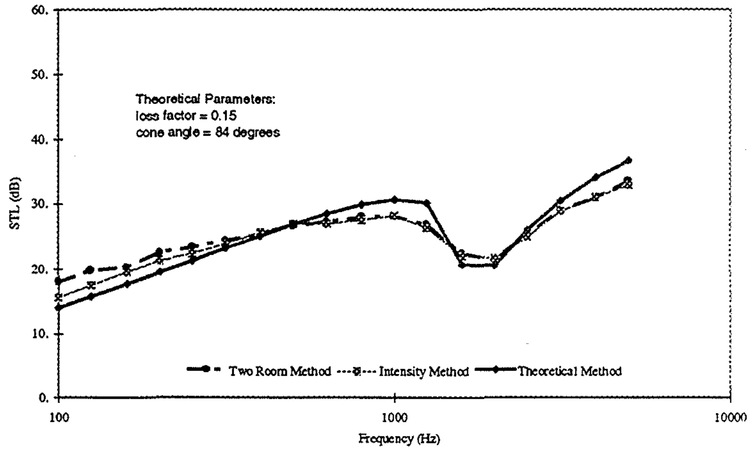


Figure 3 Sound Transmission Losses for Unstiffened 18mm MDF Partition.

λ Two Room Method, v Intensity Method, υ Calculation.

The normalised standard deviations of the one third octave band sound, that is, the F_1 field indicators, are shown in Figure 4 for the 36 'on-stiffener' locations and the 45 'off-stiffener' locations.

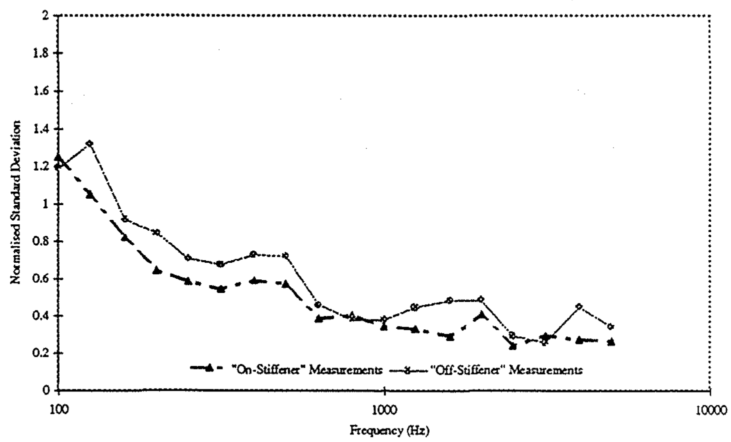


Figure 4 Normalised Standard Deviation of Intensity Levels for 36 'On-Stiffener' and 45 'Off-Stiffener' Measurements on Unstiffened Partition.

STIFFENED PARTITION RESULTS

Figures 5 and 6 give the results for the stiffened partition corresponding to the results given in Figures 3 and 4 for the unstiffened partition.

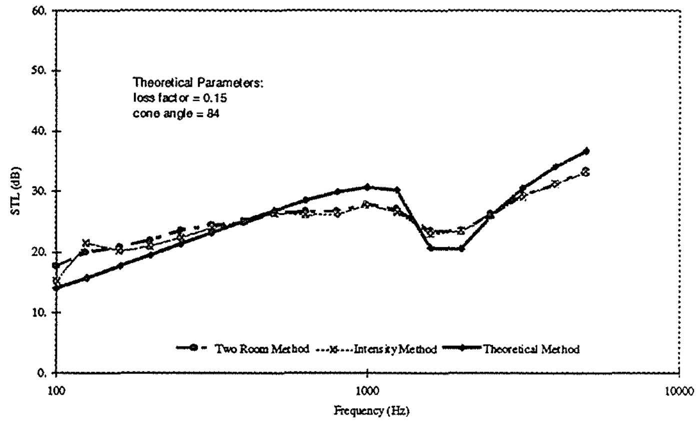


Figure 5 Sound Transmission Losses for Stiffened 18mm MDF Partition

λ Two Room Method, ν Intensity Method, υ Calculation.

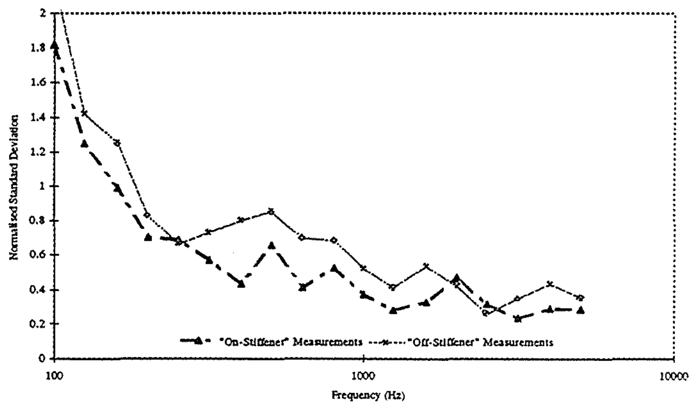


Figure 6 Normalised Standard Deviation of Intensity Levels for 36 'On-Stiffener' and 45 'Off-Stiffener' Measurements on Stiffened Partition.

The sound power flowing through the four 302mm wide by 3714mm long strips associated with the four stiffeners, expressed as a percentage of the total sound power flowing through the panel in each one third octave band is shown in Figure 7. Also shown is the corresponding result for the unstiffened panel.

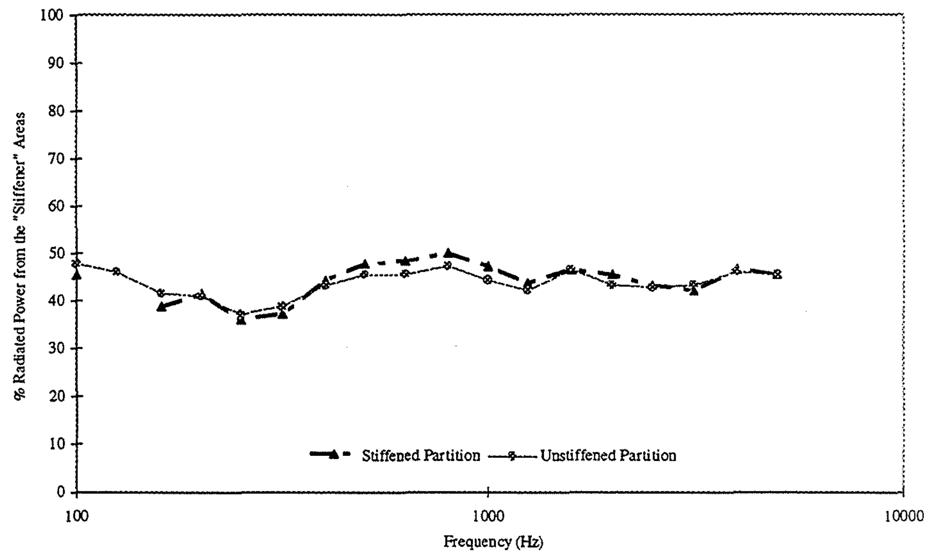


Figure 7 Percentage of Power Radiated from the Stiffener Associated Areas of the Stiffened Partition and the Unstiffened Partition.

COMMENTS ON THE RESULTS

A comparison of the results shown in Figures 3 and 5 shows that the stiffeners in fact have surprisingly little effect upon the sound transmission loss of the single leaf, 18mm thick, MDF partition. It should be noted that the channel stiffeners, being of open section are torsionally very soft and are unlikely to significantly influence the propagation of flexural waves in the MDF sheet in a direction orthogonal to that of the stiffeners. It would be expected that, for the unstiffened partition at least, the percentage of the sound power radiated from the portion of the partition surface associated with the stiffeners would be in proportion to the percentage of area associated with the stiffeners. Since the areas associated with each of the intensity measuring points are equal, the percentage would be expected to $(36/81) \times 100 = 44.4\%$. It can be seen from Figure 7 that in most one third octave bands this percentage is approximately achieved for both the unstiffened and the stiffened partitions.

ACKNOWLEDGEMENT

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EXPERIMENTAL INVESTIGATION OF ACTIVE CONTROL OF SOUND TRANSMISSION THROUGH DOUBLE WALLS

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ABSTRACT

Recently, the active noise control technology has been used to increase sound transmission loss of double wall structures. Several approaches have been put forward and explored individually. However, no comparative study on those approaches has been conducted to show which approach is more effective for given circumstances. In this paper, three different approaches for controlling sound transmission through double panel partitions to a room, ie, applying acoustic control sources in the air gap between the two panels (cavity control), applying vibration control sources on the radiating panel (panel control) and applying acoustic control sources in the receiving room (room control), are studied and compared against each other experimentally. The mechanisms involved in each approach are illustrated and the conditions for effective noise attenuation are examined. The results show that the modal overlap of each sub-system (the cavity, the radiating panel and the room) and the control mechanisms involved are the two most important factors that determine the effectiveness of each corresponding approach.

1. INTRODUCTION

Double panel partitions are widely applied in noise control engineering when relatively high values of the acoustic transmission loss have to be realised with lightweight structures. Examples include mobile office partitions, partition walls in high buildings and aircraft fuselage shells. In the low frequency band, such partitions are however less efficient due to the resonance effect of double panel systems (panel-cavity-panel coupling systems). Since active noise control is mostly efficient at lower frequencies, it is worthwhile to explore its potential in increase of transmission loss of double panel partitions. Recently a number of investigations have been conducted in this regard. These investigations can be grouped into three approaches in terms of types and locations of control sources.

The first approach is referred to as the cavity control in this paper. It attenuates sound radiation into a room by means of controlling the sound field in the air gap (the cavity) between the panels by using acoustic sources in the cavity. This approach is represented by the work of Sas and Bao (1992). They have studied the cavity control theoretically, numerically and experimentally and shown that it is very effective for certain cases. The increase of transmission loss in their experiment was as high as 40 dB. Similar work has also been conducted by Grosveld and Shepherd (1991), and by Gagliardini and Bouvet (1993). In the work of Grosveld and Shepherd, experiments were carried out on a mock-up of an aircraft fuselage. Global attenuation was achieved in a area of 50×50 inch^2 for a single frequency noise by using four control loudspeakers. In the work of Gagliardini and Bouvet, the control of broad-band noise was considered. They were able to achieve an increase of transmission loss of 10 dB over the frequency range between 80 and 160 Hz.

The second approach is active structural acoustic control (ASAC) at the partition panels (referred to as the panel control). In this case, the radiated sound is attenuated by means of controlling the vibration of the radiating panel using force sources mounted on the panel. This approach is represented by the work of Carneal and Fuller (1993). In their work, the control moments, generated by piezoelectric actuators, were applied to the panels to minimise the radiated acoustic field into a room. An increase of 20 dB in transmission loss was achieved for certain cases. A similar work has also been conducted by Thomas *et al* (1992). They achieved a reduction of sound pressure by 40 dB.

The third approach is referred to as the room control that attenuates the radiated sound by means of acoustic sources in the receiving room. Unlike the other two approaches, this approach is often not viewed as a method of controlling sound transmission but rather as an application of active noise control in enclosures or in free space.

So far, those approaches have been only explored individually and the circumstances for their effective application have remained unknown. Thus, the main objective of our study is to have a systematic comparison among different approaches. The study is conducted analytically and experimentally. The result of the analytical study has been presented elsewhere (Pan and Bao, 1996), showing the mechanisms involved in each approach and conditions for their effective application. The result of the experimental study is presented in this paper, where the comparison of the different approaches is made in terms of different criteria such as the frequency band of noise reduction, the change of panel vibration and the noise reduction in the area close to the partition, and of different circumstances such as with and without thermal insulation in the partition.

2. EXPERIMENTAL SET-UP

In the experiment, a double panel partition is mounted in a common wall between an anechoic chamber functioned as the source room and a reverberation chamber as the receiving room. The arrangement of the test chambers and the double panel partition are shown in Fig.1. The volume of the receiving room is 56.25 m^3 and its reverberation time below 200 Hz is around 1.7 seconds.

The double panel partition consists of two aluminium panels of 2 mm thickness, separated by an air cavity of 275 mm depth. The other two dimensions of the air cavity are 2150 mm and 900 mm respectively. The side walls of the cavity are concrete. The two panels are clamped to two heavy steel frames respectively, allowing no vibration transmission between the two panels. The unclamped area of the panels is 2035×780 mm^2. The incident panel is excited by the loudspeakers placed in the anechoic chamber and 1.5 m away from the panel. Once excited, the incident panel radiates energy into the air cavity, thereby exiting the radiating panel, which in turn radiates energy to the receiving room. Unless otherwise stated, harmonic excitation is used in all the experiments described, as it serves better in revealing the control mechanisms involved in each approach.

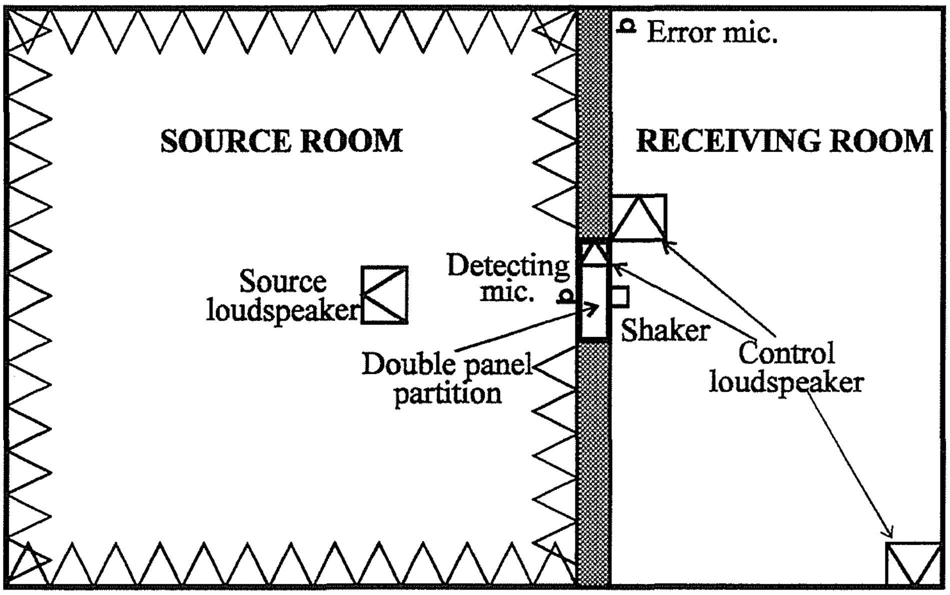

Fig.1. Schematic presentation of the experimental set-up.

An adaptive feedforward controller is used for the control. The reference signal of the controller is taken from a detecting microphone placed in the source room close to the partition, and the error signals from error microphones placed normally in the receiving room. As indicated in the analytical study, the quantitative understanding of the control mechanisms and performance comparison can be obtained from systems with any number of control sources. Thus, for simplicity, a single control source is used in all the experiments except for the case where the effect of thermal insulation is investigated. For the cavity control, the control loudspeaker is placed in a corner of the cavity. In the panel control, a mini-shaker is amounted at the center of the radiating panel as the control source. As for the room control, two locations are selected for the control loudspeaker: one is next to the radiating panel and the other in the far corner of the receiving room. A single error microphone is usually placed in a corner of the receiving room. Occasionally, two error microphones in other locations are also used.

3. COMPARISON IN TERMS OF FREQUENCY BAND OF REDUCTION

In this set of experiments, the performance of each control arrangement is evaluated by measuring the controlled and uncontrolled sound pressure. The sound pressure level (SPL) was measured at 15 selected points distributed over the receiving room and processed into two measures to provide the basis for the evaluation. One measure is the number of SPL increase out of 15 locations, which indicates if the noise reduction is global. It is regarded as global reduction if this number is less than three. The other measure is the averaged reduction level derived from comparison of the controlled and uncontrolled SPL over 15 locations. This measure indicates the amount of reduction.

3.1 Room control

The experiments described in this sub-section is designed to reveal the mechanism of the room control. As suggested in the analytical study (Pan and Bao, 1996), the necessary condition for global reduction of the room control is that the two modal vectors due respectively to the primary (the radiating panel) and secondary (the loudspeaker) sources are mathematically proportional. The above condition can only be satisfied in two situations. One is to place the secondary source as close as possible to the primary source, allowing a similar generation of all adjacent modes. The other happens at those frequencies where the modal overlap is low: ie, the sound field is dominated by a single mode and the contribution of non-resonance modes is small. In the latter situation, it is not necessary to have the secondary source close to the primary source as long as the secondary source can excite the same modes as the primary source does.

In the experiments, both situations are observed. In one of the control arrangements, the loudspeaker is placed just next to the radiating panel (the next-to-source control). While in the other arrangement, the control loudspeaker is in the far corner of the room (the away-from-source control), which enables to excite all the room modes. Table 1 summarises the experimental results. As expected, the away-from-source control is indeed quite effective at 26 Hz where the sound field is dominated by the first room mode. A noise reduction of 15 dB is achieved. Due to the damping of the room and the close spacing between the higher room modes (which results in a higher modal overlap), the sound field after 29 Hz is never clearly dominated by a single mode. Consequently, the away-from-source control is only effective in the low frequency range up to 29 Hz. Whereas for the next-to-source control, since the control loudspeaker is very close to the radiating panel (which allows a similar generation of the sound field up to certain frequencies) the controllable frequencies extend up to 64 Hz. Due primarily to the different nature between the primary (panel) and secondary (loudspeaker) sources and also to the inevitable distance between them, the similarity between the two corresponding vectors is quickly diminished as the frequency increases. Consequently, global control is no longer achievable at higher frequencies, even with the next-to-source control. It is interesting to note the control spill-over at 59 Hz where the modal overlap is not higher than that at 62 Hz. The modal analysis of the room indicated that the room modes that were excited by the panel at 59 Hz could not be properly excited by the control loudspeaker due to its location. This again reveals that the mechanism of the room control is purely the modal suppression.

Table 1. Comparison between the two room control arrangements

Freq.(Hz)		26	29	51	59	62	
Next-to-source	ARL(dB)	**15.9**	**5.3**	**13.9**	**-0.9**	**10.4**	No global control
control	NOI	2	1	1	11	0	after 64 Hz
Away-from-source	ARL(dB)	**15.0**	**4.4**			No global control	
control	NOI	1	1			after 29 Hz	

* ARL: averaged reduction level over 15 locations;

NOI: number of locations where SPL increases.

In summary, the experiments described in this sub-section confirms that the mechanism of the room control *is* the modal suppression. In order to have the maximum control effect, the control loudspeaker should be placed as close as possible to the radiating panel.

3.2 Cavity control

One obvious mechanism of the cavity control is to break the noise transmission path: ie, to attenuate the sound field in the cavity. This reduces the vibration of the radiating panel and consequently the sound radiation into the receiving room. This mechanism relies on the modal suppression of the cavity modes. Another mechanism can also be involved in the cavity control: ie, to change the vibration pattern of the radiating panel through the modal rearrangement of the cavity modes to form a weak sound radiator. This is possible as the vibration pattern of the radiating panel is largely determined by the sound field in the cavity. Our analytical study (Pan and Bao, 1996) has indeed shown that each pressure component in the receiving room can be expressed as the linear combination of the pressure modal components of the cavity. This suggests that the sound pressure in the room can be attenuated by the rearrangement of the cavity modes. Therefore one of the aims of this sub-section is to experimentally confirm the two mechanisms of the cavity control. In addition to the SPL at the 15 locations in the receiving room, the SPL at six selected locations in the cavity and the vibration level at six selected positions on the radiating panel are also measured. In the experiments, four different error microphone arrangements are used. One is referred to as the external sensing where the microphone is placed outside of the cavity, in a corner of the receiving room. The other three are referred to as the internal sensing where the microphones are placed inside the cavity. They include use of a single microphone at two different locations (Internal sensing I and II) and use of two error microphones simultaneously (Internal sensing III). While the internal sensing system is more compact and less sensitive to any change in the receiving room, the external sensing system may have better performance as the error signals are related more directly to the ultimate aim of global noise reduction in the room.

Table 2. Comparison among different arrangements of the cavity control

Frequency (Hz)		51	62	100
External sensing	ARL(room,dB)	**26.2**	**13.2**	**8.8**
	ARL(cavity,dB)	11.6	7.4	2.7
	PVR	yes	no	no
Internal sensing I (single mic.)	ARL(room,dB)	**26.1**	**14.7**	**-2.5**
	ARL(cavity,dB)	13.8	6.4	0.8
	PVR	yes	no	no
Internal sensing II (single mic.)	ARL(room,dB)	**25.2**	**6.8**	**11.3**
	ARL(cavity,dB)	11.0	9.7	0.3
	PVR	yes	no	no
Internal sensing III (two mic.)	ARL(room,dB)	**25.8**	**8.9**	**5.6**
	ARL(cavity,dB)	8.3	9.1	1.4
	PVR	yes	no	no

* ARL: averaged reduction level;
PVR: panel vibration reduction.

Some typical results from the experiments are summarised in Table 2, where the averaged noise reduction in the room as well as in the cavity and the change in the panel vibration are listed for each arrangement. From the experiments, the cavity control is effective in the frequency range up to 102 Hz. This range can be divided into three sub-ranges to show the control mechanisms involved.

In the lower frequency range up to 58 Hz, the global reductions are achieved both in the cavity pressure and the panel vibration (see the result at 51 Hz in Table 2). Thus, the control mechanism involved for the noise reduction in the receiving room is clearly the modal suppression. The modal analysis of the cavity shows that the sound field in this frequency range is controlled by the Helmholze mode. Consequently, the reduction level in the room is not sensitive to the location of the error microphone, and very good reduction in the receiving room is achieved by using both the internal (26 dB at 51 Hz) and the external (26 dB at 51 Hz) sensing.

From 58 Hz to 80 Hz, the global reduction is achieved in the cavity pressure but not in the panel vibration (see the result at 62 Hz in Table 2). This is quite surprising, as it implies that the global reduction of the sound pressure in the cavity does not necessarily lead to the global reduction of the panel vibration. The modal analysis of the cavity shows that the sound field is now dominated by the first cavity mode. The global noise control in the cavity is expected by using a single control source with the modal suppression. As other cavity modes may have some influence on the sound field in this frequency range and the control spill-over effect, the controlled (residual) field is still strong and complex enough to excite the previously-unexcited vibration modes. This explains why the global reduction in the cavity pressure does not lead to the global reduction in the panel vibration. Also because of the participation of other cavity modes, different distributions in the cavity pressure are resulted due to different locations of the error microphone. Moreover, these different pressure distributions correspond to different reduction levels in the room (eg, 14.7 dB reduction of internal sensing I compared to 6.8 dB reduction of internal sensing II, at 62 Hz). Clearly, the result suggests that the modal rearrangement is involved. The control mechanism in this case is really the combination of the modal suppression and the modal rearrangement, with the modal rearrangement playing a more important part. It should be noted that the external sensing system can always achieve a high level of reduction (eg, 13.2 dB at 62 Hz).

From 80 Hz to 102 Hz, the global reduction cannot be achieved in both the cavity pressure and the panel vibration (see the result at 100 Hz in Table 2). For some frequencies, reduction in the room is even not achievable. The modal analysis shows that the cavity field is now dominated by at least two cavity modes. The reduction in the room is clearly due to the rearrangement of those cavity modes, which changes the radiating panel into a weak radiator. As for the internal sensing, unlike in the lower frequency range, the panel vibration patterns with different error microphone's locations can be very different. Consequently, the reduction in the room is very sensitive to the location of the error microphone. The arrangement that produces a similar pressure distribution in the cavity as that of the external sensing yields the largest reduction, while those with different distributions from that of the external sensing achieve no reduction or even increase the noise level in the room.

In summary, the external sensing can always be associated with the most appropriate mechanism for the cavity control over the whole frequency range. While for the internal sensing, only when the residual sound pressure distribution is similar to that of the external sensing can good noise attenuation in the room be obtained. Using more error microphones for the internal sensing can avoid the noise increase in the receiving room, as it ensures only the modal suppression at work. However, the best reduction cannot be obtained by this strategy at higher frequencies where the modal rearrangement is actually required to gain reduction.

3.3 Panel control

It is well known that two mechanisms are involved in the panel control (Pan, 1988, Fuller *et al*, 1991). One is the modal suppression which suppresses the panel vibration thereby reducing the radiated sound. For a single channel control, the modal suppression is often associated with the attenuation of sound transmission near the resonance frequency of a panel controlled mode where the transmitted energy is carried mainly by one uncoupled panel mode. The other mechanism is the modal rearrangement which rearranges the magnitudes and phases of the panel modes thereby change the radiating panel to a weak radiator. This can often lead to an increase of overall panel vibration. The modal rearrangement is associated with the sound transmission into the room controlled modes where the energy is carried by several panel modes.

Table 3. Typical results of the panel control

Freq.(Hz)	26	29	51	62	74	80	85	110	116	132	155
ARL(dB)	**14.5**	**8.2**	**27.9**	**12.7**	**9.9**	**3.2**	**4.7**	**1.8**	**4.2**	**1.1**	**6.1**
NOI	0	0	0	0	0	4	1	5	2	7	3

* ARL: averaged reduction level over 15 locations;
NOI: number of locations where SPL increases.

Some typical experimental results from the panel control are listed in Table 3. It can be seen that the controllable frequency range of the panel control can go up to 155 Hz. This is because the modal density of the panel does not increase as the frequency increases. Therefore, the modal overlap of the panel (the major factor that determines how high the frequency of the modal control can be) depends only on the damping factor which increases with the frequency. The controllable frequency range can further be divided into two sub-ranges. Up to 42 Hz, the global reduction in the room is always achievable, thanks to a low modal overlap of the panel. Beyond that, the reduction can be achieved only at certain discrete frequencies where the modal overlap is low (usually at the resonance frequencies of the panel). In the higher frequency range, the performance is sensitive to the location of the error microphones. The experiment in this regard shows that using multiple error microphones can avoid the problem.

3.4 Comparison among the three approaches

Table 4. Typical results of the three approaches

	Room control		Cavity control		Panel control	
Freq.(Hz)	ARL(dB)	NOI	ARL(dB)	NOI	ARL(dB)	NOI
BB35-55	**7.2**	0	**9.1**	0	**1.6**	4
BB45-85	**1.8**	4	**5.7**	0	**0.2**	7
26	**15.1**	1			**14.5**	0
48	**7.9**	4	**17.2**	1	**15.1**	0
51	**13.9**	1	**26.2**	0	**27.9**	0
55	**11.6**	1	**14.8**	0	**15.5**	0
62	**10.4**	0	**13.2**	0	**12.7**	0
74			**16.2**	0	**9.9**	0
80			**6.0**	0	**3.2**	4
85			**2.4**	1	**4.7**	1
89	No reduction after 64 Hz		**1.9**	5	**2.6**	3
100			**8.8**	1	**2.2**	2
116					**4.2**	2
155			No reduction after 102 Hz		**6.1**	3
					No reduction after 155 Hz	

* ARL: averaged reduction level over 15 locations;
NOI: number of locations where SPL increases;
BB35-55: band limited noise from 35 to 55 Hz;
BB45-85: band limited noise from 45 to 85 Hz.

As demonstrated in the previous subsections, the next-to-source arrangement in the room control and the external sensing arrangement in the cavity control yield better results than other arrangements in the two corresponding approaches respectively. Thus, hereafter when the room control and the cavity control are mentioned, the results from those two arrangements are used for the comparison.

Table 4 summarises the typical results of the three control approaches. It can be seen that the cavity control and the panel control are much better than the room control in terms of global reduction, averaged reduction level and the frequency band of reduction. This can be explained as follows. First, both the cavity and the panel controls are applied to the transmission path, while the room control to the radiated sound in the room. The better global control and larger amount of reduction can be expected when the transmission path is effectively controlled. Secondly, both the cavity and the panel controls work on the basis of two mechanisms of the modal suppression and the modal rearrangement, while the room control only on a single mechanism of the modal suppression. Finally, the modal overlap of the room is much higher than that of the cavity due to the much larger size of the room, and also higher than that of the panel at higher frequencies due to the fact that the modal density of the panel does not increase with frequency while that of the room does. The latter two reasons result in that the cavity and the panel controls can achieve noise reduction in a wider frequency range.

Hereafter, the focus of the comparison will be between the cavity control and the panel control. For this case, the comparison is based on the frequency band of reduction, the averaged reduction level and the effect on the panel vibration.

Although the panel control is effective at higher frequencies (up to 155 Hz) thanks to constant modal density of the panel, global noise reduction can only be achieved at discrete frequencies. On the other hand, the cavity control results in global reduction continuously up to 85 Hz. Thus, the cavity control is effective in attenuations of board-band noise but the panel control is not. In this respect, the panel control is even worse than the room control, as shown by the first two rows in Table 4. This can be explained as follows. The panel control relies mainly on the modal rearrangement as the panel vibration is normally dominated by more than one panel mode at most frequencies. If only a point control source is applied, how well the mechanism works is very sensitive to the source location. While for the cavity control, the cavity pressure is dominated by either the Helmholze mode or the first cavity mode up to 80 Hz. Thus, a single loudspeaker in an appropriate location is able to control the whole frequency range. Although after 80 Hz the loudspeaker location become important for control of the cavity pressure, the mechanism is really about the rearrangement of the panel vibration. As the cavity control applies the pressure force on the whole panel instead of a single point, it allows a single loudspeaker to work for a wider frequency range. In summary, for control of board-band noise or the tonal noise with a variable frequency, the cavity control is superior to the panel control.

The noise reduction level achieved by the panel control is as high as that by the cavity control at controllable frequencies, as both approaches are applied to the transmission path and the modal rearrangement is as effective as the modal suppression.

The panel control relies mainly on the modal rearrangement when the modal density of the panel is relatively high. In this case, the panel vibration level is likely to increase. While for the cavity control, the panel vibration is always reduced in the low frequency range up to 58 Hz, because the modal suppression is the mechanism in this frequency range. Beyond that frequency range, the increase of panel vibration level can often be observed. Thus, in terms of the effect on the panel vibration, the cavity control does better than the panel control at lower frequencies and has a similar behaviour at higher frequencies.

4. COMPARISON IN TERMS OF OTHER ASPECTS

As demonstrated in section 3, both the cavity and the panel controls can achieve better noise attenuation than the room control in many aspects. Therefore, the comparison in this section is conducted only between the cavity and the panel control approaches.

4.1 Near field reduction

The effect of control on the sound field close to the panel is quite important for certain applications such as in aircraft where passengers have to sit close to fuselage (trim panels). To investigate this effect, sound pressure measurements were conducted at a distance of 3, 10 and 36 cm from the panel respectively. For each distance, SPLs were measured at nine points in a 3×3 array on a 60 cm x 120 cm rectangular area.

Table 5 summarises the typical results of the experiments. It should be noted that the global noise reductions in the room were achieved at all frequencies listed in the table. From the experiments, the noise reduction in the area close to the radiating panel (near field reduction) is closely related to the control mechanism involved. For those frequencies where the modal suppression is involved, global reduction can be achieved even at the distance of 3 cm from the panel, as shown by the results obtained in the low frequency range up to 58 Hz in the cavity control. On the other hand, if the control mechanism involved is the modal rearrangement, the near field reduction cannot be guaranteed any more. In this case, the near field reduction is very much dependent on how the panel vibration is rearranged. For some rearrangements (eg, at 62 Hz in the panel control), the near field reduction is still achievable. While for some other rearrangements (eg, at 85 Hz in the panel control), the near field reduction cannot be achieved. In the experiments, the cavity control tends to give a better result when the mechanism of modal rearrangement is involved. For example, at 85 Hz global reduction can be achieved to some extent at the distance of 36 cm from the panel by the cavity control but not by the panel control (in fact, the SPL has been globally increased). This indicates that the rearrangement of the panel modes through the cavity control may be more effective, as it applies the pressure force on the whole panel in stead of at some discrete points. For this reason and also due to its wide frequency range of the modal suppression, the cavity control has achieved global reduction at the distance of 3 cm in the frequency range up to 58 Hz, at 10 cm up to 80 Hz and at 36 cm in the whole controllable frequency range (up to 102 Hz). While for the panel control, even at as low as 26 Hz the global reduction is not possible at the distance of 3 cm. At the distance of 10 cm, the global reduction can be achieved in the frequency range up to 74 Hz. As the frequency increases, even at the distance of 36 cm the global reduction is not possible. The experiments also show that the near field reduction is very sensitive to the location of the control source for the panel control. In summary, the cavity control yields better results in terms of global noise reduction in the near field.

Table 5. Typical near field noise reductions

	Cavity Control						Panel Control					
Freq.	3 cm		10 cm		36 cm		3 cm		10 cm		36 cm	
(Hz)	ARL	NOI	ARL	NOI	AR L	NOI	ARL	NOI	ARL	NOI	ARL	NOI
26							**4.3**	3	**9.5**	2	**14.2**	0
51	**19.0**	0	**19.0**	0	**19.5**	0	**1.2**	3	**8.9**	0	**20.5**	0
62	**11.2**	0	**12.9**	0	**15.6**	0	**7.8**	0	**10.2**	0	**12.3**	0
74	**2.3**	2	**6.1**	0	**11.4**	1	**2.3**	3	**5.8**	1	**7.5**	0
85	**0.3**	5	**3.8**	4	**6.0**	2	**-3.1**	8	**-2.6**	8	**-1.7**	7
100	**3.1**	4	**4.9**	3	**11.0**	1						
116							**2.3**	3	**4.2**	1	**3.2**	1

* ARL: averaged reduction level in dB over 9 locations;
NOI: number of locations where SPL increases.

4.2 Reduction with thermal insulation

In many double wall applications, thermal insulation materials are filled in the gap between the two walls to have a better thermal insulation. The effect of thermal insulation on both the cavity and the panel controls has been investigated experimentally. In the experiments, a 100 mm thick polyester blanket was padded in the gap of the double panels. This kind of thermal insulation material has a density of 7.2 kg/m^3 and is often used in the building industry.

BUILDING CODE OF AUSTRALIA 1990 - IMPACT SOUND INSULATION OF BUILDING PARTITIONS

PAPER 2 - SHORTCOMINGS OF THE CODE IN MEASUREMENT AND INTERPRETATION

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INTRODUCTION

When the Building Code of Australia was produced in 1990, one of its basic objectives was to "ensure that acceptable standards of amenity are maintained for the benefit of the community"[1]. Included in Part F5 of the Code are specifications for the sound insulation offered by partitions from impact sound. Table F5.5 is provided, detailing the construction of three walls which have been deemed satisfactory in reducing impact sound.

In order that a new partition will possess the required impact sound insulation properties, the Code specifies a Test of Equivalence. It then becomes mandatory to test the partition in a laboratory and then to compare the results with those of a Table F5.5 partition. This allows the new partition to be certified as "deemed to-satisfy". The method of testing is described by the Code.

As demonstrated in the first paper, the determination of the impact sound insulation of a partition by use of the Code method has been straight-forward in many respects. It has also shown that these properties can be found by use of a single impact in preference to the tapping machine.

In order that measurement methods might be applied in any suitable laboratory, it will be necessary to show that the interpretation of results is unambiguous as regards the intention of the Code. At the same time the constructor/provider of a partition must be confident that the partition will create an acoustic climate in a receiving room which is amenable to the occupant. This paper will bring to notice several factors in the Code, with respect to impact insulation of partitions, which create difficulties for both the measuring laboratory and the provider of a partition in a real construction.

SHORTCOMINGS OF CODE

1. Testing of a construction

(a) **Location of the impact.** For some uniformity between testing laboratories it is felt essential that the horizontal steel platform bearing the tapping machine be at a pre-determined fixed distance above the lower edge of the sample partition. If this be stated (not currently) then any support to keep the plate horizontal would require adjustments. This would overcome any variations between the constructions of laboratories where the distance between the floor of a source room and the base of the aperture containing the sample differs.

The Code requires continuous contact between the impact plate and the sample. However, plane surfaces of the sample may not eventuate in actual construction. The Code does not take into account some, admittedly unusual, samples where the surface may be quite irregular. These two departures from plane will accordingly reduce the area of the sample actually impacted.

Plasterboard partitions comprise stud constructions. The effect of the impact will then vary between those impacts adjacent to studs and those between studs. It is therefore necessary to specify the number, and perhaps nature, of locations for the steel impacting plate.

The number of microphone locations to detect the resulting sound pressure level in receiving room has not been specified. Alternatively, the number needed is that required to produce a designated 95 per cent confidence interval.

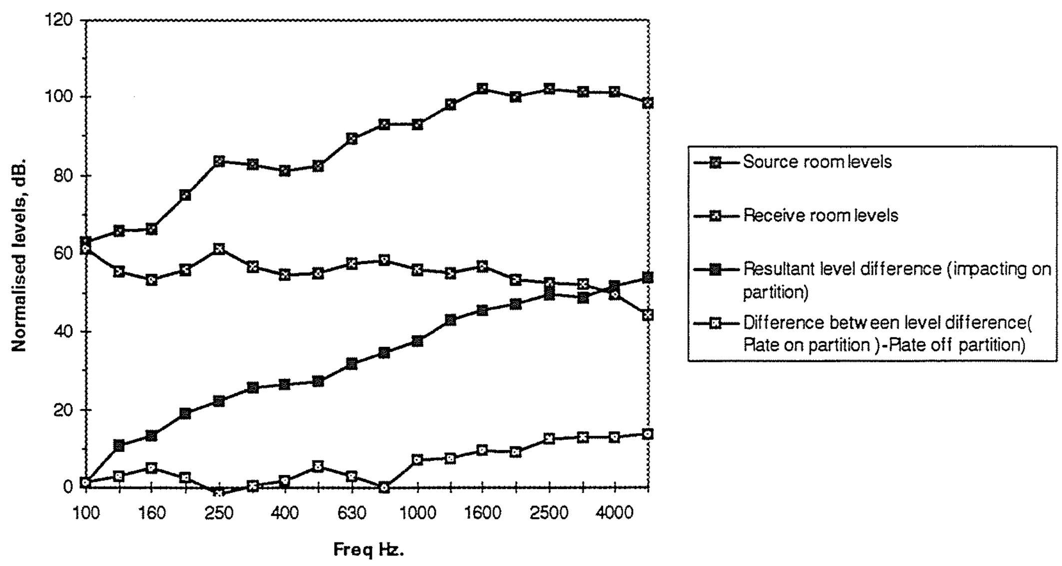
(b) **Nature of the impact.** The tapping machine was first used for the impact on floors to simulate footstep noise, and was surely not intended for walls. Indeed, by placing the tapping machine on a steel plate, the impact produced on a (vertical) partition is not a direct one but one indirectly via the steel plate. Discussion on this aspect is continued under Interpretation.

2. Qualification of measuring laboratory

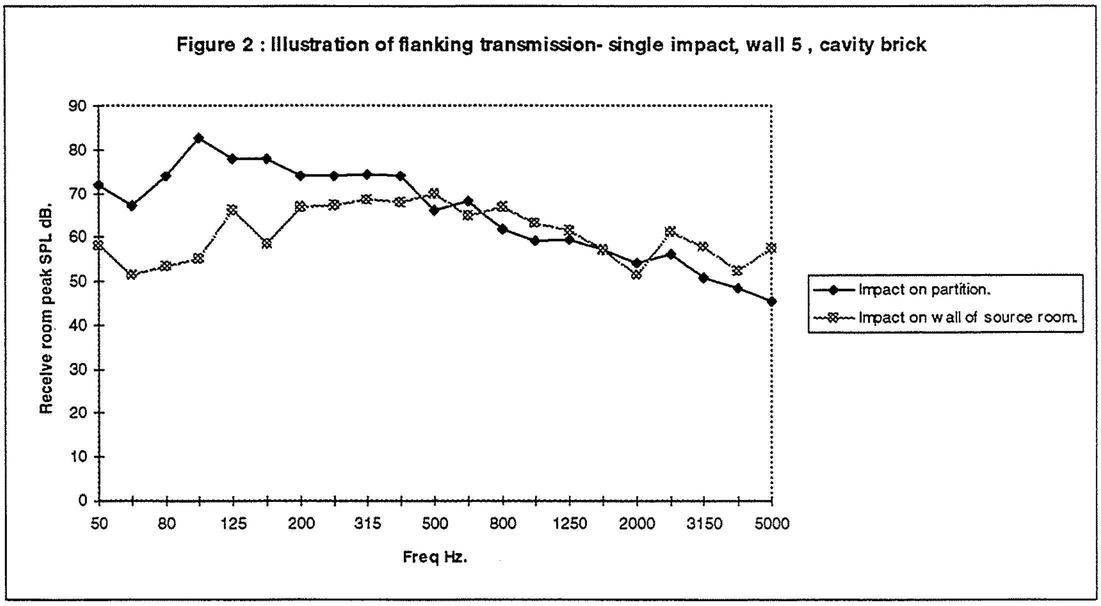
For impact testing the Code requires the laboratory to comply with AS 1191. Measurements have been conducted in such a laboratory which satisfy requirements for flanking noise. Nevertheless, some disturbing results have been produced in impact measurements.

Using the tapping machine, levels have been measured in the receiving room. These measurements are repeated, but moving the impact plate slightly away from contact with the wall. However, the results in the other room did not change significantly, indicated by Figure 1.

Figure 1 : Influence of indirect paths, with tapping machine.



When using the rod to measure the effect of a single impact, results have been given in the first paper. The impact rod is then shifted to make an impact on a wall of the source room, a distance of 3 metres from the aperture containing the wall. The consequent results in the other room, as shown in Figure 2 are quite disturbing. They indicate that impacts are capable of producing significant sound pressure levels in the adjoining room, so there are flanking path problems which have not occurred when the transmission suite has been used for measurement of airborne noise.



Consequently, the validity of measurements of the effect of impacts on a wall, whether from the tapping machine or due to a single impact, is questionable. It will be necessary to carry out extensive work to determine whether the intrusion of sound due to an impact in the source room is a function of the type of wall being tested. Should a dependence be found then it nullifies any attempt to compare a new wall with one of the complying walls shown in Table F5.5 of the Code.

3. Deemed-to-satisfy provisions

Interpretation of "no less resistant". In order to comply with the Code, a new partition is to be no less resistant to impact sound than a wall listed in Table F5.5. In the first place, the frequency range of measurement of impact sound has not been specified. That aside, suppose 18 one-third octave frequency bands were used. The Code seems to imply that the new partition would be deemed Fail if just in one band the normalised sound level receiving room were just 1 dB above a corresponding Table F5.5 partition!

INTERPRETATION OF THE CODE

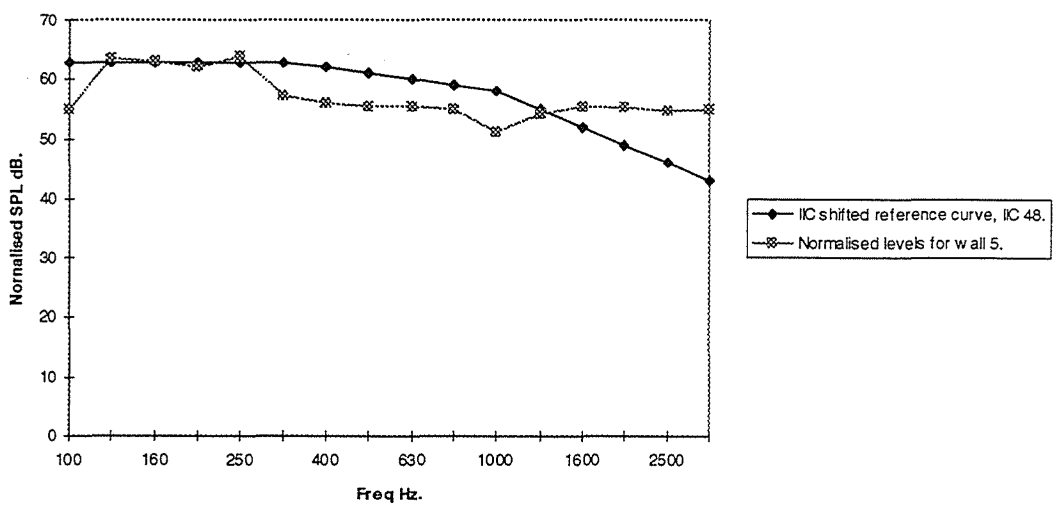
1. Possible single-value of impact insulation. Use of a single number to express the insulating properties of a partition would overcome the puzzle above, in a like fashion to Sound Transmission Class, STC for the properties of a partition for airborne sound transmission. Intended for floors, this single-number term is the Impact Insulation Class, IIC as set out in ASTM E989[2], alternatively as the weighted normalised impact sound pressure level, $L'_{n,w}$ by ISO/DIS 717-2.2[3]. It is emphasised however that IIC and $L'_{n,w}$ are intended for the impact rating of floor-ceiling assemblies, and its determination involves use of the tapping machine in the direct-impact application.

Because of differing exciting mechanisms in the impacting of walls, especially if the impact is a single rather than a repetitive one, the IIC or $L'_{n,w}$ reference curve is not considered the most appropriate one.

By way of illustration, consider a cavity brickwork wall, Wall 5 which is described in Table F5.5 of the Code as being suitable for reducing impact sound. It comprises two leaves of 90mm brick masonry. The results of the measurement of impact sound levels using the tapping machine in the manner stated in the Code, and the variation of normalised impact sound levels in the receiving room with frequency is shown in Figure 3. In the first instance, let us confine the frequency range to between 100 and 3150 Hz.

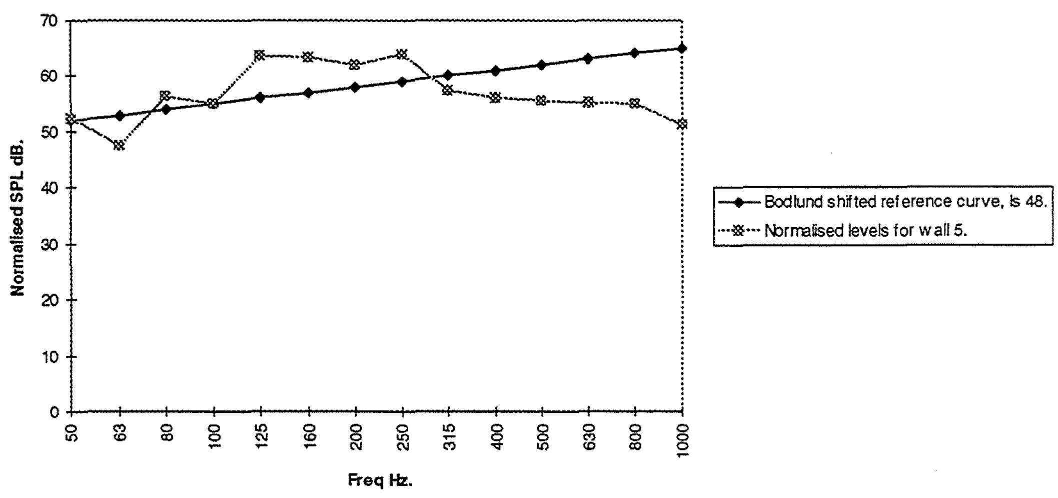
The single-number value is found by subtracting from 110 dB the shifted reference curve value at 500 Hz. This reference curve is also shown in Figure 3. The sum of unfavourable deviations must not exceed 32 dB, but in line with ISO/DIS 717-2.2 a single unfavourable deviation may have any value. The resulting IIC rating is IIC 48. As a comment, the resistance offered by this wall to impact sound is quite poor at high frequencies, alternatively that the reference curve is not of the most appropriate shape for application to walls. This behaviour is found to be similar for other walls tested.

Figure 3 : Impact insulation of cavity brick (wall 5), IIC shifted reference curve.



Work has been carried out by Bodlund[4] on alternative reference curves. Although the work included a study of party walls, the majority of data presented confines itself to floors. However, Bodlund concluded that his modified reference curve yields a new single index, I_S and a better correlation was found between it and the subjective rating of the effects of intrusive impact sound. The important difference for this index is the modification of the reference curve frequency range to between 50 and 1000 hertz, and the shape of the curve. Figure 4 shows the measured normalised impact levels for the cavity brickwork wall stated above. Also shown is the Bodlund shifted, modified reference curve. The single-value I_S for the wall is 48.

Figure 4 : Impact insulation of cavity brick (wall 5), Bodlund shifted reference curve.



As a suggestion then, for a partition to meet the Code, its appropriate single-number value would need to at least match that of the above cavity brickwork wall. The problem remains to adopt the most appropriate type of single-number value for partitions.

2. A more-appropriate type of impact

(a) **Possible single impact.** A possible and more appropriate impact on a wall, explained in the first paper, is the use of the single impact. It would be possible to standardise the mechanism in order that the energy of impact would be known and controllable. Extensive work has been carried out in this respect by the Japanese, using a rubber ball and applied to their standard JIS A418[5,6], though work has been confined to impact on floors. Work has also been carried out by Craik[7], using a plastic-headed hammer.

(b) For other intrusive noises such as plumbing, the first paper has discussed impact noise produced by a hose and tap. Due to difficulties this method would be confined to comparison of values for various walls, which would be applicable to the Code intentions.

An additional complication has arisen in the possible use of single impacts. From work reported, the measurement of the effect of sounds due to and impact due a single event has been measured close to the partition, and as such have not been normalised as with IIC or $L'_{n,w}$ sound pressure levels.

3. Subjective effect in receiving room

The Code assumes that the effect of intrusive sounds in the receiving in terms of the normalised sound pressure levels. However, previous work has pointed out[8] that the effect of intrusive sounds should be in terms of the C-weighted levels. The various terms are discussed by Schultz[9] who raises relevant points: The A-weighted sound levels in the receiving room yield poor correlation with subjective judgements by occupants. He suggests measurement of peak impact levels using impulse characteristic of time constant 35 milliseconds.

There has been later discussion by Kumagai et al[10] on improved techniques on level measurement, while others have suggested the use of C-weighting in the measurement of levels in the receiving room due to intruding noise due to impacts. Nevertheless, there appears agreement that the type of level to be measured depends on the type of intrusion due to impacts. That is, a repetitive type of impact should be assessed differently from one due to a single impact.

CONCLUSION

Much more detail must be provided in the Code in order that is a true code in preference to its deemed current status of just a guide.

It is obvious that a great deal of research work is necessary in order to make the Building Code of Australia a valid one for the provision of walls suitable of providing acceptable standards of amenity for the occupants of adjoining rooms.

NOTES

1. Building Code of Australia, 1990

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TRAFFIC NOISE MODEL FOR INTERRUPTED FLOW URBAN ROADS

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ABSTRACT

This paper presented the analysis and simulation of urban traffic noise model for the roadway network in city center of Bangkok. Road network in the central part of Bangkok which is surrounded by the Gordon line of Rajchadapisek Ring Road was used as the study area of this research project. The mathematical model for traffic noise which was generated from interrupted flow condition of traffic on this road network was presented in this paper. Data for this study was collected from 60 uniformly distributed locations in this central Bangkok road network which included traffic characteristics and traffic noises. The geometrical dimensions of these road sections at the locations of collecting traffic characteristics and traffic noises were also measured. Road traffic noises were measured by using the precision integrated sound level meters in L_{eq} index with A weighting scale of decibel unit (dBA) for one hour period. Characteristics of traffic noise from different type of vehicles on this roadway network were also analyzed. These vehicle types included two popular vehicles in Bangkok, namely, tuk-tuk (motortricycles taxi) and motorcycles in addition to other typical vehicle types of automobile, truck, and bus. Characteristics of traffic noise sources together with data on traffic noise and geometrical dimension of these roadways were then used to analyze and build the interrupted traffic noise simulating model. Statistical analyses were applied for significant test of the model. The results of this interrupted traffic noise model for Bangkok traffic condition and its statistical test for the goodness of fit of the model were finally presented in this paper.

1. INTRODUCTION

Interrupted traffic flow or the so called stop-and-go traffic flow in urban area generates traffic noise, which is different in characteristics and also in the technique for traffic noise modeling from that of the uninterrupted flow or free flow traffic condition which generally appeared on the normal condition of highway and expressway. Therefore, this study project was aimed at the building of interrupted flow traffic noise model for the urban area of the city of Bangkok. So that this model could be used for forecasting and analysis of urban traffic noise together with the investigation for traffic noise mitigation measures for interrupted flow traffic in this capital city of Thailand.

2. STUDY AREA AND MATERIALS

Urban road network in the central part of Bangkok which surrounded by the Gordon line of Rajchadapisek Ring Road was used as the study area of this research project. Data for this study was collected from 60 uniformly distributed locations in this central Bangkok road network which included traffic characteristics, traffic noise, and geometrical dimensions of the cross-section of these roadway locations. On each location, sound level meters were set on both sides of the road. Three periods of time were used for data collection on each location, namely, the morning peak hours period (6.00 a.m. - 9.00 a.m.), the afternoon peak hours period (3.00 p.m. - 6.00 p.m.), and the off peak period. Data was collected for 1 hour during each of these three time periods for both of traffic characteristics and traffic noise simultaneously. This data collection contributed to the total of 360 data sets for being used in the analysis of this research project.

Data on traffic characteristics consisted of traffic volume and composition of traffic for vehicles in the urban traffic stream, namely, automobile, truck (light, medium, and heavy), and bus (mini, and normal). This study also included two more types of popular vehicles in Bangkok, namely, motorcycles, and tuk-tuk (motortricycles taxi). Another traffic characteristics was the average spot speed of traffic vehicles in roadway.

For traffic noise data, it was collected simultaneously with the traffic characteristics data by using precision integrated sound level meters. This traffic noise was measured in L_{eq} index with A weighting scale of decibel unit (dBA) for one hour period at each selected location at the same time of collecting traffic volume, and traffic spot speed. These meters were set at the distance of 1 m. from curb side with the height of sound level meters of 1.20 m. from ground surface.

Measurement of each location's geometrical dimensions were also done in term of number of lanes, lane width, median width, sidewalk width, and curb to facade width. Distance from location of sound level meter to the nearest intersection was measured together with the notification of direction of traffic flow on each side of road's median.

3. CHARACTERISTICS OF URBAN INTERRUPTED TRAFFIC NOISE

Interrupted traffic noise from the stop-and-go condition of urban roads created traffic noise characteristic which was quite different from traffic noise of the free flow or uninterrupted traffic flow on general highway or expressway. This stop-and-go condition came from traffic signals on the urban roadway network which resulted in the deceleration noise and acceleration noise of vehicles when they were approaching red traffic signal at the intersection or leaving the intersection stop line at the green traffic light. These deceleration and acceleration traffic noise were not only different from each other, but also different from the cruising traffic noise in the middle of green light period. This different characteristics through out the noise measurement period of this interrupted traffic noise in the urban area created the great difficulty in building the theoretical formula traffic noise model for this kind of traffic condition. Therefore, most of the research works were done in the direction toward building the empirical or semi-empirical interrupted flow traffic noise model.

4. ANALYSIS AND MODELING

4.1 Approach of the Analysis

This study project, therefore, worked in the direction of formulating the empirical model of interrupted flow traffic noise for the city of Bangkok. Two analytical approaches were investigated in this research, firstly, the single model analysis, and secondly, the two separated models analysis. The analyses and testing of models in these two approaches were described in the following parts of this paper.

4.2 Analytical Parameters

Several parameters which might have an effect on the generating interrupted traffic noise were tested for their correlation with traffic noise level measuring on the sites in this study. These parameters consisted of vehicle volume which were classified into different types of vehicle appeared on Bangkok's roadway, average spot speed of vehicles on traffic stream, roadway width, distance from curb to building facade, and distance to nearest intersection. The data for these parameters was separated into near side roadway parameters and far side roadway parameters in the analysis.

Correlation test was done in order to test for the correlation of these parameters and traffic noise level in L_{eq} and the colinearity among these parameters if there was any. The sets of highly correlated parameters to the L_{eq} were further input to the multiple regression analysis. Step-wise analysis technique was also applied in this multiple regression analysis processes of this study.

4.3 Noise Level from Different Type of Vehicles

Since the different type of vehicles on the roadway created the different noise levels as shown in Figure 1 in the study of noise level from each type of motor vehicles in Bangkok in relating to the spot speed of that vehicle. This study utilized this individual type of vehicle's noise characteristic in order to identify for the proportional weighting scale of noise level generated by a unit of each vehicle type in comparison to that of the automobile unit.

The overall spot speed range from field survey data of this study together with the overall mean value of vehicle speed were superimposed on these vehicle noise characteristics. Since there were too many types of vehicle in Bangkok, therefore, vehicle types were grouped into a smaller number of classifications based on their similarity of noise level characteristics within the speed range which were observed by this study.

From this analysis, the proportional weighting scale of noise level generated by each class of vehicles could be given as shown in Table 1. Therefore, the volume of traffic for being used in the traffic noise model could be described in term of noise generating ratio of each type of vehicle in Bangkok urban traffic in comparison to a unit of automobile noise as the following.

$$\text{Volume of Traffic} = (AU) + 1.04\,(LT) + 1.12\,(MT+TT) + 1.14\,(HT) + 1.09\,(MC+BU+MB) \qquad (1)$$

where :

AU	=	Automobile	HT	=	Heavy Truck
LT	=	Light Truck	MC	=	Motorcycles
MT	=	Medium Truck	BU	=	Bus
TT	=	Tuk-Tuk or Motortricycles	MB	=	Mini Bus

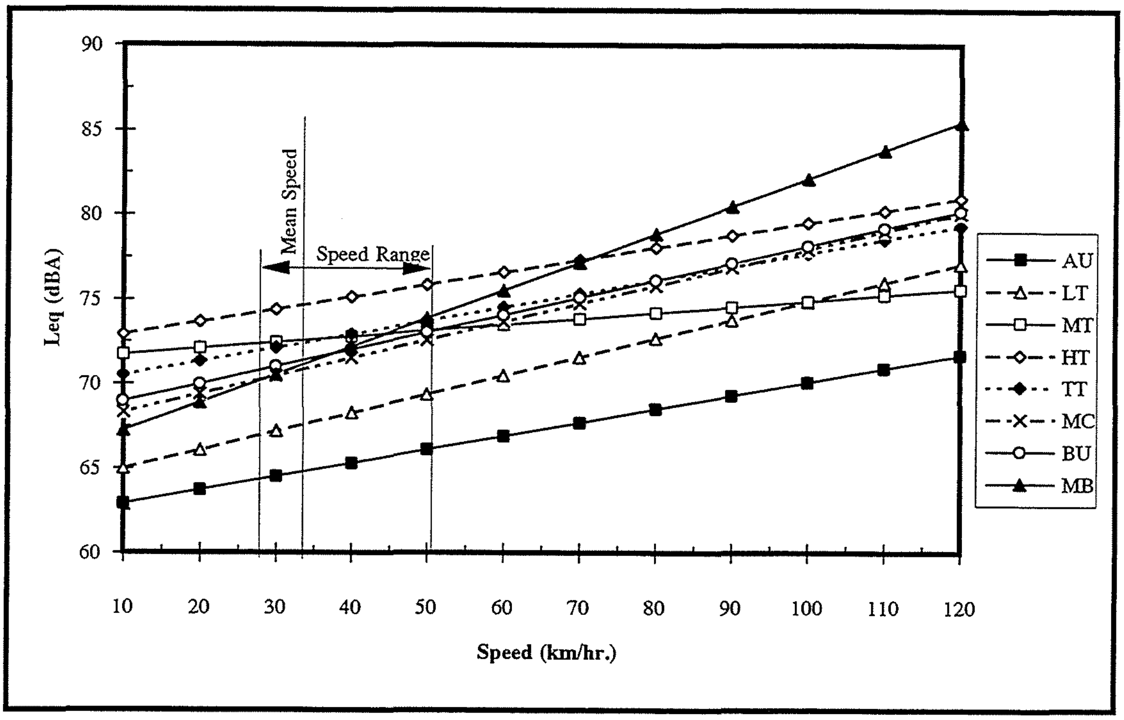

Figure 1. Spot Speed Range and Mean Value of Overall Spot Speed Overlay on the Relationship between Leq and Speed of Different Types of Vehicle

Table 1. Noise Level of Each Class of Vehicle at the Overall Traffic Mean Speed

	Each Class of Vehicle							
Traffic Mean Speed (Km/hr)	**AU (dBA)**	**LT (dBA)**	**MT (dBA)**	**TT (dBA)**	**HT (dBA)**	**MC (dBA)**	**BU (dBA)**	**MB (dBA)**
33.00	65.00	67.50	72.50	72.50	74.60	71.00	71.00	71.00
Ratio to AU	1.00	1.04	1.12	1.12	1.14	1.09	1.09	1.09
Vehicle Grouping	AU	LT	(MT + TT)		HT	(MC + BU +MB)		

4.4 Single Model Analysis

This single model approach was the first analysis applied to build the unique Bangkok interrupted flow traffic noise model which could be applied to both side of the urban roadway. The model analysis was done by utilizing the overall data collected from roadway sections for every parameters in the analysis. Plan view layout of road section for data collection on every parameter and its description for the single model approach was shown in Figure 2.

Sets of highly correlated parameters to the L_{eq} which were measured simultaneously on the road way locations were input into the multiple regression analysis with stepwise regression approach. In this analysis, data sets from both sides of roadway were input into the analysis at the same time in order to build the single unique traffic noise model.

Every parameter and its coefficient which selected from each step of modeling process was also tested for its logic in the traffic and noise characteristics. The final model from this single model analysis approach could be mathematically described as the following.

$$L_{eq} = 71.31 + 0.10\ Sn + 0.91\ \text{Log}\ Vn + 0.04\ Sf + 0.0028\ \text{Log}\ Vf - 0.149\ Dg \qquad \text{\_\_\_\_\_\_\_\_(2)}$$

Coefficient of multiple determination (R^2) = 0.534
Standard Error (SE) = 0.690
Number of Data = 254

where :

L_{eq} = Equivalent traffic noise level in 1 - hour (dBA)
Sn = Mean speed of traffic on near side of observer (both side of road) (km/hr)
Sf = Mean speed of traffic on far side of observer (both side of road) (km/hr)
Vn = Volume of traffic for near side of observer (both side of road) (veh/hr)
Vf = Volume of traffic for far side of observer (both side of road) (veh/hr)
Dg = Geometric mean of road section (m)
= $\sqrt{df \times dn}$
df = Distance from observer to center line of far side roadway (m)
dn = Distance from observer to near side roadway curb (m)

Even though the R^2 value of this model was the highest value which this research project could get from all of the logical single models in the analysis process, it was still not statistically acceptable in term of the low R^2 value. Therefore, this study looked into the other analysis approach of building two separated models on the basis of acceleration lane and deceleration lane models for the urban roadway.

4.5 Two Separated Models Analysis

This analysis approach acknowledged the difference in traffic noise characteristics between acceleration lane and deceleration lane on both side of the urban road when vehicles leaving intersection on the green traffic light and when they coming to stop on the red traffic light at the intersection respectively. The acceleration lane model was built by considering noise level meter was placed on the sidewalk by the side of acceleration lane of roadway when traffic leaving the intersection. Data for near side and far side parameters from all of the data collecting location under this acceleration lane condition was applied to build the model. Every parameter with the potential of generating traffic noise under this condition was tested against the observed traffic noise level in order to see their correlation with the generated L_{eq} in the correlation test, and to see whether there was any colinearity among these noise generating parameters. Sets of highly correlated parameters the traffic noise level were input into the multiple regression analysis with stepwise regression approach in order to analyze for this acceleration lane traffic noise model. Several modifications were applied to the set of parameters, namely, application of log scale to those parameters, and adjustment of the weighting of noise level created by each group of vehicle in comparison to automobile. The logic of every coefficients from the model were also investigated in order to see their logical meaning in traffic engineering and traffic noise characteristics.

In case of declaration lane traffic noise model, the same procedure as previously mentioned were also given by using traffic noise data sets from those roadway locations, but this time with the condition that noise level meter was set up by the side of the deceleration lane. All of the parameters were measured for near side and far side of roadway under this deceleration lane condition.

The plan view layout of road sections and position of each parameter which were used in the analysis of acceleration lane model and deceleration lane model were also shown in Figure 2.

Definition of these parameters could be described as the followings.

A = Location of sound level meter on acceleration side (A)
B = Location of sound level meter on deceleration side (B)
df(a) = Distance from observer (A) to center line of roadway on far side (m)
dn(a) = Distance from observer (A) to roadway curb on near side (m)
df(b) = Distance from observer (B) to center line of roadway on far side (m)
dn(b) = Distance from observer (B) to roadway curb on near side (m)
J = Distance from observer to the nearest junction (m)
Sf(a) = Mean speed of traffic on far side of observer (A) (km/hr)
Sf(b) = Mean speed of traffic on far Side of observer(B) (km/hr)
Sn(a) = Mean speed of traffic for near side of observer (A) (km/hr)
Sn(b) = Mean speed of traffic for near side of observer (B) (km/hr)
Vf(a) = Volume of traffic for far side of observer (A) (veh/hr)
Vf(b) = Volume of traffic for far side of observer (B) (veh/hr)
Vn(a) = Volume of traffic for near side of observer (A) (veh/hr)
Vn(b) = Volume of traffic for near side of observer (B) (veh/hr)

The final model from this analysis for both of acceleration lane and deceleration lane model could be mathematical described as the followings

Acceleration Lane Interrupted Traffic Noise Model :

$$L_{eq} = 56.94 + 0.09\ Sn(a) + 5.20\ Log\ Vn(a) + 0.04\ Sf(a) + 0.029\ Log\ Vf(a) - 0.048\ dg(a) \qquad (3)$$

Coefficient of multiple determination (R^2) = 0.726
Standard Error (SE) = 0.597
Number of Data = 127

Deceleration Lane Interrupted Traffic Noise Model :

$$L_{eq} = 71.10 + 0.075\ Sn(b) + 0.45\ Log\ Vn(b) + 0.085\ Sf(b) + 0.40\ Log\ Vf(b) - 0.048\ dg(b) \qquad (4)$$

Coefficient of multiple determination (R^2) = 0.640
Standard Error (SE) = 0.533
Number of Data = 127

These two separated models of acceleration lane and deceleration lane could provided the high value of R^2 of 0.726 and 0.640 respectively. These R^2 values were higher than that of the single model which meant that all of the independent variables or noise generating parameter of these two separated models could explain a larger percentage of variations of dependent variable or the Leq than those of the single model. All of the coefficient in these acceleration lane model and deceleration lane model also showed the logical meaning in traffic and noise characteristics.

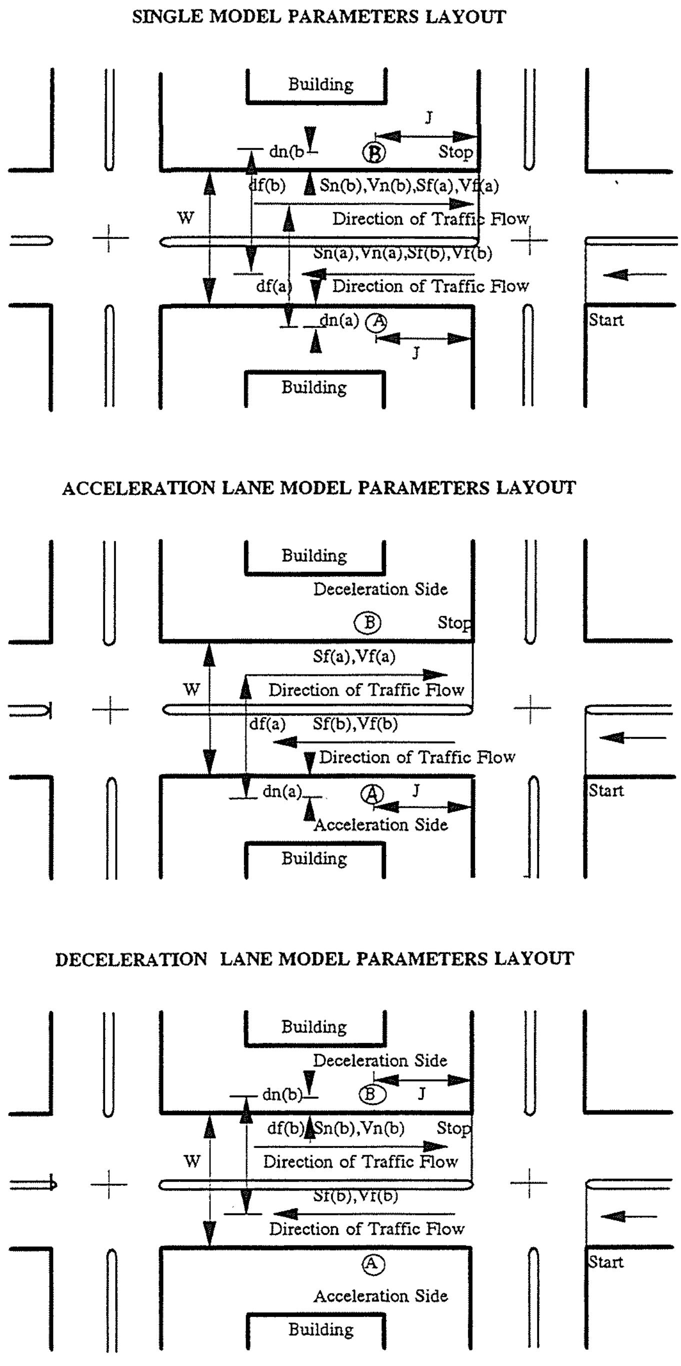

Figure 2. Layout of Road Section and Parameters for Each Type of Modeling Approach

5. TESTING OF THE MODELS

In order to observe how good the model in predicting interrupted flow traffic noise in Bangkok urban area, the statistical paired t-test was applied to test for the goodness of fit of both of the acceleration lane model and the deceleration lane model to the observed traffic noise data on these two lanes condition. The results of this paired t-test for the acceleration lane model and deceleration lane model were shown in Table 2 and Table 3 respectively. These results showed that at degree of freedom 126 and $\alpha = 0.05$, both of the models gave t-statistical values less than that of t-critical value from t-table, which statistically meant that the predicted values from these two models were significantly fitted to the measured ones from the survey sites.

Table 2. Paired T-Test for Separated Acceleration Lane Model at Alpha 0.05

Description	Leq (measured)	Leq (predicted)
Mean	78.926	78.926
Variance	1.255	0.911
Observations	127	127
Pearson Correlation	0.852	
Hypothesized Mean Difference	0	
df	126	
t Stat	-9.072×10^{-13}	
t Critical (one tail)	1.657	
t Critical (two tail)	1.979	

Table 3. Paired T-Test for Separated Deceleration Lane Model at Alpha 0.05

Description	Leq (measured)	Leq (predicted)
Mean	78.896	78.896
Variance	0.759	0.481
Observations	127	127
Pearson Correlation	0.795	
Hypothesized Mean Difference	0	
df	126	
t Stat	-3.942×10^{-13}	
t Critical one tail	1.657	
t Critical two tail	1.979	

6. CONCLUSION

Several conclusion points could be drawn from this study as the followings. From two modeling approaches of the single model analysis, and the two separated lane models analysis, the separated acceleration lane model and deceleration lane model analyses which acknowledged the different characteristics of traffic noise between the acceleration and deceleration conditions of vehicular traffic in the stop-and-go situation could provide the best fit models to these separated lane conditions. These models could be mathematically stated as the followings.

Acceleration Lane Traffic Noise Model :

$$L_{eq} = 56.94 + 0.09\ Sn(a) + 5.20\ Log\ Vn(a) + 0.04\ Sf(a) + 0.029\ Log\ Vf(a) - 0.048\ dg(a)$$

Deceleration Lane Traffic Noise Model :

$$L_{eq} = 71.10 + 0.075\ Sn(b) + 0.45\ Log\ Vn(b) + 0.085\ Sf(b) + 0.40\ Log\ Vf(b) - 0.048\ dg(b)$$

These two models provided the high values of coefficient of determination R^2 of 0.726 and 0.640 for acceleration lane model and deceleration lane model respectively. The R^2 values of these two model were also higher than that of the single noise model which was tested earlier in this project. Both models also gave the highly significant in the goodness of fit test by using paired t-test technique in order to see how good the predicted values from these two models could be fitted to the observed ones. Therefore, these two separated models of acceleration lane and deceleration lane models could be significantly used for the forecasting of interrupted flow traffic noise in urban road network in the city of Bangkok.

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QUEENSLAND NOISE LAW

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1.0 INTRODUCTION

The areas of law in Queensland relating to noise include:

- the Environmental Protection Act 1994 (the "EPA");
- the Noise Abatement Act (1978) (the "NAA");
- the Local Government (Planning and Environment) Act 1990 (the "PE Act"); and
- common law nuisance.

This paper will cover the legislative aspects, and accordingly, will not address common law (judge-made law) nuisance.

The issues to be covered include:

- how the EPA deals with noise;
- the type of EPA licence conditions that have been imposed;
- how the EPA and NAA interact;
- how conditions are imposed on town planning approvals under the PE Act; and
- future directions in noise legislation in Queensland.

Reference will be made to:

- licences issues under the EPA; and
- decisions of the Planning and Environment Court concerning noise.

2.0 THE ENVIRONMENTAL PROTECTION ACT 1994 ("the EPA")

The EPA has two major components:

1. it requires a person carrying out environmentally relevant activities ("ERAs") to hold a licence or approval; and
2. it creates an umbrella concept of the general environmental duty.

2.1 ERAs

There are 85 listed ERAs. These range from power stations to motor vehicle workshops. A licence or approval is required to carry out an ERA.

Conditions are imposed on a licence or approval when it is issued. Conditions generally cover:

- air discharges;
- water discharges;
- stormwater management;
- land application of contaminants;
- noise control;
- waste management;
- self monitoring and reporting;
- approved documents; and
- definitions.

2.2 Conditions

A typical noise condition is:

(F1) The environmentally relevant activity must be carried out by such practicable means necessary to prevent or minimise the emission of noise.

(F2) The emission of noise from the licensed place must not result in levels greater than those specified in Table 1 of the Noise Schedule.

SCHEDULE F TABLE 1

Noise Limits at a Noise Sensitive Place	
Period	Noise level at a Noise Sensitive Place measured as the Adjusted Maximum Sound Pressure Level LAmax adj,T
7am - 6pm 6pm - 10pm 10pm - 7am	Background noise level plus 5 dB(A) Background noise level plus 5 dB(A) Background noise level plus 3 dB(A)
Noise Limits at a Commercial Place	
Period	Noise level at a Commercial Place measured as the Adjusted Maximum Sound Pressure Level LAmax adj,T
7am - 6pm 6pm - 10pm 10pm - 7am	Background noise level plus 10 dB(A) Background noise level plus 10 dB(A) Background noise level plus 8 dB(A)

This table has become the definitive criteria (this was not intended)

often not possible to comply in practice

Definitions commonly included in licences and approvals are:

- "Noise sensitive place" means:

  (a) a dwelling, mobile home or caravan park, residential marina or other residential premises;

  (b) a motel, hotel or hostel;

  (c) a kindergarten, school, university or other educational institution;

  (d) a medical centre or hospital;

  (e) a protected area (being a protected area under the nature Conservation Act 1992, a marine park under the Marine Parks Act 1982, or a World Heritage Area); or

  (f) a park or gardens. —— can't be a buffer.

- "Background noise level" means either:

  LA90,T being the A-weighted sound pressure level exceeded for 90% of the time period not less than 15 minutes, using fast response; or

  LAbg,T being the arithmetic average of the minimum readings measured in the absence of the noise under investigation during a representative time period of not less than 15 minutes, using fast response.

An alternative set of noise conditions that has been imposed is:-

(F4) The emission of noise from the licensed place must not result in levels greater than those specified in Schedule F, Table 2.

SCHEDULE F TABLE 2

NOISE LIMITS AT A NOISE SENSITIVE PLACE	
Period	Noise limits at a Noise Sensitive Place measured as the Adjusted Maximum Sound Pressure Level LAmax adj,T
Monday to Saturday 7am - 6pm	Background noise level plus 5 dB(A)
Monday to Saturday 6pm to 10pm	Background noise level plus 5 dB(A)
Monday to Saturday 10pm to 7am	Background noise level plus 3 dB(A)
All other times and public holidays	Background noise level

NOISE LIMITS AT A COMMERCIAL PLACE	
Period	Noise level at a commercial place measured at the Adjusted Maximum Sound Pressure level LAmax adj,T
Monday to Saturday 7am - 6pm	Background noise level plus 10 dB(A)
Monday to Saturday 6pm to 10pm	Background noise level plus 10 dB(A)
Monday to Saturday 10pm to 7am	Background noise level plus 8 dB(A)
All other times and public holidays	Background noise level plus 5 dB(A)

The value of background noise level to be used in conjunction with the noise limits in Table 2 above is to be obtained from Table 3 below using the following methodology:-

(i) If the measured background noise level is above the upper limit of the appropriate range, the upper limit of the appropriate range must be used.

(ii) If the measured background noise level is below the lower limit of the appropriate range then the lower limit of the appropriate range must be used.

(iii) If the measured background noise level is within the appropriate range then the measured value must be used.

This is a problem for the 24hr - 7 Days a week industries.

SCHEDULE F TABLE 3

Range of volumes of Background Noise to be used in conjunction with Noise Limits in Table 2.

TIME PERIOD	RANGE OF BACKGROUND NOISE LEVELS dB(A)
Monday to Saturday 7.00am - 6.00pm	30-50
Monday to Saturday 6.00pm - 10pm	30-40
Monday to Saturday 10pm - 7.00am	30-35
At all other times and public holidays	30-35

2.3 Legal Effect of Conditions

A breach of a licence condition attracts a maximum penalty of $750,000.00 for a company or $150,000.00 and 2 years imprisonment for an individual.

There are two alternatives when a person cannot comply with a condition:

- reduce the noise to comply with the condition; or
- alter the condition.

2.3.1 Reducing the Noise

A person may enter into an environmental management program ("EMP"). An EMP provides a person with a period of time over which to reach compliance with the conditions.

An EMP to run for more than three years must go through a public notice procedure. This allows third parties to make submissions. These submissions are the basis for third party rights of internal review and appeal against the issuing of an EMP and its conditions. EMPs to run for three years or less are not required to go on public notice.

The benefit of an EMP is that a person cannot be prosecuted for causing unlawful environmental harm if they comply with an EMP. An EMP also provides a person a certain period of time in which to reach compliance.

An EMP can be created by:-

- a person voluntarily lodging a draft EMP;
- the administering authority (either the DoE or Local Government) requiring a draft EMP, such as in a condition of a licence; or
- a person submitting a draft EMP subsequent to lodging a program notice.

A program notice is a statutory notice which is given by a person who has caused or threatened environmental harm. A person is protected from prosecution for a continuation of the offence after the notice is given. A draft EMP must be submitted after a program notice. The person will have the period specified by the administering authority to prepare and submit the draft EMP. This will be a maximum period of three months.

A person loses the protection of the program notice if:-

- the person fails to submit the draft EMP in time;
- the draft EMP is rejected; or
- the person fails to comply with the EMP.

A draft EMP should be prepared with regard to the DoE guidelines.

2.3.2 Altering the Condition

Licence conditions may be altered by:-

- lodging an internal review application; or
- applying for an amendment of a licence.

(a) Internal Review Application

An internal review application is commonly lodged when a licence is issued. It serves as a negotiating mechanism. It also maintains the option of an appeal. In most cases an appeal can only be lodged if an internal review application has first been made.

An internal review application must be:

- made in the approved form; and
- supported by enough information.

An internal review application should be lodged within fourteen days after:

- the day on which notice is received of the decision; or

- the day by which a decision should have been made, where the administering authority fails to make a decision within the prescribed time frame.

A longer period is allowed to lodge an internal review application when "special circumstances" exist. At this time, it is uncertain what will amount to "special circumstances".

Usually a conference is then held to discuss the internal review application. If no decision is made, or the decision is unsatisfactory, then a notice of appeal may be filed in the Planning and Environment Court.

(b) Application to Amend a Licence

A person can apply to amend the licence at any time. The application must be:

- made in the approved form;
- supported by enough information; and
- accompanied by the application fee which is currently $150.00.

If the application to amend the licence is refused, or if no decision is made within the prescribed time, then a person may lodge an internal review application as discussed above.

There are provisions in the EPA requiring public notice to be given of amendment applications in certain circumstances. These provisions have not taken effect. If they take effect, then third parties will have rights of submission and appeal to certain types of amendment applications.

2.4 Third Party Rights

(doesn't have to be neighbour).

The third party rights under the Act are:-

1. rights of submission and appeal about:

   (a) licence applications;

   (b) certain types of amendment applications; and

   (c) draft EMPs to run for more than three years; and

2. **Remedy or restraint orders.**

Only the rights shown above in bold type are currently in force.

The public notice of draft EMPs is discussed above.

Section 194 provides a power for any person to apply to the Court for an order to remedy or restrain an offence or a threatened or anticipated offence against the Act. Section 194 provides safeguards against unfounded actions. These safeguards include:

- the power to require security for costs;

- the power to require a person to give an undertaking as to damages; and

- the obligation to order costs against a plaintiff when the Court is satisfied that the proceedings were brought for obstruction or delay.

2.5 The Environment Protection (Noise) Policy

A draft of the Environment Protection (Noise) Policy ("the Noise EPP") has been released for public submissions. A large number of submissions were made and substantial changes to the draft of the Noise EPP have been made.

A commentary on the draft of the Noise EPP released for public consultation can be found in Volume 2 Issue 8 of the Queensland Environmental Practice Reporter at Page 79. Due to the changing nature of the Noise EPP it is not discussed in detail here.

3.0 THE NOISE ABATEMENT ACT 1978 ("THE NAA")

3.1 Summary of the NAA

3.1.1 When does the NAA Apply?

The operation of the NAA is excluded in certain circumstances. For example section 29 provides that a Local Government cannot competently provide by by-laws (now called local laws) for the abatement of excessive noise emitted from industrial and commercial premises.

3.1.2 Noise Abatement Orders

A noise abatement order may be issued. However, the person issuing the noise abatement order shall consider the following factors, were relevant:-

- the sound pressure level of the noise;
- the type and characteristics of the noise and in particular whether it is a continuous noise at a steady level or whether it is of a fluctuating, intermittent or impulsive nature;
- the frequency components associated with the noise;
- the degree of interference that the noise is likely to cause to the conduct of activities ordinarily carried on on premises other than those from which the noise is emitted;
- the nature of the lawful uses permitted for premises in the neighbourhood of the premises from which the noise is emitted and the date of establishment of particular lawful uses;
- the topographical features of the area in which are situated the premises from which the noise is emitted;
- the number of complaints received concerning the alleged excessive noise;
- other noises ordinarily present in the neighbourhood of the premises from which the noise is emitted;
- if the complaint on which the person required form the opinion is acting has been made by an owner or occupier of premises who has become such owner or occupier at a date subsequent to the date when the noise complained of first came to be emitted, the action taken in relation to such premises to limit the effect of noise emitted from other premises in the neighbourhood.

A right of appeal lies against a noise abatement order. When an appeal is instituted, the effect of the noise abatement order is automatically stayed until the proceedings are determined. In addition, the emission of noise to which the noise abatement order relates may continue in the interim.

3.1.3 The EPA and the NAA

Noise is currently regulated by the EPA and the NAA. The EPA contains more detailed offence provisions and higher penalties than the NAA.

The DoE has an internal policy which provides that when a noise complaint or nuisance is covered by both Acts, then the DoE will use the NAA rather than the EPA.

3.2 Future Directions

It is anticipated that the NAA will be replaced by the Environmental Protection (Noise) Policy (the "Noise EPP"). The Noise EPP has been released for public consultation and significant changes have been made based on submissions that were lodged.

4.0 TOWN PLANNING APPROVALS

Town Planning approvals are regulated by the Local Government (Planning and Environment) Act 1990 ("the PE Act"). Town Planning approvals are usually subject to conditions. The general rule is that the conditions must be "relevant or reasonably required in respect of the proposal to which the application relates".

- In *Mt Marrow Blue Metal Quarries Pty Ltd v. Moreton Shire Council* [1994] QPLR 290 and [1995] QPLR 182, a condition restricted the location of buildings away from a quarry. The dust and noise levels were considered to be acceptable on this basis.

- The interaction of the EPA and the PE Act was discussed by Quirk J. in *Chin and Chee v. Maroochy Shire Council and Another* [1996] QPLR 229.

  Chin and Chee centred around a proposal to rezone lane to the extractive industry zone to allow for sand extraction. Noise impact was one of the environmental issues raised.

  Quirk J stated at page 232:

  "...it was submitted by the Applicant that the Environmental Protection Act *might be regarded as a code in respect of control over environmentally relevant activity and that it might be inappropriate for a Local Authority or the Court to usurp the function of the administering authority by seeking to define and impose environmental control over activities covered by the Act. For the purposes of this case I do not regard it as necessary to decide this point. Both the Applicant and the Respondent are content to see the following conditions imposed upon the rezoning approvals.*

  9. Environmental Management Program

  Prior to commencement of the use, the Applicant (and/or its successors in title) shall comply with all requirements of the Environmental Protection Act 1994 and any regulations, licences, or authorities pursuant thereto which may apply to the development as an environmentally relevant activity within the meaning of the Act. In the event that the administering authority does not require an environmental management program, the Applicant will, in any event, prepare and carry out such a program to the satisfaction of the Chief Executive Officer.

  10. Upon termination of the extractive industry and rehabilitation of the site in accordance with the environmental management program, transfer proposed lot 5 on drawing 3153-10 is offered, to Council for park purposes."

It appears that an environmental management program under the EPA will be required and that this will be an EMP to run for more than three years (assuming that the extraction operations will operate for more than three years). Accordingly, the developer will be required to give public notice of the EMP and third parties will have rights of submission and appeal. This may result in another hearing before the Planning and Environment Court.

This complication could have been avoided by requiring a (non-statutory) environmental management plan, rather than the statutory EMP.

5.0 FUTURE DIRECTIONS

5.1 NAA and the Noise EPP

Finalising the Noise EPP and the repeal of the NAA will rationalise noise laws in Queensland. At this time, the final form of the Noise EPP is not known.

5.2 EPA Prosecutions

There have been prosecutions under the EPA for:

- pollution of a creek by stormwater runoff (conviction and fine of a company);

- pollution of a stormwater drain by fuel (conviction and fine of an individual); and

- air pollution (decision pending)

It is reasonable to expect that prosecutions for noise pollution are not far away.

5.3 Town Planning

Integration of town planning approvals and environmental authorities are expected under the new planning legislation. In the meantime, it appears that the Planning and Environment Court is leaning towards maintaining PE Act and EPA applications as separate procedures.

6.0 SUMMARY

In summary:

- Noise conditions imposed under EPA licenses are generally onerous.

- Mechanisms are available to meet compliance with licence conditions or to alter licence conditions to be more achievable. These mechanisms should be adopted prior to a person being prosecuted for noise emissions.

- The Noise EPP and the new planning legislation will have a substantial impact on noise law in Queensland when they take effect.

NOISE IN NEW ZEALAND LEGISLATION AND STANDARDS - AND A CASE FOR EVOLVING THE NZ NATIONAL BUILDING CODE

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ABSTRACT

Noise as used in NZ standards and legislation appears to have an implied rather than a defined meaning. Without a firm definition the aims with respect to noise are not clear and precise. A proposal is made for a definition which allows a distinction to be made between different categories of noise and emphasises the need to distinguish between public and individual "health". The goal of the NZBC is investigated and the suitability of laboratory determined STC ratings of building components is questioned as a means of acheiving this goal. From a review of the trends in building acoustics regulations overseas a proposal is made for developing Section G6 of the New Zealand building code and instituting a quality rating for the acoustical performance of NZ residential buildings. Finally we report briefly on a project to investigate possibilities for a "friendly" source for conducting field measurements in buildings.

1. INTRODUCTION

Several acts of Parliament make reference to NOISE [1,2,3,4]. In addition there are four NZ standards in the 6800 series which mention NOISE [5,6,7,8]. Although it is common practice to begin Acts and Standards with a list of definitions none contains a definition for noise. It could be that the quantity is considered to be self evident but on reading the texts and considering the content of associated guidelines (e.g. the guidelines to the old Noise Control Act [9]) it seems that the more likely explanation is that the uses made of the term are so wide that a precise definition is far from obvious.

1.1 Noise and Compound Terms with Noise.

1.1.1 In Standards

The first version of NZS 6801 prepared in 1977 was entitled *Methods of Measuring Noise*. In the 1991 revision the title was changed to *Measurement of Sound* If my memory serves me well this change was not prompted by the potentially embarassing claim made in the *Guidelines to the Noise Control Act 1982* that "noise cannot be measured"[9] but by the realisation that it was not obvious that all sounds were, or were going to create, noise. The definitions list in NZS 6801:1977 carried the following -

NUISANCE,ANNOYANCE OR INTRUSIVE NOISE. Noise which is undesired by the recipient - and Part 2 of the text was dedicated to "Measurement of Nuisance Noise". It is a point to note that in this definition the focus is on the individual recipient whereas in the Measurement part the emphasis is on community response.

The revised NZS 6801:1991 rather pointedly attempts to remove 'noise' from its text but it still creeps in in various places. There is no discernible pattern to the places where it has still been allowed. These same comments apply to the 1991 revision of NZS 6802:1977 *Assessment of Noise in the Environment.*

However the term is back firmly in place in the 1992 Standard - NZS 6805:1992 *Airport Noise Management and Land Use Planning* even to the extent that where the standard makes reference to NZS 6801:1991 and NZS 6802:1991 their titles revert back to their superceded 1977 versions using the term Noise instead of Sound. This is perhaps understandable given that in the interim the Resource Management Act 1991 (RMA) had appeared which, with its unequivocal use of noise, recalled us back to a less discriminating usage!

In a variety of places the approach taken in NZS 6802:1991 indicates that its primary concern is the likely community reaction to sound rather than the plight of individuals (see comment C3.1.4) but it lists that the possible (negative) assessments of "noise"are that it is "unreasonable", "excessive" or "offensive or injurious to health" (later referred to as being a "nuisance").

1.1.2 In Legislation

The Noise Control Act 1982 (now repealed) provided a list of "Interpretations" (undoubtedly a term used to avoid the tighter requirements involved if the term "definitions" had been used instead). The interpretations involving noise were transferred (with slight modifications) to the RMA 1991 and we now find the following -

> "NOISE' includes vibration.
>
> 'EXCESSIVE NOISE' means any noise that is under human control and of such a nature as to unreasonably interfere with the peace, comfort and convenience of any other person (other than a person in or at the place from which the noise is being emitted)

1633.

the term "excessive noise" may include any noise emitted by any -

(a) Musical instrument; or
(b) Electrical appliance; or
(c) Machine, however powered; or
(d) Person, or group of persons; or
(e) Explosion or vibration. "

Section 16 of the RMA also identifies another type of noise (presumably somewhat lesser than excessive noise) when it speaks about "unreasonable noise". This section requires that occupiers and users of land "adopt the best practicable option to ensure that the emission of noise from that land does not exceed a reasonable level".

The implication is that there exists a condition of 'Reasonable noise'. The RMA gives no guidance as to what comprises this condition but the Guidelines to the Noise Control Act (presumably having no official standing now that the act has been repealed) try to give some help by including a section entitled 'WHAT IS A REASONABLE LEVEL OF NOISE?'. There is, however, a subtle but very significant difference between 'reasonable noise' and 'reasonable level of noise'. The latter suggests a quantity which when measured on a single dimension (i.e. its LEVEL, - a technical term used to indicate a measure expressed in dB's) can be designated to have reasonableness based on the size of that parameter. The former is less preordaining.

Key points from this section of the Guidelines are -

- noise is defined as "unwanted sound"
- reasonableness is claimed to vary from 'case' to 'case'
- whilst the implication appears to be that what is being assessed are the effects on <u>people</u> the factors that Noise Control Officers are directed to consider are all <u>objective</u> factors, and a clear assumption is made that what is to be assessed is the degree of effect from a public (i.e. a population mean) point of view rather than an affected individual's.

The "unwanted sound" definition reappears in the Draft Code of Practice for Management of Noise in the Workplace [10] and this is one place where we can see the difficulties posed by a simplistic definition. The Code of practice carries on to define the quantity "Noise exposure" and there we find it has been necessary to revert to using the word <u>sound</u> in place of <u>noise</u> so that we have -

" 'Noise Exposure' means the amount of sound energy a person is exposed to during a representative day" otherwise the definition for noise, if it were used in this second definition, would potentially allow an exemption from the requirements of the act for those workers and employers who might claim that what they were exposed to was <u>wanted</u> sound.

The Draft Regulations for the Management of Noise in the Workplace [11] make mention of 'excessive noise' but since they are only concerned with the impact on hearing acuity within the workplace then this is clearly not the same 'excessive noise for which provisions are made in the RMA. A further significant difference (cf the approach in the old Noise Control Act - and therefore that also of the RMA - and the approach in NZS 6802:1991) is that the Guidelines to the Health and Safety in Employment Act [12] make it plain that the intention is the protection of all individuals from "hazard" not just the setting of a standard suitable to protect a mean population (i.e. protection suitable for those demonstrating an average susceptibility to damage)!

1.2 The Status of Noise

We might fairly say that no formal definition of Noise has been given. The only place one appears is in the non-mandatory Code of Practice for Management of Noise in the workplace which presents the view that "noise" means unwanted sound. Since there is a need to distinguish at least the three types of noise dealt with in the Standards and Acts discussed i.e. "reasonable noise", "unreasonable noise" and "excessive noise" any definition must sit happily with these uses and whilst it may seem fair that "unwantedness" could be quantified to allow distinctions between "unreasonable" and "excessive" it does not seen fair that a level of "unwantedness" should be adjudged "reasonable".

1633 In view of the current effort to have common standards for Australia and New Zealand it is relevant to look at Australian usage. Australian Standard AS 1633-195 [13] presents a Glossary of Acoustical Terms and the following appear amongst the entries -

" NOISE -

(a) Sound which a listener does not wish to hear

Notes: 1. 'Noise' is applied as a prefatory modifier for many terms used in noise control, a branch of applied acoustics, e.g. noise exposure. For some of the quantities it would be equally acceptable to use the more general modifier 'sound'.

2. Undesired electrical disturbances in a transmission channel or device may also be termed 'noise' in which case the qualification 'electrical' should be included unless it is self-evident.

3. The use of 'noise' to include sound of all kinds is deprecated in acoustics.

(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.

Note: Modifying words or phrases may be prefixed to the term 'noise' to suit the particular conditions

NOISE EXPOSURE - see Sound Exposure

NOISE LEVEL - A term used in lieu of sound level when the sound concerned is being measured or ranked for its undesirability in the contextual circumstances.

Whilst these entries show that thought has been given to covering the range of uses for 'noise' there is no immediate help towards a logical distinction between "reasonable", "unreasonable" and "excessive" noise. However, it is clear that there is accord with New Zealand over the need for some value judgement, where listeners are involved, in order to decide desirability or wantedness.

2. Sound

Our sense of hearing provides a subjective sensation which is central to life in allowing communication, enjoyment, emotional expression, warnings, etc. and it is because of this (and because of the potential for irrelevant excitations of our ears to intrude upon these desirable activities) that we are concerned by those features of our environments which will promote the desirable sensations and inhibit the undesirable.

These sensations result from our ears being worked. This requires energy and it is a flow of energy through the air - provided it has certain properties (e.g. suitable strength and falls within a suitable frequency range) - that normally works our ears. Frequently both the sensation and the flow of energy in the air are referred to as SOUND. This lack of a clear distinction between the two often results in difficulties. We can resolve this by distinguishing two types of sound -
Objective sound (an energy flow in the form of waves in a material) and **Subjective sound** (the sensation when our ears receive appropriate objective sound). It is the objective sound we manipulate and control with our materials etc. both inside and outside buildings but we must keep in mind that it is the resulting subjective sound that is the only reason for bothering! Always the subjective sound must guide our efforts, and it is crucial to keep in mind that the set of measures and descriptions applied to objective sound do not directly describe the corresponding subjective sound.

3. A Proposal for a meaning for Noise.

We can only benefit if we set ourselves to produce a scientific definition for noise. If this ends up differing from lay usage then we need to be clear on that but not necessarily alarmed. We have little difficulty in finding examples of where scientific definition differs from lay usage - and even examples of where the two are diammetrically opposed (consider, for example, the lay sense of 'elastic' and the scientific definition of 'elasticity'). However, it will benefit instinctive understanding if this an open definition (i.e. consistent with uses in other domains and disciplines).

My own contribution - to start the debate - derives from considering the way noise is used in the term '**signal-to-noise ratio**', and considering the difference in meaning of our two common verbs - **to listen** and **to hear**. S/N
Signal denotes a message or communication which we are attempting to attend to, but at the same time we receive another input which also we perceive but which does not contribute to the message i.e. the noise. If we restrict

ourselves to talking about sound and make a necessary distinction between objective sound and subjective sound then *signal and noise are not fundamentally distinguished in the objective sound domain but are strongly differentiated in the subjective sound domain.* I thus arrive at a definition for noise as sound from which we choose not to extract an audible communication. In other words **noise is subjective sound which is merely heard rather than listened to.**

obj
subj.

An act of will then allows us to take the same objective sound input and swap it between being a 'desired audible communication' (signal?) and being a 'noise' by choosing whether or not we direct our attention to it or its content. Importantly, this then allows that 'noise' can describe sounds which are desirable (e.g. the 'noise' used in masking systems or in audiological testing). Also it allows us to make a logical distinction between 'reasonable', 'unreasonable' and 'excessive' noise on the basis of the degree of effort required to render it 'merely heard'. (The relevant 'effort' to be assessed of course, is that of the exposed persons and not some hypothetical 'reasonable' or 'mean' person'. In my view the reasonable person approach is akin to establishing a single size for clothing production based on the dimensions of the 'average' person.)

Where the distinctions are required for planning purposes then it is appropriate to use politically determined figures for the degrees of effort that separate the noise categories, but where it is a specific case involving particular individuals then judgements should be based on the individual capablilites ansd sensibilities. This points to a need for research to establish techniques for measuring this effort and for setting values to use for planning purposes.

4. Would a new term be helpful?

One thing that strikes as somewhat unbalanced in all this is that we have the term 'Noise' without there being a complementary term for the sounds which are not noise. In the lay mind it is probable that 'Sound' stands opposed to noise but given the ease with which we also find these two terms being used synonymously this can only feed the confusion when we are concerned - e.g. in Planning Tribunal hearings - to distinguish the two states. The difficulties are further compounded by the lack of generally accepted terminology for making the distinction between stimulus and sensation and appreciation of its importance. In the above, the terms *objective sound* and *subjective sound* have been adopted to resolve this. Further, *noise* and *non-noise* are suggested as being distinguished on the basis of whether the objective sound is merely *heard* or *listened to*. Following the well established precedent in English of coining new words from related Latin or Greek roots we have the possibility, if we so choose, of creating a new term for our use from the Greek word meaning 'to listen'. The Greek for 'to hear' - akouein - has already entered our language and given us 'acoustic' and its derivatives. The word 'acroamai' meaning 'to listen to' is far less familiar although derivatives of it can be found in English dictionaries [14]. This verb root would logically give us the noun 'acroama' or perhaps '*acroma*' for something listened to - i.e. a non-noise subjective sound. (It is perhaps worth noting that the Romans adopted the word "acroama" [15] to describe an 'item in an entertainment', something listened to 'with pleasure'.). A parallel from another sense dimension i.e. that of smell, gives a useful example for a precedent - i.e. aroma - expressing a subset with non-neutral values.

acroma

Noise: Subj sound which is merely heard rather than listened to.

5. The New Zealand Building Code

The Building Act 1992 established the the National Building Code and provides us with a particular example of the use of 'noise' in legislation. Noise transmission is one of the concerns addressed in the performance requirements detailed in section G6 of the Code [16]. The stated "objective" is-

> "to safeguard people from illness or loss of amenity as a result of undue noise being transmitted between abutting occupancies".

This objective is followed by clauses which detail the "Functional Requirements" and the level of "Performance" to be provided in terms of objective measures. Undoubtedly the objective would be seen by most people as highly laudable but it is not clear that it provides what our population might be expected to demand i.e. satisfaction with the acoustic performance of buldings.

If we support the Code's provisions then it necessarily means that we know the meaning of "undue noise", that we can distinguish when people are being caused to suffer "illness" and "loss of amenity"and that we have evidence that the "performance" specified achieves the protection stated. A simple test serves to establish that this cannot be and that therefore we cannot support the code in its present form.

The wording "to safeguard people" makes no exceptions (there is no qualification to restrict the situation to the 'reasonable','median' or 'average' person) and so the population extremes must be provided for. No one could claim to know what these extremes are and hence the performance provisions have no foundation.

Nonetheless, the fact that the Code presents a beginning for New Zealand in promoting some quality minima for acoustic performance is to be warmly welcomed. Further the fact that it is demonstrably incomplete and has limited application has been a boon for designers and materials manufacturers allowing them a gradual lead-in with time to undertake preformance tests and to develop and optimise systems to the required performance.

5.1 Inadequacies to be addressed

(a) Although it is probably true that the residents most at risk from neighbour noise are those living in buildings with some part of the structure being common to the two dwellings, **the protection for residents of other dwelling types needs addressing** also.

(b) The exposure of affected residents results from the sum total of sound entering the dwelling and this is affected by the total area of radiating surfaces plus any transmission that flanks the separating walls and floors (as well as the inherent insulating ability i.e. TL or impact insulation of the component walls and floors). So **what should be assessed is the insulation acheived between the dwellings in the completed building** rather than the insulation per unit area of the component constructions. **Even more relevant would be a rating of the actual sound to which residents are exposed.**

(c) The wall performance requirements are given in terms of STC ratings. The STC dates from an era before the ubiquitous home Stereo and before it became common practice for certain groups within society to have amplified music or TV sound as a continuous background to home life. These provide spectra significantly different from the speech-like spectrum presumed for the development of the STC comparison contour (see Figure 1) and, in particular, they contain significant energy at low frequencies below the range of STC (and where the laws of acoustics deem the insulation of partitions to be at their weakest). Recent overseas commentators emphsise that most disatisfaction arises from the low ferquency transmitted sound below the range of the STC rating [17]. If therefore the rating of components continues as part of the building code then **an extension to, or addition to, the STC rating is required.**

for good LF required (problem)

(d) There is anecdotal evidence [18] to suggest that we must expect the same nominal constructions to give different performances in different buildings. This is likely to be caused by differences in builder technique, panel size, detailing, contingent walls and their rigidity of connection, penetrations for services etc. If the code retains its focus on component performance **it is desirable in view of this potential variability that field testing replaces laboratory testing** as the basis for certification of a product or particular type of construction.

(e) Somewhat similar observations may be made about the Tapping Machine impact tests and IIC rating as a complete indication of the performance of floors (overseas concern has been evident for some while that body impacts from children playing even when there are thick floor coverings and IIC values are very high are transmitted at annoying levels[19]). **A total omission however is the absence of a rating for horizontally (i.e. wall) transmitted impact and equipment sound.**

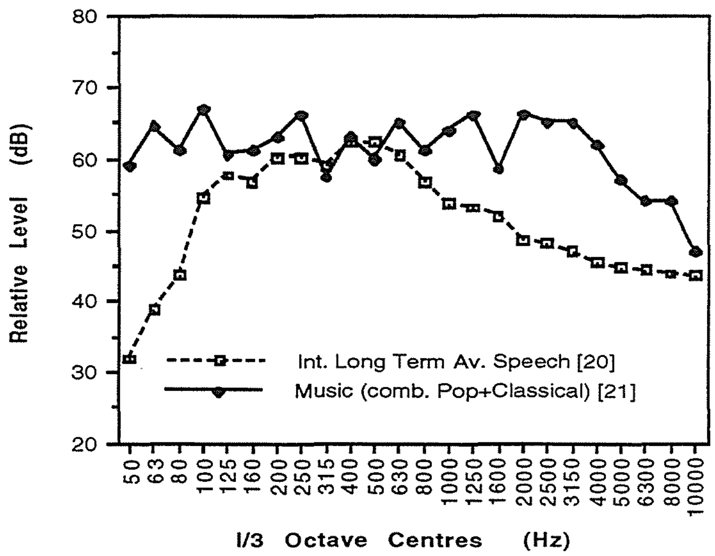

Fig 1: A comparison of the New International Long Term Average Speech Spectrum (ILTASS) [20] with a combined Music Spectrum [21]

(f) The code gives the impression that a single standard is all that is necessary. We are dealing here with a question more of comfort than a black and white issue like "will the building structure stay up or not?". **More appropriate than a single standard would be a range of levels of acoustic quality** which the resident at the time of design or purchase could select from on the basis of their sensibilities to sound. Since levels of comfort are likely to be a matter of taste, sensitivity, and past experience consideration could be given to an assessment based on a detectability criterion rather than degrees of intrusion.

(g) An increasing number of dwellings are under more severe exposure from traffic noise than from neighbour noise. Evidence from overseas surveys shows that traffic noise is the major source of pollution complaints (France for example has introduced in its new regulations a performance requirement for dwellings for the amount of insulation the facades give against traffic noise[22]). The potential for effects on health and for "loss of amenity" must be as great as from neighbour noise (even though there is some evidence that the weightings may be different) so **it is desirable to include exterior wall insulation in our code.**

(h) An implication of the code is that between-dwelling insulation is the only significant concern. However, the amenity and the quality of the dwelling may be strongly affected by a lack of sufficient within-dwelling insulation, especially where the house is well separated so that there is no significant transmission from neighbours. If the typical range of family sound-producing activities cannot be simultaneously accomodated then the dwelling has an

→ graded quality rating

inadequate quality for that family. **It is desirable therefore to establish a protocol for rating the within-dwelling insulation.**

5.2 Possible modifications and avenues for investigation.

1. The STC rating could be modernised by an ajustment to the comparison contour to account for improved knowledge about speech spectra and dynamic range, and also the provision of additional contours to rate partition performance against music and traffic noise spectra. As Figure 2 shows the current requirement of STC 55 will not ensure non-audibility for raised voices and music at strongish (but not loud) levels. Overseas the trend is for setting STC 60 as the minimum (and IIC of 62) when tested in the field [23]. This may be rather demanding for our lightweight timber type of construction but we should aim for comparable standards.

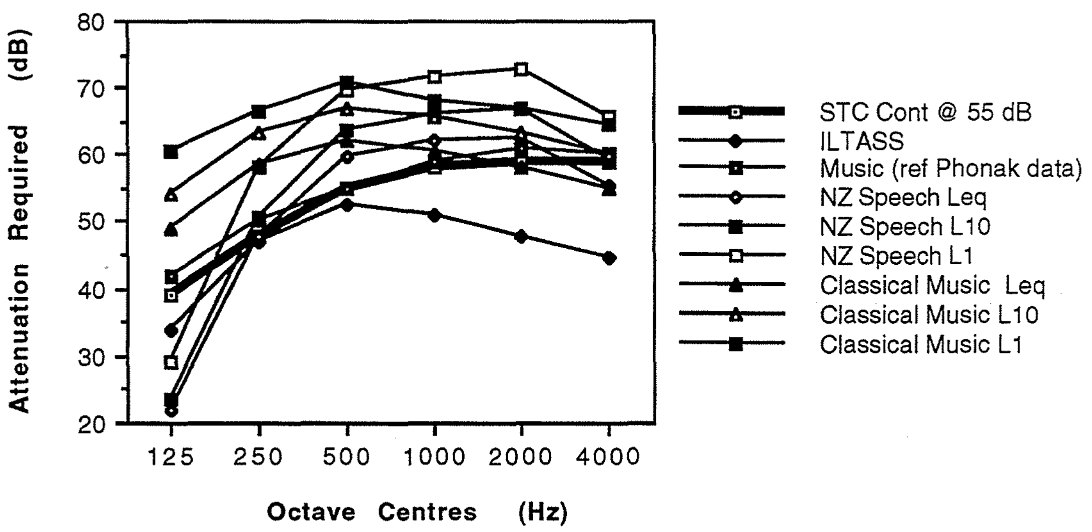

Fig 2: The relative attenuations required to reduce to zero the detectability of expected speech and music in dwellings compared with the attenuation rovided at STC 55 ([24] for speech data)

2. The performance requirements could be reformulated along the lines increasingly used in Europe where the rating is based on the normalised A-weighted level differences (D_{nAt} [22]) between source dwelling and receiving dwelling, and where the radiated spectrum is specified to account for different types of source (e.g. the French regulations use a Pink noise for internal measurements and a special 'street noise' for facade measurements).

3. The use of "soft body" impact insulation techniques (e.g. as referenced in [19]) could be trialled to develop a data base from which more firm recommendations could be made.about improving impact rating procedures. At the same

time research should be encouraged into techniques for measuring and rating horizontal impact and equipment noise transmission.

4. Consideration could be given to a graded quality rating for all types of dwelling which would provide designers, builders and purchasers with more options and certainty of customer satisfaction. The Qualitel Label of Acoustic Comfort system pioneered in France [22] with rankings from 1 star to 3 star presents a comprehensive example. It appears that Germany too has a somewhat similar scheme [23].

6. Field Measurements and Sources

6.1 We argue above that it is desirable to move from laboratory-measured performance of building components as a basis for a building code to actual measurement of the environment experienced by building occupants. Even if the present approach of focussing on component performance is retained measuring this performance in the field can only been seen as an improvement on the present proceedure.

6.2 Accreditation of a particular product or system can currently be acheived on the basis of a **single** sample tested under lab conditions. No test of its repeatability is required. This cannot be supported as being scientific, even less if the results are thought to be representative of the performance that would be acheived in normal building contexts.

6.3 In order to promote field measurements, convenient methods of performing the measurements need to be provided. Our experience of carrying out measurements with conventional techniques in accordance with ISO 140/4 has led us to believe that amongst the major drawbacks are-

(1) the volume and weight of the equipment to be transported, - and accomodated on-site in rooms that are frequently of small volume

(2) inadequate signal/noise ratios, and contamination by transient noises on sites where construction is still underway

(3) the disturbance caused to neighbours when measuring at sites already occupied. Our work towards a resolution of these difficulties began with an investigation of alternatives to the usual power amplifier and louspeaker source radiating pink noise as the excitation signal.

6.4 The ideal source for field measurements will meet, amongst others, the following criteria -

(1) it will be sufficiently small and light to fit with other items in normal "hand" luggage

(2) it will have a range of preselectable output powers with a maximum sufficient to compete adequately with high construction noise levels - allowing a minimum of measurements (and hence time) when high levels are clearly no disturbance to residents or occupants of neighbouring premises

(3) it will have signal repeatability sufficient to allow coherent averaging of low-level excitations in order to provide useable signal/noise ratios where measurements at high levels would be disturbing to neighbours

(4) it will be self-powered, cheap - both to produce and run, and be acceptable in luggage for travel by plane.

6..5 After surveying sirens, air-horns, MLS generators and explosive sources we conclude that remotely-fired explosive charges - in particular those charges used in powder-powered equipment - would satisfy more of the above criteria than any of the alternatives we have considered.

Early results [25] confirm that impulse sources produce STC values which are identical with those measured using conventional techniques in the laboratory. Individual third octave TL values differ from those obtained conventionally by amounts no bigger than the repeatabilities expected for laboratory measurements. Example results are shown in figures 3 and 4.

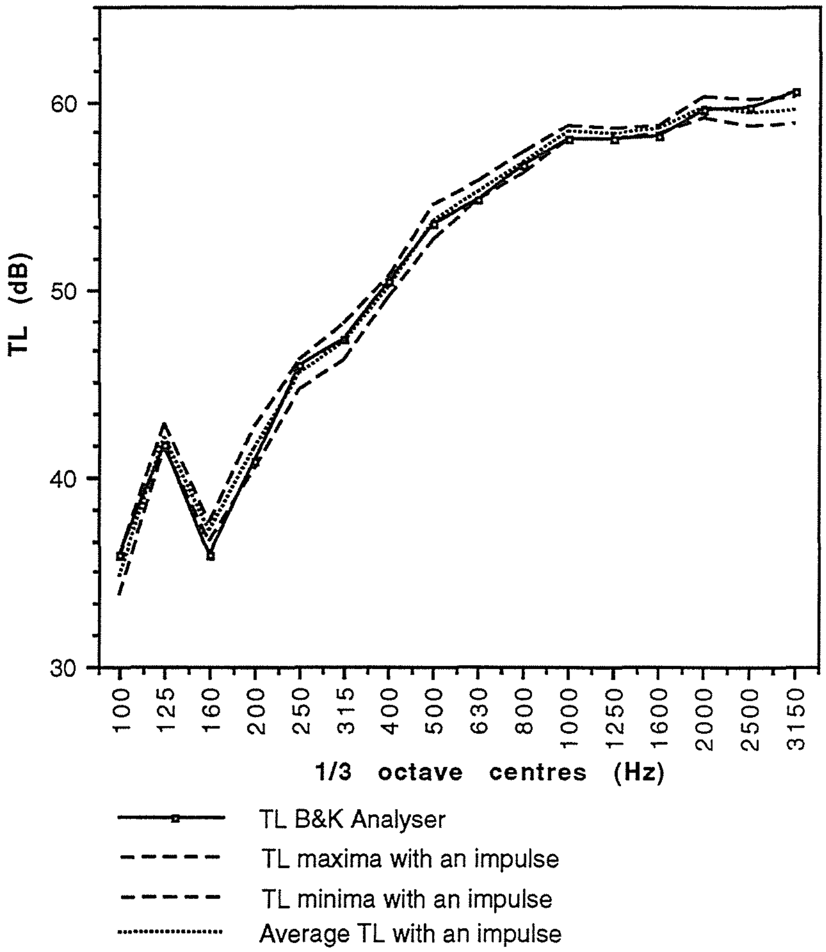

Fig.3: Comparison of Wall TL values measured using an impulse source and measured using pink noise excitation

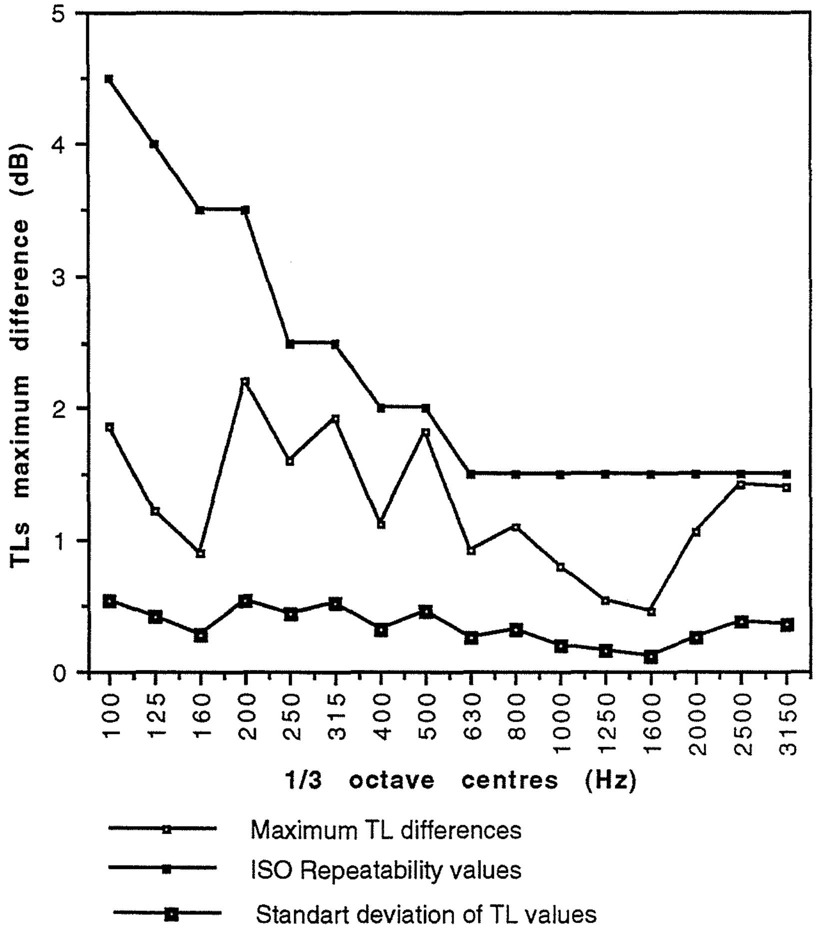

Fig. 4: Maximum differences found in 20 repeated measurements of TL (different source and receiver positions) using an impulse source compared with the repeatability values of ISO/2:1991.

7. Conclusion

Consideration has been given to the use of the term 'noise' in NZ legislation and standards and it is suggested that an open definition covering all of our uses of the term exists if we make a distinction between those subjective sounds which are "merely heard" and those that are " listened to".

Thus noise becomes those sounds we merely hear and hence a subcategory of subjective sound. A more definite emphasis is thus given to the part played by human choice and attention in determining whether or not exposures to sound result in noise. In addition more focus is placed on the individual since the same objective sound may be simultaneously *merely heard* by some and *listened to* by others.

A suggestion is made that it is helpful to have a term to specifically refer to non-noise subjective sounds i.e. those we listen to, and a possibility is identified, **acroma,** derived from the Greek verb meaning to listen.

A review of the content of the section of the NZ National Building Code dealing with sound insulation is presented and suggestions are offered for a significant evolution of the approach and content. Principal amongst these are

(1) a shift of emphasis from laboratory-based component testing to field measurement of dwelling performance, and

(2) to supplement a minimum performance requirement with an optional rating of the acoustic comfort of dwellings which addresses within-dwelling transmission of sound as well as between-dwelling transmission.

Early results from a review of alternative sources for field testing indicate that excitation using explosive charges is capable of producing the same results as those obtained using a conventional loudspeaker source.

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QUEENSLAND'S APPROACH TO NOISE LEGISLATION — THE ENVIRONMENTAL PROTECTION (NOISE) POLICY

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ABSTRACT

The proposed Environmental Protection (Noise) Policy is part of the State Government's strategy to improve the quality of the environment for the people of Queensland. The *Environmental Protection Act 1994* is a comprehensive Act that uses the concepts of environmental harm and environmental values to establish broad principles to control environmental pollution. The basic purpose of the policy is to provide guidance to the Act.

INTRODUCTION

The proposed draft noise policy provides a framework to protect Queensland's environment from noise while allowing for development that improves the total quality of life, both now and in the future. The policy has seven key concepts

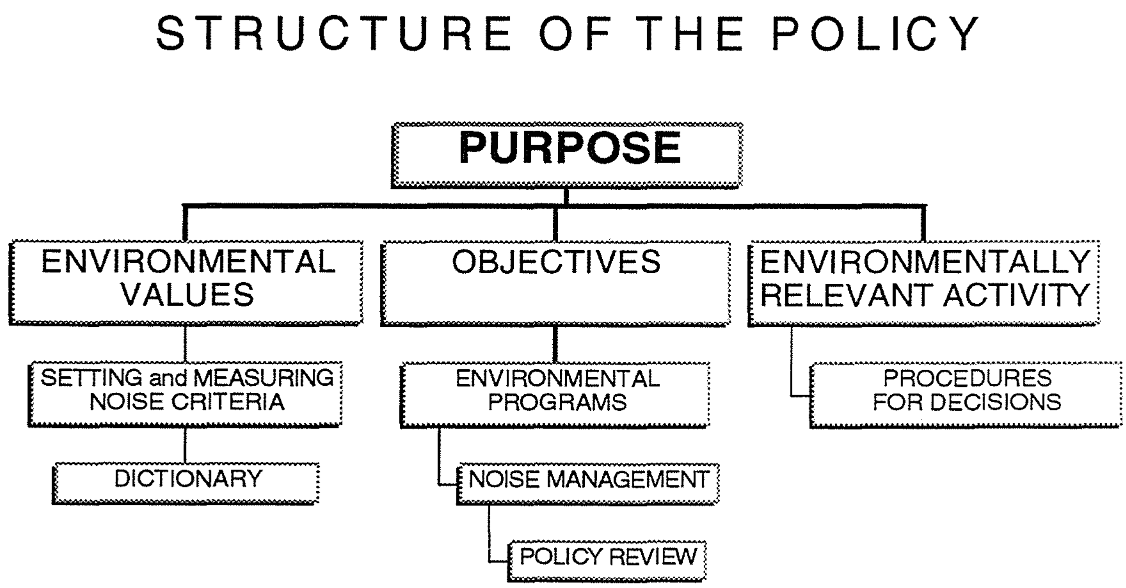

Figure 1—general structure of the revised policy

- it has a purpose and establishes procedures for its assessment and review;
- it sets objectives and a timetable for achievement;
- it states environmental quality criteria (environmental values);
- it requires the development of environmental programs that identify impacts on the environment;
- it provides procedures for licensing, approvals, complaints and dispute resolution;
- it states noise criteria for certain activities; and
- it requires product performance (noise labels) for specified noisy machinery.

The purpose

The purpose of the policy is to protect the quality of Queensland's acoustic environment by—

- identifying environmental values for Queensland's acoustic environment; and
- deciding on and stating acoustic quality guidelines and objectives to enhance or protect the environmental values; and
- promoting consistent and equitable decisions about Queensland's acoustic environment that have regard to efficient use of resources and best practice environmental management; and
- promoting planning of residential and industrial areas and transport infrastructure to prevent or mitigate the adverse effects of noise; and
- involving the community through consultation, education, research and promotion of community responsibility for environmental noise management.

WHAT IS THE "ENVIRONMENT"

The Environmental Protection Act defines 'environment' very widely. The definition in the Act says—

> "Environment includes—
>
> (a) ecosystems and their constituent parts, including people and communities; and
>
> (b) all natural and physical resources; and
>
> (c) the qualities and characteristics of locations, places and areas, however large or small, that contribute to their biological diversity and integrity, intrinsic or attributed scientific value or interest, amenity, harmony and sense of community; and
>
> (d) the social, economic, aesthetic and cultural conditions that affect, or are affected by, things mentioned in paragraphs (a) and (c)."

The Act is also concerned to ensure that 'environmental harm' does not occur. Environmental harm is—

"any adverse effect, or potential adverse effect (whether temporary or permanent and of whatever magnitude, duration or frequency) on an environmental value; and includes environmental nuisance."

Environmental values

The Environmental Protection Act defines an 'environmental value' as—

(a) a quality or physical characteristic of the environment that is conducive to ecological health or public amenity or safety; or

(b) another quality of the environment identified and declared to be an environmental value under an environmental protection policy or regulation."

The environmental values under the noise policy are qualities of the environment that provide public, community or individual amenity. The policy responds to the amenity that people want is the ability to live free of intrusive noise. In particular, they want to—

- have undisturbed sleep;
- have undisturbed passive recreation; and
- be able to converse or listen without undue interference from noise.

Nevertheless, all of us in the community at some stage make or contribute to noise, and any restrictions must be fair and reasonable. The responses from the public consultation confirm that noise is a significant issue for many people.

Noise becomes an issue because it results in unwanted or adverse effects on one or more environmental value(s). Someone, and this could be an administering authority or through dispute resolution, has the task of assessing whether the effect truly exists, having regard to the circumstances relating to the noise. Noise could, therefore, be classed as being—

- "reasonable"; or
- a nuisance or environmental harm; or
- "unreasonable interference with a person's enjoyment of a place".

The difficult issue is in qualifying 'adverse effect'. One meaning is that it is 'a consequence of opposing forces'. With noise there are clearly defined parties and clearly defined qualities of the environment enjoyed by those parties. The economic (or social) activities of one party affects the social and amenity values of another. Also to be considered is the issue that an "adverse" activity can also have a positive social, economic and cultural value to the community at large. Each party adversely affects the other. This means an assessment of opposing perspective's and a complainant does not necessarily have primacy (see figure 2).

ASSESSMENT OF ADVERSE EFFECT

OFFENCE **Unlawful Environmental Harm** an Adverse effect on, or unreasonable interference with, an environmental value	**FORMING AN OPINION** **Unreasonable Interference** ↑ Adverse or potentially adverse effect on an environmental value ↓ **Reasonable interference**	**NO OFFENCE** No interference with environmental value

Figure 2—assessment of adverse effect

The fundamental question to be asked is 'whose environment is it that is going to be adversely effected?'

WHAT ARE THE REASONABLE EXPECTATIONS OF THE MAJORITY IN THE COMMUNITY AND, HOW CAN THEY BE ACHIEVED?

By setting objectives in the policy

The proposed objectives are—

- by 1 December 1999 — completing an assessment of the ambient acoustic environment, community response and the impact of noise on the environment of Queensland;
- by 1 March 2002 — achieving an ambient acoustic environment of 55dB(A) or less for more than 60 percent of Queensland's population living in residential areas;
- by 1 March 2010 — achieving an ambient acoustic environment of 55dB(A) or less for more than 90 percent of Queensland's population living in residential areas; and
- the levels are measured as the 24-hour LAeq.

The aim of the objectives is to ensure that the community is not exposed to levels that would cause 10% or more of the urban population to be highly annoyed by noise. It is an indicator of a level of ambient sound where consideration must be given to noise reduction. The objective is progressive and long-term to the year 2010.

By developing environmental programs

An environmental program is developed to identify the actual, or potential, adverse environmental impacts on the environment from industrial activities, urban development or transportation. The program includes performance indicators, monitoring requirements and time-frames for compliance with the general environmental duty to prevent or minimise environmental harm. An industry can develop codes of practice under the policy that will assist in meeting the general environmental duty.

The policy's objectives will be achieved substantially through progressive management of noise from activities associated with transportation. Transportation and administering authorities, local governments, transportation operators and manufacturers are expected to accept responsibility for noise issues and to reduce the level of noise caused by transportation. The policy requires control over intrusive noise from aircraft, railways and road traffic. Organisations or people responsible for transportation systems must prepare environmental programs by 1 July 1999.

The policy complements and reinforces the role which land use planning plays in noise control. Separating incompatible land uses is an important step in noise prevention. Noise level prediction and reference to the noise potential of land use activities is a basic part of the planning process. Through strategic planning, land use zoning, buffer zones and the development approval process, noise problems may be minimised or prevented.

By licensing certain activities

Under the Environmental Protection Act some activities must hold an licence or approval called an environmental authority. These activities are listed in the *Environmental Protection (Interim) Regulation 1995.* The noise policy provides guidance for new and existing industries and the community about how to gain a licence or approval by stating—

- information that must be supplied before an application can be approved;
- noise criteria that are considered when the application is being assessed; and
- nuisance noise criteria that can be applied.

The administering authority must consider a number of "standard criteria" in assessing the licence application, for example—

- any applicable plans or requirements of a local government;
- any applicable impact study, assessment or report; and
- the character, resilience and values of the receiving environment.

The information that must be provided with an application can include—

- an assessment of the activity's impact on neighbours; and
- an environmental noise management plan.

By product performance (noise labelling) of noisy machines

From 1 July 1998 manufacturers, importers or wholesalers of certain machines must attach noise labels to inform customers of the machines' noise levels. The noise policy follows the guidelines used in New South Wales and Victoria, and makes the Queensland approach to this issue consistent with the national approach. The machines include—

- chainsaws;
- domestic air conditioners;
- grass-cutting machines, including lawn mowers, ride-on mowers, edge-cutters, string-cutters and brush-cutters;
- mobile air compressors;
- mobile garbage compactors; and
- pavement breakers.

By noise criteria

In response to public comment, the policy seeks to establish controls over certain nuisance noise through the setting of noise criteria. Noise criteria have been developed for—

- general criteria for steady and non-steady sound, excessive noise, background levels and building interiors;
- air conditioning plant;
- animals and caged birds;
- blasting and vibration;
- building and grounds maintenance activities (domestic);
- carparks;
- construction, renovation or demolition activities;
- emergency or stand-by generators;
- entertainment noise from an indoor venue;
- indoor sporting venues;
- measurement and assessment protocols;
- motor vehicles;
- motor vessels;
- musical instruments and sound equipment;
- outdoor concerts;

- power tools;
- public address systems;
- refrigeration plant on a vehicle;
- shooting ranges;
- spa or swimming pool pump and equipment; and
- transport terminals.

The Police have controlled some types of excessive noise (such as loud music, parties and off-road vehicles) under the Noise Abatement Act (and the Environmental Protection Act) and will continue to do so under this policy. Local governments can also develop local law policies and local laws to deal with noise nuisance.

IS THE PROPOSED POLICY CONSISTENT WITH CURRENT THINKING?

The draft Environmental Protection (Noise) Policy seeks to provide a framework for noise management to achieve a level of environmental noise acceptable to the majority of people in Queensland. Relevant state, national and international legislation, standards and research have been reviewed in the preparation of the policy.

The then Australian and New Zealand Environment Council in June 1990 assessed the noise exposure limits recommended by Organisation for Economic Co-operation and Development (OECD). The OECD considers outdoor levels of less than 55dB(A) (Leq) 'acceptable' and levels above 65dB(A) (Leq) 'unacceptable'. Noise levels between 55dB(A) and 65dB(A) are considered "undesirable". ANZEC concluded that the OECD recommended levels are appropriate environmental noise goals for Australia. It also recommended that research be extended into the problem of environmental noise, particularly traffic noise, with the view to developing both long-term and short-term strategies to improve the noise situation.

The World Health Organisation (WHO) recommends, for good speech intelligibility indoors, a level of less than 45dB(A) (Leq). WHO recommends an indoor noise limit of 35dB(A) (Leq) at night, to facilitate sleeping, and to preserve the restorative processes of sleep.

WHAT ASSISTANCE WILL THERE BE TO UNDERSTAND THE FINAL POLICY?

Guidelines

The Department of Environment, in conjunction with industry associations, is developing environmental management guidelines and industry codes of practice to support the policy. The guidelines, available from the Department, describe environmental problems associated with various premises and noise sources, and some ways to manage them.

The guidelines discuss good environmental management practice but are not intended to cover all eventualities or state any one best course of action. They are designed to help administering authorities, industry and the community meet their environmental management responsibilities. Guidelines are also being prepared to explain procedures required by the policy such as noise modelling and noise management plans.

Education and training

The Department of Environment has prepared training guides in noise assessment for Department and local government officers. Basic training materials in noise assessment and management have been prepared for educators within schools, TAFEs and universities, industry and the private sector. Guidelines for noise management and information pamphlets have been prepared for industry and the community.

Simple sound level indicators can be used in the many instances where an indication of noise levels is needed. The use of sophisticated meters is needed only when legal proceedings may be taken or when ambient or background studies are being made.

HOW CAN ACTION BE TAKEN TO CONTROL NOISE?

infringement notices

The traditional method requiring an administering authority make a decision. The revision introduces two offences and penalties as alternatives to the enforcement procedures in the Environmental Protection Act—infringement notices assist in cost-effective resolution of nuisance complaints; and clear offence provisions provide guidance for complaint.

dispute resolution

The concept of dispute resolution (and mediation) was introduced in the consultation draft to empower individuals to resolve nuisance noise complaints without resorting to the courts.

The policy creates a dual process to assess nuisance. This involves—

- intrusive noise criteria to assist people in making and sustaining a complaint; and
- objective criteria for administrative authorities or dispute resolution.

The concept of mediation was well received during consultation although there is clear concern about process and administrative ability to sustain the proposal—

- the policy currently proposes to endorse a conflict resolution process; and
- the process must occur before an administering authority has decided to take action under the Environmental Protection Act or the enforcement provisions of the policy.

WHO WILL ADMINISTER THE POLICY?

The policy is administered by the Department of Environment. The Department can delegate its administrative responsibilities to other bodies, providing they accept the delegation.

RESPONSES FROM PUBLIC CONSULTATION

SUMMARY OF THE CONSULTATION

Consultation on the policy commenced on 3 May 1996 with key stakeholder meetings and the full public consultation commenced on 20 May 1996. The consultation round closed on 23 July 1996 (key stakeholder meeting) following the final public meeting on 9 July 1996.

Approximately 7300 policies, 7000 Regulatory Impact Statements, 7000 Response Forms, 8200 Summaries and 8200 'Noise' brochures were distributed. As at 30 September 1996, 910 "personal response survey" forms and 358 detailed submissions and responses have been received. Three responses on the Regulatory Impact Statement have also been received.

Twenty-five public meetings were held over the consultation period. Each meeting was of between 1↔ and 3 hours' duration. Forty-nine meetings have been held with industry, key-stakeholders and special interest groups. A significant number of inquiries (approximately 1000) were received by telephone over the time from the end of May to the end of September. The response from the public consultation on the policy clearly identifies specific noise issues with the main concerns being licensing of environmentally relevant activities, administration of the policy and "neighbourhood" noise.

NOISE ANNOYANCE OF HOSPITAL STAFF IN BANGKOK METROPOLITAN AREA

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ABSTRACT

The study on noise annoyance of hospital staff in Bangkok Metropolitan Area was performed by measuring noise levels in buildings of five hospitals. The values of the equivalent sound level for 8 hours, $Leq_{(8)}$ in the daytime were 50.0-73.5 dB(A) and values of $Leq_{(16)}$ in the evening and night were 50.5-71.4 dB(A). The percentage of staff in the hospital who were annoyed was 69.7%. The staff who lost concentration in their work was reported at 16.4 %.

It was also found that the sources of noise were from both indoor (37.6%) and outdoor (54.5%) sources. Motorcycle and public buses are the two main outdoor noise sources.

INTRODUCTION

Noise pollution is one of the major environmental problems in the Bangkok Metropolitan Area. According to the monitoring data reported by the Pollution Control Department, the $Leq_{(24)}$ levels measured at curbside monitoring stations were 62-83 dBA and 68-83 dBA in 1993 and 1994 respectively. Not only people who work or stay outside the buildings, but also people who work inside the building suffer from the noise problem. Hospitals which should have a quiet environment are also affected by noise problems. This paper presents the study of noise effects on staff working in the hospitals which are located on the main streets in Bangkok.

METHODOLOGY

Indoor noise measurements were carried out inside 25 buildings from 5 hospitals located near streets in the Bangkok Metropolitan Area. The staff who work in each building were selected by random sampling to be interviewed by questionnaire.

The sample consisted of 396 individuals , aged 20-50. Some questions in the questionnaires used in this survey are adopted from one used in the Noise Annoyance Survey in Central London (McKennell and Hunt,1966). This study was carried out during the period September to December 1988.

RESULTS AND DISCUSSION

1. Noise Levels in the Hospitals

As shown in Tables 1 and 2, $Leq_{(8)}$ daytime levels and $Leq_{(16)}$ levels for evening - night time in the hospital building are 50 - 73.5 dB(A) and 50.5 - 71.4 dB(A), respectively. The average $Leq_{(24)}$ levels are 50.3-72 dB(A). The acceptable indoor noise level recommended by the US. EPA. is that $Leq_{(24)}$ levels should be less than 45 dB(A). Noise levels in all hospital buildings exceeded the recommended level, as many buildings have no air - conditioning. The noise sources were found to be both indoor and outdoor. The daytime and night time noise levels are almost the same.

TABLE 1: Measured Noise Levels dB(A) $Leq_{(8}$ for Daytime and dB(A) $Leq_{(16)}$ for Evening-Night Time (parenthesis value) in hospital buildings

Hospital Name	Building No.1	Building No.2	Building No.3	Building No.4	Building No.5
1. Childrens	61.1 (56.9)	53.5 (56.2)	61.4 (59.0)	59.7 (55.2)	57.5 (57.1)
2. Rama	61.1 (58.1)	50.0 (50.5)	-	-	- (54.3)
3. Central	68.8 (67.1)	71.0 (65.3)	73.5 (71.4)	72.7(69.0)	71.1(68.3)
4. Lerd Sin	65.1 (62.6)	65.0 (62.4)	66.6 (62.8)	60.6 (58.6)	63.6 (60.7)
5. Vachira	61.8 (61.3)	68.1 (62.9)	59.5 (56.1)	65.3 (59.9)	63.8 (59.5)

TABLE 2: Measured $Leq_{(24)}$ dB(A) Noise Levels in Hospital Buildings

Hospital Name	Building No.1	Building No.2	Building No.3	Building No.4	Building No.5
1.Children	58.5	55.3	59.8	56.6	57.5
2. Rama	59.1	50.3	-	-	-
3. Central	67.7	67.2	72.0	70.3	69.6
4. Lerd Sin	63.4	63.3	64.0	59.3	61.6
5. Vachira	61.5	64.6	57.2	61.9	60.5

2. Effects of Noise on Hospital Staff

The effects of noise on hospital staff are shown in Table 3. Hospital staff realized that a noise problem in their working environment can cause various effects on them. Most of them (69.7%) felt annoyance and easily lost their temper. Some (16.4%) seem to lose concentration in their work. Few suggested that noise might cause mental effects (10.1%) and also hearing loss (3.8%).

TABLE 3: Effects of Noise on Hospital Staff

Type of Effect	Percentage
1. Loss of Concentration in Work	16.4
2. Mental Effect	10.1
3. Hearing Loss	3.8
4 Annoyance and Easy Loss of Temper.	69.7

3. Types of Noise Sources

About 55 % of the staff were disturbed by outdoor noise, and 38% were disturbed by indoor noise, as shown in Table 4. There are many outdoor noise sources that cause annoyance to the hospital staff. Motorcycles are identified as being most responsible (53.9%) for noise that causes annoyance to the hospital staff. The second ranking noise source (21.3%) are public buses. Other noise sources identified are taxis (10.9%), trucks (4.9%), three-wheel motorcycles (3.5%), private cars (2.9%), trains (2.0%) and mini-buses (0.6%) as shown in Table 5 .It was also found that the traffic noise sources that cause annoyance to most of the staff came from car horns,exhaust pipes and braking.

TABLE 4: Percentage of Staff Who Suffered from Various Noise Source

Noise Source	Percentage
1. Indoor Noise Source	37.6
2. Outdoor Noise Source	54.5
3. No Disturbance	7.8

TABLE 5: Types of Outdoor Noise Source Which Cause Annoyance to Hospital Staff

Outdoor Noise Source	Percentage
1. Motorcycle	53.9
2. Three-Wheel Motorcycle	3.5
3 Public Bus	21.3
4. Taxi	10.9
5. Truck	4.9
6. Mini-Bus	0.6
7. Private Car	2.9
8. Train	2.0

CONCLUDING REMARKS

The results from this survey can be used as basic data for the government and concerned agencies. More attention should be paid to noise reduction in hospital buildings, especially in new hospital building programs. Traffic noise should be mitigated as a first priority.

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THE BRISBANE AIRCRAFT NOISE SURVEY 1995-96

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1. INTRODUCTION

Recent episodes relating to attempts to quantify aircraft noise measurement in Australia might be viewed as beginning and ending with the deliberations of two parliamentary committees: firstly, in 1970 when the House of Representatives Select Committee on Aircraft Noise called for Australia to adopt the noise exposure forecast (NEF) system which the U.S.A. had developed in the late sixties, and secondly, in 1996 when the Senate Select Committee on Aircraft Noise[1] called for new concepts to replace the NEF system (ANEF system in Australia) which is now increasingly perceived to be inadequate for the tasks for which it has been designed.

The Senate Select Committee found, inter alia, that the ANEF system is dependent for its accuracy on flawed forecasts and assumptions and does not deal adequately with noise events (ibid. p.208). The Committee therefore recommended

> "That the National Accoustic Laboratories (NAL) re-evaluate the usefulness and relevance of the existing ANEF system as predictor of long term community reaction to aircraft noise around Sydney Airport; and that the NAL explore the development of indices or other information for predicting the noise impact for communities faced with a changing noise environment." (ibid. p.209)

The catalyst for these conclusions appears to have been the failure of the Environmental Impact Statement (E.I.S.) carried out by Kinhill Engineers Pty Ltd[2] to predict the noise impact of the operations of the third runway at Sydney Airport; and as the E.I.S. relied upon the ANEF system that system was deemed by the committee to be inadequate for the task for which it was used.

1.1 Previous Publications and the Present Study

Two publications about aircraft noise dominate the history of the last 25 years. The first was Australian Standard AS-2021 published in 1977 which set out the details of the noise exposure forecast system, and the second was the 1982 study by Hede and Bullen (HB)[3] which surveyed 3,500 residents around five capital city airports in Australia and refined the NEF system to Australian conditions (ANEFs). The Hede and Bullen changes were subsequently adopted by AS-2021 in its 1985 revision.

The purpose of the present study (1995-6) conducted jointly by the QUT and the Department of Environment is to show that the ANEF concept as it is used in AS-2021 is an inadequate measure of the disturbance people in close proximity to airports suffer as a result of aircraft noise. The QUT/Department of Environment study also contends that the HB assumption that the ANEF concept provides the best indicator of community reaction to aircraft noise is no longer true. The present study takes up a suggestion by Hede (1993)[4] and develops impact descriptors which can be used in conjunction with ANEFs to assess more accurately the environmental effects of aircraft noise.

The QUT/Department of Environment survey plots contours around Brisbane airport which represent " 20% moderately affected" and " 10% seriously affected", and these may be seen later in this report.

2. THE ANEF SYSTEM

The ANEF system is made up of two parts: a series of contour lines around an airport linking areas of similar noise exposure as well as some conclusions about the disturbance effects such exposure levels have on residents living within those contours.

The development of this system owes much to the HB Report whose aims were to

- evaluate the Noise Exposure Forecast (NEF) system - the official index used to measure aircraft noise in Australia.
- and to determine the 'dose/response' relationship between aircraft noise exposure and community reaction to the noise (A.J.Hede and R.B.Bullen, 1982, op.cit. p.1)

The HB report surveyed 3,500 residents around five airports none of whom was exposed to less than 15 ANEF. Following this report the ANEF index was adopted by planning authorities throughout Australia for assessing the effect of aircraft noise on people, and has been incorporated into Australian Standard AS-2021 and "is the noise measure used in EISs for developments of and around airports" [A.J.Hede, 1993, 42]. For this reason the dose/response curve provided in AS-2021 does not go below ANEF 15, leading people to conclude that for an exposure to aircraft noise below ANEF 15, the impact is negligible. As Hede himself points out, such a conclusion is invalid since 7% of people are seriously affected and 33% moderately affected at that level.

In fact, AS-2021 is quite specific on this point.

> "In the areas outside 20 ANEF, it is generally accepted that noise exposure is <u>not</u> (my emphasis) of significant concern. Within the area from 20 ANEF to 25 ANEF aircraft noise exposure <u>starts to emerge as</u> (my emphasis) an environmental problem" (AS-2021, 1985 edit., p.35.)

The QUT/Department of Environment study rejects statements such as this and provides evidence of people "seriously affected" by aircraft noise outside 20 ANEF.

2.1 Criticism of the Use of the ANEF System

1. The HB Report concluded that 'equal energy' indices (eg. NEFs) show a significantly stronger relationship with community reaction to aircraft noise than other types of index tested, including 'peak level' indices and indices which are independent of the number of overflights per day." (A.J.Hede and R.B. Bullen, 1982,p.11). But the NEF concept was based on studies before the introduction of wide bodied jets and these aeroplanes have a different noise profile.

2. Secondly, in the Hede and Bullen study, the coefficient of determination in the "dose/response" relationship was 0.13 which is evidence of a weak relationship[5]. Thus while ANEFs cannot be discarded as indicators, other measures such as L_{max} need to be employed as well. Some evidence is

emerging both here and overseas that people are more disturbed by L_{max} levels above a critical value equal to this "annoyance level" than by the "bucket of noise" represented by ANEF contours.

3. The results of a Swedish study, published after the EIS Supplement, were said to support the conclusion that annoyance reaction "is better related to the number of aircraft and the maximum noise level than to energy equivalent levels for noise exposure". (Senate Select Committee Report, 1995. p.178). Preliminary tests indicate that this conclusion may also apply in Brisbane.

4. As the ANEF formula is based on a yearly time frame it is intuitively obvious that ANEFs are based on averages that in practise may never exist: because there are four ANEFs - one for each season - and then they are different again for different weather conditions.

5. The ANEF formula makes certain assumptions about flight paths. These assumptions may not hold since aircraft are often left or right of the centerline of the flight path and often descend or ascend more quickly or more slowly than normal.

6. In many cities ANEFs are of little relevance for residential areas. For example, in Brisbane for the most recent ANEF only about 5% of the area covered by ANEF 20 covers a residential area. By far the greater part of ANEF 20 forms a cigar shape around the runway where no one lives.

7. Finally, ANEFs tend to ignore flights paths, many of which are seasonal, since flight paths depend on the direction of the prevailing wind. The ANEF "finger" extends under the flight path but then ends. Yet the flight path continues and people directly under the flight path still experience aircraft noise. even though they may be outside the ANEF contour. Also, with the NEF system, people who live (in the area) between ANEF 20 and 25 are treated as suffering from equal aircraft noise. Yet the width of the band may be several hundred metres and clearly those people in the band directly under the flight path may be subject to nearly 3dB(A) more than those people 300 m. away for planes at a height of 300-350m.

3. METHODOLOGY

As the survey relates to people's attitudes to aircraft noise, the sampling frame chosen comprised areas of Brisbane under (or near) the existing flight paths. It was thought that this would allow a more comprehensive analysis of the problem than would be obtained if the sampling frame was confined to ANEF zones, especially given the perceived limitation of ANEF levels as a predictor of the disturbance people suffer as a result of aircraft noise. In all, 50 statistical local areas (S.L.As) were chosen out of the 164 S.L.A.s in Brisbane City. All of these were under or near the flight paths for airplanes landing or taking-off from either the 01/19 or the 14/32 runway. In addition, (2) S.L.As were chosen in Logan City as a control sample. Thus the sample comprised 52 S.L.A.s in total. S.L.As correspond closely to suburbs although some small suburbs (eg. Seven Hills, Doomben, Buranda and Stones Corner) are incorporated into larger neighbouring suburbs as an S.L.A. For ease in understanding what follows we will refer to these areas as suburbs. The 52 suburbs chosen for the survey are shown on the map with the flight paths marked in, and are also listed with their population statistics and number of dwelling units in Table 1. On 1995 population figures (A.B.S. Cat. No. 3201.3) the total number of residents in the area surveyed was 239,617 and they lived in 97,175 dwelling units (D.U.s)

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Table 1
Suburbs in Brisbane Aircraft Noise Survey

	D.U.'s	Population	SLA Code
Acacia Ridge	2453	7115	1001
Albion	1109	2381	1004
Annerley	3918	8871	1015
Archerfield	256	636	1023
Ascot	1955	4821	1026
Balmoral	1329	3216	1042
Belmont	765	3056	1057
Bowen Hills	377	692	1067
Bulimba	1546	3753	1086
Buranda	476	1066	1525
Camp Hill	3831	9328	1097
Cannon Hill	1588	3961	1102
Carina	2905	8260	1113
Carindale	2435	10750	1108
Clayfield	4335	9400	1151
Coorparoo	5841	12968	1157
Doomben	46	102	1467
East Brisbane	2277	5016	1195
Fortitude Valley	417	1283	1233
Gordon Park	91	279	1585
Grange	1381	3295	1244
Greenslopes	3528	7043	1247
Gumdale	280	950	1252
Hamilton	1895	4158	1255
Hawthorne	1516	3810	1258
Hemmant	604	1567	1265

Hendra	1440	3513	1271
Highgate Hill	2541	5090	1277
Holland Park	3041	7527	1282
Holland Park West	2208	5423	1285
Kangaroo Point	2005	4310	1304
Kedron	5024	11328	1312
Lutwyche	1196	2546	1345
Manly West	2810	8671	1367
Morningside	2151	5330	1397
Murrarie	858	2510	1413
Newfarm	4909	9032	1421
Newstead	443	923	1427
Norman Park	2571	6243	1432
Nudgee Beach	106	239	1443
Pinkenba	120	276	1467
Rochedale	451	1406	1495
Seven Hills	758	1866	1397
Springwood	2071	6742	4642
Stones Corner	510	1431	1525
Tarragindi	3657	9414	1563
Tingalpa	1756	6314	1571
Underwood	867	2801	4651
Wilston	1405	3305	1618
Windsor	2848	5916	1623
Wooloongabba	1874	4143	1631
Wooloowin	2401	5541	1634

Population statistics (1995) taken from A.B.S. Cat. No. 3201.3 March 1996.
D.U.'s (1991) taken from A.B.S. Cat. No. 2730.3 June 1993.

3.1 The Sample

When parts of the Brisbane Statistical Division are to be surveyed it is easier to work with a data file of collection districts (C.D.'s). Consequently, the C.D.'s corresponding to the 52 suburbs in the sampling frame were listed after which 128 C.D.'s were chosen on a probability proportional to size basis. This allowed each of the 128 interviewers employed in the survey to operate with one C.D. each. Field workers then segmented the 128 C.D.'s (1 segment = 10 D.U.'s) and two segments from each C.D. were drawn randomly for 100% sampling.

This two stage process ensures that each D.U. in the population has an equal chance of being chosen. That is, the sample is random. In all, 2050 interviews were carried out giving the sample an accuracy of about ±2% at two-sigma limits.

At a later stage respondents who lived under the Instrument Landing System (I.L.S.) were interviewed and included in the sample.

A field audit was carried out to ensure that interviewers conducted interviews with the people chosen by the sampling process.

Results were analysed using the S.P.S.S. program.

4. THE QUESTIONNAIRE

4.1 The Conceptual Framework

Questionnaires on Aircraft Noise owe much to the conceptual framework adopted by Hede and Bullen[6] of stimulus variables, intervening variables and response variables. Stimulus variables include the physical characteristics of the noise; intervening variables represent the filtering process and involve personal characteristics, work habits, attitude to the source of the noise and sensitivity to noise, while response variables include the type and degree of annoyance. Questionnaires then usually follow up on after effects (eg. health effects, emotional effects) as well as on changes to behaviour patterns.

The questionnaire used in the Q.U.T/Department of Environment survey follows this approach. For this presentation, we focus on part of the questionnaire only, namely those questions like Q8, Q13, Q15 and Q16 which deal with annoyance.

The suburbs listed in this report are based on means obtained from the Likert scale used in Q.8. Hede and Bullen (1982, op.cit. p.62ff) point out the pitfalls of using single rating scales and recommend using a composite index (GR). Time so far has prevented us from developing this.

4.2 Minimising Response Bias

Survey work should always avoid "cold calling" and this is usually managed by publicizing the topic of the survey in the media before interviewing is carried out. One difficulty with pre-publicizing a survey which deals with a specific environmental problem such as aircraft noise is the problem of response bias, which occurs when people who are only marginally affected by the topic exaggerate their responses if they feel strongly about the environment / noise / aircraft etc. (See Hede and Bullen, 1982, p.27). The standard way to neutralize this sort of response bias is to design the questionnaire in such a way that it begins with questions on more general environmental issues and initially only mentions the specific issue - aircraft noise - in the context of other noise sources. This was done in this survey in two ways, as follows.

Firstly, the questionnaire begins by asking people about their neighbourhood in general terms eg. "What two things do you like/dislike about living in this area?" However, responses to the question are classified only as 1. noise related or 2. non-noise related.

This approach tends to minimize response bias because the respondent is unaware of the purpose of the question and the general nature of the question helps to conceal that. Thus while the respondent is thinking about reasons as diverse as "proximity to schools, hospital etc." or "the price is right" the survey designers are only interested in whether the response is related to a noise issue or to a non-noise issue. A noise related response in answer to the question "like living in this area" usually comes from a response such as "This is a quiet neighbourhood," whereas a noise related response in answer to the question "dislike living in this area" comes mostly from complaints about noise (non specific noise at this stage but people may mention aircraft noise if that is their area of concern). It is significant that nearly three times as many people mentioned noise related responses to the question "dislike living in this area" than to the question "like living in this area" and about one half of the "dislike" noise related responses mentioned aircraft noise. We can conclude then that aircraft noise is an area of concern for many respondents but because the information was volunteered by the respondent in the context of a discussion of more general environmental matters response bias is unlikely to be significant.

The second way in which response bias is minimized comes with a later question on various noise sources ranging from traffic noise to garbage collection noise that may or may not exist in the neighbourhood. After ascertaining whether or not the respondent hears those noises in the neighbourhood he/she is questioned on the degree of annoyance that these noises cause. In the questionnaire aircraft noise is included near the middle of the list between lawn mower noise and noise from pets such as barking dogs. The purpose of this question is to avoid drawing attention to aircraft noise until one can quantify it in relation to other noise in the neighbourhood. The weighted means of the degree of annoyance from the various noise sources were significant with aircraft noise easily at the top of the list at 3.567 compared to a composite for all other sources of noise at 1.787. Traffic noise came in second at 2.662. Only after this question does the questionnaire hone in on aircraft noise. So again, we can be reasonably confident that response bias has been minimised.

5. SUMMARY OF MAIN FINDINGS

The population surveyed represented a stable group slightly more female (55.2%) than male (44.7%). Almost one third of all respondents had lived in the area for more than 15 years. Nearly 3 in 4 of all respondents either owned their residence or were in the process of buying it. More people (84.6%) heard aircraft noise than noise from other sources.

Nearly one half of all respondents reported that visitors remarked upon aircraft noise during the course of their stay. Evening was the worst time for aircraft noise although one person in twenty (5.6%) often found aircraft noise a nuisance at night. Activities which were often disturbed by aircraft noise were, in order: TV/radio, phone conversations, conversations with others in the house, relaxing, reading and studying, outdoor activities and sleep. Disturbance to sleep was of greatest concern: so it is noteworthy that 13.5% of respondents reported that their sleep was "sometimes disturbed", with 5.9% of respondents reporting that their sleep was "often disturbed".

Only about one person in eight has made a complaint about aircraft noise (mostly to the FAC) so the number of complaints cannot be taken as representative of the problem. When those who had not complained were asked why, many said that there was no point complaining because, short of changing the flight path or shifting house, nothing could be done. Only 10.6% of people had seriously considered moving because of aircraft noise but when respondents were asked if they would seriously consider moving if the amount of aircraft noise increased in the future, a surprising 32.7% of people answered yes. When asked what action the authorities should take in regard to the aircraft noise problem most people (41.1%) said that flight paths should be restructured to share the noise more evenly across the population.
In terms of numbers, about 24 000 residents in Brisbane are seriously bothered by aircraft noise.

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6. WHAT IS THE DEPARTMENT GOING TO DO ABOUT NOISE FROM AIRCRAFT?

This is not a simple question and one that is not capable of a simple answer.

The first point to be made is that it is debatable whether the Environmental Protection Act has any legal effect on any airport in the State. The Act binds all persons, including the State, and, as far as the legislative power of the Parliament permits, the Commonwealth and the other States. However, as an airport operated by a Commonwealth agency and allowing access to Commonwealth jurisdiction aircraft, it is highly likely that the measures of the Act or any environmental protection policy do not directly apply.

The Commonwealth is obliged, however, through Airservices Australia (sections 8 and 9 of the *Airservices Act 1995*) to carry out activities to protect the environment from the effects of, and the effects associated with, the operation of Commonwealth jurisdiction aircraft. This duty is "as far as is practicable", but in a real sense has been identified by the *Report of the Senate Select Committee on Aircraft Noise in Sydney*, November 1995.

The Federal Airports Corporation is also obliged to carry out activities to protect the environment from the effects of, and the effects associated with, the operation and use of aircraft operating to or from Federal airports. The legislation (section 8 of the *Federal Airports Corporation Act 1986*) does not appear to be as comprehensive as that under the Airservices Act.

The policy has identified an affected environment and the parties to that environment.
Both the Air Services Act and the Federal Airports Corporation Act are consistent in their approach to "protecting the environment".

The Environmental Protection Act defines"serious environmental harm" as "environmental harm (other than environmental nuisance) that causes actual or potential harm to environmental values that is irreversible, of a high impact, or widespread;" and "environmental nuisance" as "unreasonable interference or likely interference with environmental values caused by.... noise...."

Some critics would argue that "10% seriously affected" is equivalent to serious environmental harm and that "20% moderately affected" corresponds to nuisance value. But this has yet to be tested.

Resolution of the issue is a necessary debate between the Department, Local government, the community, Airservices Australia and the Federal Airports Corporation.

References

1. "Falling on Deaf Ears", Report of the Senate Select Committee on Aircraft Noise in Sydney, Commonwealth of Australia, November, 1995.

2. The final EIS comprised to Draft EIS (1989) and the EIS supplement (October 1991). The Draft EIS received the 1992 Engineering Excellence Award from the Institution of Engineers, Australia. Nevertheless it "appreciably underestimated" the impact of aircraft noise or residents.
   Estimations of the number of Sydney residents seriously or moderately affected by aircraft noise in the Draft EIS were as follows:

	Moderately affected	Seriously affected
Year 1988	170 100	54 700
Year 2010	82 000	26 900

   The drop in the year 2010 would mostly be done to the replacement of older noise aircraft with quieter models.

3. Aircraft Noise in Australia: A Survey of Community Reaction, NAL Report No.88., A.J. Hede and R.B. Bullen, AGPS, February, 1982.

4. "Impact Descriptors vs. Exposure Indices in Environmental Assessment", A.J. Hede, Acoustics Australia, V.21, No.2, pp41-44,August 1993.

5. "The finding that only 13% of reaction was attributable to the amount of noise only applies in 'long term equilibrium' conditions...". Senate Select Report, p.198.
   Other factors which affect noise reaction include
   - attitudes to the source of a noise
   - personal sensitivity characteristics
   - the activity (eg. sleep) being disturbed
   - the level of background noise
   - fear that an aircraft will crash in the area

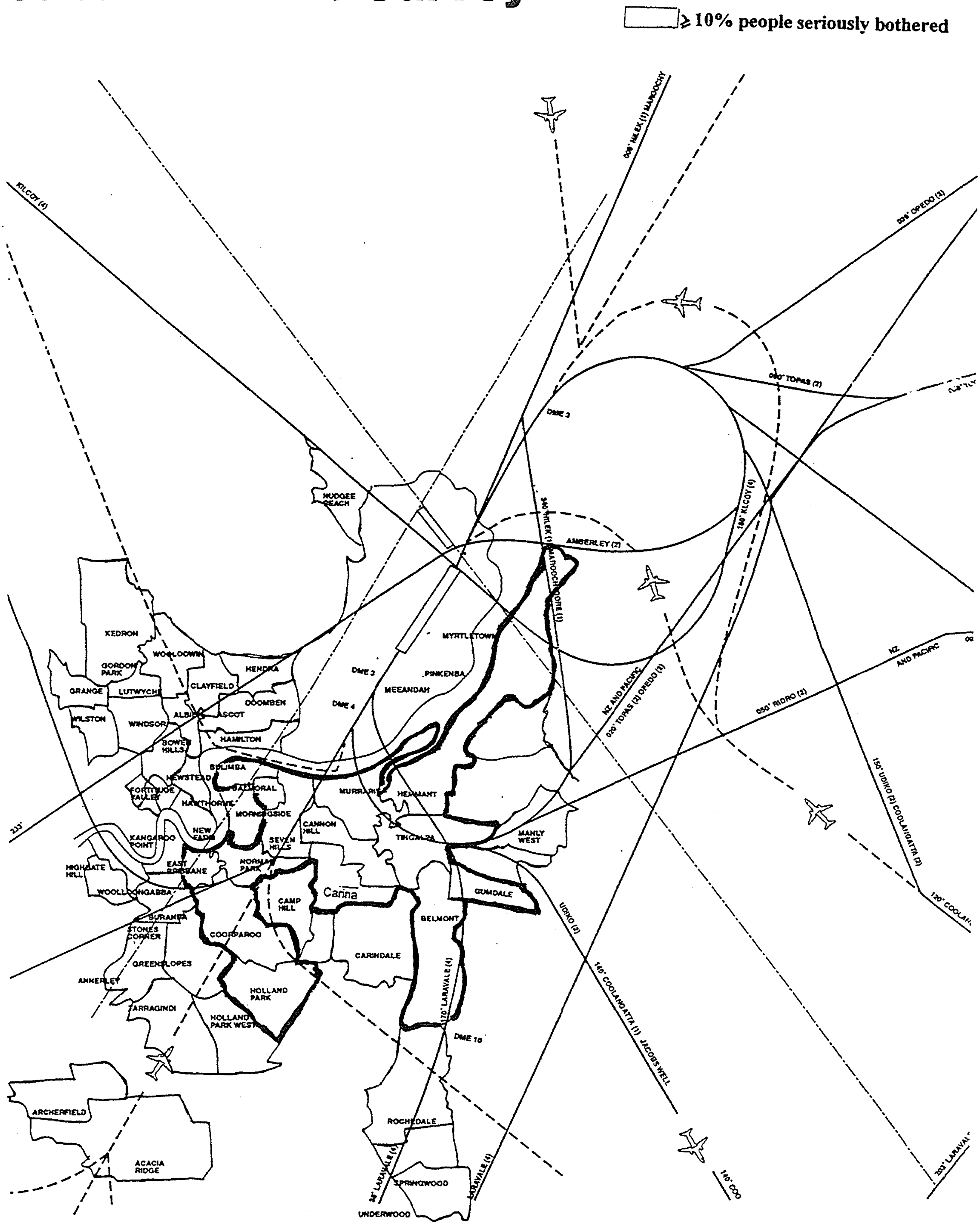
6. NAL Informal Report No.62, AJ Hede and RB Bullen

Purchase > 40 ANEF
Insulate resid 30-40
Public Bldg ≥ 25

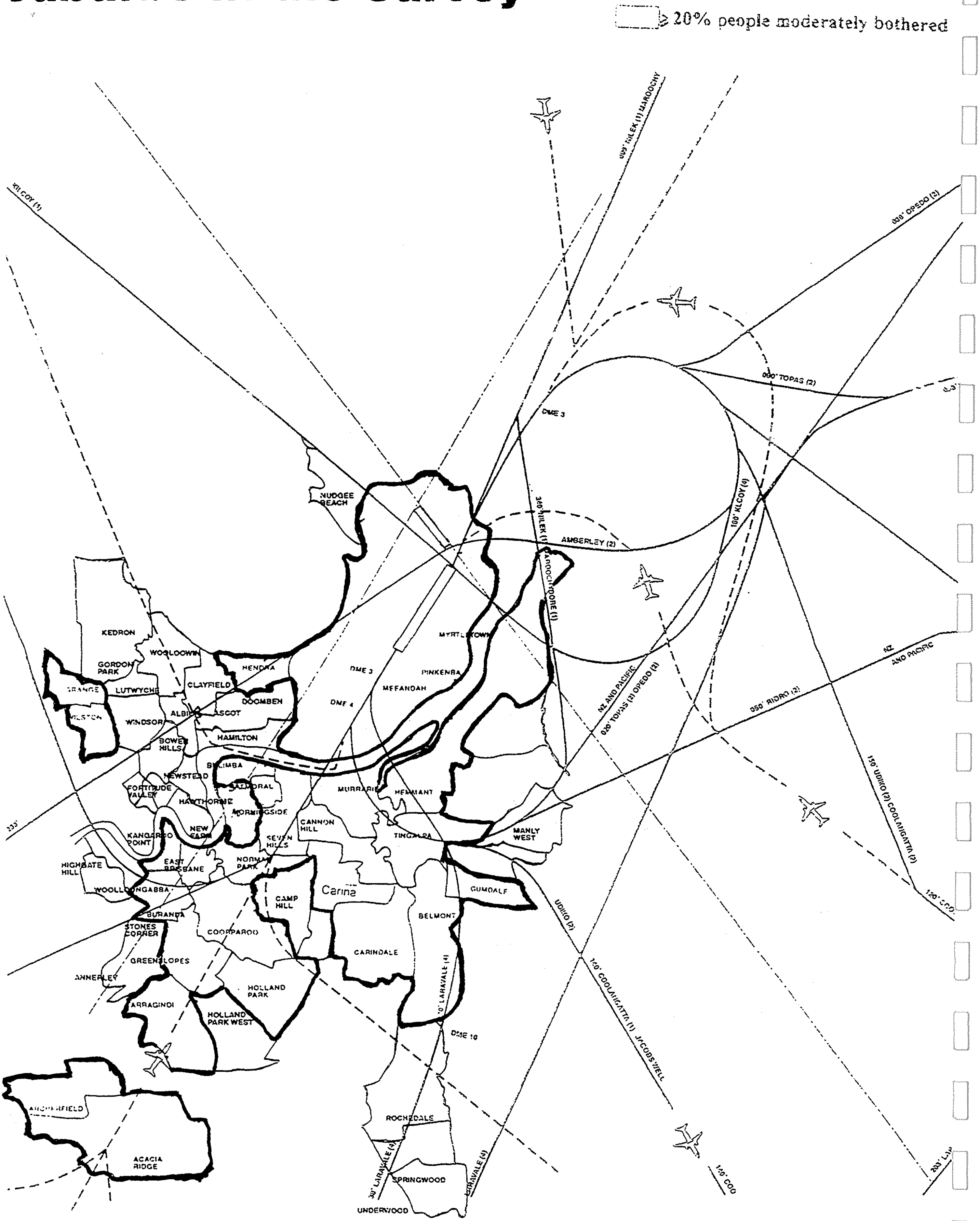
Aircraft Noise Survey

Suburb	*Aircraft noise index as perceived by residents*	*All other noise index as perceived by residents*	*% people seriously bothered*	*% people moderately bothered*	*% people who have complained*	*% people who have complained to FAC*	*% people not satisfied with complaint*	*% restructure flight paths to evenly share*	*% restrict flight paths to one area and compensate*	*% income < $40,000*	*% renting*
I.L.S.	5.33	1.90	38.9	30.6	13.9	8.3	11.1	52.8	19.4	44.4	30.6
Seven Hills	5.32	1.81	27.3	41.9	21.8	16.4	23.6	52.7	10.9	40.0	23.6
Hemmant	5.31	1.55	22.5	50.0	32.5	22.5	25.0	52.5	12.5	45.0	20.0
Norman Park	5.17	1.97	25.0	40.3	18.1	12.5	11.1	58.3	8.3	24.9	22.2
Cannon Hill	5.02	1.96	30.2	34.9	20.6	6.3	15.9	61.9	31.7	44.5	4.8
Balmoral	4.98	1.79	13.0	45.7	8.7	2.2	10.9	50.0	10.9	47.8	10.9
Belmont	4.95	2.06	13.2	54.5	21.0	7.8	18.6	49.1	21.0	22.2	17.4
Gumdale	4.72	1.77	15.8	24.6	25.2	16.6	23.1	38.5	11.5	34.6	3.8
Morningside	4.69	1.93	17.8	42.2	30.0	8.9	12.2	57.8	10.0	47.8	30.0
Tingalpa	4.61	1.57	16.3	44.2	22.1	5.8	18.6	52.3	7.0	26.8	14.0
Holland Park	4.57	1.94	21.4	28.5	21.4	14.3	21.4	57.1	7.1	28.6	28.6
Murarrie	4.51	1.99	17.9	25.7	15.4	5.1	10.3	51.3	10.3	28.3	25.6
East Brisbane	4.50	1.97	14.3	35.7	7.1	7.1	7.1	57.1	28.6	21.4	7.1
Coorparoo	4.43	1.88	14.3	40.8	18.4	6.1	10.2	46.9	12.2	34.7	22.4
Holland Park W	3.86	2.05	7.1	14.2	0.0	0.0	0.0	28.6	7.1	21.4	21.4
Carina	3.76	1.72	12.7	21.8	9.1	3.6	7.3	40.0	20.0	31.0	29.1
Pinkenba	3.75	1.28	3.1	34.4	21.9	18.8	21.9	28.1	3.1	51.1	9.4
Tarragindi	3.67	1.92	7.2	30.4	8.7	2.9	4.3	31.9	10.1	20.2	2.9
Acherfield	3.34	2.29	7.1	28.5	14.3	7.1	7.1	64.3	7.1	50.0	35.7
Kangaroo Point	3.33	2.21	8.3	20.8	12.5	4.2	0.0	29.2	29.2	41.7	37.5
Clayfield	3.20	2.04	2.2	15.5	17.8	4.4	4.0	48.9	6.7	13.3	11.1
Acacia Ridge	3.14	1.62	7.1	21.4	0.0	0.0	0.0	42.9	7.1	71.4	42.9
Bulimba	3.13	1.53	10.9	20.4	12.5	4.7	9.4	43.8	14.1	45.4	26.6
Doomben	3.12	1.97	0.0	19.2	7.7	0.0	3.8	38.5	15.4	24.0	26.9
Hamilton	3.05	1.48	3.6	17.9	16.1	10.7	7.1	44.6	12.5	7.2	16.1
Wooloongabba	2.95	2.32	0.0	20.0	0.0	0.0	0.0	25.0	25.0	45.0	70.0
Gordon Park	2.91	1.90	9.1	13.6	4.5	0.0	0.0	36.4	4.5	36.3	36.4
Camp Hill	2.91	1.19	7.5	18.9	15.1	5.7	11.3	34.0	3.8	41.5	11.3
Newstead	2.79	2.55	0.0	14.2	0.0	0.0	0.0	35.7	7.1	50.0	71.4
Carindale	2.71	1.69	5.1	20.4	6.8	1.7	3.4	32.2	11.9	22.1	5.1
Wilston	2.57	1.63	0.0	21.3	2.0	0.0	0.0	31.9	2.1	38.3	27.7
Wooloowin	2.57	2.26	0.0	0.0	0.0	0.0	0.0	71.4	7.1	50.0	50.0
Hawthorne	2.41	1.64	0.0	10.3	0.0	0.0	0.0	30.8	5.1	30.8	15.4
Hendra	2.39	1.60	1.9	46.2	11.5	5.8	5.8	48.1	11.5	50.0	17.3
Bowen Hills	2.36	1.98	7.1	0.0	0.0	0.0	0.0	35.7	0.0	78.6	64.3
Greenslopes	2.35	1.57	2.9	47.1	2.9	2.9	0.0	32.4	5.9	38.2	29.4
Fortitude Valley	2.32	1.76	0.0	9.1	0.0	0.0	0.0	9.1	9.1	22.7	63.6
Windsor	2.29	1.63	0.0	14.2	0.0	0.0	0.0	64.3	7.1	50.0	14.3
Buranda	2.29	1.68	0.0	14.3	9.5	4.8	9.5	4.8	4.8	42.9	38.1
Grange	2.28	1.66	0.0	21.0	2.0	1.0	0.0	20.9	7.0	30.3	16.3
Kedron	2.28	1.43	0.0	13.9	2.8	0.0	0.0	50.0	19.4	22.3	30.6
Annerley	2.27	2.33	0.0	8.1	2.7	0.0	0.0	21.6	2.7	64.8	37.8
Ascot	2.26	1.78	0.0	10.0	2.0	0.0	2.0	22.0	8.0	26.0	14.0
Albion	2.24	2.00	0.0	0.0	0.0	0.0	0.0	47.1	11.8	58.8	52.9
Stones Corner	2.14	1.84	0.0	0.0	0.0	0.0	0.0	35.7	7.1	57.1	35.7
Nudgee Beach	2.10	1.56	0.0	5.0	15.0	10.0	10.0	10.0	5.0	50.0	10.0
Highgate Hill	2.09	2.03	0.0	9.1	0.0	0.0	0.0	0.0	0.0	27.3	18.2
Lutwyche	2.08	1.78	0.0	16.6	8.3	0.0	0.0	8.3	8.3	41.6	33.3
Newfarm	1.92	1.50	0.0	13.1	7.9	0.0	2.6	10.5	10.5	26.3	47.4
Rochedale, Springwood, Underwood	1.70	1.49	0.0	10.6	0.0	0.0	0.0	42.6	6.4	61.7	19.1
Manly West	1.68	1.82	0.0	6.5	3.2	0.0	32.0	32.3	3.2	61.3	25.8
Total (n=2040)	**3.56**	**1.80**	**10.0**	**26.3**	**12.7**	**5.5**	**8.9**	**41.1**	**11.3**	**35.3**	**23.2**

Suburbs in the Survey



Suburbs in the Survey



DELAYED PULSE SCATTERING BY TURBULENCE

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ABSTRACT

Impulse waveforms are significantly altered by the turbulent atmosphere found outdoors. However it has been shown that the average outdoor waveform is identical to the turbulent free waveshape obtained indoors. Some individual outdoor waveforms have reduced or enhanced peaks while others exhibit delayed energy when compared to an indoor waveform. The frequency spectra for measured waveforms have been examined. As a result of this study a simple filter model modifying only the amplitude of the frequency components has been tested and shown to produce some of the observed waveshapes. However, some questions remain unanswered, in particular what mechanism in the atmosphere can produce a delayed, inverted pulse ?.

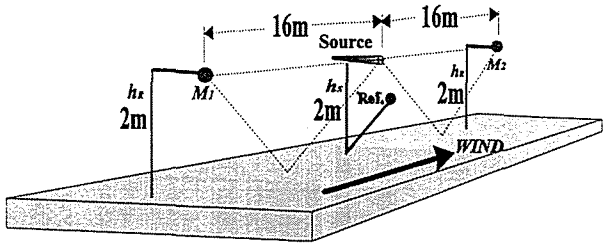
INTRODUCTION

When sound is moving through the atmosphere, the effects of turbulence modify the levels expected in the presence of simple wind and temperature gradients in the atmosphere (Gilbert K.E. et.al.). Impulses of sound provide a useful tool to investigate such effects as they propagate through a snapshot of the atmospheric conditions, and results show up as a change in the waveform. An extensive study in Norway (Kerry G.) has recently investigated effects in a forested area under a wide range of weather conditions, however, the complexity of the location and the distances involved are such that a detailed study of the individual processes occurring along the propagation path is not possible. In general, some of the pulse energy may be scattered away from the propagation path by turbulence, while at other times a component may be scattered towards the receiver from more distant regions. In the latter case, there may be a significant delay and the additional components would then appear in the tail of the pulse. The following investigation seeks to identify any such delayed components and considers mechanisms which may contribute to producing the observed waveforms.

MEASUREMENT GEOMETRY

As shown in Fig. 1, for outdoor measurements an impulse source was positioned midway between two microphones arranged along the dominant wind direction, 2m above a flat grassy surface with a source-receiver separation of 16m (Don C.G. et. al., 1993). This geometry causes about a 1.4ms delay between the surface reflected and the direct pulse, providing a clear interval in which to seek any additional delayed energy scattered by turbulence in the atmosphere. To permit adjustments to be made for shot-to-shot variations in pulse amplitude, a reference microphone was positioned 50cm from the source, in the plane at right-angles to the wind direction. Being close to the source the reference pulse did not experience significant turbulent effects, so its peak value was used to normalise the main receiver waveforms, thereby eliminating variations in the source intensity.

Figure.1: Measurement Geometry



RESULTS AND ANALYSIS

Indoors, where there is no wind, only half of the above geometry was required. Waveforms from 100 indoor shots were recorded and ensemble averaged after adjusting for source variations. Repeating this procedure for over 170 pulses obtained in the middle of a soccer field, with wind speeds varying ranging up to 8 ms^{-1}, it was found that the resultant waveform agreed very closely with the indoor average. Consequently, it was decided to use the average indoor waveform as the reference. Contours, located one and a half standard deviations from this mean (McLeod I.D. et. al.) were calculated at each instant along the time trace and used to set the range of variation expected from source fluctuations in the outdoor situation.

Individual pulse waveforms recorded outdoors were examined to see if and where differences occurred beyond the indoor deviation contours. Results from this study are shown in Table 1. Note that a "peak" in the tail is not necessarily one with a similar shape to the main pulse, rather they may be broad regions where the pressure deviates significantly above or below the contours. Incidentally, the range of reference pulse heights measured indoors and outdoors closely agreed and there was no evidence of delayed energy in the tail of the reference pulse, supporting the assumption that the sound reaching this microphone did not experience significant turbulent effects (Don C.G. et. al. 1995).

From Table 1, it is apparent that wind direction is not a major factor, although there may be a greater tendency for inverted peaks to occur in the downwind condition. About 84% of pulses experience some form of change, with 69% showing delayed energy arriving in the tail. Commonly this delayed energy corresponds to an inverted region compared to the main peak. In no case was this energy greater than 2% of the total pulse energy, compared to 14% changes which frequently occur in the main peak region.

TABLE 1

Percentage of pulses showing changes in waveform (total 178 pulses). Last column gives total of upwind and downwind results. Many pulses occur in several categories.

Categories	Upwind (%)	Downwind (%)	% of Total
No change to pulse	15	18	16
Decreased peak amplitude	43	30	37
Increased peak amplitude	32	20	26
Upright peak in tail	29	21	25
Inverted peak in tail	35	54	44

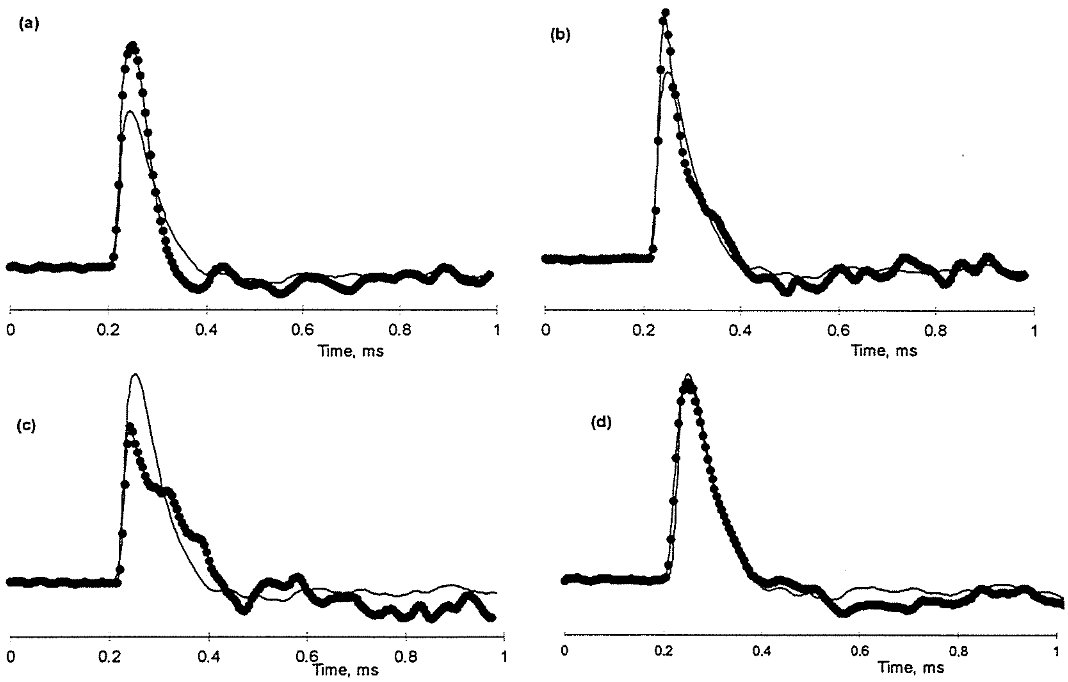
This implies that for our geometry, scattering from regions beyond about 0.6m from the direct path is relatively unimportant. Two other generalisations can be made:

- reduced amplitude pulses are generally broader than the average waveform.
- enhanced amplitude pulses are generally narrower than the average.

Some examples of enhanced and reduced pulse waveforms, compared to the average waveform, are given in Fig. 2 It should be noted that a peak in the tail is not necessarily associated with a significant change in the main pulse shape, Fig. 2(d), nor is a delayed inverted peak in the tail necessarily associated with a positive region in the same pulse.

It is not difficult to visualise a turbulent region, or turbule, which scatters energy out of the most direct path between source and receiver, thereby reducing the peak value and/or broadening the pulse, perhaps producing a shape such as the main peak in Fig. 2(c). However, what mechanism will increase the energy and narrow the pulse to cause those in Fig. 2(a) and (b)? It is even more difficult to visualise a mechanism, in space away from solid reflecting surfaces, which will cause pressure inversion, as required to explain the small inverted peaks in the tail found in about 44% of cases. To gain some insight into what properties a model must have to explain the above phenomena, an investigation of the relative shapes of the frequency spectra of the pulses was undertaken. In particular we sought to determine if it was possible to manipulate the amplitude and phase of a pulse to arrive at the various observed waveforms.

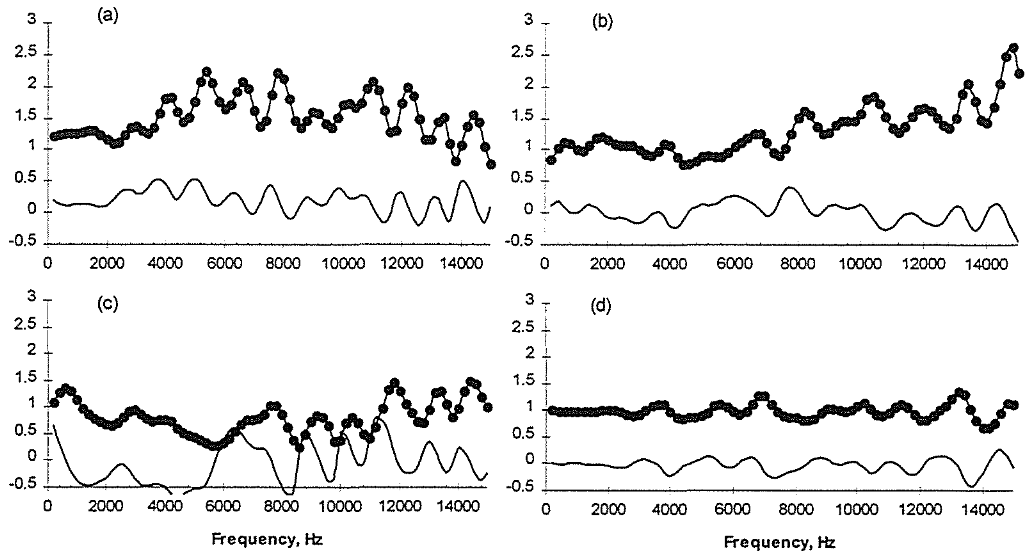
Figure 2: Four Examples of Measured Outdoor Waveforms Superimposed on the Averaged Indoor Waveform (thin line).



By dividing the Fourier components of an individual pulse by the corresponding components of the average pulse, the change in amplitude and phase required to produce a measured waveshape can be displayed. Results for the pulses in Fig. 2(a) to (c) are shown in Fig. 3(a) to (c), while Fig. 3(d) is an example of a comparison between the spectra of two indoor pulses. The variations in the phase difference spectrum for any indoor pair is generally within ± 0.5 radians, while the amplitude ratio is relatively constant about unity. Outdoors, the fluctuations in the phase difference spectrum are similar to the indoor range, however, the amplitude changes are much greater with variations by a factor of two being common. Thus it appears that, at least to a first approximation, we can model propagation through the atmosphere by filtering the amplitudes rather than altering the phase of the frequency components.

Fig. 4 indicates the result of modelling a turbulent atmosphere as a notch filter applied only to the amplitude. The average pulse waveform, shown by the dotted line, has been frequency analysed and the indicated filter applied before reconstituting the waveform. In cases (a) to (c), the filters all have a 2kHz wide bandwidth, with the centre frequency set progressively higher. Partially eliminating the lower frequencies has a more marked effect on the peak height, however, a filter centred around 5kHz, case (b), produces significant broadening along with some reduction in peak height. The effect can be increased by expanding the bandwidth, as in case (f). This produces a resultant waveform not unlike many of the observed pulses, for example waveform (c) in Fig. 2. If a band of frequencies are enhanced, as in cases (e) and (g) of Fig. 4, a peak with an increased height can be produced, however, the resultant pulse does not exhibit the degree of narrowing seen, for example, in waveform (b) of Fig. 2.

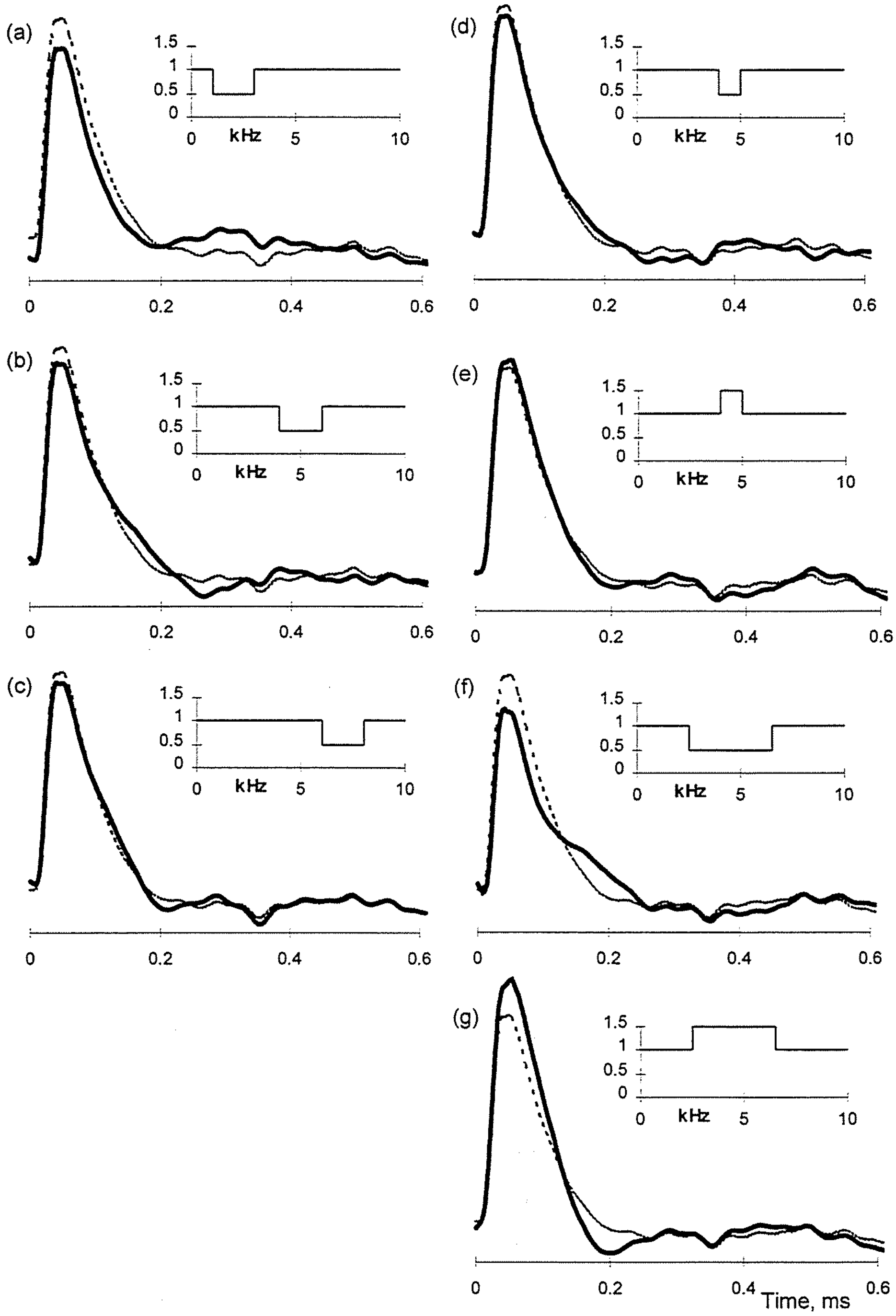
Figure 3: Normalised Relative Amplitude (dotted line) and Phase Difference in Radians (thin line) for Four Waveforms. Cases (a) to (c) Correspond to Pulses (a) to (c) in Fig. 2 Compared with the Average Indoor Pulse while (d) is a Typical Result for Indoor Pulses



Inspection of Fig. 4 indicates that changes in the tail of the pulse do occur as a result of tis simple filtering, although most of the changes are within the limits set by the standard deviation contours described earlier. Using the filter model, changes in the tail are always associated with a change in the peak, so pulses similar to Fig. 2(d) can not be explained using this model. Furthermore, there is a periodic nature to the changes, i.e. a region above the average curve is followed by one below then another above, etc.. Outdoor waveforms rarely exhibit such a pattern. Admittedly, the filters used in Fig. 4 were simple notch ones, however, more complicated shapes have been tried but they have not produced all the required waveshapes.

Attempts have also been made to modify the phase difference spectrum of an indoor pulse: once again resulting shapes do not exhibit the features required by the observed pulse waveforms. None of these modifications to the frequency spectra have been able to invert the main peak which is required in order that a scattered pulse from a region distant from the most direct path to be inverted as is occasionally observed, Fig. 2(d). A further complication is, if such a modification were to be found then why is an inverted direct pulse never detected.?

Figure 4: The Averaged Indoor Pulse (dotted) Compared with a Reconstructed Waveform after being Modified by the Indicated Filter



CONCLUSION

Attempts to model a turbulent atmosphere by a simple filter acting only on the amplitude, and not the phase, of the frequency components of a pulse have produced many of the required waveform features. However, the model leaves much unresolved. Scattering by a turbule can be considered as filtering by deflecting some frequency components out of the direct path. Components scattered from an adjacent but longer path will be delayed causing broadening but it is difficult using a purely scattering model to explain how an enhanced peak with the same rise time as the direct can be formed. Furthermore, no filtering mechanism will leave the main peak unaltered while producing an inverted delayed peak. The delay implies scattering from a region away from the direct path but, as yet, there is no mechanism to explain how the inversion is produced.

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ACOUSTIC PROPERTIES OF A SINGLE-CYLINDER ENGINE EXHAUST

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ABSTRACT

The exhaust noise of a single-cylinder spark ignition (SI) engine was experimentally investigated in order to apply active control to reduce the engine exhaust noise. A pressure transducer was used to measure the exhaust noise pressure in lieu of a conventional microphone. This allowed the exhaust noise to be distinguished from the noise generated from other sources in the engine. The exhaust pressure was measured at a position 10 cm from the exit of the exhaust pipe, which is the position selected for installing the actuator in the control system. The periodicity of the exhaust noise was examined with the ensemble-average of the exhaust pressure signals. The results of ensemble-standard deviation show that the exhaust noise investigated was basically quasi-periodic and its variation between cycles was insignificant. The properties of the exhaust pressure in the frequency domain were analysed using FFT spectral analysis. The power spectra of the exhaust noise pressure varying with engine load are presented and discussed.

INTRODUCTION

The concept of active noise attenuation has become a reality in recent years because new technological developments are providing the opportunity for the advantages to be realised (Leith and Tokhi, 1987, and Stevens and Ahuja, 1991). It has been suggested that a control system based on this idea to reduce the exhaust noise generated by internal combustion (IC) engines is more effective than a conventional muffler.

In order to apply the active control to the engine exhaust noise in a appropriate way, it is necessary to understand the acoustic characteristics of the exhaust gas as well as the causes. Gasper et al. performed frequency and time domain analysis to the exhaust pressure pulses from a small Wankel engine (Gasper et al., 1980). Their objective was to investigate the possibility of using empirical methods as aids in exhaust system design. Measured static pressure of the exhaust gas at different engine speeds was used to derive a simple linear equation for the number of harmonics in the frequency band. The equation indicated that the number of significant harmonics decreased as engine speed increased. They concluded that footholds available in the non-linear theory might eventually bring about improvements in the design of exhaust systems. Prasad and Crocker used a transfer function method with a random excitation source to study acoustic source impedance of a multi-cylinder IC engine (Prasad and Crocker, 1983). In their work, two piezoelectric transducers were used instead of microphones to measure the exhaust acoustic pressure at two locations of the exhaust pipe. The measured pressure at two locations was used to determine the acoustic transfer function between two pressure signals. The effect of signal to noise ratio on the transfer function was discussed. The measured engine internal source impedance was used successfully to predict insertion loss of a muffler and radiated sound pressure from the exhaust system. Desmons, Hardy and Aurgan applied a least square method to characterise an IC engine considered as a noise source (Desmons et al., 1995). In their experiments, only one external microphone was used together with a set of calibration loads. The engine with manifolds was considered as the source to greatly simplify the model. The model gave good agreement between the prediction and experimental results of the dimensionless source impedance and velocity. They concluded that a linear theory could predict the noise level at the output of the exhaust system when the transfer matrix of the silencer was known.

The previous work reviewed above aimed at predicting the exhaust system characteristics for the design of mufflers. A model used to predict the exhaust pressure in an active control system is different from that used in current numerical simulation which is practically impossible for active control due to the long computation time. The prediction in a control system must be completed in a very short time interval to match the control algorithm. On the other hand, the on-line prediction of the exhaust pressure is ideal if the prediction is based on measurements of the sensors already installed on engines such as the speed and load measurements. Lack of such a model requires further understanding of the acoustic characteristics of exhaust gas. The simplification of the model and the methods of prediction may also need to be different from those in the previous work. In the recent course of studies, the authors have experimentally investigated the exhaust noise of a single-cylinder spark ignition (SI) engine for use in active control.

MEASUREMENTS AND DATA ACQUISITION

The schematic description of the test rig is shown in Figure 1. A Magnum Kohler 8 engine was tested in the experiments. It was a four-stroke single cylinder SI engine with 305 cc displacement and 6.0 kW at 3600 RPM. The exhaust system was an instrumented pipe with two bends of 90 degrees, one horizontal at a position 110 mm from the exhaust port and the other vertically at 220 mm from the first bend and 900 mm from the exit of the exhaust pipe. The exhaust pipe was 1230 mm long and the inside diameter was 30 mm. The pipe length was necessary for discharging the exhaust gases to the outside area through a main exhaust pipe. A thermocouple of type K Chroml / Alumeh was installed near the pressure transducer to measure the time average temperature of the exhaust gas. The exhaust pressure was measured by a piezo-electric pressure transducer (model 112B11) produced by PCB Piezotronics Inc. This pressure transducer can operate under high temperature up to $316^{\circ}C$. A cooling jacket was fabricated and the running cold water was used to cool the transducer.

A LabView program was developed for data acquisition and primary data processing. LabView is the software produced by National Instrument Corporation, U.S.A. and provides a general-purpose programming environment. With this software, a virtual instrument can be established to emulate the real instrument. The developed LabView program was run during the experiments to digitise analogue signals, calibrate, perform Fourier transformation and save data. The engine ignition signal was used as a trigger to initiate the data acquisition. The trigger signal and the output of the pressure transducer were digitised and acquired by the LabView program. The selected sample rate was 3 kHz and each realisation had 3000 sample points. The output of the pressure transducer was converted to pressure in kPa and the power spectrum of each realisation was calculated in the LabView program. The LabView program also provided options for saving the data of trigger, exhaust pressure and power spectrum into files in a spreadsheet format.

Measurement of exhaust pressure was conducted at a designated range of engine operating conditions. The engine speed was varied from 1920 to 3000 RPM, and the power output of the engine was varied from 200 to 3100 W. In the designated engine operating conditions, the exhaust pressure was measured at two locations, 110 mm from the exhaust port and 100 mm from the exit of the exhaust pipe as shown in Figure 1.

DISCUSSION

The time series of the exhaust pressure and its power spectrum at engine speed of 3000 RPM and power output of 3200 W are presented in Figure 2. The corresponding noise level measured by a sound meter outside the exhaust pipe and near the exit end was 104 dB. Experience from previous work showed that the spectral characteristics of the exhaust signals were normally dominated by an extensive sequence of discrete tones that were harmonically related to the engine firing frequency (Davies, 1996). The spectrum shown in Figure 2(b) supports the previous work. As the engine tested was four-stroke and the measurement was taken at 3000 RPM, the first peak of amplitude appears at 25 Hz, which is the firing frequency. The sequential harmonics are at $n \times 25$ Hz, where n is an integer greater than 1. The major acoustic energy of the exhaust gas is distributed among the lower frequency components represented by the first six peaks. These frequency components are below 200 Hz and are well known to be difficult to control with conventional mufflers. This can also be seen in Figure 3 which shows a comparison of two spectra of the exhaust pressure measured at the same position, but Figure (b) with a muffler installed at the exit end of the exhaust pipe and Figure (a) without. As shown in the figure, the noise at the frequency band above 300 Hz was reduced by the muffler. However, the amplitude of the high frequency components in the original exhaust pressure without a muffler is relatively small. Therefore the reduction of the exhaust noise in the high frequency band with the muffler is in fact insignificant. On the other hand, the amplitudes of the low frequency harmonics with a muffler are greater than that without a muffler due to the resistance to the exhaust flow caused by the installation of the muffler.

This experimental investigation also shows that the harmonic frequency of the exhaust pressure is independent of the positions within the pipe. In Figure 4 are the spectra of exhaust pressure measured at two positions (see Figure 1) along the exhaust pipe. The pipe length between the two positions was 1030 mm. As shown in the figure, the amplitude of the harmonic components decreases downstream but the harmonic frequencies remain unchanged along the exhaust pipe.

It is proposed by the authors that the signals driving the actuator in the active control system be generated by predicting the original exhaust pressure in terms of engine speed and load. This will allow the removal of the transducer detecting the exhaust noise, which exists in a conventional active control system. One of the fundamental aims of the present work was to acquire the information from experiments for deriving the mathematical models which describe the change of exhaust pressure with engine operating conditions. This model will be used to numerically generate the signals to drive the actuator in the active control system.

There are two ways to describe the acoustic characteristics of the exhaust noise emission (Davies 1996). The first method is to perform the noise measurements in the frequency domain with the associated subjective and legislative assessments. The second method is to classify the observed sound emission into a sequence of harmonic orders of the engine rational frequency, each number covering the whole range of operating speed at either full or part load. Such a description remains specific to each particular engine configuration and does not necessarily directly represent the overall acoustic performance of exhaust systems. The second method has been adopted in the present work with consideration for the requirements of the control system.

The power spectra of the exhaust pressure in low frequency range at a steady engine speed and in a range of varying engine loads are presented in Figure 5. It can be seen that the low frequency harmonics of the exhaust pressure dominate the exhaust noise. Since the frequency of each of the harmonic components is an integer times the engine firing frequency, the exhaust pressure pulse during one engine cycle can be described as follows:

$$p(t) = R_0 + \sum_{i=1}^{N} R_i\,(\cos\omega_i t + \phi_i) + Z_0$$

Where *p(t)* is the instantaneous exhaust static pressure as a function of time or crank angle, R_0 the steady component of the exhaust pressure, R_i the amplitude of the harmonic component with frequency of ω_i and phase shift of ϕ_i. N is the number of harmonic components and Z_0 a stationary random noise. R_0, R_1 and ϕ_i vary with engine operating conditions such as engine speed and load, while ω_i depends only on the engine speed and is determined by:

$$\omega_i = \frac{i \cdot RPM}{120}\ (1/s) \qquad i = 1, 2, 3, \ldots, N$$

The amplitude of the harmonic components R_i varies with engine load at a constant engine speed and this relation is shown in Figure 5. With the change of engine load, the frequency components of the exhaust pressure remain unchanged and only depend upon the engine speed. At 3000 RPM, the frequency component of 50 Hz (twice the engine firing frequency) has the greatest amplitude. The amplitudes of harmonic components in the power spectra are plotted against the engine load in Figure 6 for 3000 RPM. The non-linear relation between the amplitudes and engine load should be noted. The gradient of the amplitude is small at light engine load but becomes large when engine load is heavier than 2500 W. The amplitude gradient of the frequency components of 100 Hz, 125 Hz and 150 Hz is small until engine load is heavier than 2500 W.

As the active control system is proposed to predict the exhaust pressure from engine operating conditions, the prediction of exhaust noise at steady engine operating condition will be concentrated in one engine cycle if the temporal trace of the exhaust noise is reasonably repeatable from cycle to cycle. The cyclic variation of the exhaust pressure is examined by ensemble-averaging exhaust pressure of twenty engine cycles. Figure 7 shows the ensemble-average and ensemble-standard deviation of the exhaust pressure at engine speed 2800 RPM and load 2480 W. The traces in grey colour are the twenty exhaust pressure time series to be ensemble-averaged and the thick black trace the ensemble-average. The maximum standard deviation is about 1.7 kPa, which appears at the phase of the peak value of the ensemble-averaged exhaust pressure. By observing the whole cycle, the standard deviation is mostly less than 0.5 kPa. The cyclic deviation is not significant, as shown by both ensemble-average and ensemble standard deviation.

CONCLUSIONS

The frequency properties of the engine exhaust noise generated by a single-cylinder small engine are characterised by harmonic components related to engine operating conditions. The frequencies of the harmonic components structuring the exhaust pressure pulses depend only on the engine firing frequency. The amplitudes of the harmonic components decrease along the exhaust pipe but the frequencies of the harmonics remain the same.

The analysis of experimental results demonstrates that quantitatively, the amplitude of the harmonic frequencies and the engine load at a steady engine operating condition have a certain non-linear correlation. The cyclic variation of the exhaust pressure pulses at a steady engine speed is insignificant. As active control is most applicable to periodic or quasi-periodic signals, the feasibility of the application of active control to exhaust noise reduction has been confirmed. The prediction of the pressure pulses constructed by harmonic components in terms of engine operating conditions is promising.

ACKNOWLEDGEMENT

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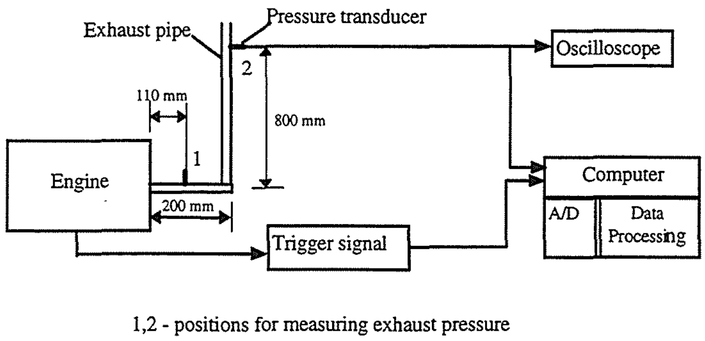


Figure 1: Schematic view of the test rig

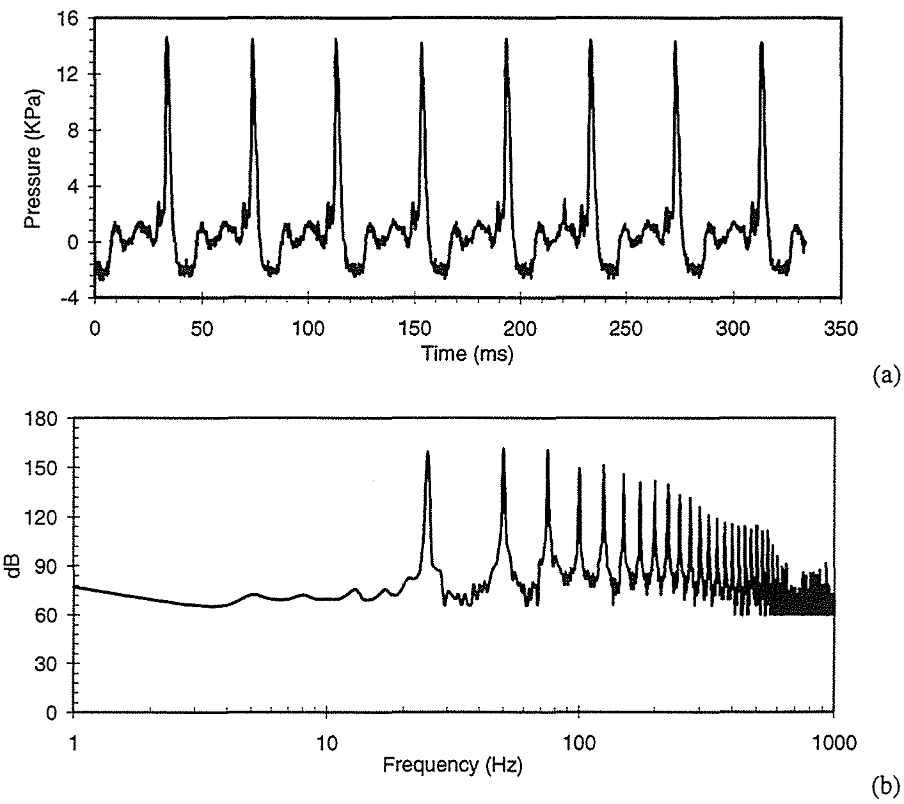
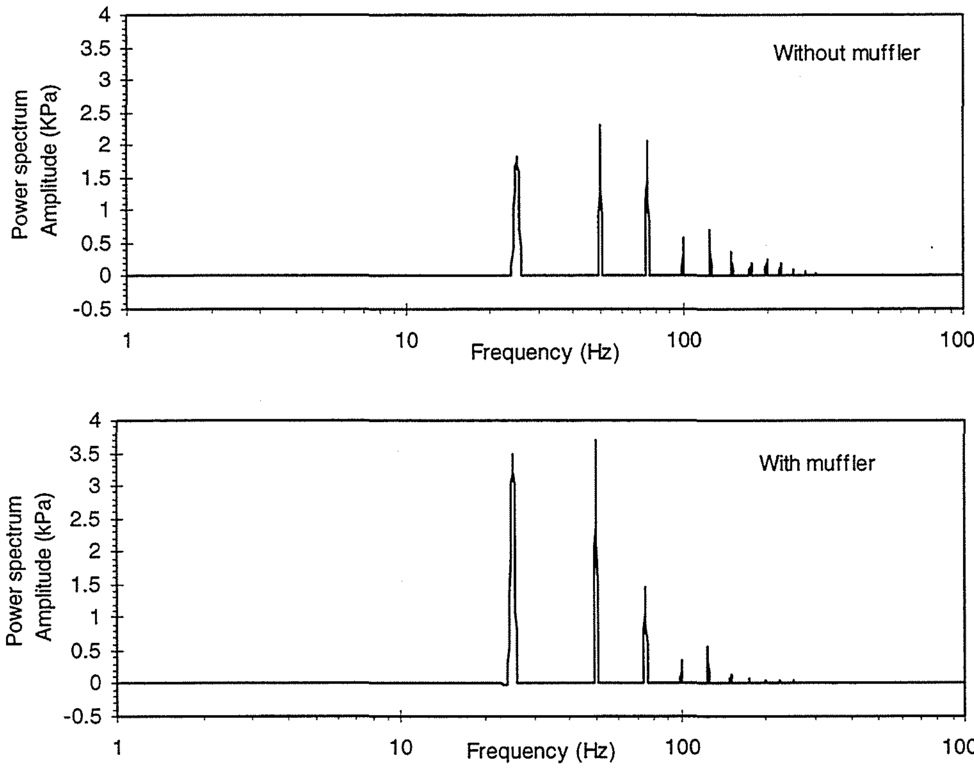


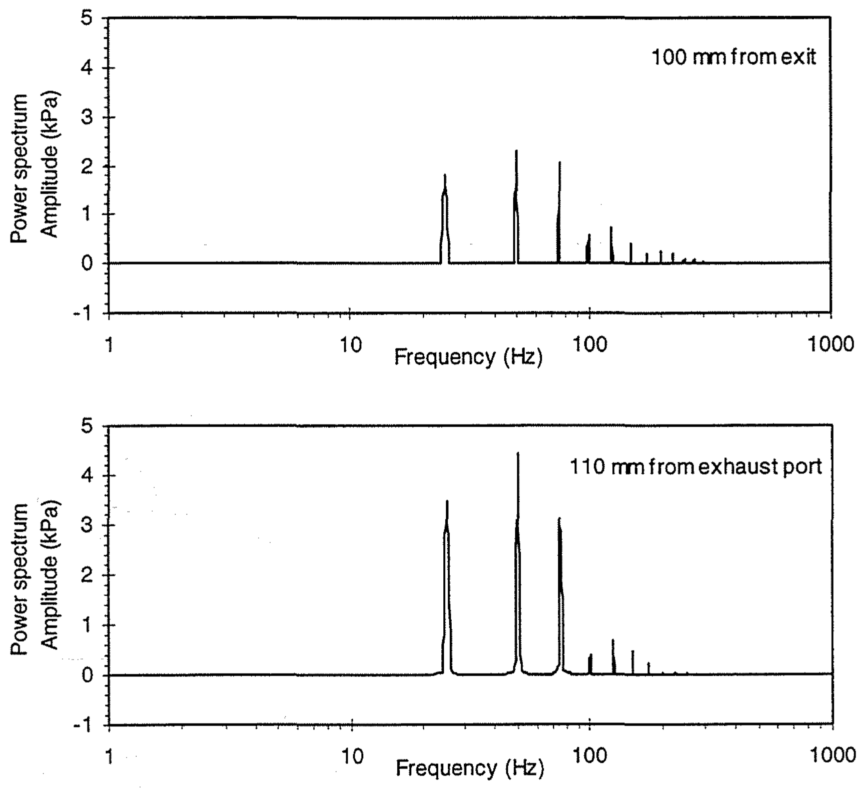
Figure 2: Exhaust pressure and its power spectrum at 3000 RPM and 3200 W, 100 mm from the exit of the exhaust pipe without a muffler



(a)

(b)

Figure 3: Spectra of exhaust pressure at 3000 RPM and 3200 W,
(a) without muffler, (b) with muffler



(a)

(b)

Figure 4: Spectra of the exhaust pressure measured at two positions of the exhaust pipe under the same engine operating condition: 3000 RPM and power output of 3200 W

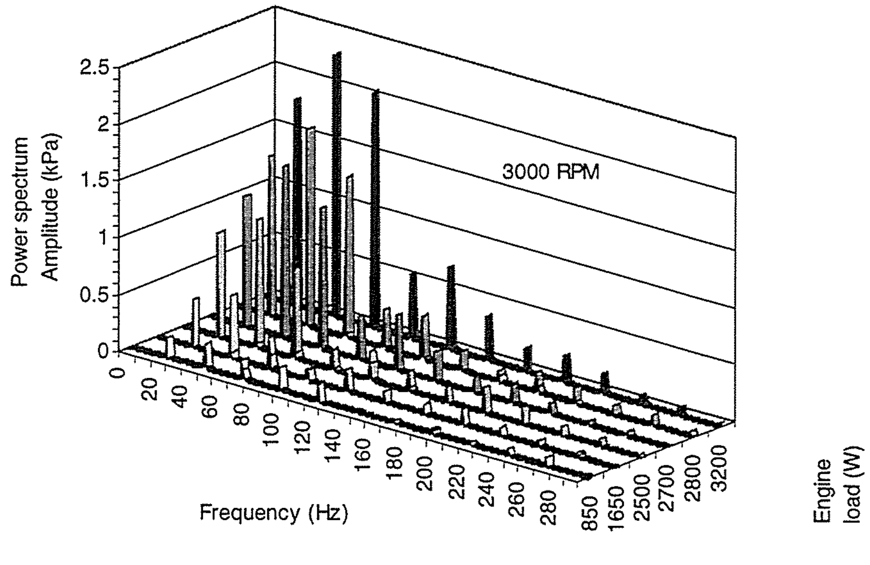


Figure 5: 3-D spectra of exhaust pressure at 3000 RPM and varying engine load without muffler

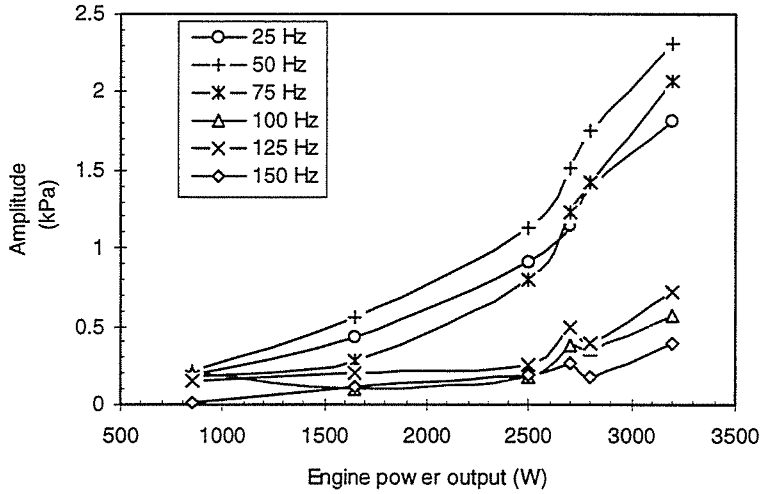


Figure 6: Variation of peak harmonics with engine load

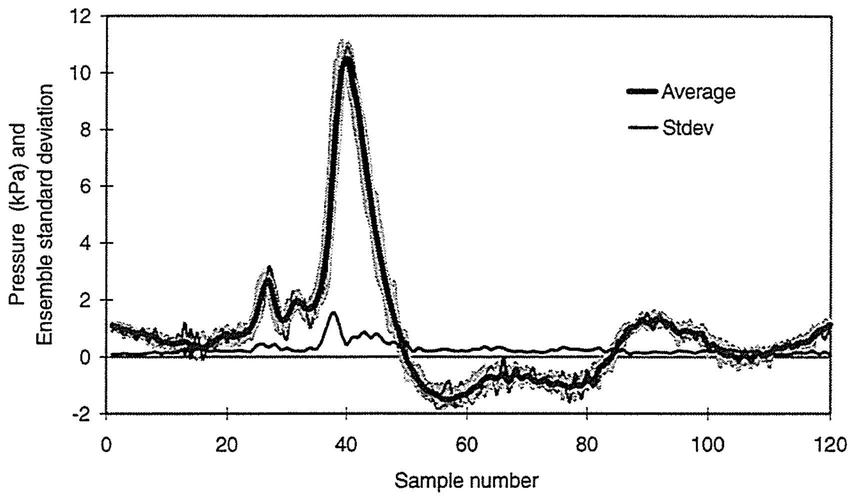


Figure 7: Cyclic variation of exhaust pressure signals at 2800 RPM and 2500 W

RECENT PROGRESS IN THE PREDICTION OF VIBROACOUSTIC RESPONSE

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ABSTRACT

The most recent progress in the development of methods for prediction of vibroacoustic response is reviewed in this paper. An analytical method which is computational efficient is proposed for predicting band-limited response of acoustic-structural coupled systems.

INTRODUCTION

Accurate modelling of vibroacoustic systems is significantly dependent on the accurate description of boundary conditions and coupling between mechanical and acoustical media. In the low frequency range, subsystems in a vibroacoustic system have low modal densities and vibrational wavelengths are also larger. The system response can then be modelled using relatively few modes and well established deterministic techniques such as the Finite Element Method (FEM) and the Boundary Element Method (BEM) are commonly applied for this frequency range. Beyond this frequency range which are the medium and high frequency ranges, subsystems are modally-dense. Details of boundary conditions are now becoming significant and accurate modelling of the system response requires information of a large number of modes of the subsystems. Consequently, substantial computational effort is required if low frequency techniques such as FEM and BEM are still employed. In addition, vibrational wavelengths are also shorter and a large number of elements are required for resolving the details of the system response. This will be very inefficient in terms of computational time as well as computer storage requirement (Pietrzyk, 1995).

In recent years, the prediction of vibroacoustic response has gained a considerable interest. Many investigations have been aimed at developing computational efficient analytical, experimental, statistical or numerical techniques capable of handling the large number of modes beyond the low frequency range. Recently, a subdomain decomposition technique (Ramiandrasoa et al, 1996) has been proposed for computing eigenproperties and frequency response of modally-dense acoustic cavities. A large acoustic cavity is decomposed into smaller sub-cavities of lower modal densities and each sub-cavity is separated from the others by fictitious interfaces. Eigenproperties of each sub-cavity subject to continuity at all interfaces are then obtained numerically and eigenproperties of the original large cavity are synthesised by using a modal synthesis approach. Acoustical response of the cavity can then be predicted. Similar subdomain decomposition approach has also been used by Terao and Sekine (1996) but in a later step, they used partial Gauss-Jordan elimination technique to numerically solve the sound pressure and particle velocity at the fictitious interfaces. Computer storage has been shown to be reduced if the subdomain decomposition technique is employed but the computational time remains the same as for the prediction of response of the original large cavity without subdomain decomposition.

A new numerical method called the coupled wave method (Desmet et al, 1996) has also been developed recently for the prediction of vibroacoustic response. In contrast to the classical finite element technique where the structural and acoustical domains of the vibroacoustic system are divided into small elements and the dynamic equations within each element are solved, the entire subsystems are now described by complex structural and acoustical wave propagation functions. These functions are solutions of the structural and acoustical wave equations. By applying boundary conditions and using a weighted residual formulation where the acoustical plane wave functions are chosen as weighting functions, a set of algebraic equations can be obtained and the system response can then be predicted. By comparison to the finite element method, improvement in calculation time has been achieved if this coupled wave method were employed for the prediction of vibroacoustic response.

A number of simplified approaches have also been developed to reduce the computational effort. For example, analytical expressions for the prediction of vibroacoustic response were derived for Asymptotic Modal Analysis (AMA)(Kubota et al, 1988). However results obtained were of Statistical Energy Analysis (SEA) type and were based on averaged quantities or parameters. Thus AMA is more adapted to the prediction in high frequency range. Application of this method is uncertain in both low and medium frequency ranges because there are insufficient overlapped modes and the analysis frequency band contains too few resonance modes.

Recently, Bonilha and Fahy (1995) have proposed a technique which combines a probabilistic model with a deterministic model for vibroacoustic modelling. The technique called hybrid probabilistic-deterministic method was established and a probabilistic description is employed for a subsystem with modal density high enough to justify an SEA model whilst a deterministic description is employed for the subsystem with sparsely distributed modes and amenable to FEM modelling. This approach allows compression of the required information for characterising the dynamics of the vibroacoustic system by using an approximation on the modally-dense subsystem.

About two decades ago, band-limited power flow expressions have been established by Pope and Wilby (1977, 1980) for sound transmission into enclosures but the development was aimed for low frequency analysis and for analysis in the high frequency extreme. An analytical method is proposed in this paper for predicting band-limited energies of subsystems in acoustic-structural coupled systems. The present method is based on the classical modal coupling analysis but it is comparatively more computational efficient especially for analysis beyond the low frequency range. For analysis in the medium and high frequency ranges, manipulation of large complex matrices as in the classical modal coupling analysis will be avoided if the present method is employed. Improvement in calculation time can also be achieved with the proposed method since calculations of energies at each single frequency and subsequent frequency averagings for band-limited energies have been avoided. Instead the band-limited energies can be predicted straight away in one calculation for each subsystem given the lower and upper frequency limits and centre frequency of the band. In this paper, the application of the proposed method for both narrow and broad bands analyses is shown with some numerical examples and its computational efficiency and prediction accuracy are demonstrated.

MODAL COUPLING ANALYSIS

In an acoustic-structural coupled system which consists of an enclosed acoustical medium and a structural medium, the sound pressure $p(\vec{r},\omega)$ describes the steady-state acoustical response and the structural velocity $v(\vec{\sigma},\omega)$ describes the steady-state structural response. To use the modal coupling method, the responses are expressed in orthogonal functions (mode shape functions) as follows :

$$p(\vec{r},\omega) = \sum_{i=1}^{N} P_i(\omega)\Phi_i(\vec{r}), \tag{1}$$

$$v(\vec{\sigma},\omega) = \sum_{j=1}^{M} V_j(\omega)S_j(\vec{\sigma}) \tag{2}$$

where P_i and V_j are the complex pressure and velocity amplitudes of the ith acoustic mode and jth structural mode. $\Phi_i(\vec{r})$ and $S_j(\vec{\sigma})$ are acoustic and structural mode shape functions. $\vec{r}$ and $\vec{\sigma}$ are respectively the location vectors in the enclosed sound field and on the structural surface. Substituting Eqs. (1) and (2) into the corresponding acoustic and structural wave equations and applying the Green's function method (Dowell et al. 1977), it can be shown that :

$$\begin{bmatrix} P_1 \\ M \\ P_N \end{bmatrix} = \begin{bmatrix} 1+\sum_{j=1}^{M}\dfrac{B_{1,j}B_{j,1}}{\chi_{a1}\chi_{sj}} & \Lambda & \sum_{j=1}^{M}\dfrac{B_{1,j}B_{j,N}}{\chi_{a1}\chi_{sj}} \\ M & O & M \\ \sum_{j=1}^{M}\dfrac{B_{N,j}B_{j,1}}{\chi_{aN}\chi_{sj}} & & 1+\sum_{j=1}^{M}\dfrac{B_{N,j}B_{j,N}}{\chi_{aN}\chi_{sj}} \end{bmatrix}^{-1} \begin{bmatrix} Q_1 \\ M \\ Q_N \end{bmatrix}, \tag{3}$$

$$\begin{bmatrix} V_1 \\ M \\ V_M \end{bmatrix} = -\frac{1}{\rho_0 c_0} \begin{bmatrix} \chi_{s1}+\sum_{i=1}^{N}\dfrac{B_{1,i}B_{i,1}}{\chi_{ai}} & \Lambda & \sum_{i=1}^{N}\dfrac{B_{1,i}B_{i,M}}{\chi_{ai}} \\ M & O & \\ \sum_{i=1}^{N}\dfrac{B_{M,i}B_{i,1}}{\chi_{ai}} & & \chi_{sM}+\sum_{i=1}^{N}\dfrac{B_{M,i}B_{i,M}}{\chi_{ai}} \end{bmatrix}^{-1} \begin{bmatrix} \sum_{i=1}^{N} B_{1,i}Q_i \\ M \\ \sum_{i=1}^{N} B_{M,i}Q_i \end{bmatrix}. \tag{4}$$

In above, the sound field is excited by a distributed steady-state acoustic source inside the enclosure and the structure is driven through coupling with the sound field. Similar derivation can also be performed for the case where the structure is excited by some mechanical forces and the sound field driven through coupling but will not be considered here. The quantities in Eqs. (3) and (4) are as follows :

$$\chi_{ai} = \frac{jM_{ai}}{\omega\rho_0 c_0 A_s}(\omega_{ai}^2 - \omega^2 + j\eta_{ai}\omega_{ai}^2), \tag{5}$$

$$\chi_{sj} = \frac{jM_{sj}}{\omega\rho_0 c_0 A_s}(\omega_{sj}^2 - \omega^2 + j\eta_{sj}\omega_{sj}^2), \tag{6}$$

$$Q_i = \frac{-\rho_0 c_0}{A_s\chi_{ai}}\int_{V_0} q\Phi_i \;\, dV, \tag{7}$$

$$B_{j,i} = \frac{1}{A_s}\int_{A_s} \Phi_i(\vec{r})S_j(\vec{\sigma}) \;\, d\sigma \tag{8}$$

where M_{ai}, M_{sj}, ω_{ai}, ω_{sj} η_{ai}, η_{sj} are acoustic and structural modal masses, resonance frequencies and loss factors respectively. $B_{j,i}$ is the modal coupling coefficient between the ith acoustic mode and jth structural mode, $q(\vec{r},\omega)$ is volume velocity of the sound source per unit volume of the enclosure, ρ_0 is air density, c_0 is speed of sound in air and A_s is surface area of the structure.

Steady-state acoustic energy of the sound field and vibrational energy of the structure can then be obtained once the sound pressure and velocity vectors are solved from Eqs. (3) and (4) and are given by :

$$E_a = \frac{1}{2\rho_0^2 c_0^2}\sum_{i=1}^{N} P_i P_i^* M_{ai}, \tag{9}$$

$$E_s = \frac{1}{2}\sum_{j=1}^{M} V_j V_j^* M_{sj}. \tag{10}$$

* denotes complex conjugate of the quantity, V_0 is volume of the enclosure and m_s is mass of the structure.

BAND-LIMITED ENERGIES

To formulate analytical expressions for band-limited energies of the system, a number of approximations will be employed. Detailed derivations, explanation and justifications of all approximations are given by Sum and Pan (1996) and only a summary of results will be presented here.

For the sound field, ignoring non-diagonal terms in the pressure transfer matrix in Eq. (3) (ie. the matrix to be inverted), expanding each complex pressure amplitude, substituting all amplitudes into Eq. (9) and then performing an averaging over a frequency band, it can be shown that the band-limited acoustic energy is given as :

$$(E_a)_{\Delta\omega} = \frac{\rho_0^2 c_0^2}{2\Delta\omega}\sum_{i=1}^{N}\frac{[\int_{V_0} q\Phi_i dV]^2}{M_{ai}}\int_{\Delta\omega}\frac{\omega^2}{[(\overline{\omega}_{ai}^c)^2 - \omega^2]^2 + (\overline{\eta}_{ai}^c \omega_{ai}^2)^2}\, d\omega \tag{11}$$

where $\Delta\omega$ is the frequency bandwidth. $\overline{\omega}_{ai}^c$ is the coupled or effective resonance frequency and $\overline{\eta}_{ai}^c$ is the effective loss factor of the ith acoustic mode due to coupling and are given by :

$$(\overline{\omega}_{ai}^c)^2 = \omega_{ai}^2 - \frac{(A_s\rho_0 c_0)^2\omega_0}{M_{ai}}\sum_{j=1}^{M}\frac{B_{j,i}^2}{\Delta\omega_{sj}^{3dB} M_{sj}}\left[\frac{\overline{\varepsilon}_{sj}}{\overline{\varepsilon}_{sj}^2 + 1}\right], \tag{12}$$

$$\overline{\eta}_{ai}^c = \eta_{ai} + \frac{(A_s\rho_0 c_0)^2\omega_0}{M_{ai}\omega_{ai}^2}\sum_{j=1}^{M}\frac{B_{j,i}^2}{\Delta\omega_{sj}^{3dB} M_{sj}}\left[\frac{1}{\overline{\varepsilon}_{sj}^2 + 1}\right]. \tag{13}$$

$\Delta\omega_{sj}^{3dB}$ is the half-power bandwidth of uncoupled jth structural mode, ω_0 is the centre frequency of the band and

$$\overline{\varepsilon}_{sj} = \frac{2}{\Delta\omega_{sj}^{3dB}}\left(\omega_{sj} - \omega_0\right). \tag{14}$$

Both Eqs. (12) and (13) are invalid for the first acoustic mode with natural frequency 0 Hz and instead, the Helmholtz resonance frequency and loss factor (Pan and Elliott, 1995) should be used when evaluating band-limited acoustic energy from Eq. (11).

For analysis beyond the low frequency range, shift in natural frequency of higher order acoustic modes due to coupling with structural modes is relatively weak and thus it will be assumed that $\overline{\omega}^c_{ai} \approx \omega_{ai}$. Complex integrations can be performed analytically for Eq. (11) which yield :

$$\int_{\Delta\omega} = I(\omega_U, \omega_{ai}, \overline{\eta}^c_{ai}) - I(\omega_L, \omega_{ai}, \overline{\eta}^c_{ai}) \tag{15}$$

where

$$\begin{aligned} I(\omega, \omega_{ai}, \overline{\eta}^c_{ai}) = & \frac{(c_{ai}A_{ai} + d_{ai}B_{ai})}{2(A^2_{ai} + B^2_{ai})} \ln\left[\frac{(\omega + A_{ai})^2 + B^2_{ai}}{(\omega - A_{ai})^2 + B^2_{ai}}\right] + \\ & \frac{(c_{ai}B_{ai} - d_{ai}A_{ai})}{2(A^2_{ai} + B^2_{ai})} \tan^{-1}\left[\frac{4B_{ai}\omega(\omega^2 - A^2_{ai} - B^2_{ai})}{(\omega^2 - A^2_{ai} - B^2_{ai})^2 - (2B_{ai}\omega)^2}\right], \end{aligned} \tag{16}$$

$$A_{ai} = \frac{\omega_{ai}}{\sqrt{2}}\sqrt{1 + \sqrt{1 + \overline{\eta}^{c2}_{ai}}}, \tag{17a}$$

$$\approx \omega_{ai} \quad \text{if} \quad \overline{\eta}^c_{ai} << 1, \tag{17b}$$

$$B_{ai} = \frac{\omega_{ai}}{\sqrt{2}}\sqrt{\sqrt{1 + \overline{\eta}^{c2}_{ai}} - 1}, \tag{18a}$$

$$\approx \frac{\overline{\eta}^c_{ai}\omega_{ai}}{2} \quad \text{if} \quad \overline{\eta}^c_{ai} << 1, \tag{18b}$$

$$c_{ai} = -0.5, \tag{19}$$

$$d_{ai} = 0.5/\overline{\eta}^c_{ai}. \tag{20}$$

ω_U and ω_L are respectively the upper and lower frequency limits of the frequency band under consideration. Value of the inverse tangent term in Eq. (16) varies from 0 to 2π.

Similar derivation can be performed for the structural part of the vibroacoustic system. By ignoring non-diagonal term in Eq. (4), expanding each complex velocity amplitude, substituting all amplitudes into Eq. (10), neglecting acoustic cross-product terms and performing an averaging over a frequency band, the band-limited vibrational energy of the structure can be written as :

$$\begin{aligned} (E_s)_{\Delta\omega} = & \frac{(\rho_0 c_0)^4 A^2_s}{2\Delta\omega} \sum_{j=1}^{M} \frac{1}{M_{sj}} \sum_{i=1}^{N} \frac{B^2_{j,i}[\int_{V_0} q\Phi_i dV]^2}{M^2_{ai}} \\ & \int_{\Delta\omega} \frac{\omega^4}{\left[[(\overline{\omega}^c_{sj})^2 - \omega^2]^2 + (\overline{\eta}^c_{sj}\omega^2_{sj})^2\right]\left[(\omega^2_{ai} - \omega^2)^2 + (\eta_{ai}\omega^2_{ai})^2\right]} d\omega. \end{aligned} \tag{21}$$

$\overline{\omega}^c_{sj}$ is now the coupled or effective resonance frequency and $\overline{\eta}^c_{sj}$ is the effective loss factor of the jth structural mode due to coupling where

$$(\overline{\omega}_{sj}^{c})^2 = \omega_{sj}^2 - \frac{(A_s\rho_0 c_0)^2\omega_0}{M_{sj}}\sum_{i=1}^{N}\frac{B_{j,i}^2}{\Delta\omega_{ai}^{3dB}M_{ai}}\left[\frac{\overline{\varepsilon}_{ai}}{\overline{\varepsilon}_{ai}^2+1}\right], \tag{22}$$

$$\overline{\eta}_{sj}^{c} = \eta_{sj} + \frac{(A_s\rho_0 c_0)^2\omega_0}{M_{sj}\omega_{sj}^2}\sum_{i=1}^{N}\frac{B_{j,i}^2}{\Delta\omega_{ai}^{3dB}M_{ai}}\left[\frac{1}{\overline{\varepsilon}_{ai}^2+1}\right]. \tag{23}$$

$\Delta\omega_{ai}^{3dB}$ is the half-power bandwidth of uncoupled ith acoustic mode and

$$\overline{\varepsilon}_{ai} = \frac{2}{\Delta\omega_{ai}^{3dB}}(\omega_{ai} - \omega_0). \tag{24}$$

Again by neglecting shift in natural frequency of higher order structural modes due to coupling ($\overline{\omega}_{sj}^{c} \approx \omega_{sj}$) for medium and high frequency analyses and performing analytical integration for Eq. (21),

$$\begin{aligned}\int_{\Delta\omega} = {} & I_1(\omega_U,\omega_{ai},\omega_{sj},\eta_{ai},\overline{\eta}_{sj}^{c}) - I_1(\omega_L,\omega_{ai},\omega_{sj},\eta_{ai},\overline{\eta}_{sj}^{c}) + \\ & I_2(\omega_U,\omega_{ai},\omega_{sj},\eta_{ai},\overline{\eta}_{sj}^{c}) - I_2(\omega_L,\omega_{ai},\omega_{sj},\eta_{ai},\overline{\eta}_{sj}^{c})\end{aligned} \tag{25}$$

where

$$\begin{aligned}I_1(\omega,\omega_{ai},\omega_{sj},\eta_{ai},\overline{\eta}_{sj}^{c}) = {} & \frac{(c_{j,i}A_{sj} + d_{j,i}B_{sj})}{2(A_{sj}^2 + B_{sj}^2)}\ln\left[\frac{(\omega + A_{sj})^2 + B_{sj}^2}{(\omega - A_{sj})^2 + B_{sj}^2}\right] + \\ & \frac{(c_{j,i}B_{sj} - d_{j,i}A_{sj})}{2(A_{sj}^2 + B_{sj}^2)}\tan^{-1}\left[\frac{4B_{sj}\omega(\omega^2 - A_{sj}^2 - B_{sj}^2)}{(\omega^2 - A_{sj}^2 - B_{sj}^2)^2 - (2B_{sj}\omega)^2}\right],\end{aligned} \tag{26}$$

$$\begin{aligned}I_2(\omega,\omega_{ai},\omega_{sj},\eta_{ai},\overline{\eta}_{sj}^{c}) = {} & \frac{(e_{j,i}C_{ai} + f_{j,i}D_{ai})}{2(C_{ai}^2 + D_{ai}^2)}\ln\left[\frac{(\omega + C_{ai})^2 + D_{ai}^2}{(\omega - C_{ai})^2 + D_{ai}^2}\right] + \\ & \frac{(e_{j,i}D_{ai} - f_{j,i}C_{ai})}{2(C_{ai}^2 + D_{ai}^2)}\tan^{-1}\left[\frac{4D_{ai}\omega(\omega^2 - C_{ai}^2 - D_{ai}^2)}{(\omega^2 - C_{ai}^2 - D_{ai}^2)^2 - (2D_{ai}\omega)^2}\right],\end{aligned} \tag{27}$$

$$A_{sj} = \frac{\omega_{sj}}{\sqrt{2}}\sqrt{1 + \sqrt{1 + \overline{\eta}_{sj}^{c2}}}, \tag{28a}$$

$$\approx \omega_{sj} \quad \text{if} \quad \overline{\eta}_{sj}^{c} << 1, \tag{28b}$$

$$B_{sj} = \frac{\omega_{sj}}{\sqrt{2}}\sqrt{\sqrt{1 + \overline{\eta}_{sj}^{c2}} - 1}, \tag{29a}$$

$$\approx \frac{\overline{\eta}_{sj}^{c}\omega_{sj}}{2} \quad \text{if} \quad \overline{\eta}_{sj}^{c} << 1, \tag{29b}$$

$$C_{ai} = \frac{\omega_{ai}}{\sqrt{2}}\sqrt{1 + \sqrt{1 + \eta_{ai}^2}}, \tag{30a}$$

$$\approx \omega_{ai} \quad \text{if} \quad \eta_{ai} << 1, \tag{30b}$$

$$D_{ai} = \frac{\omega_{ai}}{\sqrt{2}}\sqrt{\sqrt{1+\eta_{ai}^2}-1}, \tag{31a}$$

$$\approx \frac{\eta_{ai}\omega_{ai}}{2} \quad \text{if} \quad \eta_{ai} << 1, \tag{31b}$$

$$c_{j,i} = \frac{-2\eta_{ai}\overline{\eta}_{sj}^{c}\omega_{ai}^2\omega_{sj}^2\left[\omega_{ai}^2(1+\eta_{ai}^2)-\omega_{sj}^2(1+\overline{\eta}_{sj}^{c2})\right]}{G_{j,i}}, \tag{32}$$

$$d_{j,i} = \frac{\eta_{ai}\omega_{sj}^2\left\{\omega_{sj}^4(1+\overline{\eta}_{sj}^{c2})^2+\omega_{ai}^2\left[\omega_{ai}^2(1+\eta_{ai}^2)(1-\overline{\eta}_{sj}^{c2})-2\omega_{sj}^2(1+\overline{\eta}_{sj}^{c2})\right]\right\}}{G_{j,i}} \tag{33}$$

$$e_{j,i} = -c_{j,i} \tag{34}$$

$$f_{j,i} = \frac{\overline{\eta}_{sj}^{c}\omega_{ai}^2\left\{\omega_{ai}^4(1+\eta_{ai}^2)^2+\omega_{sj}^2\left[\omega_{sj}^2(1-\eta_{ai}^2)(1+\overline{\eta}_{sj}^{c2})-2\omega_{ai}^2(1+\eta_{ai}^2)\right]\right\}}{G_{j,i}} \tag{35}$$

$$G_{j,i} = 2\eta_{ai}\overline{\eta}_{sj}^{c}\left\{\omega_{ai}^8(1+\eta_{ai}^2)^2+\omega_{sj}^8(1+\overline{\eta}_{sj}^{c2})^2-2\omega_{ai}^2\omega_{sj}^2\left[2(\omega_{ai}^2-\omega_{sj}^2)^2+ 2(\eta_{ai}^2\omega_{ai}^4+\overline{\eta}_{sj}^{c2}\omega_{sj}^4)+\omega_{ai}^2\omega_{sj}^2(1-\eta_{ai}^2)(1-\overline{\eta}_{sj}^{c2})\right]\right\}, \tag{36a}$$

$$\approx 2\eta_{ai}\overline{\eta}_{sj}^{c}\omega_{sj}^8(\eta_{ai}^2-\overline{\eta}_{sj}^{c2})^2 \quad \text{if} \quad \omega_{ai}\approx\omega_{sj}. \tag{36b}$$

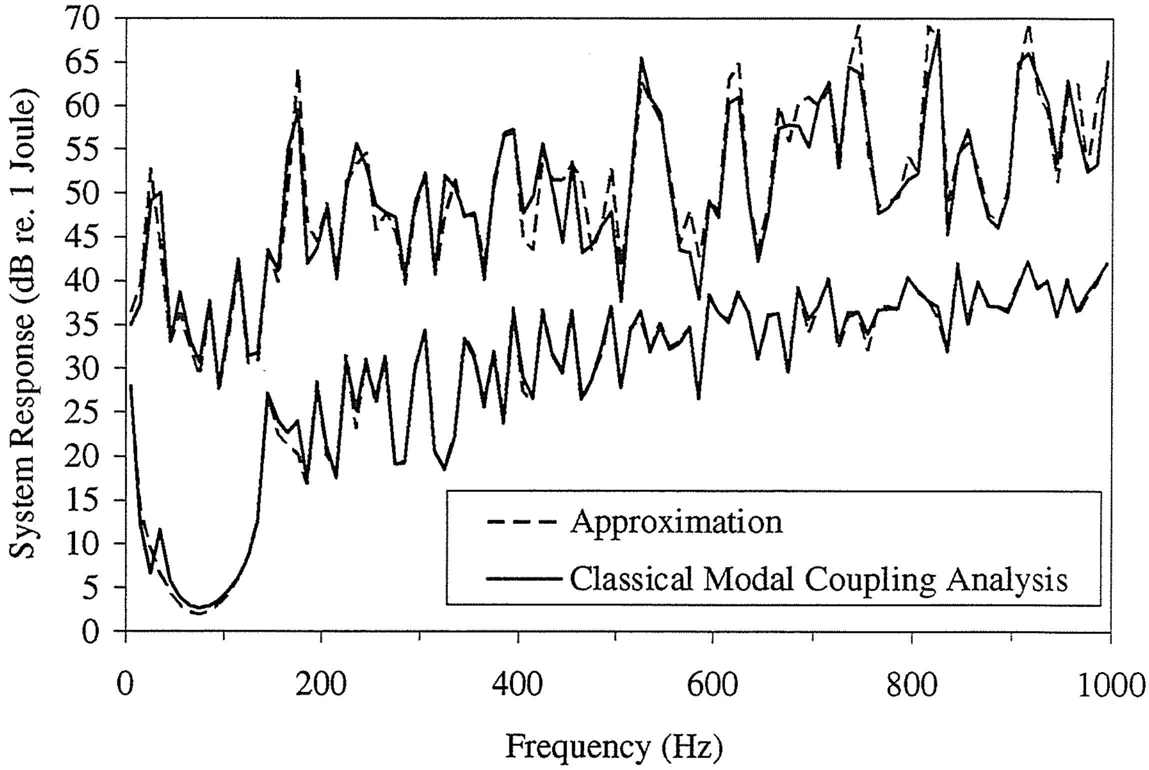
Value of the inverse tangent terms in Eqs. (26) and (27) varies from 0 to 2π.

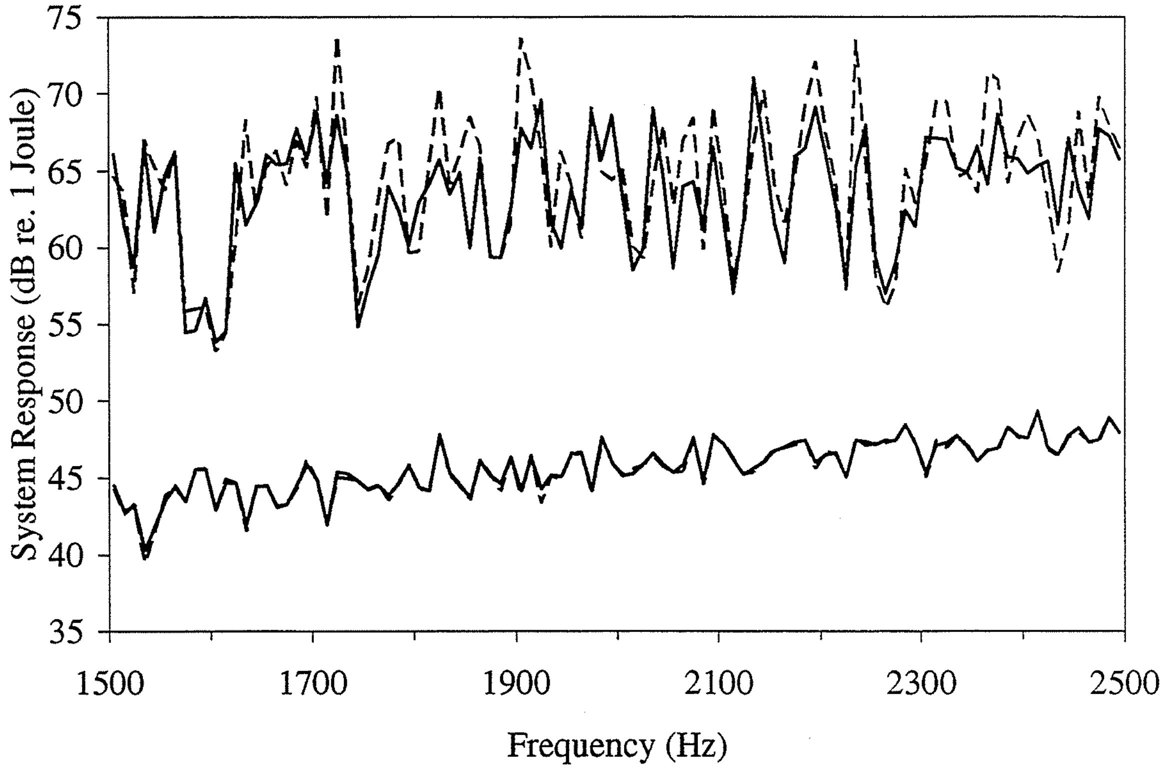
NUMERICAL EXAMPLES

Application of the present analytical method is shown here with some numerical examples. A rectangular simply-supported 0.868m x 1.150m aluminium plate backed by a rectangular parallelepiped acoustic enclosure of dimensions (0.868, 1.150, 1.000)m is considered. A loudspeaker located at a corner of the enclosure is used to drive the enclosed sound field. Analyses are done for narrow frequency bands and 1/3 octave bands and results obtained from the present method are compared to those obtained from classical modal coupling analysis where numerical integrations are required for band-limited energies ie. Eqs. (9) and (10). In each analysis, sufficient acoustic and plate modes have been used in calculations to avoid modal truncation errors.

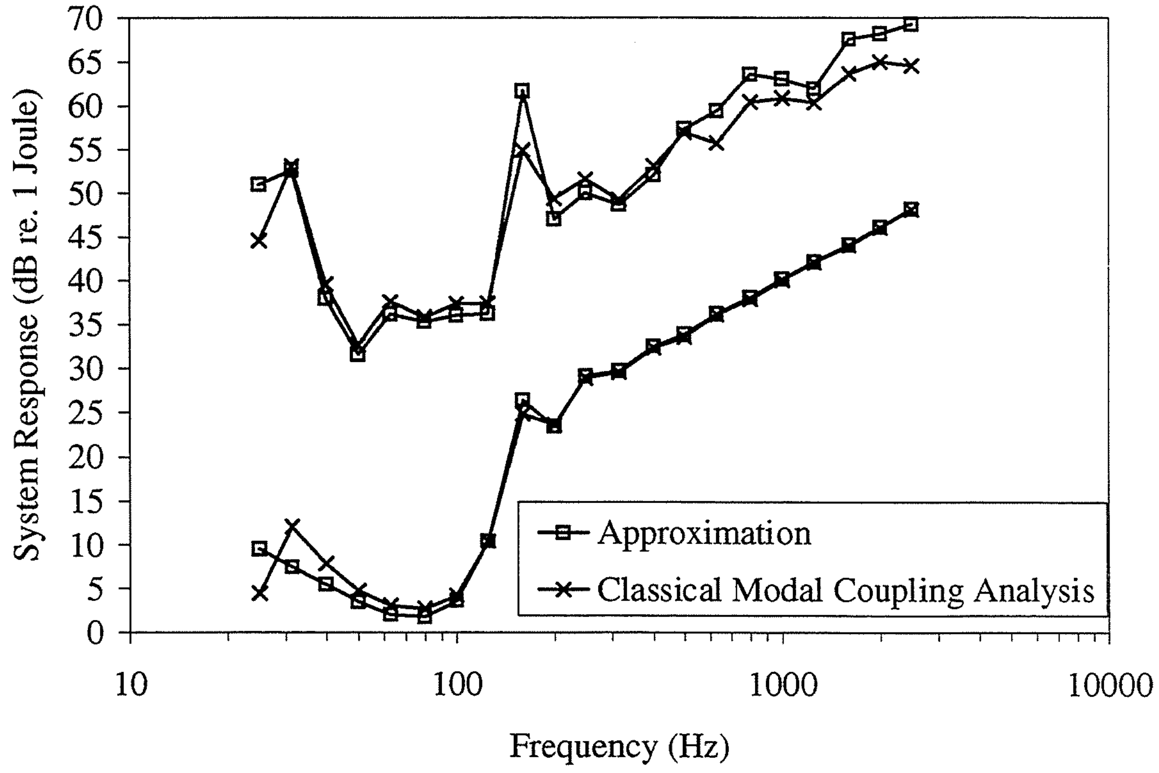
Narrow band results are shown in Fig. 1. In this example, the plate thickness used is h=5.8mm, acoustic modal decay time is Ta=1.0s and plate modal decay time is Tp=1.0s. Two subfigures are used to cover the energies in the low-medium frequency range and medium-high frequency range. The third subfigure shows the corresponding results in 1/3 octave bands. Fairly good agreement between the results obtained from the present method and the classical modal coupling analysis is achieved. All calculations were performed on a

DEC Alpha 3000 300X workstation and recorded computational time was found to improve remarkably by about 50 times if the present method is employed for the prediction of band-limited energies.





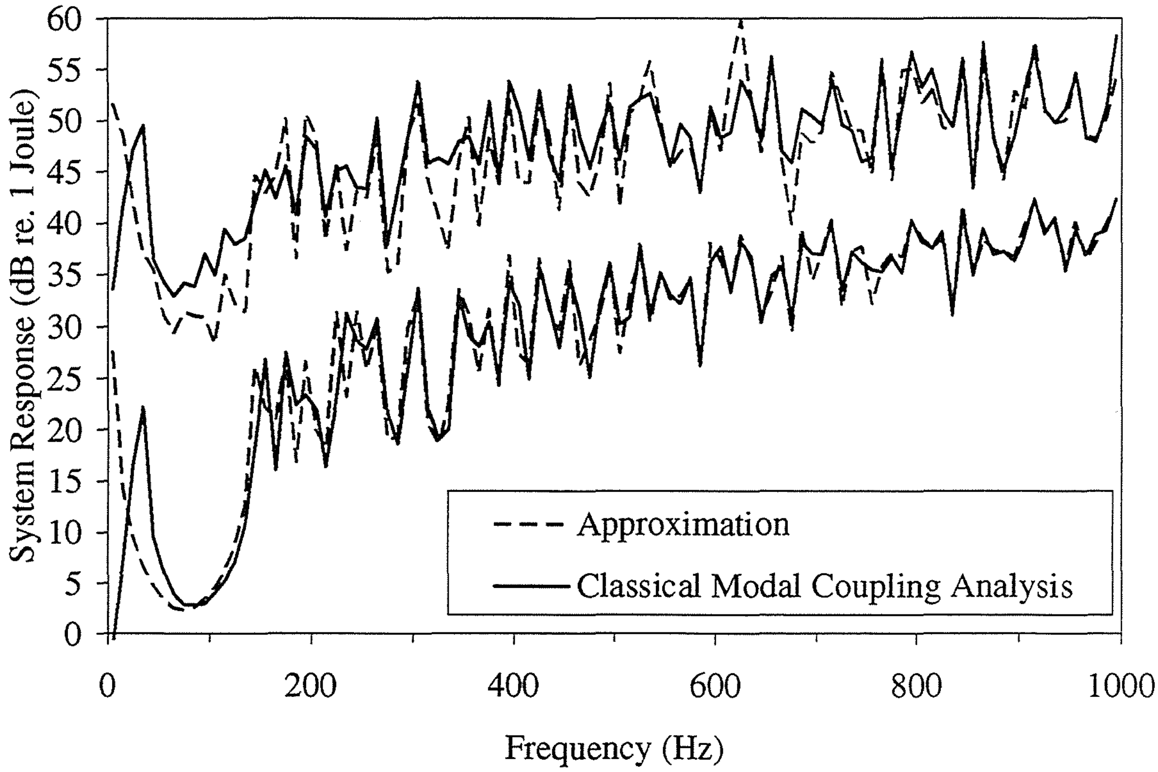
System Response (dB re. 1 Joule)
75
70
65
60
55
50
45
40
35
1500
1700
1900
2100
2300
2500
Frequency (Hz)



System Response (dB re. 1 Joule)
70
65
60
55
50
45
40
35
30
25
20
15
10
5
0
10
100
1000
10000
Frequency (Hz)
Approximation
Classical Modal Coupling Analysis

Figure 1: Acoustic and vibrational energy levels for Ta=1.0 s, Tp=1.0 s and h=5.8 mm in (a)low-medium frequency range and (b)medium-high frequency range ; (narrow band analysis) and (c)1/3 octave band analysis.

A thinner plate of h=1.0mm is now used in place of the h=5.8mm plate and all calculations were repeated and the corresponding narrow and 1/3 octave band results are presented in Fig. 2. Results obtained from both methods do not agree well in the low frequency range because more plate modes are now distributed towards the low order acoustic modes and approximations made in the previous analytical derivations become significant. However fairly good agreement can still be observed in the medium and high frequency ranges and the present method is capable of predicting 1/3 octave band energies to within 6 dB for frequency bands above 250 Hz. The results indicate that the present method can be useful for analysis beyond the low frequency range.



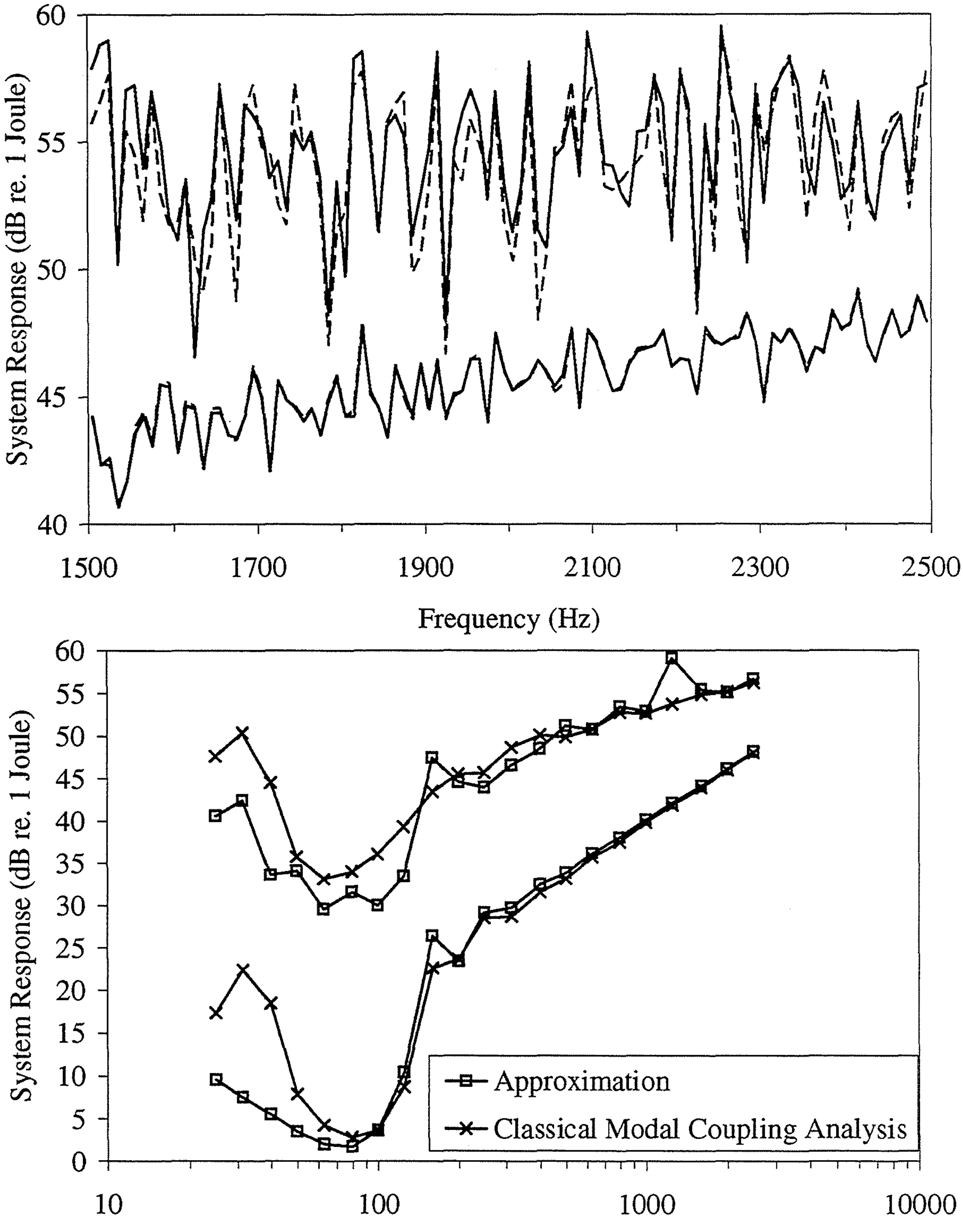


Figure 2: Acoustic and vibrational energy levels for Ta=1.0 s, Tp=1.0 s and h=1.0 mm in (a)low-medium frequency range and (b)medium-high frequency range ; (narrow band analysis) and (c)1/3 octave band analysis.

CONCLUSIONS

Based on a review of recent progress in the field of vibroacoustic, an analytical method for the prediction of energies in vibroacoustic system has been proposed. It is more computational efficient than the classical modal coupling analysis for medium and high frequency analyses where the classical approach has to cope with manipulation of large complex matrices. The proposed method can also be useful for narrow band analysis where there are no or insufficient resonance modes in the band for reliable statistical application. Computational efficiency and prediction accuracy which are the main features of the present method have been demonstrated.

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BUILDING CODE OF AUSTRALIA 1990
IMPACT SOUND INSULATION OF BUILDING PARTITIONS

PAPER 1 - MEASUREMENT OF IMPACT INSULATION
PAPER 2 - SHORTCOMINGS OF THE CODE IN MEASUREMENT AND INTERPRETATION

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ABSTRACT

Part F5 of the Building Code of Australia (BCA) 1990, "Noise Transmission and Insulation" sets out performance requirements for the sound transmission properties of building partitions. Specifically, Clause F5.5 states that a wall separating a non-habitable room in one sole-occupancy unit from a habitable room in an adjoining unit must have an STC not less than 50, and provide a satisfactory level of insulation against impact sound. Specification F5.5 contains the test method by which an impact sound test can be conducted. The first paper will discuss contrasts in measurement results carried out in a sound transmission suite of the impact sound insulation of various partition walls. In addition to the BCA method for impact insulation, results will be presented on two methods employed utilising an alternative impact noise source. No attempt was made to understand the mechanisms or characteristics of the transmitted impact noise by the three methods. Airborne sound performance will be reported but not discussed. The second paper will point out the shortcomings of the BCA method of wall impact testing, both in the measurement of parameters and in the interpretation of results. It will not attempt to resolve the dilemma but will offer suggestions requiring further study. The BCA impact test method has lain somewhat dormant since 1988, not only in its use but also any impending revisions. It has been the plethora of inner city apartment buildings constructed in recent times that has been the impetus which has prompted a re-evaluation.

INTRODUCTION

Growing concerns by the building industry about the noise radiated due to impacts on walls has been highlighted with the increasing trend of apartment type buildings throughout the country. Apart from new build construction, old office blocks and woolstores have been recycled into multi-residential apartments. Typical forms of construction for party wall is concrete, masonry or drywall, or combinations thereof, and range in thicknesses anywhere from nominally 90 to 200 mm. One of the tests to evaluate the acoustic integrity of a (party) wall is an impact sound test as outlined in the Building Code of Australia (BCA). Apart from the BCA method, attempts have been made to emulate two other common sources of impact noise encountered in the domestic environment. Measurements were taken on four different wall constructions to compare trends within and between the three methods. Common sources of impact noise that are radiated through party walls come from cupboard doors being closed, light switches being turned on and off, chopping up of vegetables and use of mixers or coffee grinders on the kitchen bench, hydraulic noises from plumbing fittings, and clothes dryers fixed to the wall.

The concern of footfall noise a number of years ago had resulted in much work being done to understand and resolve floor-ceiling impact transmission mechanisms. This saw appropriate Standards for test method and classification devised and implemented. In contrast, wall impact sound transmission has been little studied. As such, no Australian or International Standard exists for testing or rating impact noise performance of a wall. The only existing method is that which appears in the BCA.

The acoustic integrity of the partition separating neighbours is of prime importance to ensure a harmonious environment is maintained. The BCA was implemented to ensure that acceptable standards of buildings are maintained in regards to safety, health , and amenity for the benefit of the community.

Background

Attempts to ratify the events that lead to the inception of the impact method into the BCA was largely unsuccessful. Also unsuccessful was finding results and conclusions from an unpublished study by Weston who did much of the preliminary work on devising this impact test method for walls. Put up as a proposal by CSIRO to the regulatory body at the time, the wall impact method was incepted into (the first) national building code in 1988 and still remains in its inaugural form in the BCA 1990.

SCOPE OF TESTING

Measurements were carried out in two phases:

Phase 1 Determination of sound transmission loss by the method of Australian Standard 1191 - 1985, and

Phase 2 Investigation by empirical methods of the resistance of the partitions to incident impact sound using three different impact noise sources.

The transmission loss chambers at RMIT, a NATA accredited laboratory, were utilised to conduct full scale acoustic testing. Phase 1 had shown the laboratory to comply to the requirements of AS 1191 - 1985. Although not discussed in this paper, the purpose of Phase 1 measurements was twofold. Firstly, to ascertain by test the ability to achieve a minimum STC 50 as stipulated by the BCA for those walls requiring impact sound insulation and, secondly to provide test results to the sponsor.

DESCRIPTION OF SPECIMENS

Wall systems tested were chosen as being practical building partitions utilised in the building industry today. Three double leaf plasterboard walls of timber frame construction and one masonry wall system were the basis of this investigation. A specific parameter for these walls was to achieve a minimum Sound Transmission Class (STC) 50.

The following provides a brief description of the partitions, in order of testing.

Wall 1 Staggered timber stud system - 2 layers 13 mm Boral Firestop plasterboard each side of 70 x 45 mm staggered timber studs at 600 mm centres each side of 90 x 45 mm timber plates - 50 mm thick fibreglass cavity insulation - sheets nail attached - joints staggered - perimeter sealed Nom. surface density 42 kg/m^2.

Wall 2 Drywall masonry system - 190 mm thick Boral concrete block masonry wall with:

(i) each face of the blocks fitted with 50 mm x 50 mm timber battens at nom. 600 mm centres, screw fixed into Rawlnuts;

(ii) the space between the battens completely filled with 50 mm thick fibreglass cavity insulation;

(iii) the outer face of the battens finished with 10 mm Boral plasterboard.

Nom. surface density 192 kg/m^2.

Wall 3 Timber stud system - 2 layers 13 mm Boral Firestop plasterboard each side of 90 x 45 mm timber studs at 600 mm centres - 50 mm thick fibreglass cavity insulation - sheets nail attached - joints staggered - perimeter sealed.
Nom. surface density 42 kg/m^2.

Wall 4 Resilient timber stud system - 2 layers 13 mm Boral Firestop plasterboard each side of 90 x 45 mm timber studs at 600 mm centres - resilient channel fixed at 600 mm centres to one face of framing only - plasterboard fixed to resilient channel - 50 mm fibreglass cavity insulation - sheets nail/screw attached - joints staggered - perimeter sealed.
Nom. surface density 42 kg/m^2.

The nominated wall systems were also chosen for their classification under the BCA. Refer to **Table 1** and the pertinent clause from BCA 1990 reproduced below for definition.

F5.5 *Walls between a bathroom, sanitary compartment, laundry or kitchen and a habitable room in an adjoining unit:*

(a) *A wall separating a bathroom, sanitary compartment, laundry or kitchen in one sole-occupancy unit from a habitable room (other than a kitchen) in an adjoining unit must:*

(i) *have an STC of not less than 50; and*

(ii) *provide a satisfactory level of insulation against impact sound*

(b) *A wall satisfies (a)(i) and (a)(ii) if it is:*

(i) *in accordance with Table F5.5, or*

(ii) *for other than masonry, in 2 or more separate leaves without rigid mechanical connection except at their periphery, or*

(iii) *identical to a prototype that is no less resistant to the transmission of impact sound when tested in accordance with Specification F5.5 than a wall listed in Table F5.5.*

TABLE 1

Relationship of each wall to clause F5.5 of the BCA

Partition	Section Satisfied in Part F5.5	Comments
Wall 1	(b)(ii)	Staggered stud construction provides two separate leaves
Wall 2	(b)(i)	Construction in accordance with Table F5.5 - 'Concrete blockwork'
Wall 3	N/A	Basic construction used as a benchmark reference for Walls 1 and 4
Wall 4	(b)(iii)	Prototype testing is the only option to ascertain compliance for insulation against impact sound

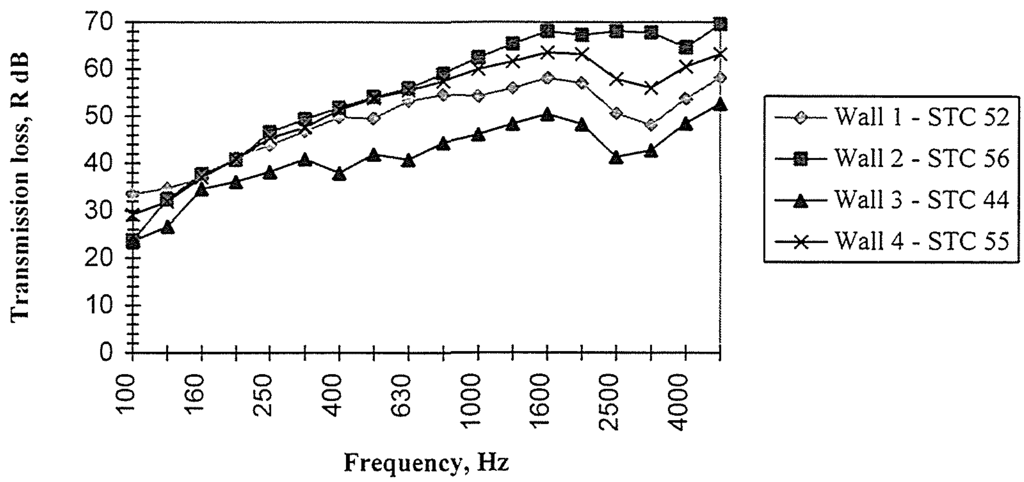
Note that Wall 2 is one of the three reference walls listed in Table F5.5, and Wall 4 a prototype wall. Although strictly not a prototype, Wall 1 is included for the purpose of assessing the performance of a 'deemed-to-satisfy' system when actually tested.

PHASE 1 - DETERMINATION OF SOUND TRANSMISSION LOSS

Results

For each Wall the measured airborne sound transmission loss, R dB, in each third octave bandwidth of centre frequencies between 100 and 5000 hertz is represented graphically in **Figure 1**. The stated STC values have been determined in accordance with AS 1276 -1979.

Figure 1: Sound Transmission Loss of the Four Tested Walls



PHASE 2 - DETERMINATION OF IMPACT INSULATION

The purpose was to examine contrasts in measurement results of three different impact methods conducted on the wall specimens used in this investigation. Comparisons were made between wall specimens subjected to the same impact method, and conversely, between the different impact methods conducted on the same wall.

The different impact methods are listed below. Each method was assigned a number which was also the order of testing.

Impact method 1 'Impact Sound - Test of Equivalence' as outlined in the Building Code of Australia 1990, Specification F5.5.

Impact method 2 Impact sounds generated by a single event eg furniture/hammer impacts

Impact method 3 Plumbing noise; a "real life" situation that is a common cause of annoyance

Impact method 1 is a (required) testing procedure prescribed by the BCA for comparative evaluation of a complying wall listed in Table F5.5 to that of a prototype wall. Impact methods 2 and 3 are attempts to measure 'real life' situations whose impacts are known causes of annoyance.

All three methods were not applied to all four partitions as shown in **Table 2.**

TABLE 2

Impact Methods That Were Performed on Each Wall System

Wall	Impact Method 1	Impact Method 2	Impact Method 3
1	Yes	Yes	No
2	Yes	Yes	Yes
3	Yes	Yes	Yes
4	Yes	Yes	No

In all three impact methods conducted, the test facilities and equipment were that as for Phase 1 which comply to the requirements of AS 1191 - 1985. A fundamental difference exists for expressing measurement levels between the procedures of impact testing and AS 1191 testing, namely the absolute sound pressure levels in the receiving room are used to determine the impact sound pressure level (SPL), whereas the sound pressure level difference between the source and receiving rooms is used to determine the sound transmission loss (R).

As for Phase 1, four independent microphone positions were used in the reverberation room for Impact methods 1 and 3. Impact method 2 however, had its microphone positions close to the specimen face.

For Impact methods 1 and 3 the sound pressure levels were normalised using the following equation:

$$L_n = L_p - 10 \log (A_o / A) \text{ dB} \qquad (1)$$

where,

L_p = average sound pressure level in the receiving room, in decibels.

A_o = 10 m^2.

A = equivalent absorption area in receiving room, in square metres.

The value of A was determined from the reverberation times measured in Phase 1, and the value of A_o adopted as 10 m^2 as specified under Part F5 Specification F5.5 of the BCA.

IMPACT METHOD 1 - BCA TEST OF EQUIVALENCE

Method

The measurements were performed to comply with the requirements of Part F5 Specification F5.5 of the BCA, the method of which is outlined below as it appears in the BCA.

(a) *The wall constructions to be compared must be tested in a laboratory complying with AS 1191.*

(b) *A horizontal steel platform 510 mm x 460 mm x 10 mm thick must be placed with one long edge in continuous and direct contact with the wall to be tested on the side of the wall on which the impact sound is to be generated.*

(c) *A tapping machine complying with ISO 140/VI-1978 (E) must be mounted centrally on the steel platform.*

(d) *The sound transmission through the wall must be determined in accordance with AS 1191 except that the tapping machine as mounted on the steel platform must be used as the source of sound.*

(e) *The impact sound pressure levels measured in the receiving room must be converted into normalised levels using a reference equivalent absorption area of 10m^2.*

Equipment

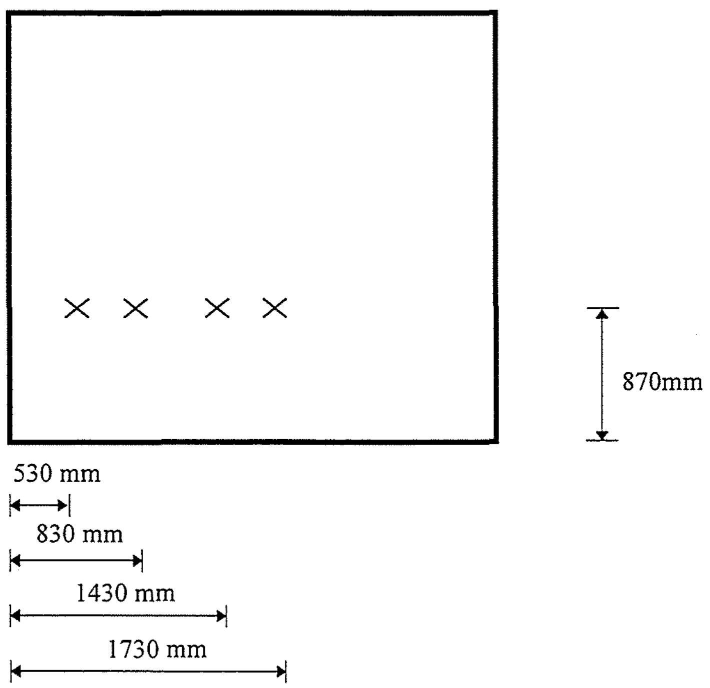
The 'standard' tapping machine used was a Bruel & Kjaer Type 3204 (Serial No. 91082) which fulfils the requirements of ISO 140/VI-1978 (E). A rotating cam allows five steel feet, each weighing 500 grams, to be raised then dropped under gravity through 40 mm, at a rate of 10 impacts per second. It was these feet striking the steel plate that generated the impact noise source.

Procedures

Details that are not provided in the BCA method include the height of the impact plane above the lower surface of the partition, and the number of impact locations for the plate. Adopted for this investigation, and shown in **Figure 2**, is an impact plane height of 870 mm and four different locations used for the impact source.

The time of sampling the impact sound pressure levels in the receiving room and number of microphone positions, also not covered in the BCA, have been chosen to comply to AS 1191 - 1985 viz. four independent microphone positions with the signal temporally averaged for the sampling time of 128 seconds.

Figure 2: Impact sound testing with B&K Tapping machine on wall specimens showing the locations of impact in the sound source room of the transmission loss chambers. The locations marked are to the centre of the impact plate. The impact height is measured from the base of the wall.

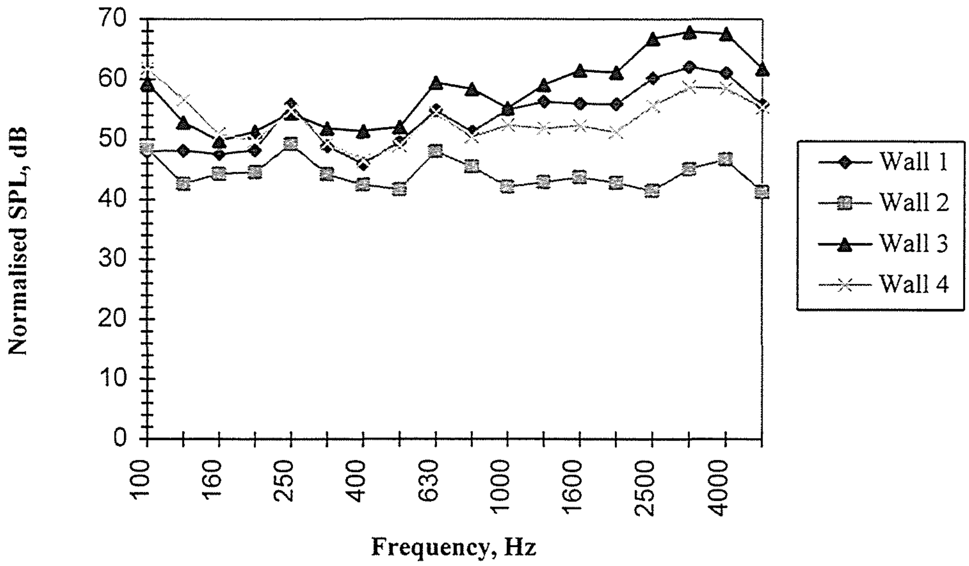


ELEVATION - Source Room

Results

For each Wall the normalised sound pressure levels (SPL) in each third octave bandwidth of centre frequency between 100 and 5000 Hz is represented graphically in **Figure 3**.

Figure 3: Normalised SPL Measurements of Walls 1 to 4 Subjected to Impact Method 1



Comments

With similar trends in slope, and similar results obtained in some frequency bands, as seen in **Figure 2**, the comparisons of the walls are made simpler by looking at the average values in the low (100 - 250 Hz), medium (315 - 1000 Hz) and high frequency (1250 - 5000 Hz) ranges.

Low Frequency Range

The average values in the normalised sound pressure levels for Walls 1 to 4 in the low frequency range are 50, 46, 53 and 55 dB respectively. This clearly shows that Wall 2 (drywall masonry system) attenuates impact sound better than the double-leaf plasterboard systems.

Wall 1 and 2 follow a similar trend; at 100 Hz they are almost identical and from 125 to 250 Hz Wall 2 consistently lower by 3 - 7 dB. Walls 3 and 4 are closely associated with shifts in performance; levels are within 4 dB of one another. All Walls experience a peak at 50 Hz.

Medium Frequency Range

The average values in the normalised sound pressure levels for Walls 1 to 4 in the medium frequency range are 51, 44, 55 and 50 dB respectively. Performance of Walls 1, 2 and 3 are almost identical in this to the low frequency range with values being within 2 dB of one another. Wall 4 experiences a 5 dB increase in performance, now making it the best performing double-leaf plasterboard wall. Wall 2 (drywall masonry system) continues to perform better than the double-leaf plasterboard walls.

Walls 1 and 4 are identical in performance with curves somewhat superimposed. Wall 3 begins to decline in performance. Wall 2 is consistently better than Walls 1 and 4 by 4 to 13 dB. Another peak is encountered at 630 Hz for all walls.

High Frequency Range

The average values in the normalised sound pressure levels for Walls 1 to 4 in the high frequency range are 58, 44, 64 and 55 dB respectively. Wall 2 has an invariable performance over the frequency ranges with a maximum deviation of 2 dB. Walls 1, 3 and 4 (double-leaf plasterboard walls) experience a significant loss in performance in this range.

The largest contrast in levels occur in the high frequency range where differences of between 16 and 25 dB occur. There are marked differences in performance between all walls. At 2000 Hz, Walls 1, 3 and 4 experience a sudden increase in level with a peak culminating at 3150 Hz. For Wall 2 this same phenomena begins at 2500 Hz, peaking at 4000 Hz.

Overall

Clearly, Wall 2 (masonry system) performs better than Walls 1, 3 and 4 (drywall systems) over the entire frequency range measured. It is difficult to ascertain a grading of performance for the drywall systems (Walls 1, 3 and 4) because they perform quite differently to one another in each frequency range. Generally, it can be said however that Wall 3 is the worst performer.

Wall 2 has, for all intensive purposes, a flat frequency response over the entire frequency range as seen in **Figure 3**. The same can be said for Walls 1, 3 and 4 with the exception in the high frequency range. Interestingly, performance trends for walls subjected to this impact method closely resembles that of sound transmission loss testing from Phase 1.

OTHER EXPERIMENTAL METHODS

Impact Method 2 - Impact Sounds Generated by a Single Event

The test impactor used for this study was basically a circular polycarbonate (plastic) rod so chosen not on any scientific basis but that "it looked it would do the job". The method was so adopted however, as it would enable the energy imparted to the partition to be calculated.

Since the impact sound measurements were undertaken on a comparative testing basis, the emphasis was not so much on the physical properties of the impact rod to represent a 'worst case' situation as was establishing a repeatable test method by imparting a fairly constant impact energy .

Equipment

The set-up basically consisted of a plastic rod, with a flat impact face, suspended at two points by sisal string.

The length of the pendulum arm, which comprised of the sisal string was 280 mm.

The impact rod was of solid polycarbonate material 375 mm long, 30 mm diameter and a mass of 335 grams.

Procedures

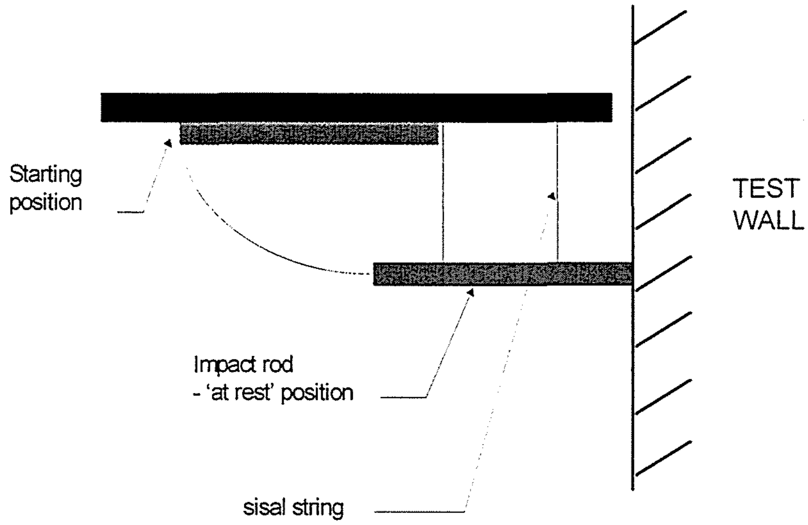
The impact rod was initially located so that the impact face was just touching the test wall with it (the impact rod) at rest in its vertical position..

The impact rod was released at an angle of 90 degrees from vertical and was allowed to fall under its own weight. Refer **Figure 4a**.

After completing a single impact on the test wall, the rebounded impact rod was prevented from striking the wall again.

The maximum sound pressure levels recorded were not normalised. Basically, normalising is not possible because absorption in the (receive) room is not taken into account. Due to the nature of the impact there is an insufficient level build-up to take time averaging over the reverberation enclosure. A method has been adopted whereby an exponential averaging time of 1/32 second using maximum hold was used with levels captured by a microphone in a plane close to the partition.

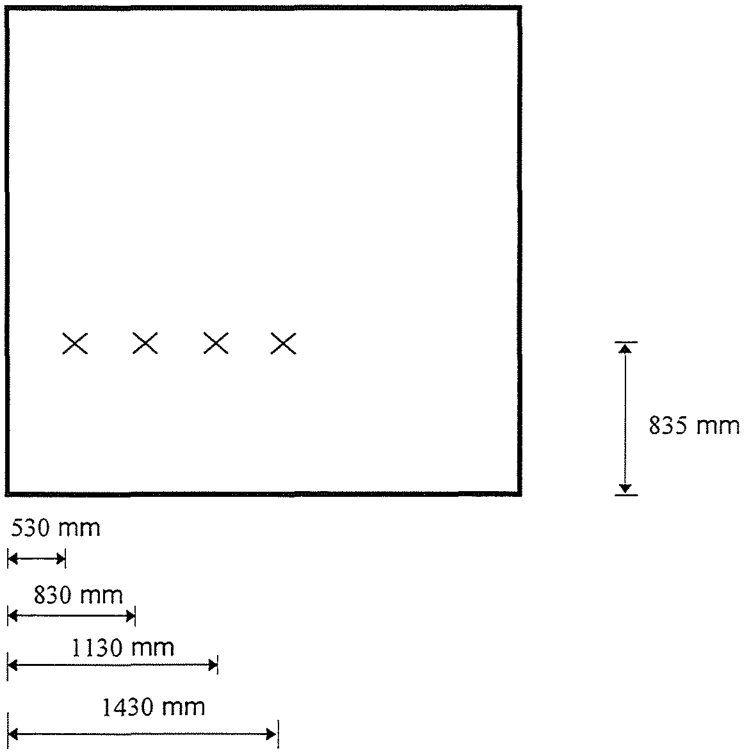
Figure 4a: Set-up of Impact method 2 showing the impact rod at initial and finish positions. The dotted line indicates the rod swings as a pendulum when released.



Four different locations for the impact source and microphone positions were used as seen in **Figures 4b and 4c**.

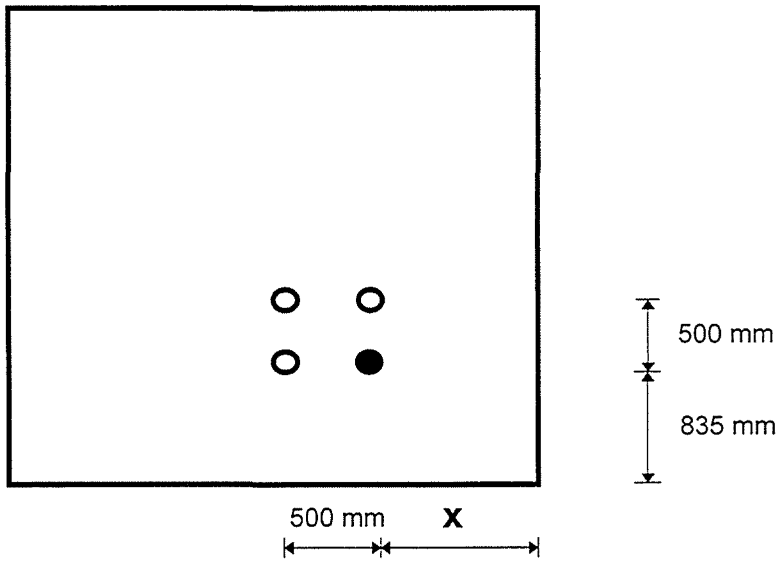
Figure 4b: Impact sound testing with polycarbonate rod on wall specimens showing the locations of impact in the sound source room of the transmission loss chambers.

The impact height is measured from the base of the wall.



ELEVATION - Source Room

Figure 4c: Impact sound testing with polycarbonate rod on wall specimens showing microphone positions in the receive room of the transmission loss chambers. Microphone positions were in a plane 535 mm away from the specimen surface.



X - varies accordingly with impact locations

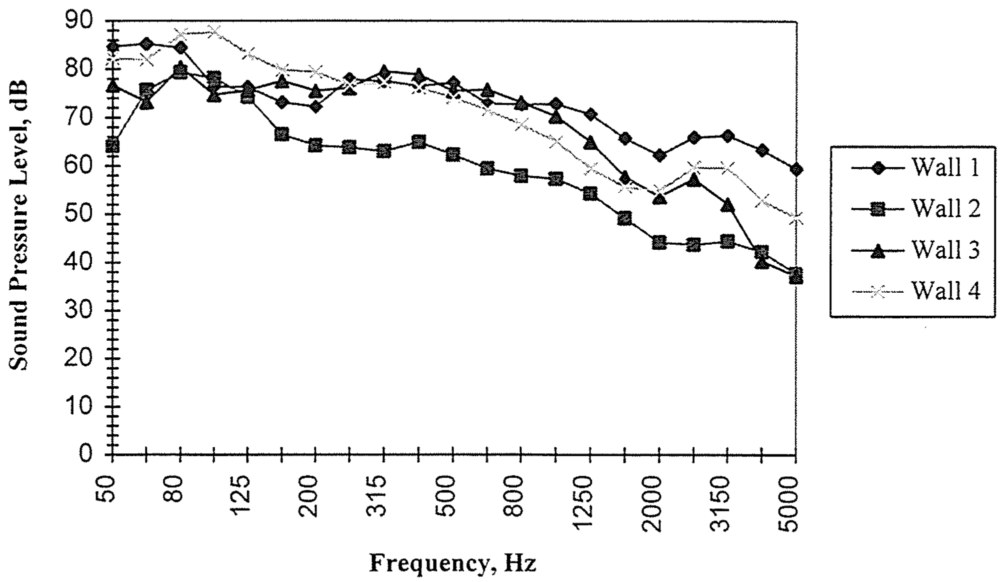
● - microphone position behind point of impact for each impact location

<u>ELEVATION - Receive Room</u>

Results

For each wall the maximum sound pressure levels in each third octave bandwidth of centre frequency between 50 and 5000 Hz is represented graphically in **Figure 5.** Results shown are absolute sound pressure levels which are not normalised.

Figure 5: Results of the Four Walls Tested to Impact Method 2



Comments

With similar trends in slope, and similar results obtained in some frequency bands, as seen in **Figure 4**, the comparisons of the walls are made simpler by looking at the average values in the low (50 - 250 Hz), medium (315 - 1000 Hz) and high frequency (1250 - 5000 Hz) ranges.

<u>Low Frequency Range</u>

The average values in the non-normalised sound pressure levels for Walls 1 to 4 in the low frequency range are 79, 71, 76 and 82 dB respectively. This again shows, as with Impact method 1 that Wall 2 (drywall masonry system) attenuates impact sound better than the double-leaf plasterboard systems.

Wall 2 and 3 follow a similar trend from 63 to 125 Hz where they are almost identical, thereafter Wall 2 suddenly increases in performance and maintains the trend. Wall 4, the poorest performer in this frequency range experiences a peak at 100 Hz.

Medium Frequency Range

The average values in the non-normalised sound pressure levels for Walls 1 to 4 in the medium frequency range are 75, 60, 76 and 72 dB respectively. Wall 3 has no change in performance. Wall 4 experiences a 10 dB increase in performance, now making it the best performing double-leaf plasterboard wall. Wall 2 (drywall masonry system) continues to perform better than the double-leaf plasterboard walls, which is personified by an 11 dB increase in performance.

Walls 1, 3 and 4 are identical in performance with curves somewhat superimposed. Wall 4 begins to increase in performance. Wall 2 is consistently better by up to 17 dB.

High Frequency Range

The average values in the non-normalised sound pressure levels for Walls 1 to 4 in the high frequency range are 65, 45, 52 and 56 dB respectively. Another shift in performance occurs with Wall 3 becoming the best performing double-leaf plasterboard wall once again. Wall 2 continues to dominate and is only surpassed at 4000 Hz and 5000 Hz by Wall 3.

At 2000 Hz, Walls 1, 3 and 4 experience a sudden increase in level with a peak culminating at 3150 Hz. For Wall 2 this same phenomena occurs as a plateau within the same range.

Overall

Clearly, Wall 2 (masonry system) performs better than Walls 1, 3 and 4 (drywall systems) over the entire frequency range measured. It is difficult to ascertain a grading of performance for the drywall systems (Walls 1, 3 and 4) because they perform quite differently to one another in each frequency range. Generally, it can be said however that Wall 3 is the best performer.

The general trend for all Walls is of a (negative) straight line slope which basically means that performance becomes progressively better from the low to the high frequency ranges.

IMPACT METHOD 3 - PLUMBING NOISE

Plumbing noise was measured in the receiving room by running water through a garden tap which was fixed to the wall on the source room side.

Equipment

The set-up comprised 3/4 inch garden hose which fed water through,

1. a pressure regulator and gauge.
2. a flow meter - 3 to 16 litres per minute. Another gauge attached (5 to 60 litres per minute) was not used.
3. a pressure regulator.
4. a garden tap.

Procedures

A section of plaster was cut away to allow the tap to be fixed at a nominal height of 800 mm from the base of the wall and directly to the timber framing.

Measurements were taken for Walls 2 and 3 only. Attempts were made on Wall 1, however, the diameter of the hose initially used, ie. 1/2 inch, failed to create any measurable sound pressure levels in the receiving room. Measurements on Wall 4 were not required since the framework was the same as for Wall 3.

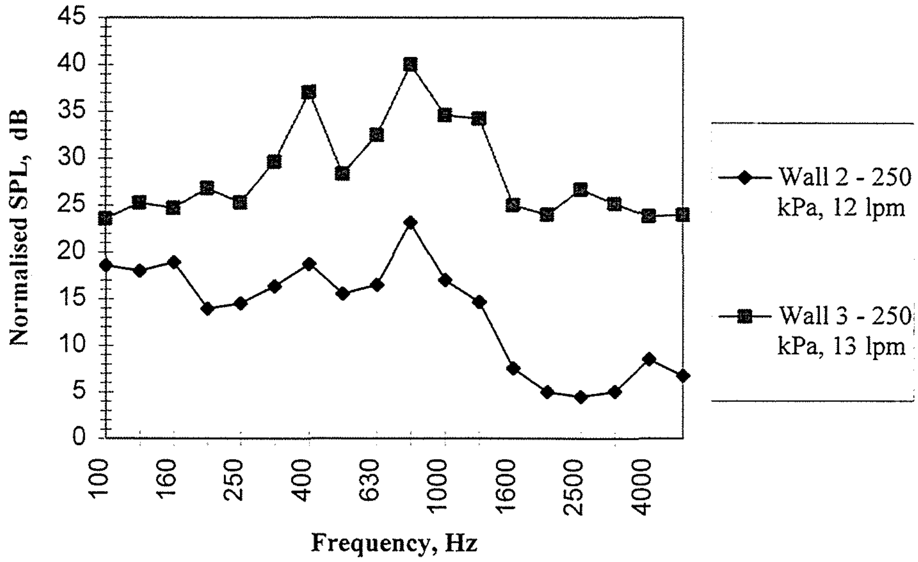
Three microphone positions were used.
Testing was conducted by varying pressure and flow.

As with Impact method 1, the impact sound pressure levels measured in the receiving room were converted into normalised levels using a reference equivalent absorption area of $10m^2$; not a specified requirement but elected to do so since method produces pressure build-up.

Results

For Walls 2 and 3 the normalised sound pressure levels (SPL) in each third octave bandwidth of centre frequency between 100 and 5000 Hz is represented graphically in **Figure 6**.
Although Identical pressures were measured, there was a slight variation in flow rate of 1 litre per minute between the two Walls.
Wall 2 - Measured pressure 250 kPa, measured flow rate 12 litres per minute.
Wall 3 - Measured pressure 250 kPa, measured flow rate 13 litres per minute.

Figure 6: Results of Walls 2 and 3 Tested to Impact Method 3



Comments

Low Frequency Range

The average values in the normalised sound pressure levels for Walls 2 and 3 in the low frequency range (100 - 250 Hz) are 17 and 25 dB respectively. This shows the masonry wall system is superior at attenuating faucet noise than a double-leaf plasterboard system.

Beginning with an almost flat response, Wall 2 experiences a sudden rise in performance with a peak at 200 Hz. Wall 3 remains somewhat flat.

Medium Frequency Range

The average values in the normalised sound pressure levels for Walls 2 and 3 in the mid frequency range (315 - 1000 Hz) are 18 and 34 dB respectively. The gap in performance has widened between the two Walls; Wall 2 only marginally worse by 1 dB whereas Wall 3 deteriorating by 9 dB. Both Walls experience troughs in performance at 400 Hz and again at 800 Hz.

High Frequency Range

The average values in the normalised sound pressure levels for Walls 2 to 3 in the high frequency range are 7 and 26 dB respectively.

Another shift occurs with Wall 3 with a performance increase of 8 dB. Wall 2 increasing by 11 dB.

Overall

Clearly, Wall 2 (masonry system) performs better than Walls 3 (drywall systems) over the entire frequency range measured.

Most of the energy transmitted is in the mid frequency range as evidenced by two large troughs that both Walls experience.

OBSERVATIONS AND COMMENTS

Sound Transmission Loss

The reference wall (Wall 2) was shown to satisfy the requirements of the Building Code of Australia (BCA) Part F5 Section F5.5, in that it achieved the STC value of not less than STC 50. For that reason it was valid to compare Walls 1, 3 and 4 to check whether they came under the classification of "deemed-to-satisfy" in the BCA Specification F5.2.

Wall 3 was shown to have the STC rating only 44, so failing to satisfy the BCA. On the other hand, both Walls 1 and 4 had STC values above 50, so satisfying the Code. In comparison with Wall 2 however, particularly above 200 Hz, the sound transmission loss values both for Walls 1 and 4 were significantly lower than those of the reference wall.

Impact Insulation

BCA method

Section F5.5 of the BCA provides a method of comparison between a reference wall and a sample wall. The method employs the use of a standard tapping machine. The BCA infers that the impact insulation values set in terms of the normalised sound pressure levels in the receiving room, become the reference requirements. In other words, should only one of the normalised sound pressure levels measured in any of the nominated frequencies exceed the corresponding value for that of the reference wall, then the prototype wall is rejected. Adopting this as the criteria, then Walls 3 and 4 fall short of the BCA requirement. Wall 1 also falls short, but as a deemed-to-satisfy construction can legally be used.

All methods

A comparison is made between the three methods of measurement, using Figures 3, 5 and 6. Such comparisons are made between;

(a) Reference Wall 2 with Wall 4

Only above 1600 Hz are the results using the tapping machine method (method 1) and impact rod method (method 2) similar between the two walls. At lower frequencies the tapping machine method indicates the reference wall to be a much less improved impact insulator than that shown using the impact rod method, and

(b) Reference Wall 2 with Wall 3

At frequencies between 125 and 1000 Hz the tapping machine method shows the reference wall to be a much more improved impact insulator than when using the impact rod method; above 1000 Hz the reference wall is not such a significantly better insulator. The plumbing noise method (method 3) shows similar results to those by the impact rod method up to 630 Hz, above which it shows the reference wall to be a much improved impact insulator than Wall 3.

In terms of the values of precision of measurement using the three methods of producing the impact on the sample, using the results for wall 3 as an illustration, the plumbing noise was seen on the average to be about as reliable as the tapping machine method of measuring impact insulation. The impact rod method was seen to be slightly inferior in precision compared with the other two methods.

DISCUSSION

It is very difficult to make any firm conclusions when comparing the different methods of producing the impact on the partitions because of the small number of tests carried out. Audibly there are distinct differences in the nature of the sound radiated through the walls between the methods, each producing unique measurement trends. In view of this, it is important that the governing body and for users of the BCA to make a firm stand on the type and nature of impacts on the partition in practice which are likely to cause complaints between occupiers of adjacent tenancies.

A possible, and most practical method to rank a wall in regards the impact insulation provided would be utilising a single number rating along the style of an STC. The second Paper addresses this issue.

REFERENCES

1. The Building Code of Australia
2. AS 1191 "Acoustics - Method for Laboratory Measurement of Airborne Sound Transmission Loss of Building Partitions".
3. AS 1045 - 1988, "Acoustics - Measurement of Sound Absorption in a Reverberation Room", Appendix A
4. AS 1276 - 1979, "Methods for Determination of Sound Transmission Class and Noise Isolation Class of Building Partitions".

VERY LOW FREQUENCY CALIBRATION OF ACCELEROMETERS.

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ABSTRACT

To enhance the accelerometer calibration facility at the National Measurement Laboratory, an apparatus and technique have been developed to extend the frequency range to very low frequencies. This is a practical implementation of the method of calibration of an accelerometer by rotation in the Earth's local gravity field. The accelerometer is attached to a flywheel, which is supported on low-friction air bearings. The accelerometer's sensitive axis is normal to the flywheel's axis of rotation, which lies in the plane of the accelerometer mounting surface. The flywheel is driven up to speed by an electric motor, then the motor is disconnected and switched off. Samples are taken of the accelerometer output as the flywheel slows down. Amplitude and phase of the accelerometer response are determined for frequencies from 20 Hz down to 0.1 Hz.

INTRODUCTION

Accelerometers are most commonly used for measuring vibration. There are a few instances where such applications need a good response to low frequencies, and where the measuring systems need to be calibrated down to low frequencies. In particular, transducers for measuring vibration affecting the human body need to be calibrated down to 4 Hz (hand/arm vibration), 1 Hz (whole-body vibration), or 0.1 Hz (motion sickness) (eg AS 2973-1987, and AS 2670-1983)

In addition to vibration measurements, accelerometers are commonly used for measuring mechanical "shock" or impact. Examples of this are in motor vehicle crash testing, and production line testing of safety helmets and seat belts. Instrumentation for such tests is usually required by Standards (eg AS 2512, AS 1801, AS 1698) to conform to the specifications of SAE J211b, which includes a frequency response within ±0.5 dB down to 0.1 Hz. The equivalent -3 dB point is about 0.035 Hz. Another way of looking at this is that the "time constant" of the measuring system must be long compared to the duration of the shock or impact, otherwise the true peak value of the shock will be underestimated by the measurement.

Calibration methods are well established for the frequency range 20 Hz to 10 kHz. For accelerometers which have a response down to zero hertz, the local gravity field of Earth can be used to calibrate by the familiar "2g turnover" method. (Clark 1995)

Calibration in the in-between frequency range, $0 < f < 20$ Hz, can in principle be done by rotation in the Earth's gravity field. This was demonstrated by White and Kerstner (1950), and elaborated on by Wildhack and Smith (1955), but has not been generally adopted for various reasons. This paper describes an implementation of the technique which has been developed at the National Measurement Laboratory (NML) in order to extend the calibration facilities down to 0.1 Hz or lower.

ROTATION VERSUS RECTILINEAR VIBRATION

Many commercial vibration exciters can provide good rectilinear motion suitable for accelerometer calibration over a range of frequencies down to about 20 or 30 Hz. One of the difficulties with lower frequencies is the large displacement required. For example, for a peak acceleration of 1.g (9.8 $m.s^{-2}$) at 80 Hz, the peak-to-peak-displacement is about 78 μm, but vibration at 0.1 Hz with the same peak acceleration has a peak-to-peak displacement of about 50 metres!

Wildhack and Smith (1955) showed that a rotated accelerometer is subjected to a sinusoidal acceleration of peak value 1 g, at the frequency of rotation, plus a centrifugal acceleration which has the effect of changing the apparent frequency response. They showed that as a consequence the frequency of resonance for an undamped accelerometer under rotation is $1/\sqrt{2}$ times the natural undamped frequency f_0.

The amplitude of the frequency response function is different for the two types of excitation, but it can be shown that for frequencies less than 0.1 f_0, the difference is less than 1 percent, and for frequencies less than 0.01 f_0 the difference is negligible. Thus if we restrict the rotational method to frequencies not greater than about 20 Hz, it should be suitable for accelerometers with $f_0 > 2$ kHz. Piezoelectric accelerometers used in shock testing generally have f_0 greater than 20 kHz.

Also, with rotational excitation the 90° phase shift occurs at $1/\sqrt{2}\ f_0$, and the phase at other frequencies also differs from the rectilinear case. However, again the difference is negligible (<<1°) for the frequency range which we are considering.

DETAILS OF NML DESIGN

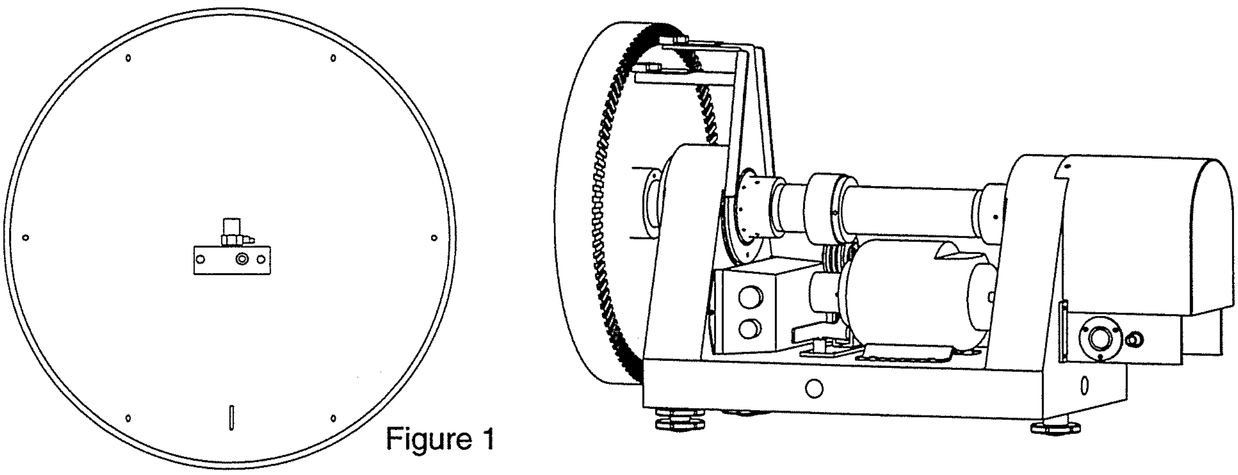

Figure 1

A steel flywheel of diameter 300 mm is constrained to rotate in a vertical plane by a pair of airbearings. The bearings, which support a 50 mm shaft and are spaced 300 mm apart, are supplied with filtered workshop compressed air at about 250 kPa. When run up to speed, the flywheel provides enough momentum to keep the wheel spinning for over an hour before friction eventually halts the system. Deceleration is exponential (Figure 2): it takes about 15 minutes for the speed to fall below 5 Hz, about 30 minutes to fall below 1 Hz, and about 70 minutes to stop, which gives plenty of time for low-frequency measurements. The speed of rotation is measured by a precision frequency counter monitoring an LED/Photodiode detecting the passage of 100 evenly spaced notches on the outer rim of the flywheel. As an accurate frequency reading is essential for calibration work, the photodiode signal is compared to a 10 MHz reference derived from one of NML's atomic clocks.

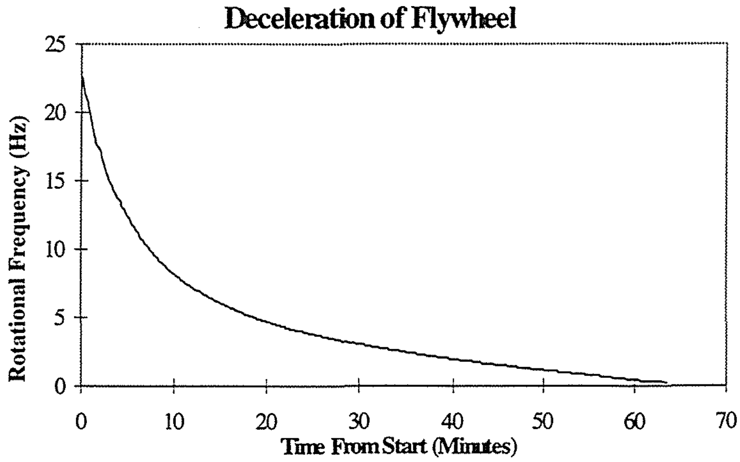

Figure 2

Two additional LED/Photodiode pairs placed 180 degrees apart act as positive and negative phase references by detecting the passing of a single small mirror embedded in the back of the flywheel.

Attached to the face of the flywheel is a mounting block for the accelerometer, arranged such that the accelerometer sensitive axis is normal to the flywheel's axis of rotation, which lies in the plane of the accelerometer mounting surface. Thus, when the flywheel is rotated, the accelerometer is subjected to a sinusoidally varying acceleration, along its sensitive axis, of amplitude equal to the local value of gravity (at the NML site this is about 9.7964 $m.s^{-2}$). From a connector on the mounting block, a coaxial cable takes the accelerometer signal to sliprings at the other end of the hollow shaft. Contact with the sliprings is by a set of silver brushes.

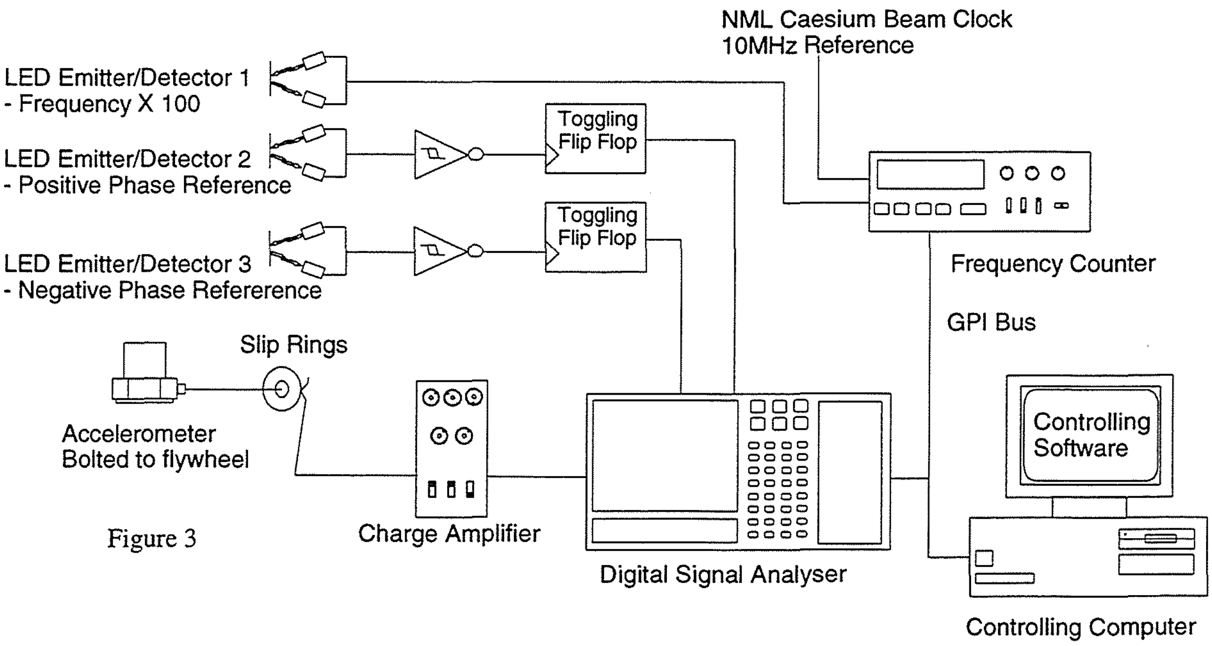

Figure 3

Data processing of the accelerometer signal and the two phase references is handled by a Data Precision DP6100B Digital Signal Analyser, utilising a calibrated 4 channel acquisition module. A diagrammatic layout of these details is shown above in figure 3.

The flywheel is driven up to speed by an electric motor via a friction drive on the shaft, midway between the two bearings. There is provision for balancing the flywheel with bolt-on masses. The whole assembly is mounted on a concrete inertia block. Background vibration is quite low at the ground floor location of the apparatus.

OPERATION

After the accelerometer is mounted and all connections made, the flywheel is statically balanced. Then it is run up to the required maximum speed, when the motor is switched off and the drive is disconnected.

As the flywheel slows down, the speed is measured and samples are taken of the outputs from the accelerometer and from the two phase reference photodetectors. The analyser and the frequency counter are controlled, and data is transferred, via a IEEE488 GPI bus from a computer. The program optimises the sampling rate as the speed decreases.

For each frame of data acquired, accelerometer voltage is determined by taking an FFT over an integral number of cycles. An algorithm is included to correct for the so-called "picket-fence " error. Average phase is determined over several cycles, comparing both positive and negative peaks with the two phase reference signals. The data is filed, and retrieved at the completion of a run for further calculations. From this data can be plotted phase, and sensitivity in volts/$m.s^{-2}$ or volts/g, corrected to ISO standard g (9.80664 $m.s^{-2}$), against frequency ranging from the maximum, which can go up to about 25 Hz, down to approximately 0.05 Hz.

RESULTS FROM TYPICAL CALIBRATIONS

To demonstrate the capabilities of the rotation apparatus, calibrations have been performed on several piezoelectric accelerometers. Figure 4 shows results for a Bruel & Kjaer 4367 with a 2651 charge amplifier set at 0.03 Hz LLF.

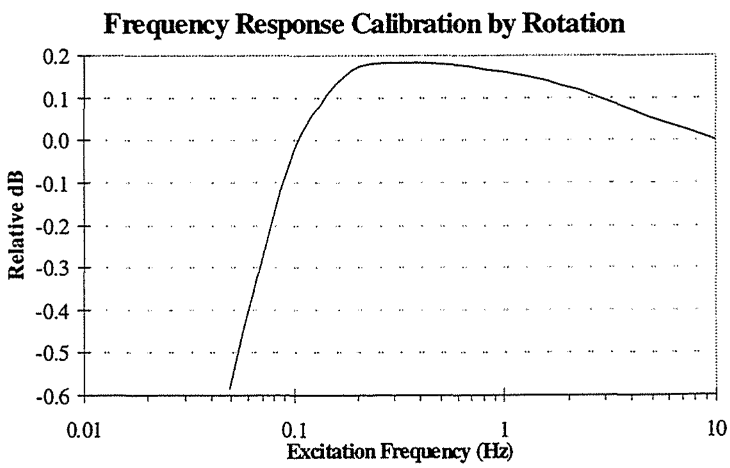

Figure 4

Figure 5 is for a Bruel and Kjaer 4371 with a B&K 4634 charge amplifier. This combination has a specified operating range from 1 Hz to about 11 kHz. These results are from an extended calibration using 3 methods: (i) calibration by rotation, from 1 Hz to 20 Hz, (ii) 2 Hz to 160 Hz rectilinear SHM on the NML horizontal air-bearing exciter, using a technique described in reference 1, and

(iii) 20 Hz to 5 kHz calibration by comparison with a reference accelerometer, on a commercial electrodynamic shaker. Figure 6 shows the corresponding phase calibration results.

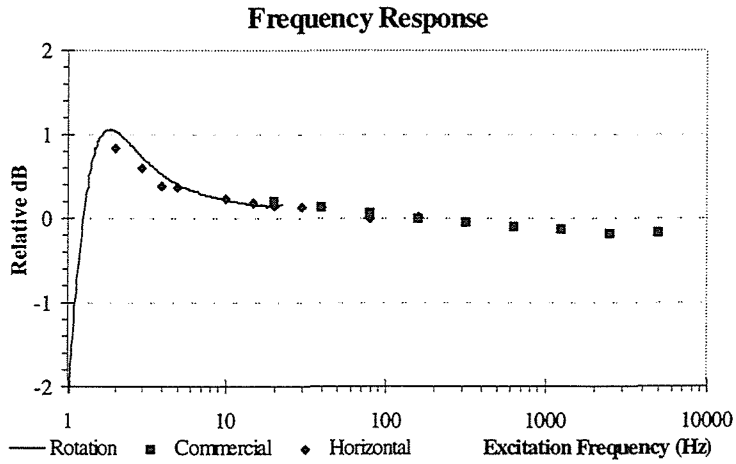

Figure 5

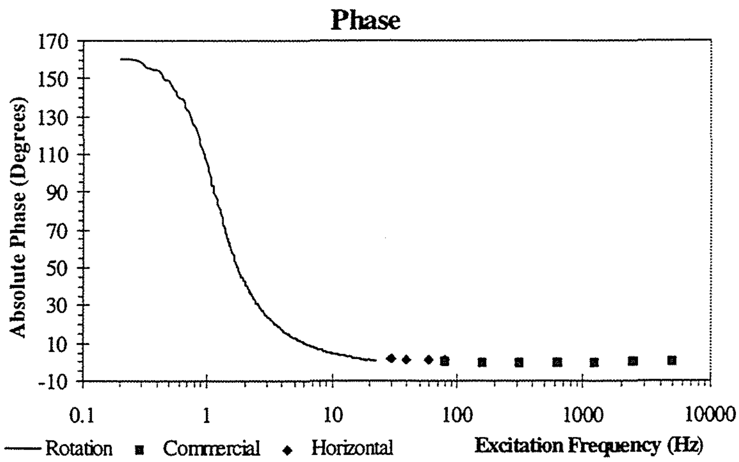

Figure 6

DISCUSSION

It has been said that the rotational method is only suitable for accelerometers which have "negligible" transverse sensitivity, because of the large transverse effect at that part of the cycle when the accelerometer sensitive axis is close to horizontal. However, most modern accelerometers have transverse sensitivity ratio (TSR) <5%, and such transverse sensitivity is usually ascribed to misalignment between the geometric axis and the true axis of maximum sensitivity. If this is so, then aligning the accelerometer such that the plane of maximum transverse sensitivity is normal to the plane of rotation should minimise the transverse error. In such a case, for a TSR of 5% the error is only about 0.13%, and the phase error is zero. If the TSR max plane is in the plane of rotation, the phase error is sine^{-1}(TSR%/100); for TSR=5% this is about 2.9 degrees (+ or -, depending on orientation).

A standard way of measuring time constant involves measuring the rate of decay following a step input. With an accelerometer, a step input can be applied by quickly standing the accelerometer up from an initially prone position. If the accelerometer system is represented as simply a capacitance C and a resistance R, the low frequency response Resp(f) can be estimated from the time constant t (=RC) from

$$\text{Resp}(f) = 20Log_{10}\left|\frac{1}{\sqrt{1+\dfrac{1}{(2\pi f)^2}}}\right|$$

Real acceleration measuring systems are less simple, and although the above can give a rough estimate of the -0.5 dB or -3 dB frequency, it cannot predict the response at other frequencies. For example, a system consisting of a Bruel & Kjaer 4367 accelerometer and a 2651 charge amplifier was subjected to a step input, and an exponential was fitted to the results. From this, $\tau = 3.8$ sec, from which $f_{(-0.5dB)}$ is estimated as 0.12 Hz. The 2651 LLF setting was 0.03 Hz. From the rotation calibration (see Figure 4) the -0.5dB frequency is 0.05 Hz.

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APPLICATION OF ACTIVE NOISE CONTROL TO NOISE BARRIERS

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ABSTRACT

Previous work has shown that the active noise control technology may improve the low frequency performance of noise barriers. In this paper, such possibility is confirmed. A multi-channel active control system has been used to create the quiet zones on the top of a barrier in order to reduce the diffraction along the barrier top and increase the insertion loss of the barrier. Both the simulation and experimental results obtained show that the barrier assisted with active noise control device achieves extra noise attenuation when the control system is optimally arranged. The results also demonstrate that the active noise control is effective at low frequencies in increasing insertion loss of a noise barrier. This feature of the active noise control overcomes the weakness of the noise barrier at low frequencies.

INTRODUCTION

Using barriers to control noise is a traditional and useful method. When screens, buildings or other large rigid barriers are interposed between the noise sources and the receivers, the propagating noise will be blocked. Only diffracted noise will be transmitted to the area behind the barrier. The diffracted field is relatively weak compared with the direct one in the area behind the barrier, thus sound attenuation occurs in this area, which is also called 'dark' area.

The effectiveness of a barrier in blocking the noise depends upon many factors such as the character of noise source, the shape and dimensions of the barrier, and environmental conditions. It has been found that while the barrier is very effective to attenuate high frequency noises, it becomes little useful or even useless to low frequency noises where the wavelength of the noise is comparable with the height and length of the barrier. Increasing the height of the barrier can improve the low-frequency performance of the barrier, but it is usually not practical. As a result, the improvement of the performance of a barrier, especially for the low-frequency noise, has been a research topic in the field of acoustics for more than 20 years.

Although the idea of using noise to cancel noise is not new (Lueg, 1936), the recent developments of control technique have made the implementation of active noise control practically possible. Because the active noise control (ANC) technique is very effective to attenuate low frequency noise (Nelson and Elliott, 1992), it is reasonable to believe that the low-frequency performance of the barrier may be improved by this technique.

Ise (1991) applied an adaptive control system into a 1/2 scale model of a passive barrier. In Ise's system, a speaker was used as a monopole control source and positioned on the top of the barrier, the error microphone was set in the desired space behind the barrier. He got a "quieter" area around the error microphone at very low frequencies (125 Hz or lower). Omoto (1993) used a multiple channel adaptive controller in his control system. Different from Ise's arrangement, Omoto put all the error microphones on the barrier top. As the sound pressure at the diffraction edge behaves as virtual sources of the diffracted field, the mechanism of this arrangement was to cancel the diffractive noise around the barrier top. For his specific configuration, Omoto concludes that when the interval of the error microphones on the diffraction edge is less than half of the wavelength, the active noise barrier works effectively.

The authors (1995 and 1996) have thoroughly investigated the active noise control in open space. It is found that large area (in terms of the wave-length) of noise attenuation can be obtained when the control system is optimally arranged. In this paper, we will apply our findings in active noise control in open space to the noise barrier, and to illustrate the effectiveness of the ANC technique in improving the low-frequency performance of noise barrier.

INSERTION LOSS OF NOISE BARRIER

Many theories may be used to predict and describe the sound insertion loss of noise barriers. The basic ones are the Huygen's principle and the Kirchhoff's diffraction formulation (Sommerfeld, 1954 and Born, 1959). For the reflective noise barrier shown in Fig. 1 and a point noise source with pressure field of

$$P_0 = \frac{A}{kr} e^{ikr}, \tag{1}$$

the diffracted field can be approximately expressed as (Bowman, 1969)

$$P_d = -\sqrt{\frac{2}{\pi k R_1}} A e^{-i\pi/4} \left\{ \mathrm{sgn}(\pi + \alpha - \phi) \frac{e^{ikR}}{\sqrt{k(R_1 + R)}} F\left[\sqrt{k(R_1 - R)}\right] + \mathrm{sgn}(\pi - \alpha - \phi) \frac{e^{ikR'}}{\sqrt{k(R_1 + R')}} F\left[\sqrt{k(R_1 - R')}\right] \right\} \tag{2}$$

for $kR_1 >> 1$, where k is the wave number of the sound, R and R' are the distances from the receiver directly to the source and to the source mirror image in the barrier. $R_1 = r + r_0$ is the shortest distance from the source to the receiver through the barrier top, $A = -iZ_0 q$ where q is the strength of the source and $Z_0 = \omega^2 \rho_0 / 4\pi c_0$, and

$$F(\mu) = \int_{\mu}^{\infty} e^{i\chi^2} d\chi \tag{3}$$

is the Fresnel integral.

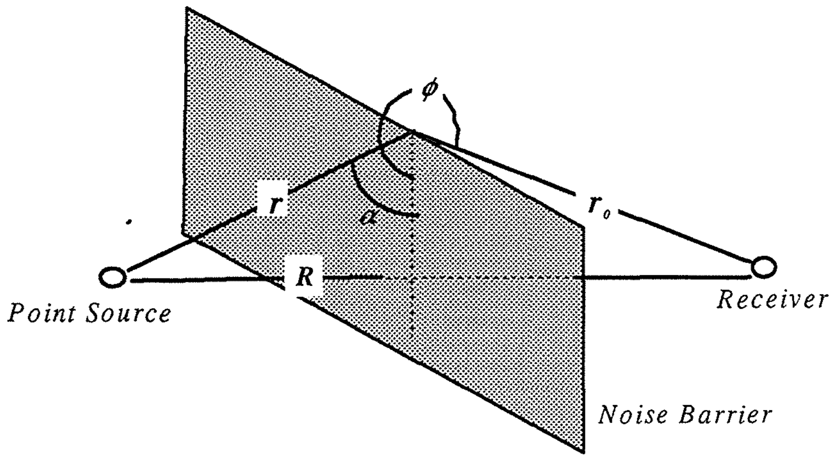


Figure 1. Demonstration of a noise barrier.

If the conditions $k(R_1 - R) >> 1$ and $k(R_1 - R') >> 1$ are also satisfied, the above expression can be further simplified as

$$P_d = \sqrt{\frac{2}{\pi k R_1}} \frac{A e^{i(kR_1 + \pi/4)}}{\sqrt{kr_0}\sqrt{kr}} \frac{\cos\frac{1}{2}\phi \cos\frac{1}{2}\alpha}{\cos\phi + \cos\alpha}. \tag{4}$$

The sound insertion loss caused by the barrier then can be given as

$$\Delta L = 20\log(|P_d|/|P_0|), \tag{5}$$

where P_0 is the sound pressure at the position of receiver when the barrier is absent, as expressed by Eq. (1). A widely used engineering approximation for the sound insertion loss of the barrier is Maekawa's asymptotic expression

$$\Delta L = -10\log(3 + 20N). \tag{6}$$

where N is Fresnel's zone number of the barrier, expressed as

$$N = \frac{2}{\lambda}\delta, \tag{7}$$

$\delta = r + r_0 - R$ is the path difference and λ the wavelength of the diffractive sound. It is obvious that the insertion loss of the noise barrier is directly proportional to the frequency of the sound.

The insertion losses of the barrier described by Eqs. (5) and (6) are shown as a function of frequency in Fig. 2 for a typical case, where the barrier is 1 m high. Located at different side of the barrier, the noise source and receiver are both 0.5 m high, and 2 m away from the barrier.

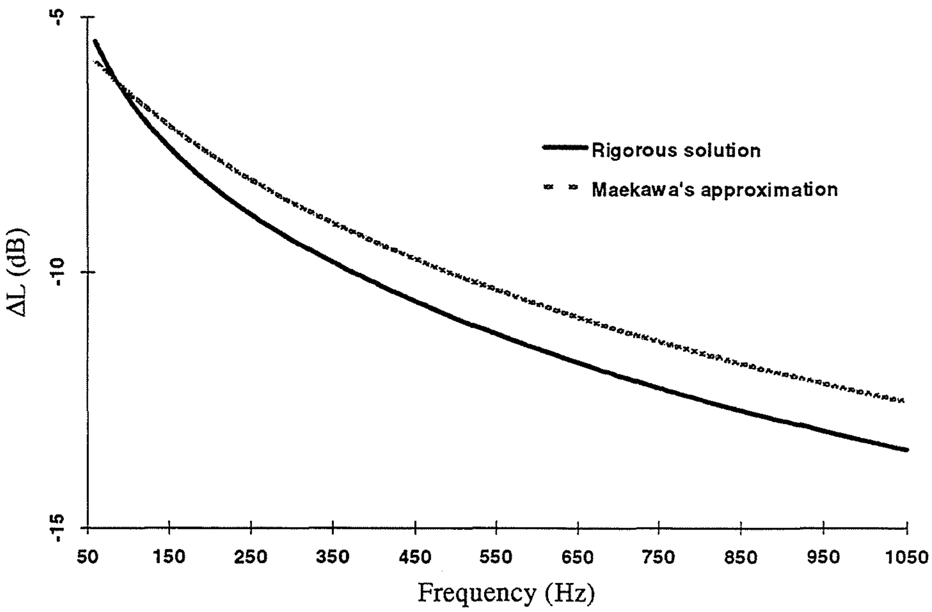


Figure 2. Sound insertion loss of a specific barrier as a function of frequencies.

It is shown in Fig. 2 that the sound attenuation of the barrier at a point in the 'dark' area is only 5 dB at the low frequencies, while the insertion loss at the high frequencies is more than 10 dB. These indicate that the effort on improving the performance of noise barrier should be focused on the low frequency range.

ACTIVE NOISE CONTROL IN OPEN SPACE

The active noise control in open space can be implemented by global control or local control. It has been found that global control can only be achieved when the control sources and the primary sources are closely located. In practical applications, this condition may not always be satisfied. Then the local control strategy seems to be the only choice. The objectives of local control of sound pressure field are (1) to create large quiet zones at required positions (where error sensors are located) and (2) to minimise the increase of sound pressure at other locations (or to minimise the increase in total power flow from all sources). Quiet zone is defined as the area where the primary sound pressure level is attenuated by more than *10* dB. If there are *M* primary sources and N secondary control sources distributed in the free field, and the corresponding source strengths are respectively:

$$\mathbf{q}_p = \left[q_{p1}, q_{p2}, \Lambda\Lambda, q_{pM}\right]^T, \tag{8}$$

$$\mathbf{q}_s = \left[q_{s1}, q_{s2}, \Lambda\Lambda, q_{sN}\right]^T, \tag{9}$$

where T denotes the transpose, the sound pressure at an observation position in the space can be expressed as:

$$P = P_p + P_s = \mathbf{Z}_p\mathbf{q}_p + \mathbf{Z}_s\mathbf{q}_s, \tag{10}$$

where P_p and P_s are the complex sound pressures at the particular field point position due to primary sources and secondary sources respectively. $\mathbf{Z}_p$ and $\mathbf{Z}_s$ are row vectors of acoustic transfer impedance from primary sources and from secondary sources to the observation position, respectively.

If L error microphones (error sensors) are used, the complex pressures at the L error microphones are $\mathbf{P} = \mathbf{Z}_{pe}\mathbf{q}_p + \mathbf{Z}_{se}\mathbf{q}_s$. The sum of the squared sound pressures at microphone positions is selected as the cost function, J, for local control:

$$J = \mathbf{P}^H\mathbf{P} = (\mathbf{Z}_{pe}\mathbf{q}_p + \mathbf{Z}_{se}\mathbf{q}_s)^H(\mathbf{Z}_{pe}\mathbf{q}_p + \mathbf{Z}_{se}\mathbf{q}_s), \tag{11}$$

where $\mathbf{Z}_{pe}$ is an $L \times M$ matrix of acoustic transfer impedance from the primary sources to the error sensors, $\mathbf{Z}_{se}$ an $L \times N$ matrix of acoustic transfer impedance from the secondary sources to the error sensors, and the superscript H denotes the Hermitian transposed. The strengths of the secondary source array are adjusted to minimise the cost function. This yields

$$\mathbf{q}_{s0} = -(\mathbf{Z}_{se}^H\mathbf{Z}_{se})^{-1}\mathbf{Z}_{se}^H\mathbf{Z}_{pe}\mathbf{q}_p. \tag{12}$$

If L=N, and with properly arranged secondary source and error sensor locations, $\mathbf{Z}_{se}$ can be nonsingular. For this case, the optimal secondary source strengths are simplified as

$$\mathbf{q}_{s0} = -\mathbf{Z}_{se}^{-1}\mathbf{Z}_{pe}\mathbf{q}_p. \tag{13}$$

This enables the sound pressures to be zero at all error sensors. Thus the quiet zones around these L positions can always be guaranteed. Then the sound pressure at any position of the space after control becomes

$$P = \mathbf{Z}_p\mathbf{q}_p - \mathbf{Z}_s\mathbf{Z}_{se}^{-1}\mathbf{Z}_{pe}\mathbf{q}_p. \tag{14}$$

The total radiated acoustic power of the system can be written as

$$W_T = \tfrac{1}{2}\left[\mathbf{q}_p^H \operatorname{Re}(\mathbf{Z}_{pp})\mathbf{q}_p + \mathbf{q}_s^H \operatorname{Re}(\mathbf{Z}_{ss})\mathbf{q}_s + \mathbf{q}_p^H \operatorname{Re}(\mathbf{Z}_{ps}^T)\mathbf{q}_s + \mathbf{q}_s^H \operatorname{Re}(\mathbf{Z}_{ps})\mathbf{q}_p\right], \tag{15}$$

where $\mathbf{Z}_{pp}$ is an $M \times M$ transfer impedance matrix among the primary sources, $\mathbf{Z}_{ss}$ an $N \times N$ transfer impedance matrix among the secondary sources, and $\mathbf{Z}_{ps}$ the $M \times N$ transfer impedance matrix between the primary and secondary sources. The principle of acoustic reciprocity applies in this discussion, ie. $\mathbf{Z}_{sp} = \mathbf{Z}_{ps}^{T}$.

It has been shown that the total sound power output of the local control system usually increases after control (Guo & Pan, 1995). The optimal configurations of the control system are those with the least increase of total power output and largest quiet zones. Figure 3 is a typical configuration of control system with multiple secondary sources, where the equally spaced secondary sources and error sensors are placed in two parallel lines. A monopole primary source is located in the central axis of the arrays of secondary sources and error sensors. The distance between the primary source and the secondary source array in the y direction is r_{ps}, and that between the secondary source array to the error sensor array is r_{se}. The secondary sources and the error sensors are separated respectively by r_{ss} and r_{ee}, with $r_{ss}=r_{ee}$ in this arrangement.

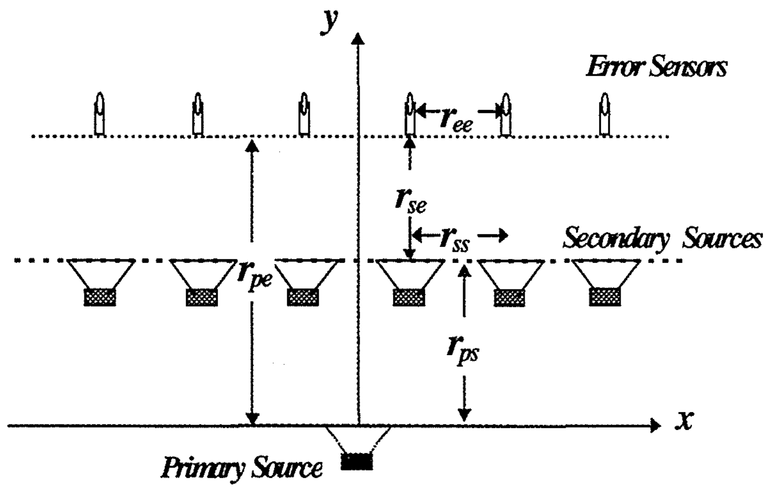


Figure 3. A typical arrangement of a multi-channel active noise control system in open space.

When the distances from the noise source to the control sources and from the control sources to the error microphones are given, there exists an optimal range of intervals among the adjacent control sources and error microphones. Within this range, the least increase of total sound power output and largest area of quiet zone can be obtained, which resembles a wedge with its edge along the error microphones. The upper and lower limits of this range are expressed as (Guo, Pan & Bao, 1996)

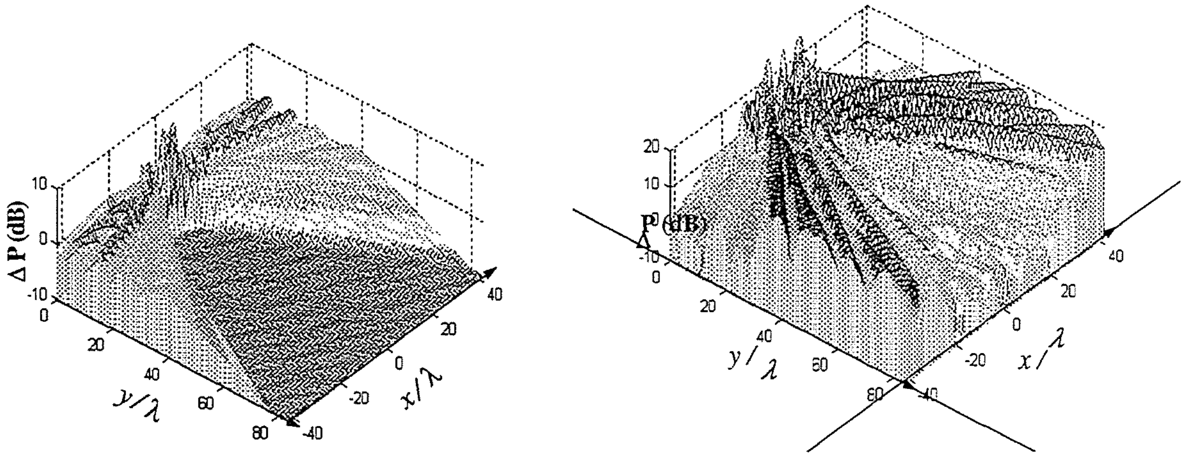
$$
\begin{aligned}
r_{ss-\max} &\cong \frac{\lambda}{2}\sqrt{1+\frac{4r_{se}}{N\lambda}} \\
r_{ss-\max} &\cong \frac{\lambda}{2}\sqrt{1+\frac{N+1}{N-1}\frac{4r_{se}}{N\lambda}}
\end{aligned}
\qquad
\begin{cases}
N = 2n \;\; (n = 1,2,3\Lambda\,) \\
N = 2n+1 \;\; (n = 1,2,3\Lambda\,)
\end{cases}
\tag{16}
$$

and

$$\begin{cases} r_{ss-\min} \cong \dfrac{5\lambda}{2} e^{-\left[\frac{3(\lambda+0.04 r_{ps})}{2r_{se}-\lambda}+\frac{20\lambda}{15\lambda+r_{ps}}\right]} & N = 2n \ (n = 2,3\Lambda\ ) \\ r_{ss-\min} \cong \dfrac{3\lambda(N+1)}{N} e^{-\left[\frac{\lambda+2r_{ps}}{2(2r_{se}-\lambda)}+\frac{12\lambda}{5\lambda+r_{ps}}\right]} & N = 2n+1 \ (n = 1,2,3\Lambda\ ) \end{cases} \quad . \tag{17}$$

It has also been found that for the configuration with the intervals outside the above range of the limits, the system is not be able to create a large area of quiet zone, and large sound power output is often observed for this case.

Figure 4 are the examples of the actively created quiet zones by a multiple control system with 21 control sources and 21 error microphones. The arrangement of the system is the primary source at the position *(0, 0)*, the *21* secondary sources at $(5\lambda\ (i\text{-}11)r_{ss})$ $(i=1,2...,21)$, and the 21 error sensors at $(10\lambda\ (i\text{-}11)r_{ss})$ $(I=1,2...,21)$, where $r_{ps}=5\lambda$ and $r_{se}=5\lambda$ The upper limit and the lower limit of the system can be calculated by Eqs. (16) and (17) as $r_{ss\text{-}min}=0.5\lambda$ and $r_{ss\text{-}max}=0.715\lambda$ Fig. 7(a) is the configuration within the optimal range ($r_{ss}= r_{ss\text{-}max}$) and Fig. 7(b) is the configuration outside the



optimal range (r_{ss}=0.88λ> $r_{ss\text{-}max}$).

Figure 4. Sound pressure attenuation of a control system with *21* secondary sources and *21* error sensors when $r_{ps}=5\lambda$, $r_{se}=5\lambda$, and (a) $r_{ss}=0.715\lambda$ and (b) $r_{ss}=0.88\lambda$

It can be seen that the quiet zone created by the optimal configuration is quite large, with no significant increase of sound pressure level outside the quiet zone. For the configuration just outside the limits, there is hardly any evident quiet zone and most of the areas suffer from a large increase of sound pressure level, as shown in Fig. 4(b). It can also be seen that the area of quiet zone created by optimally arranged control system is in terms of the wavelength of the noise, which means, in the practical application, the lower the frequency of the noise, the larger the area of quiet zone.

ACTIVE NOISE BARRIER

As the sound around the barrier top contributes diffractive sound field in the 'dark' area behind the barrier, it is reasonable to believe that if the sound around the barrier top can be cancelled, the diffractive sound in the 'dark' area behind the barrier should also be attenuated. This approach should have the equivalent effect to the increase of height of the noise barrier. Consequently the insertion loss of the barrier can be increased.

The multi-channel system used for the open space noise control is applied to the noise barrier. The control system consists of N control sources and the same number of error microphones, as shown in Fig. 5. The control sources and the error microphones are equally spaced in two parallel lines. The array of error microphones is located just on the top of the barrier. The control source array is located between the primary source and error microphone array, and in the same plane containing both primary source and error microphone array.

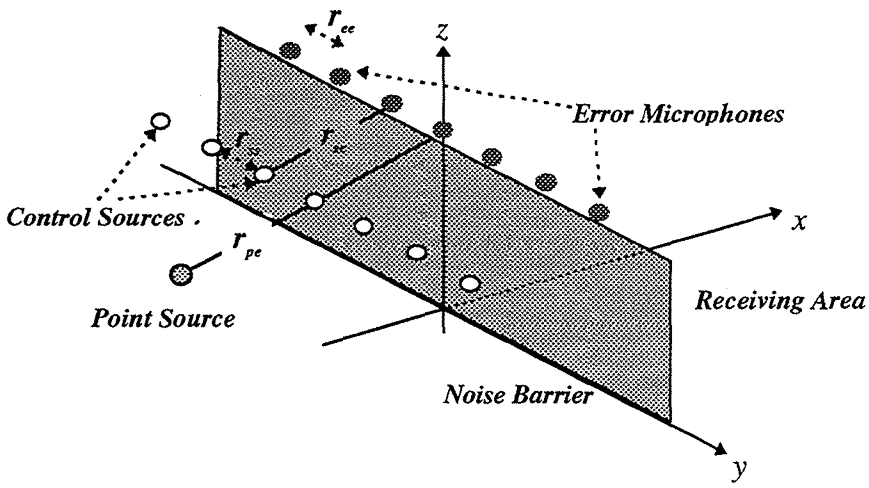


Figure 5. Active noise control system on a noise barrier.

To cancel the noise in the positions of error microphones and then create the quiet zones along the barrier top, which will increase the noise attenuation in the receiving area behind the barrier, the source strengths of the control sources should be adjusted to the values as expressed by Eq. (13). The total diffractive sound pressure becomes

$$P = P_{pd} + \sum_{i=1}^{N} P_{sd}^{(i)} . \tag{18}$$

P_{pd} is the diffraction caused by the primary noise source only, which also represents the diffractive sound while the active control system is off, and $P_{sd}^{(i)}$ the diffraction caused by the i control source. Both P_{pd} and $P_{sd}^{(i)}$ are expressed in the form of Eq. (2). Note that the strengths of the control sources are related to the strength of primary source by Eq. (13). The extra sound insertion loss created by the active multiple control system can then be described as

$$\Delta L = 20\log(|P|/|P_{pd}|) . \tag{19}$$

When applying the multi-channel active noise control system to the noise barrier, the configuration of the control system, such as the intervals of the adjacent control sources and the adjacent of the error microphones, is extremely important. It has been found that the optimal configurations of the control system in open space also apply to the active noise barrier shown in Fig. 5. The specific active noise barrier used in this analysis is 1 m high and located along the y axis. The location of the primary source is *(-1.376,0,0.5)* and the control sources are located at $(-0.688,(i-(N+1)r_{ss}/2,0.75)$. The error microphones are located at $(0,(i-(N+1)r_{ss}/2,1)$. For the control system with 3 control sources and 3 error microphones and the operating frequency at 500 Hz, the optimal range of r_{ss} is [0.22λ, 0.98λ] according to Eqs. (16)-(17). The extra sound attenuations of two configurations (one with r_{ss} within the limits as r_{ss}=0.75λ, another with r_{ss} outside the limits as r_{ss}=1.75λ) are given in Fig. 6. For this case, the grounds on both sides of the barrier are assumed as non-reflective.

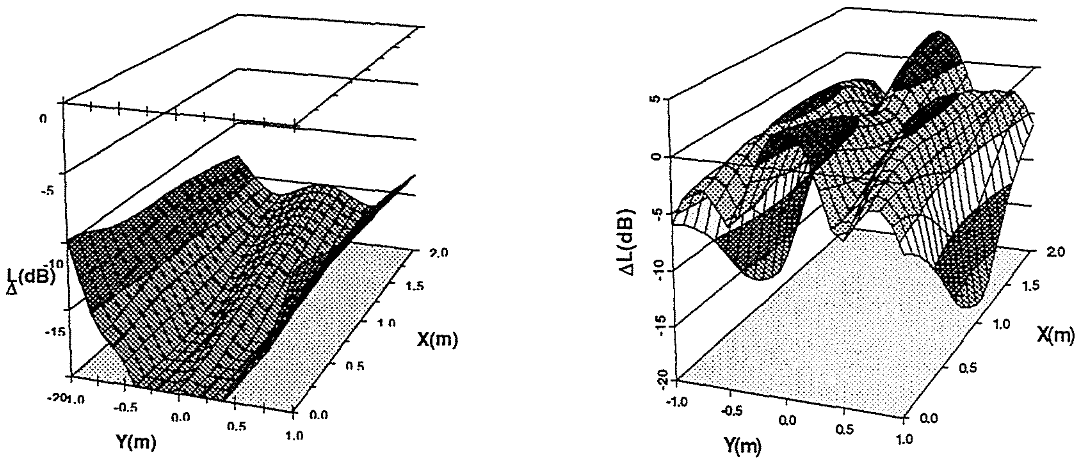


Figure 6. Extra sound attenuation due to the active noise control system when (a) r_{ss}=0.75λ and (b) r_{ss}=1.5λ.

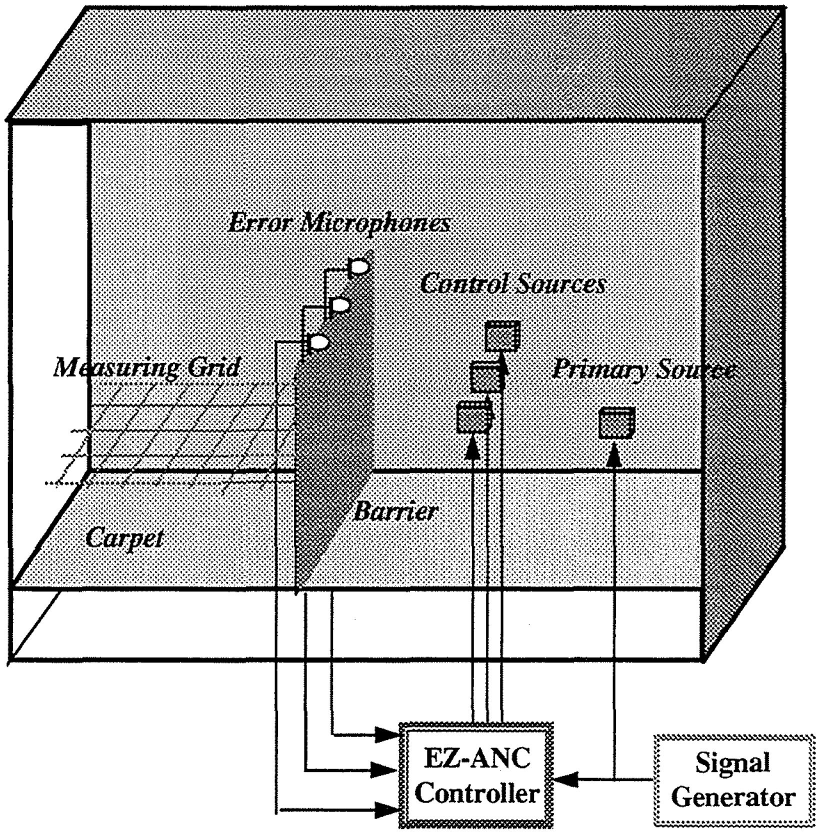


Figure 7. Experiment setup in an anechoic chamber.

Both the theoretical and experimental insertion losses of the noise barrier to the primary noise source in a measuring plane (0.5 m above the floor) are shown in Fig. 8. It can be seen that the sound attenuation by the barrier in experiments is smaller than that from the theoretical prediction. This is due to the reflections from the carpet covered floor and the walls of the anechoic chamber. It will be shown later that those reflections will decrease the effectiveness of the active noise barrier too.

Although the idea of using noise to cancel noise is not new (Lueg, 1936), the recent developments of control technique have made the implementation of active noise control practically possible. Because the active noise control (ANC) technique is very effective to attenuate low frequency noise (Nelson and Elliott, 1992), it is reasonable to believe that the low-frequency performance of the barrier may be improved by this technique.

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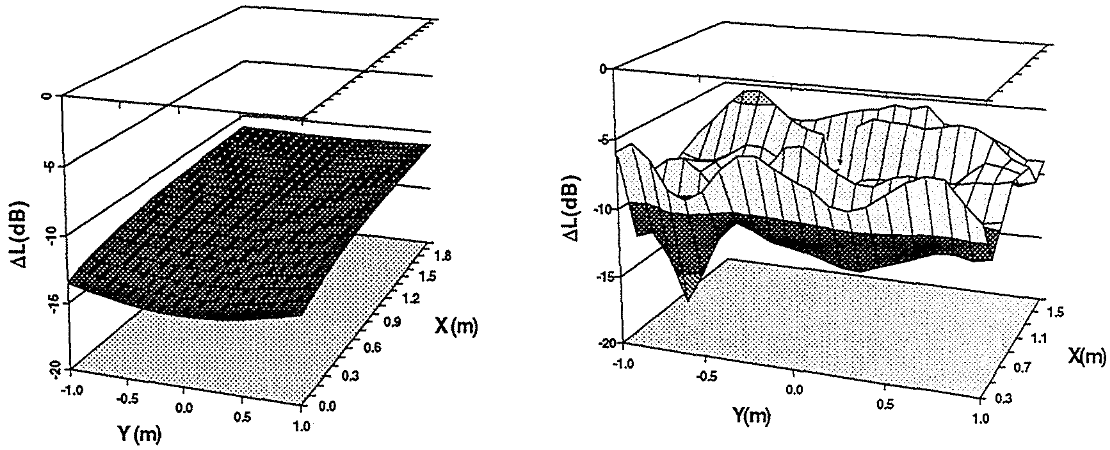

Figure 8. Insertion losses of the barrier for (a) theoretical prediction and (b) experimental case.

The coordinates of the control sources and error microphones are the same as that for the computer simulation discussed above [$(-0.688,(i-(N+1)r_{ss}/2,0.75)$ and $(0,(i-(N+1)r_{ss}/2,1)$ respectively, where $i=1,2,3$, and $N=3$]. Two different intervals of the control sources, one is within the optimal range as r_{ss}=0.75λ, another outside the optimal range as r_{ss}=1.75λ, were used to test the effectiveness of the active noise barrier. The extra sound attenuations by these two configurations of active noise control system are given in Fig. 9.

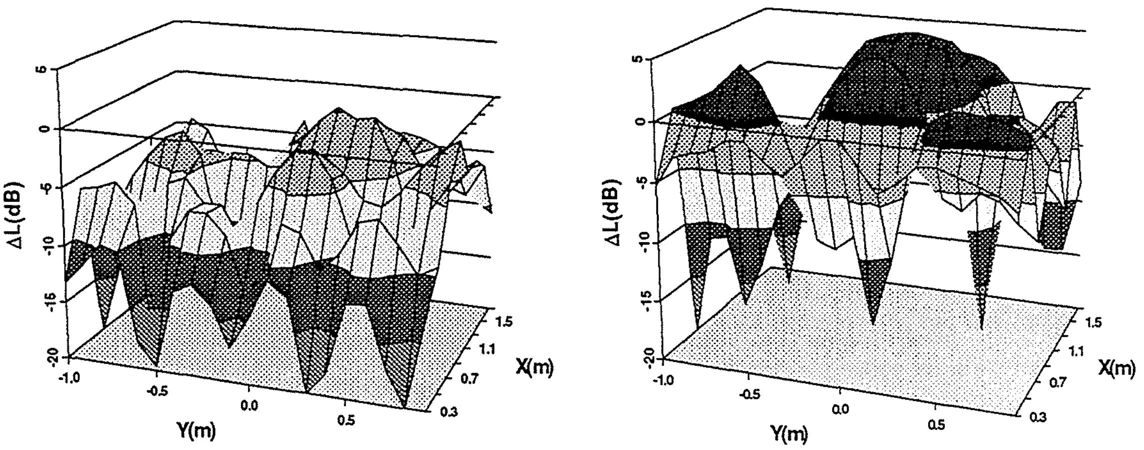

Figure 9. Extra attenuations due to the active noise control system when (a) r_{ss}=0.75λ and (b) r_{ss}=1.5λ.

Figure 9 shows a significant difference in extra sound attenuation of the active noise barrier for the different configurations of the control system. When the system is optimally arranged, the extra sound attenuation has been achieved at every position in the measuring plane (shown in Fig. 9(a)). When the system is arranged outside the optimal range, the active noise control system may even decrease the insertion loss of the passive barrier in some locations, as shown in Fig. 9(b).

Comparing Fig, 9 with Fig. 6, it is easy to find that the extra sound attenuation of active noise barrier in the experiments is not as big as that of the theoretic analysis. This is due to the reflection from the carpet, as well as from the walls of the anechoic chamber. In practice, some treatment in reducing the reflections from grounds and side walls is needed in order to increase the effectiveness of the active noise barrier.

CONCLUSIONS

The effectiveness of applying active noise control system to improve the sound insertion loss of the barrier has been demonstrated in this paper. To cancel the sound pressure at multiple positions and then create a large area of quiet zone around the diffraction edge of the barrier is a good approach to increase the sound insertion loss of the barrier. An optimal multi-channel active control system developed from the active noise control in free space can be successfully applied to the barrier. Similar to the cases of control in open space, the configuration of the control system in active noise barrier is extremely important. The extra sound attenuation due to the active noise control system can be very big only when the system is optimally arranged, otherwise, the active control system may be useless or even deteriorate the insertion loss of the barrier.

For the multi-channel active control system, the more control sources are used, the larger size of the improved 'dark' area. As the size of the improved 'dark' area is in terms of the wavelength of the noise, it can be concluded that the active noise barrier is more useful in low frequency range where the passive noise barrier is not effective.

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A SIMPLE TEST RIG FOR ILLUSTRATING FAN NOISE CHARACTERISTICS

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ABSTRACT

A simple portable test rig has been designed and constructed to illustrate some of the features that can have a significant influence on the noise levels radiated by fans. These include the effect of varying flow rates and motor power. More importantly the effect on noise of flow condition caused by disturbances such as flow separation and non-uniform velocity profiles can be demonstrated convincingly. The changes can be measured but are easily discerned by listening. The rig has proved useful for teaching purposes and has drawn attention to the caution needed when applying the familiar "fan laws" in acoustic design.

INTRODUCTION

The normal "fan law" equations, eg ASHRAE, Cory, are well known as a basis for designing duct and ventilation systems. They are available in a variety of forms and typically involve parameters such as flow rate, pressure rise and fan efficiency. The fan type is included also since these determine the various "specific sound power levels" which are the starting point for the calculations. The blade passing frequency is usually relevant giving rise to the blade passing correction.

All of the above is presented to students in the final year of the mechanical engineering undergraduate program of this university. They are warned as well of the necessity to consider the aerodynamic design of the ducting system and the need to minimise power requirements by careful attention to detail. The choice of the fan type and the benefits derived from operating at reasonable efficiency are also discussed.

While some 2 hours of lectures and calculations are given over to the above, it was considered that a **demonstration** of the principles would be a very effective way of conveying, with more impact, the same information. Thus a design project was set in which the students were asked to: "design a portable demonstration rig to illustrate the effect on fan noise of as many of the following features as possible: (i) Flow rate, (ii) mechanical imbalance of the fan, (iii) obstructions to the air flow, (iv) the effects of streamlining, (v) vibration isolation, (vi) excitation of acoustic resonances."

The total cost was set at $400 and the students advised that the best design would be constructed, evaluated and, if successful, used for demonstrations. An added bonus was that the rig could be used to illustrate aerodynamic principles in matching fan characteristics to duct system requirements.

This paper discusses briefly, the characteristics of the best design. Most of the above criteria were met, although further development is required to meet the imbalance/isolation criteria. The changes in noise levels, from 10 to 40 dB(A), can be easily discerned by simple listening.

FAN TEST RIG

Figure 1 shows the design of the test rig. It consisted basically of several sections of clear perspex tube (150mm diameter) into which was mounted a propeller driven by a variable speed electric motor. Several sliding joints were used so that the duct length and the position of the motor could be altered. Bell mouthed entrances and exits could be used. The motor was supported by 3 streamlined struts through which the power leads passed. A streamlined afterbody could be attached to the motor and a slot was provided in the duct upstream of the fan, into which obstructions could be inserted. The slot was normally sealed and airtight. The electric motor was continuously variable in speed up to about 10,000 rpm which could produce air speeds up to 10 m/s and pressure rises of the order of 140 Pa. Power consumption was about 70 W. Reversal of power allowed the flow direction to be changed. The noise radiated from the inlet of the duct could be measured or alternatively the power radiated could be obtained by measuring acoustic intensity. The usefulness of the rig is indicated in the following section.

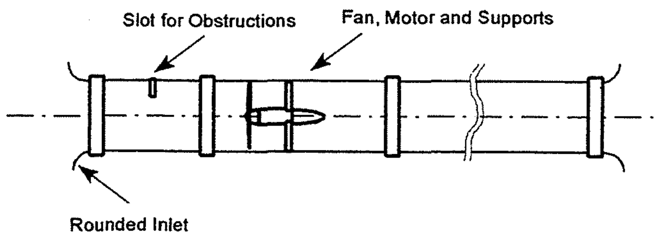

Figure 1

STREAMLINED DUCT

Tests were carried out initially with the duct in the fully streamlined condition. Both aerodynamic and acoustic parameters were measured. The former are only briefly referred to while the latter are discussed in more detail. For the moment only the sound pressure levels at 1.5 m from the duct entrance, and at 45° to the duct axis are reported but are considered useful in indicating general trends.

For this configuration, the flow could be varied as follows:

Velocity (m/s)	Pressure (Pa)	Power to Motor (W)	Fan Speed rpm
1 to 10.9	1 to 90	0.7 to 62	1270 to 9840

Figure 2 shows the variation in the "A" weighted noise level as a function of the power supplied to the electric

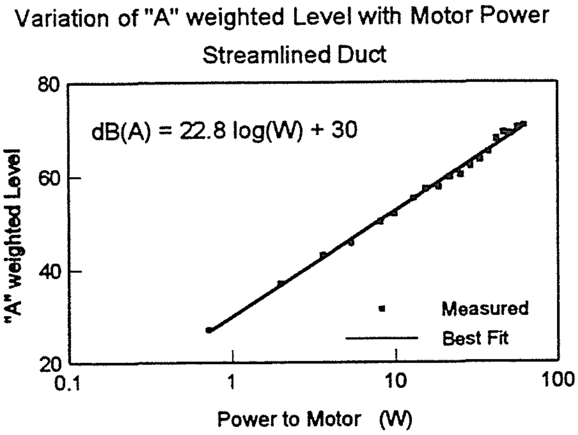

Figure 2

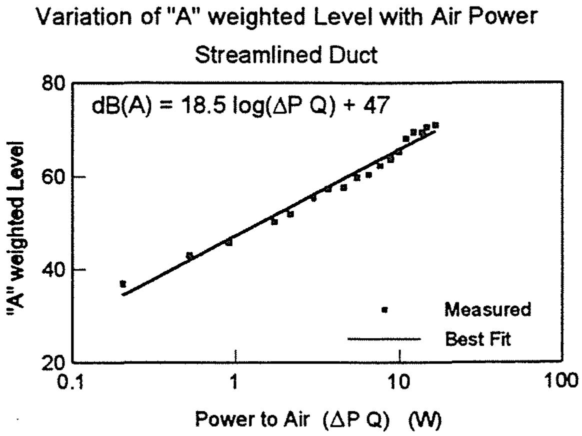

Figure 3

motor. It is clear that there was a logarithmic relationship and that a doubling of the power to the motor caused about a 7 dB(A) increase in propeller noise. A similar logarithmic relationship existed between the noise levels and the useful power imparted by the propeller to the air (measured as the product of the pressure rise and the flow rate). In this a doubling of the useful power caused a $5^1/_2$ dB(A) increase in noise levels. Note that the propeller was not operating very efficiently in that, say at 60 dB(A), the motor power was 20W while the power transferred to the air was about 5W. The air speed was 7.6 m/s.

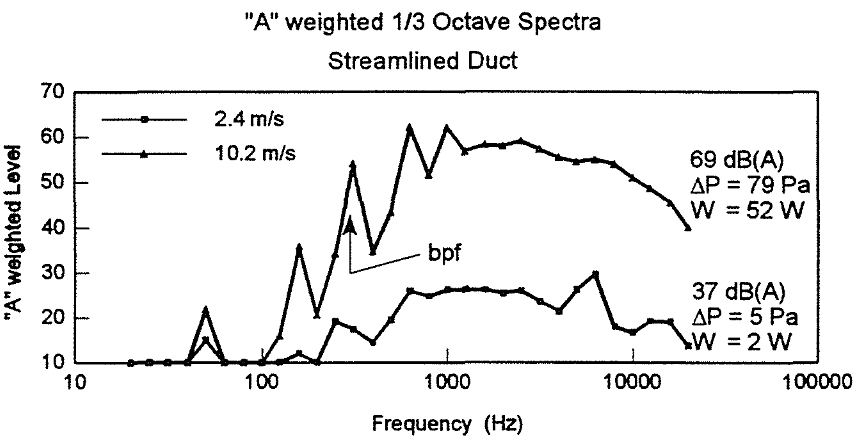

Figure 4

An indication of the variation in spectra can be seen by reference to Figure 4. At 2.4 m/s the spectra was relatively flat above about 500 Hz and only marginally above the background noise level. The blade passing frequency (90Hz) was not visible. At the upper speed, 10.2 m/s, the blade passing frequency was evident and there was also a substantial increase in spectral level - about 40 dB(A). The variation in noise level and the greater dominance of the blade passing frequency was obvious to even a casual listener.

ABRUPT INLET

The removal of the bell mouth entrance allowed a demonstration of the effect on noise of a disturbed air flow entering the propeller. For this configuration the aerodynamic parameters could be varied as follows:

Velocity (m/s)	Pressure (Pa)	Power to Motor (W)	Fan Speed (rpm)
1.0 to 9.2	6 to 139	2 to 63	2460 to 9954

Compared to the streamlined case, the maximum air speed had reduced and the pressure rise across the propeller had substantially increased. The variation in the "A" weighted level with motor power is given in Figure 5 and has the same logarithmic characteristic of the previous case. The same applied to the variation with air power. Each of these was similar in level also to the streamlined case. Here a noise level of 60dB(A) related to a motor power of about 16W and a propeller power of almost 5W. The air speed in this case was 5.4 m/s - compared with 7.6 m/s for the streamlined case.

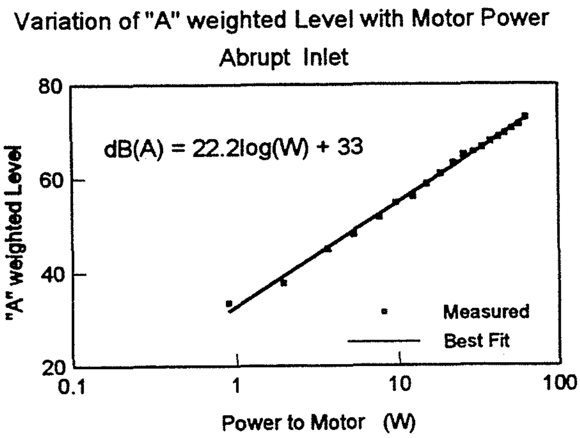

Figure 5

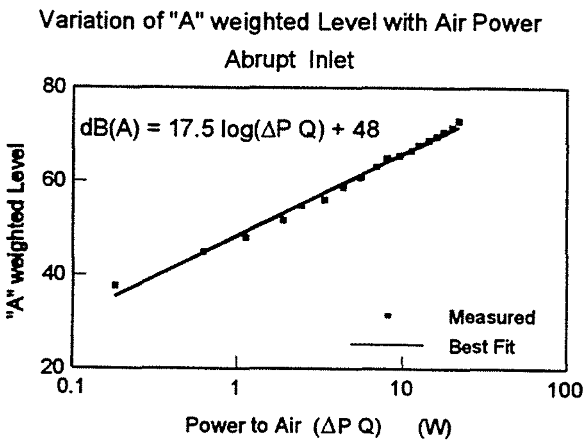

Figure 6

The spectral variation of the radiated noise is given in Figure 7. The pattern was similar to the previous case although there was a general increase in spectral level for comparable air speeds.

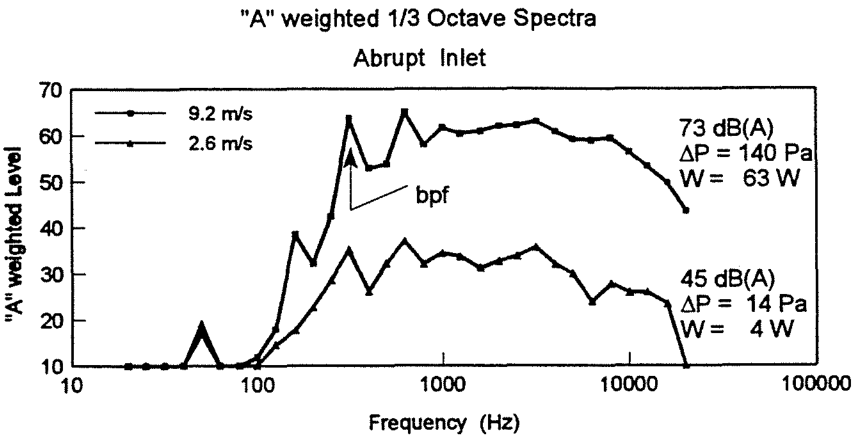

Figure 7

Several other configurations were investigated but due to space limitations, only the most severe will be reported, namely the abrupt inlet followed by obstructions immediately upstream of the propeller.

ABRUPT INLET AND OBSTRUCTIONS

The range of aerodynamic values possible for this configuration was as follows:

Velocity (m/s)	Pressure (Pa)	Power to Motor (W)	Propeller Speed (rpm)
0.3 to 3.0	1 to 93	2.0 to 65	1367 to 10359

Note that while the power supplied to the motor was comparable to those of the previous tests, the flow velocity was substantially less.

The variation of the "A" weighted sound pressure level with power supplied to the motor again followed the logarithmic pattern of the previous tests although the noise levels were a little higher, Figure 8.

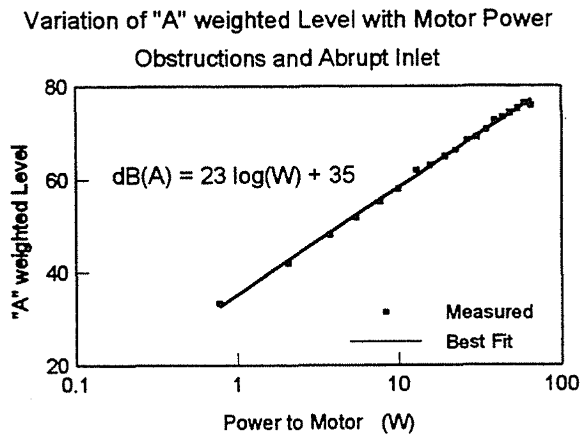

Figure 8

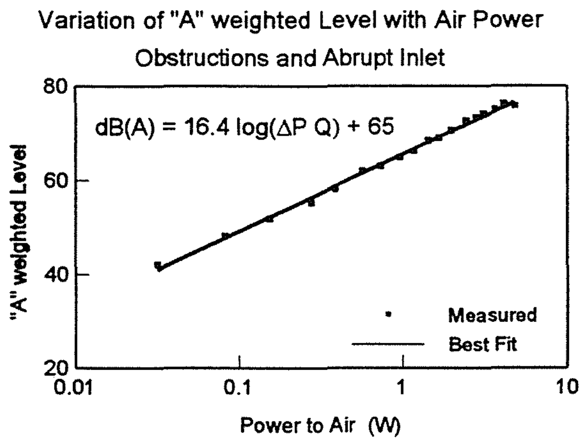

Figure 9

The logarithmic relationship occurred also when the noise levels were compared with the power delivered by the propeller, Figure 9, but were substantially higher than in the previous cases. In this case a noise level of 60dB(A) related to a motor power of 12 W and a propeller power of 0.5 W. The air speed for this case was only 1 m s^{-1}.

The spectral shapes and levels were similar to those of the previous configurations but occurred at much lower flow speeds.

GENERAL TRENDS

A summary of the results for all the various tests carried out is given in Figure 10 for the motor power variations and in Figure 11 for the propeller power variations.

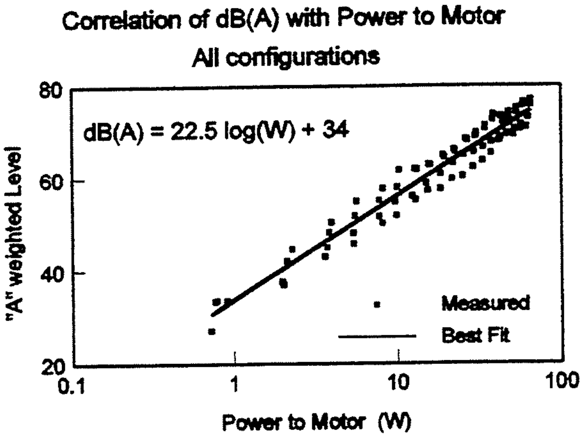

Figure 10

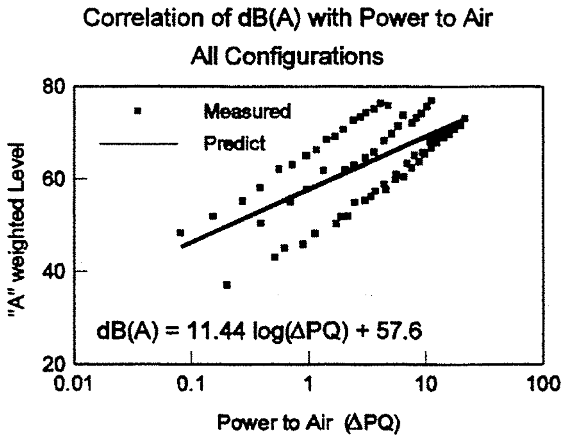

Figure 11

The former produced a fairly convincing relationship with relatively small scatter implying that the measurement of power consumed by the motor was a good indicator of the noise levels. In contrast the measurement of the useful power provided by the propeller varied significantly and indicated that pressure rise and flow rate alone were not reliable indicators of noise levels. The condition of the flow entering the propeller could have a very significant effect on noise levels.

Some further indications of how flow conditions could affect noise levels are given in Figure 12 where the "A" weighted spectra of the radiated noise are shown for an air speed of about 3.3 m/s. Four cases are presented and indicate that variations of several decibels can occur and as much as 30 dB(A) for extreme cases. Power consumed by the motor ranged from 4 watts for the streamlined case, 6 watts for the abrupt inlet and reversed flow state and about 60 watts for the obstructed case.

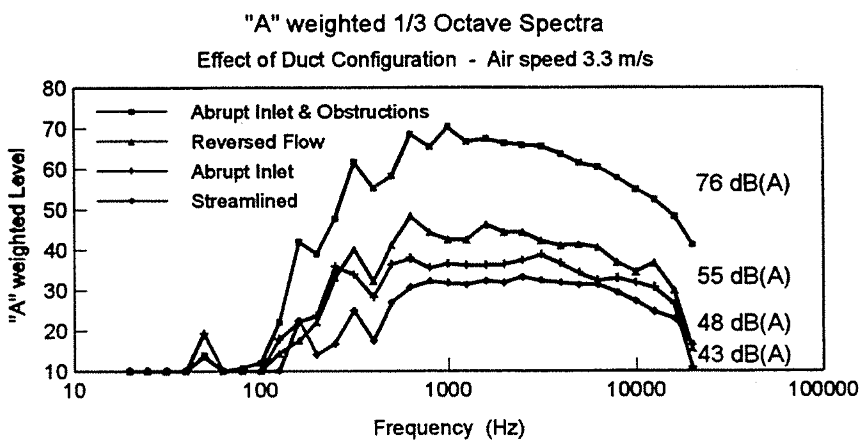

Figure 12

TONAL EXCITATION

Removal of the cover from the obstruction slot allowed air to be drawn into the duct at that position. This gave rise to the possibility of exciting a flute like sound when the tube resonances and blade passing frequencies could reinforce.

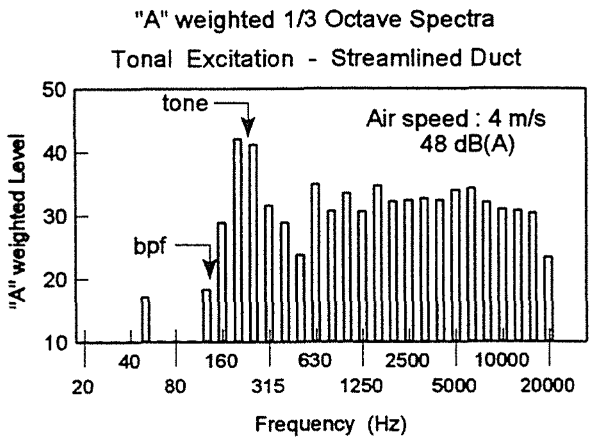

Figure 13

By varying the motor speed it was a relatively simple matter to excite a resonance which could easily be detected by simple listening. The corresponding "A" weighted spectra, Figure 13, shows that a resonance at about 220Hz had been excited and was probably coupling with an harmonic of the blade passing frequency, 120Hz.

CONCLUDING COMMENTS

The student designed test rig has proved to be very useful for illustrating various factors that affect fan noise. The changes were of a magnitude that classroom demonstrations do not require careful measurements of sound pressure level but could rely simply on subjective listening. Thus the effect of varying speed, flow conditions, reversal of flow and the presence of obstructions could be immediately appreciated. The benefits in minimising power requirements by attention to aerodynamic detail were clear. While one picture may be worth 1000 words, one ten minute demonstration was better value than two hours of lectures.

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PRACTICAL IMPLEMENTATION OF AN ACTIVE NOISE CONTROL SYSTEM IN A HOT EXHAUST STACK

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ABSTRACT

Practical issues associated with the installation of an active noise control (ANC) system in an 80m high exhaust stack, containing a hot, wet and dirty air flow, are discussed. The noise problem to be controlled was a 165Hz tone generated by the fan at the bottom of the stack and radiated by the stack into the surrounding community from which complaints were received on a regular basis, sometimes from residents living more than 1km away. The stack was split into three parallel axial sections in the vicinity of the ANC system to ensure that only plane waves were present. The use of the ANC system resulted in reductions of the tonal noise of 10dB inside one section of the duct and 20dB inside the other two sections. It is expected that similar reductions in the tonal noise would be measured in the community. However this latter measurement was not done because it was found that the tonal noise was masked by the background noise, even with the ANC system switched off. This was a result of the installation of the axial splitters which divided the duct into 3 sections such that the acoustic path length difference was 1/3 of a wavelength between adjacent sections. However, in the event that further complaints of tonal noise are received from the community, the work reported here indicates that an ANC system will be a viable and cost effective solution.

INTRODUCTION

The project described here was undertaken to reduce the 165Hz tonal noise radiated by an 80m high exhaust stack to the surrounding community. A large, 4m diameter centrifugal fan at the base of the exhaust stack forces air into the stack and generates tonal noise at the blade pass frequency (BPF). The fan had 10 blades and rotated with a speed of about 993rpm under normal plant operating conditions.

The stack is characterised by a normal operating temperature of 100°C which rises to 180°C at times. The exhaust from the fan consists of very moist and abrasive clay dust which sticks to non-vertical surfaces, forming a thick sludge. Previous attempts at the installation of a passive muffler had not been successful on a long term basis, partly because of the abrasive and sticky nature of the exhaust flow. With community complaints increasing, those responsible for the stack were left with the option of installing a passive muffler (of uncertain life) at a cost of over $200,000 plus considerable plant downtime and inconvenience or giving active noise control a try. The system design and performance corresponding to the latter option is described here. The design specification was a noise reduction of 10dB to 15dB in the community at the frequency of the blade pass tone which varied from 152Hz to 166Hz, depending on plant operation conditions.

The project began by installing two axial splitters in a section of duct about 2 metres upstream from the fan so that the duct was divided into 3 parallel axial sections (see Figure 1). This required waiting until a scheduled plant shut down, then removing the duct insulation, cutting off one side of the duct wall (6.5m long by 3m wide, which was constructed of 10mm thick mild steel), welding in the splitters so there was no acoustic connection from one duct section to the next and then welding back the duct wall.

The reason for splitting the duct was to ensure that only plane waves were propagating in the vicinity of the ANC system. A system without splitters would have resulted in the propagation of higher order acoustic modes in the duct and would have required considerably more effort to develop a suitable ANC system. Thus the splitters were installed to ensure that no duct section dimension exceeded 0.5 wavelengths (about 1.23m - see below). An added benefit of the splitters was the possibility of additional noise reduction due to passive cancellation as a result of the sound path differences between adjacent sections of the duct being fortuitously close to one third of a wavelength (at the blade pass frequency).

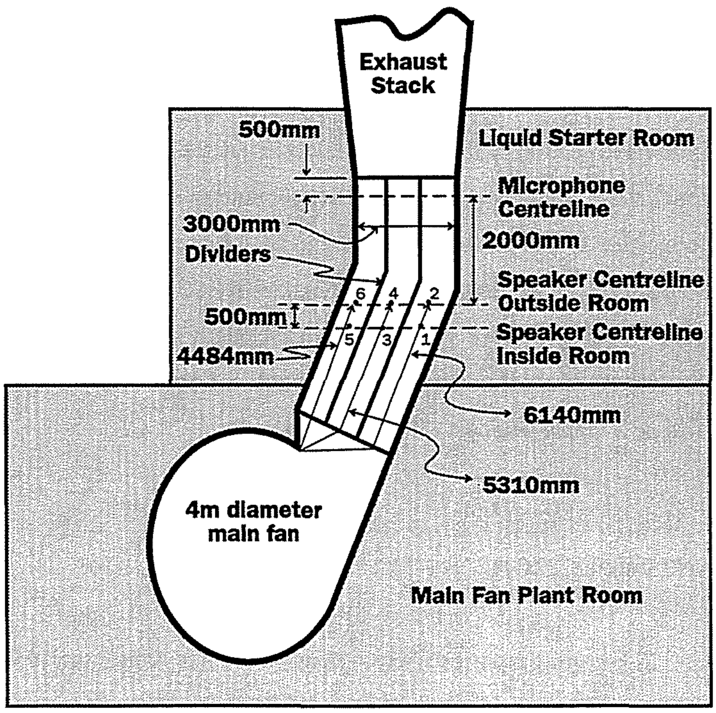

Figure 1: Main fan, microphone and loudspeaker locations, numbers represent loudspeaker locations (see Table 1).

At 100°C, the speed of sound is 388m/s. Assuming an air flow speed in the duct of 20m/s, the effective speed of sound is 408m/s and the wavelength at 165Hz is 2.47m. One third of this is 0.82m which is the path length difference between adjacent duct sections. However, the passive cancellation effect was not expected to replace the ANC system because the passive system would not be able to track temperature, fan speed and flow speed variations, all of which affect the effective wavelength of sound at the blade passage frequency.

The design of feedforward active noise control systems may be divided into two separate tasks; the design of the physical system and the design of the electronic control system as shown in Figure 2.

The physical system consists of:

- the fan which is responsible for the unwanted tonal the noise,
- the exhaust stack along which the noise propagates,
- a tachometer for measuring the fan rotational speed,
- microphones which measure the noise inside the exhaust stack,
- loudspeakers which provide additional sound to "cancel" the BPF.

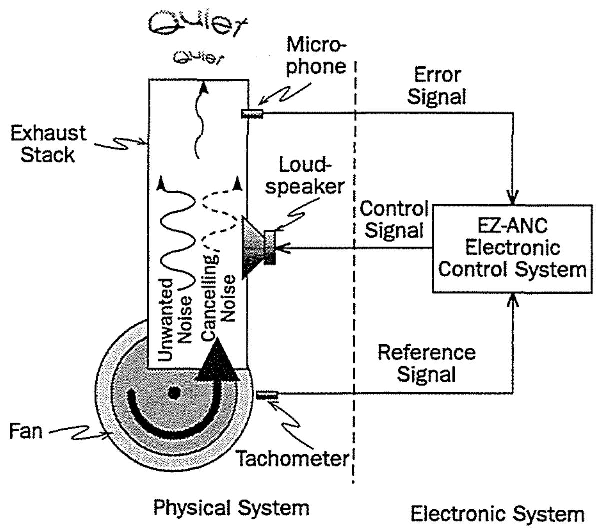

Figure 2: Basic components of the active noise control system.

The physical system includes transducers (loudspeakers and microphones) which convert physical properties into electrical signals. These electrical signals are used by the electronic system which comprises:

- a digital controller which generates an appropriate control signal based on the microphone and tachometer signals,
- power amplifiers (not shown) which amplify the control signal for the loudspeakers,
- instrumentation filters and amplifiers (not shown) which are used to condition the electrical signal so they are suitable for the digital controller.

The design of each of the system components will now be discussed in detail. This discussion will be followed by a summary of the results achieved.

PHYSICAL SYSTEM DESIGN

Before discussing the design of the components of the physical system, it is worth mentioning the factors that limit the performance of the overall system and the hierarchical importance of each factor. The first factor which limits control system performance is the location of the control sources. Once these locations have been optimised, the error sensor locations will determine the maximum achievable noise reduction. The next factor is the quality of the reference signal. If this is contaminated with frequency components which need not be controlled, then the achievable control of the components which do require control will be reduced. In fact, in an ideal situation, the relative strength of each frequency component in the reference signal should reflect the relative desired amount of reduction of each of the components in the error signal. Perhaps the hierarchical nature of active control can best be understood by reference to Figure 3.

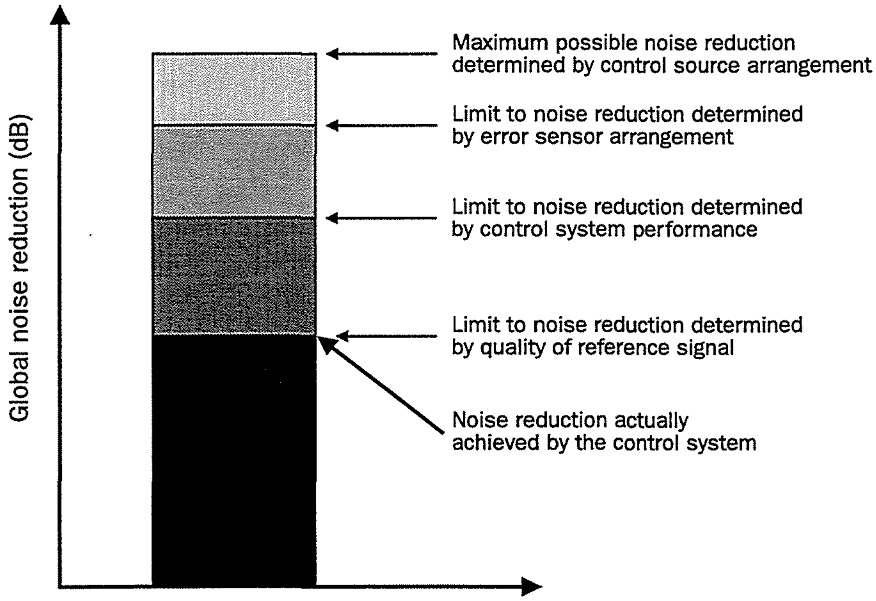

Figure 3: Performance hierarchy for active noise control

Control source location

The location of the control source in terms of wavelengths of separation between the control source and effective location of the offending noise source is crucial to the success of the installation, in terms of how hard the loudspeaker will have to be driven and the maximum amount of control achievable. The optimum locations can be calculated exactly for a constant pressure source or a constant volume velocity source (Snyder, 1991).

Unfortunately, the worst locations for a constant volume velocity source are the best locations for a constant pressure source and most industrial noise sources are somewhere between these two ideal cases, so the ideal location requires some trial and error testing to ascertain it with any degree of certainty. As a centrifugal fan is close in nature to a constant pressure source, this idealisation will be used here for illustrative purposes. Another complicating factor is the effective impedance of the primary source in terms of the phase and amplitude of the reflection of upstream propagating acoustic waves. This also affects the optimum control source location.

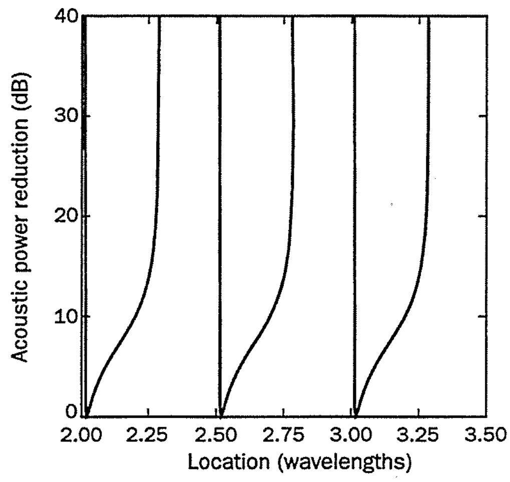

Figure 4: Total acoustic power reduction as a function of primary/control source separation for a constant pressure non-rigid primary source (Snyder, 1991).

Figure 4 shows the maximum achievable noise reduction for a constant pressure source and a non-infinite impedance at the plane of the source. Note that the maximum achievable reduction for an infinite impedance plane at the primary source is theoretically infinite.

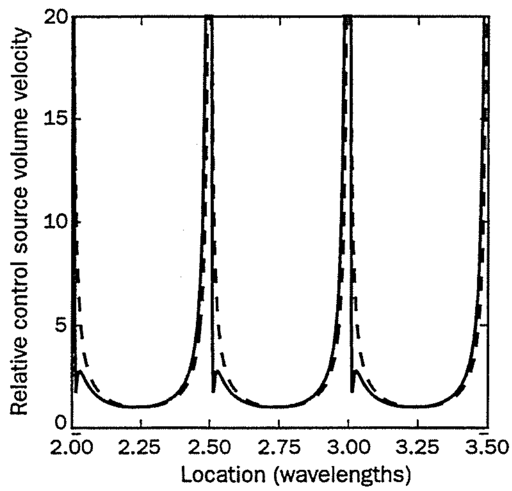

Figure 5: Relative control source volume velocity as a function of primary/control source separation for optimal control and constant pressure primary source (Hansen and Snyder, 1996).

Corresponding required volume velocities from the control source are shown in Figure 5 for both types of impedance condition. It can be seen from these figures that the achievable control as well as how hard the loudspeakers have to be driven is strongly dependent on the control source/primary source separation.

For the case under consideration, the duct temperature varies from 100°C to 180°C and for a flow speed of 20m/s, this corresponds to a wavelength variation from 2.47m to 2.72m. To cover all possibilities it is clear that three speakers in each duct section separated axially by about 0.45m will be needed. However, in the trial installation, two speakers on opposite sides of each duct section and separated axially by 0.55m were used.

Control source equipment

The control signals from the electronic controller are supplied by way of power amplifiers to the control loudspeakers to generate the cancelling acoustic signals in the three sections of exhaust stack. The loudspeaker cones were sprayed with lacquer to prevent deterioration in the moist environment which existed. The loudspeakers were rated at 250W and were mid-range JBL type 2123H, 10-inch speakers with a sensitivity of 101dB SPL 1W, 1m.

The enclosures housing the loudspeakers have provision for cooling air flow through the backing cavity and purging air flow in the front of the loudspeaker cone and then into the duct to keep the speaker as clean as possible. The original enclosure design (with no airflow in the speaker backing enclosure) is illustrated in Figure 6. The filter shown in the figure has since been removed as it was soon clogged with wet dust and reduced the acoustic efficiency of the loudspeaker. For similar reasons, the mylar seal and seal screens were replaced with kevlar reinforced mylar.

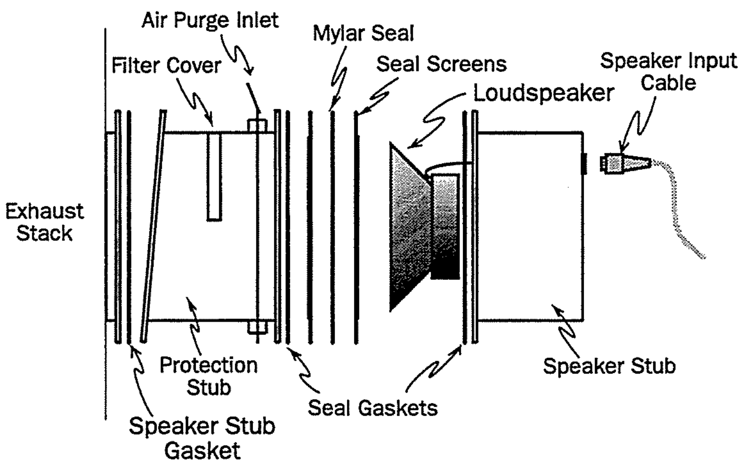

Figure 6: Loudspeaker enclosure configuration.

Very early in the trials, loudspeakers were failing on a regular basis, even though they were not being driven at more than half their maximum rating. The loudspeaker enclosures are heated by conduction and radiation from the duct walls and by convection from the hot air in the duct. Also, the power injected into the loudspeaker coil results in a significant heat load in the loudspeaker backing enclosure. When the plant operating condition is such that the duct temperature rises to 180°C, the diaphragm which supports the loudspeaker cone becomes soft and easily distorts. This has two effects. First, when the cone distorts, the voice coil attached to the cone rubs on the loudspeaker magnet. The rubbing removes the insulation on the coil and then an electrical "short circuit" occurs which destroys the voice coil. The rubbing also occurs when no control signal is applied to the loudspeaker because the high noise levels in the duct result in a significant movement of the loudspeaker cone. This voice coil failure occurred with several loudspeakers and caused significant delays in the progress of the project. Second, the when the diaphragm is soft, the loudspeaker cone is pulled into the exhaust stack by the suction effect generated by the large air flow up the stack. When the air temperature returns to normal operating temperature, the diaphragm is permanently displaced towards the end of the cone's traverse. This damage reduces the efficiency of the loudspeaker in converting electrical power into sound power. Thus the original design of the loudspeaker enclosure shown in Figure 6 was modified to include cooling air flow through the backing cavity. However it is anticipated that a chilled water jacket will be needed around the loudspeaker enclosures and loudspeaker driving magnets if the loudspeakers are to have a reasonable life.

Error sensor location

The error sensor locations were governed by the following constraints:

- At least one was necessary for each duct section.
- They had to be as far away from the control sources as possible to minimise the effect of the near field on the overall control system performance.
- They had to be well below the top of the splitters (500mm) to minimise contamination from sound propagating in adjacent duct sections.
- They should not be near a node in the standing wave in the duct section caused by reflection from the end of the splitter. The 500mm criterion also satisfied this criterion.

Error signal equipment

Error microphones were mounted in sets of three in a microphone stub as shown in Figure 7. The stub includes a filter cover to protect the microphone from airborne contaminants, a microphone holder which holds three microphones and an air-line coupling which forces cooling air over the microphones.

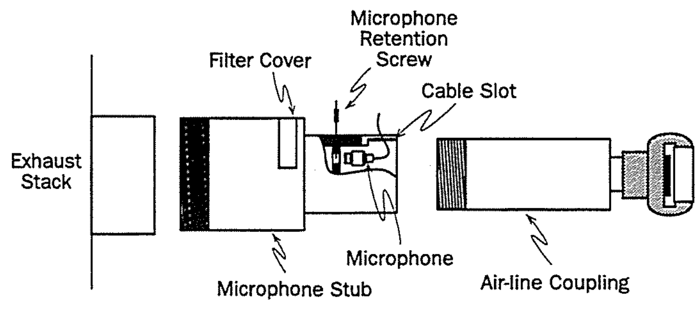

Figure 7: Microphone stub assembly.

The three microphones in each stub are connected to a microphone preamplifier and a summation circuit mounted in the ANC System Control Box, where the signals are combined into a single 'error signal' for each microphone stub. In this way, the error sensing system is triple-redundant. The ANC system will still function upon failure of up to two of the microphones in each stub.

A typical frequency spectrum of the noise level measured by an error microphone in the duct section closest to the control box is shown in Figure 8. It can be seen that the noise level at the BPF (165Hz) is approximately 120dB (linear). However this signal varies from 116dB up to 130dB, depending on the location of the microphone along the duct axis.

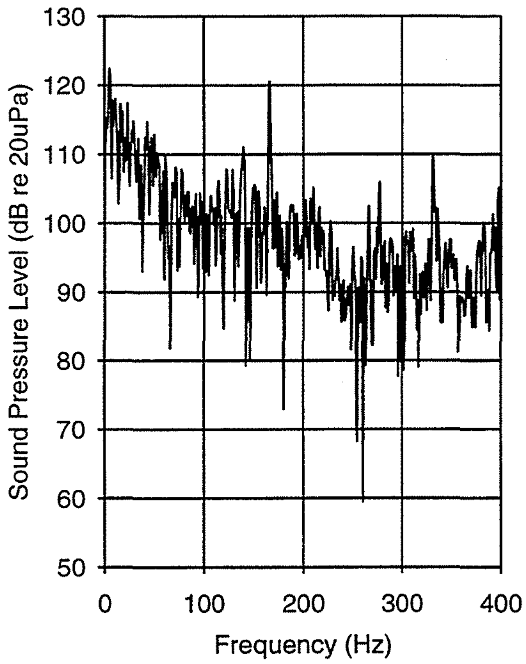

Figure 8: Sound pressure level in the exhaust stack, measured in the duct section closest to the control box.

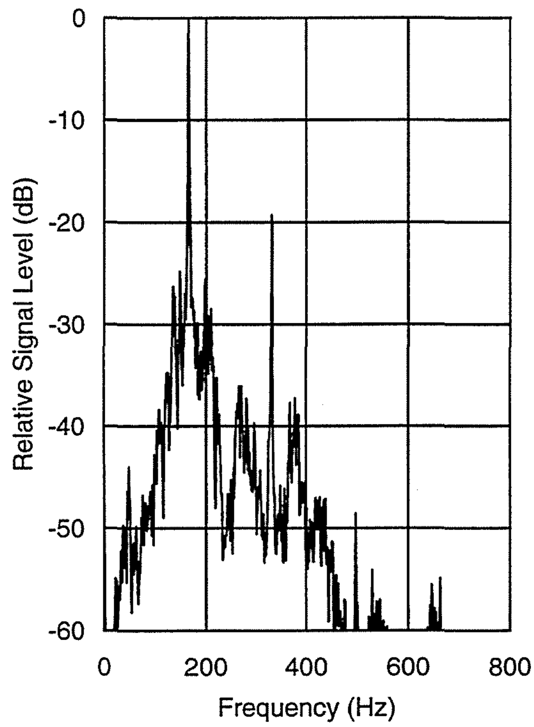

Figure 9: Typical filtered error signal frequency spectrum.

The electrical signal from the microphone passes through an analog band pass filter to reduce the signal levels at frequencies other than at the BPF. The reason for filtering the error microphone signal is to provide the digital controller with a measure of the noise at the BPF and to disregard the contributions of noise at other frequencies. A typical frequency spectrum of the filtered error signal is shown in Figure 9. The peak in the spectrum at the blade pass frequency (165Hz) is clearly evident and the second harmonic (330Hz) is nearly 20dB less. The amount by which the tonal peak exceeds the background noise varies by about ±5dB and the background noise, including that due to the air cooling, is always at least 25dB below the peak, thus providing a suitable error signal to the control system. The delay through the error sensor filter has no measurable effect on the performance of the control system.

The reliability of the microphones has proved to be excellent. At present, the cooling air which is used to reduce the temperature of the microphones has been obtained from the plant compressed air line. Oil in the compressed air line has entirely covered all the microphones and they are all still functioning after several months of use.

Reference signal considerations

There were two choices for the reference signal; a tachometer on the fan shaft or reference microphone upstream of the control sources. The advantage of the tachometer is that it is relatively straightforward to implement and is much more reliable. The disadvantage is that only frequencies corresponding to the fan blade pass frequency and its harmonics will be derived for control by the electronic controller. Although the noise causing the problem did appear to be tonal in nature, it did not appear as a sharp spectral peak, probably as a result of slight speed variations of the fan. However if the lack of sharpness of the peak were a result of noise being generated by an instability phenomenon, then the signal from the tachometer would be an unsuitable reference signal.

The advantage of using a microphone reference signal is that the noise generating mechanism is unimportant and the dominant part of the spectrum will be controlled regardless. The disadvantage is that the microphone signal will be contaminated with fluid pressure fluctuations which propagate at the speed of flow and not the speed of sound. Also any filtering of the reference signal to remove the unwanted signals is likely to result in unacceptable delays through the filter with the result that the controller is unlikely to receive the reference signal in time to generate the required control signal.

It was felt that on the balance of probabilities, it was prudent to use a tachometer reference signal, especially considering the high quality of signal that could be obtained.

Reference signal equipment

The tachometer system is used to provide a reference signal to the digital controller is illustrated in Figure 10. It consists of a digital inductive pickup mounted close to the notched shaft encoder disk. The notched encoder disc has the same number of evenly spaced notches as there are blades on the fan.

The pickup head thus supplies a square wave signal of frequency equal to the blade pass frequency to the tachometer amplifier in the ANC System Control Box which incorporates a power supply and signal conditioner. The digital signal from the pickup is converted into a sinusoidal reference signal at the same frequency as the noise produced by the fan, by filtering out all multiples of the BPF with a low pass filter.

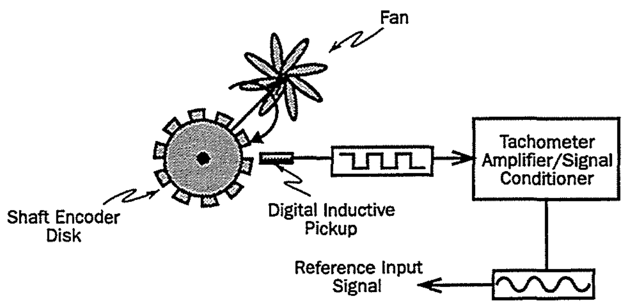

Figure 10: Reference signal equipment layout.

A typical frequency spectrum of the filtered tachometer signal is shown in Figure 11. The BPF can be seen clearly at 165Hz. The first harmonic and side bands are approximately 35dB lower; thus resulting in a high quality reference signal being provided to the digital controller.

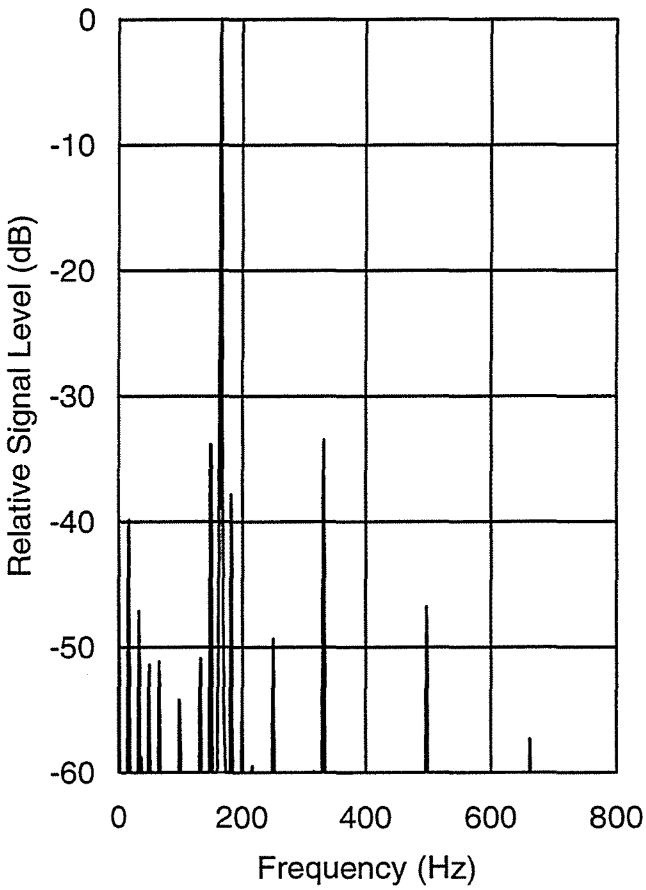

Figure 11: Reference signal frequency spectrum.

ELECTRONIC CONTROLLER

The electronic control system used to process the incoming tachometer and microphone signals was the Causal System's EZ-ANC. The control algorithm used was the standard "filtered-u" operating on an IIR filter with 30 forward taps and 20 backward taps. An IIR filter was found to be more stable than an FIR filter, probably because of the presence of axial resonances in the duct.

It was necessary to measure the transfer function between the output to the loudspeakers and the input from the error microphones on-line on a continuous basis to maintain algorithm stability. The algorithm was used to adapt the weights of an FIR filter which simulated this transfer function and which was easily incorporated in the control algorithm. It was found that best results were obtained when the controller was configured as three 2-channel systems, as shown in Figure 12. Each system had one error signal input and two control outputs and operated on one of the three duct sections. A significant amount of leakage was used in the control algorithm to even out the driving signals to the two loudspeakers in each duct section. This was necessary to prevent the loudspeaker in the poorer location in the duct (from the control viewpoint) from being over driven.

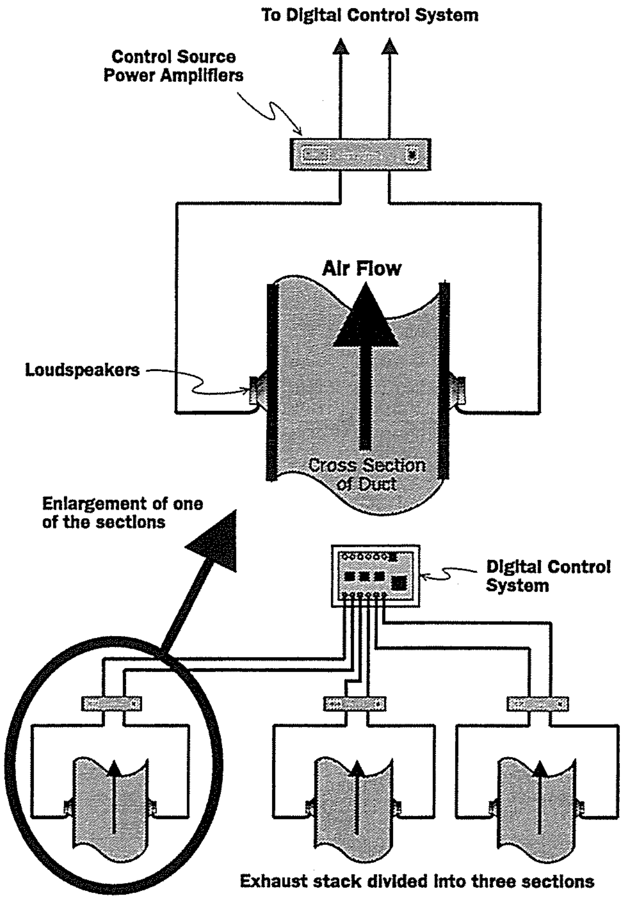

Figure 12: Control system configuration.

An important practical point which must be observed to prevent overdriving the loudspeakers as a result of unstable operation of the controller, is to make the reference signal voltage equal to the maximum allowed voltage into the power amplifiers driving the loudspeakers. The digital controller converts the reference signal voltage into a number between ±32768. This number determines the maximum voltage of the control signal.

For example if the digitised reference signal has a number of 10000, then the maximum voltage of the control signal is potentially three times the reference signal voltage. To prevent the power amplifiers from clipping or overdriving the loudspeakers, the value of the digitised reference signal voltage should be close to 32600.

To prevent the digitised error signal voltage becoming too large, which results in the digital controller becoming unstable, the value of the input gain for the error signal is adjusted so that the error signal is at approximately half of the maximum allowed value to allow for possible fluctuations.

ACTIVE NOISE CONTROL TRIALS

The ANC system described in the preceding sections has been used to reduce the noise levels in the exhaust stack. At this stage of the project, the trials have demonstrated the effectiveness of an ANC solution to the noise problem.

The active noise control trials involved using the system described in the preceding sections, to simultaneously reduce the noise levels in each of the three sections of the exhaust stack. Figures 13 to 15 show the spectrum of the filtered error microphone voltage in each of the three sections in the exhaust stack, with no active control and when active control is used. The duct section numbers are identified in figure 1, where duct section 1 corresponds to loudspeakers 1 and 2, duct section 2 corresponds to loudspeakers 3 and 4 and duct section 3 corresponds to loudspeakers 5 and 6.

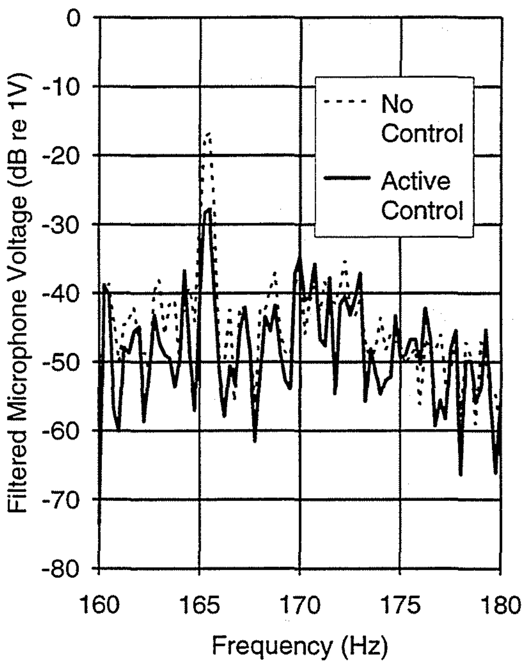

Figure 13: Filtered error microphone voltage in duct closest to control box (duct section 1).

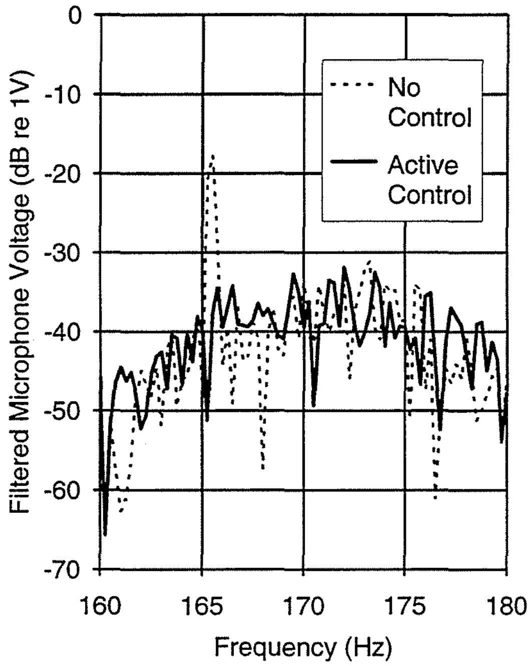

Figure 14: Filtered error microphone voltage in the middle section (duct section 2).

The differences between the two spectra shown in each figure are directly comparable to the expected reductions in the in-duct sound pressure levels at the BPF. Figure 13 shows the poorest result of all three sections. Only 10dB reduction was possible at the error microphone. Figure 13 and 14 show noise level reductions of about 20 dB. The reason for the poor result in duct section 1 is due to the non-optimum placement of the loudspeaker enclosures in the axial direction along the duct.

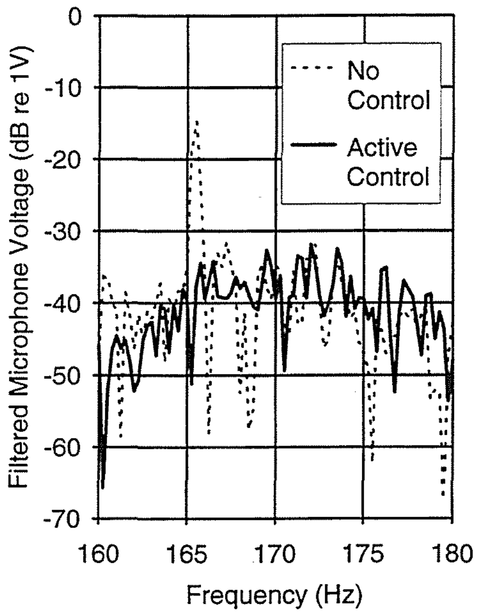

Figure 15: Filtered error microphone voltage in duct which is furthest from the control box (duct section 3).

Table 1: Loudspeaker voltage across terminals during active control trials.

Speaker number	Duct section number	Voltage (Vrms)	Power (Watts)
1	1	33	140
2	1	26	85
3	2	29	105
4	2	23	65
5	3	33	140
6	3	21	55

The required loudspeaker outputs to achieve the control shown in the figures are listed in table 1.

The loudspeakers are numbered as shown in figure 1 with the odd numbered loudspeakers located inside the liquid starter room (one in each of the three sections) and with the even numbered speakers outside on the opposite side of the duct (with number 2 opposite number 1, etc).

SYSTEM ADDITIONS

The following two extensions to the current system are desirable to provide remote control and monitoring of the ANC system performance.

Time averaged Sound Pressure Levels in the stack. Filtered outputs from the microphones can be rectified then low pass filtered to provide calibrated DC levels (5-20mA) which can be integrated with the current plant control system. This will enable the continual observation and tracking of the sound levels in the stack. Such information could be correlated with other process information to identify potential process conditions where noise emitted by the stack is at a minimum. In addition, in the unlikely event that the system becomes unstable, the operator could be alerted.

A remote reset for the EZ-ANC control system. This would allow remote reset of the control system from the process control room should the controller become unstable.

CONCLUSIONS

An active noise control system has been shown to be effective in attenuating the noise generated by a large fan at the blade pass frequency and radiated by an 80m high exhaust stack, in spite of the harsh environmental conditions. In-duct reductions of up to 20dB were obtained after special treatment of the loudspeakers and microphones. The installation of axial splitters in the duct to cut out cross modes for the active noise control system had the added benefit of providing some passive cancellation due to the propagation path in each of the three sections being one third of a wavelength different to that in adjacent sections.

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TRANSFER FUNCTIONS OF THE VOCAL TRACT CAN PROVIDE REAL TIME FEEDBACK FOR THE PRONUNCIATION OF VOWELS

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ABSTRACT

The first two resonances of the vocal tract are measured non-invasively at the mouth using a real time impedance spectrometer. These provide visual feedback to adult human subjects attempting to pronounce unknown vowel sounds. The success rates are compared with those achieved using auditory feedback, i.e. listening to and imitating sounds. Subjects who had had a maximum of two hours training using this visual feedback about their vocal tract performed better than those using auditory feedback alone. The new method therefore holds promise as a technique in speech therapy and language laboratories.

INTRODUCTION

Children learn to pronounce speech sounds using auditory feedback: they hear others speak, and then imitate certain features in the sounds they hear. People with severe hearing difficulties cannot use auditory feedback. Speech reading or lip-reading is a partial substitute, but while it provides visual feedback about the lips, it offers little information about the position of the tongue and the palate. Another case where auditory feedback is not completely adequate is in foreign language learning by adults. Most adults fail to acquire authentic accents using auditory feedback because of the process of categorisation: they hear a foreign phoneme, interpret it as a similar phoneme in their maternal language, and then reproduce the phoneme of their own language (Landercy and Renard, 1977). Feedback in which the relevant information is presented visually should overcome these difficulties to some extent.

We have recently developed a technique which measures the acoustic impedance and other transfer functions of the vocal tract in real time (Wolfe et al, 1994, 95; Dowd et al, 1996). An acoustic current comprising a large number of discrete frequency components is produced at the subject's mouth, and the pressure signal measured with a microphone determines the acoustic response of the tract, in parallel with the external field. In the work reported here, this technique was used to supply visual feedback about the first two resonances of the vocal tract to teach volunteers to pronounce a range of vowel sounds (hereinafter called vocal tract feedback). We have limited the study to vowel sounds because they may be sustained and thus allow subjects time to use the feedback to converge on a target. Vowels are also a class of phonemes which are strongly subject to categorisation and therefore might benefit most obviously from non-auditory feedback.

The broad resonances of the vocal tract are often called formants in acoustical phonetics. "Formant" is also used to describe a broad frequency band of emitted power. Because our technique allows the two quantities to be measured independently, we are careful to distinguish the two. We reserve "formant" for the broad frequency band of emitted power and the terms F1 and F2 for the first two formants. We use R1 and R2 to represent the frequencies of the first two resonances of the vocal tract. This study used and reports only data about the first two resonances, which are the most important in pronunciation of vowels, and which are approximately monotonically related to the extent of opening of the mouth and the position (forward or back) of the tongue.

We have previously used this technique to teach a group of volunteer subjects to pronounce vowel sounds using vocal tract feedback (Dowd et al, 1996). Vocal tract feedback was compared with two other feedback methods. In one of these, subjects were shown photographs of the mouth of the target speaker pronouncing the target phoneme. This method, which we call visual feedback, was a model for speech reading or "lip reading". In the other method which we call auditory feedback, subjects listened to a recording of the target sound and attempted to match it. Success was measured both by the fidelity of reproduction of the vocal tract resonances, and by recognition of recordings of the speech sound by members of a listening panel. After two hours training, subjects using a combination of vocal tract feedback and visual feedback performed as well as (judged by recognition) or better than (judged by resonance fidelity) they did using auditory feedback, even though they had been using auditory feedback since early childhood.

In that previous study, we used vowel sounds from Australian English as the target sounds, and most of the subjects spoke that language and therefore tended to perceive and to produce vowels into the categories of that language. In the work reported here we used vowels from the French language as targets for the following reasons: i) French has a relatively large number of well defined vowels (three or four nasalized and ten to twelve non-nasalized vowels, Rey and Rey-Debove, 1985); ii) it divides the (F1, F2) plane in a way different from English; and ii) we were able to find volunteer native speakers for both the target group and a listening panel.

To keep measurement sessions relatively short, we chose to study a subset of French vowels: /e/, /ɛ/, /a/, /ɑ/, /u/ and /y/. These form pairs which are sometimes confused by anglophones: /e/ and /ɛ/ are acoustically similar and are not always distinguished by native speakers; /a/ and /ɑ/ are less similar but are also sometimes confused; /u/ and /y/ are very different acoustically but are nevertheless often confused by English speakers.

MATERIALS AND METHODS

Acoustic measurements.

Details of the impedance spectrometer and its calibration procedure are given by Wolfe et al (1994, 1995). A waveform comprising hundreds of discrete frequency components is synthesized and input to a sealed loudspeaker which is matched via an exponential horn to an acoustic resistor (Fig 1). During calibration the acoustic signal is input to a reference load and the spectrum of a microphone signal is calculated (a). This spectrum includes the frequency dependent gain of the amplifier, speaker, horn, microphone and load. A new waveform is synthesized with components inversely proportional to those of the measured spectrum, and is input to the amplifier (b). For a completely linear system, this would produce an output spectrum which was independent of frequency. In the presence of non-linearities, the procedure may be repeated to obtain a frequency-independent spectrum in the reference load. This acoustic signal is then applied to an experimental load whose impedance has a much lower value impedance than the output resistance, and therefore does not affect the acoustic current. The pressure spectrum measured here is thus the ratio of the experimental impedance to that of the reference. The amplitude spectrum is calculated with a digital signal processor and displayed in real time (c).

Previous studies using this apparatus have used a resistive load as the reference so that the measured pressure spectrum gives directly the acoustic impedance of the system studied. Here we use the laboratory field, measured at the subject's lower lip with the mouth closed. There are two advantages. The impedance ratio in this case shows the effect of opening the mouth and thus putting the vocal tract in parallel with the external field. Further, the free field impedance increases with frequency, and so the acoustic current must be larger at low frequencies to give a frequency independent pressure spectrum. The larger currents at low frequencies improve the signal to noise ratio in the range where there is greatest background noise. The frequency range 200 to 2500 Hz was used. This comfortably covers the range of the first two resonances of the vocal tract. The frequency resolution was 25 Hz, which is the result of a compromise between resolution and signal to noise ratio.

Target speakers.

Eight female adult native speakers of French were studied to provide the target data. The choice of sex was not ours: only one male speaker responded to our invitation. His vocal tract data were recorded for comparison, but were not included in the rest of the study because of the systematic difference between the vocal tracts of males and females. The target speakers were first taught how to raise the velum (soft palate) without phonating, initially using a mirror as feedback, and then using the measured spectrum. The current source and

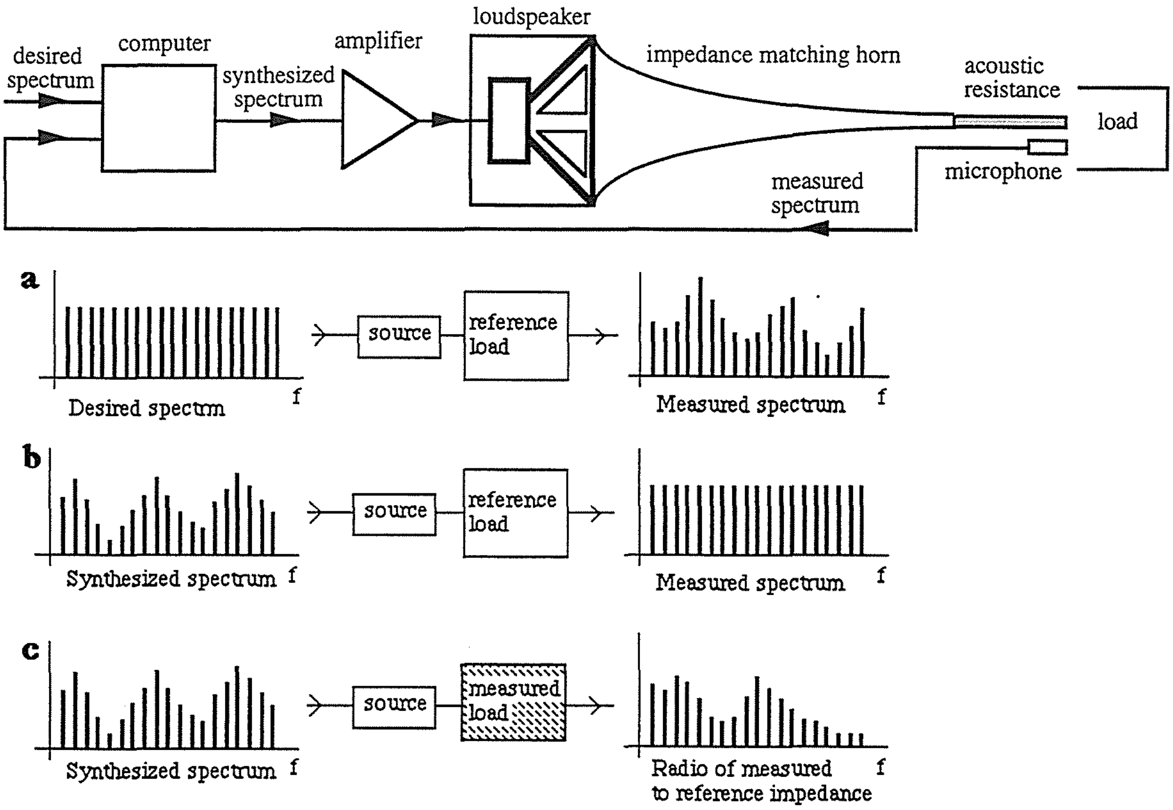

Figure 1 shows the impedance spectrometer used here and the calibration procedure. The reference load was the free field of the laboratory measured at the subject's lower lip with the mouth closed. The measured load was the same geometry but with the mouth open and the vocal tract therefore in parallel with the free field. (Reproduced from Acoustics Australia, with permission.) This technique is the subject of patent application PCT/AU95/00729.

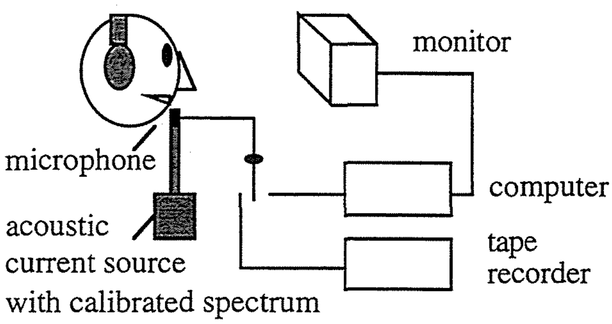

Figure 2. The acoustic current source and the microphone are positioned at the subject's lower lip. The feedback about the target phoneme may be auditory feedback (sound from headphones), or vocal tract feedback, consisting of target frequency values superimposed on his/her own impedance spectrum.

microphone were placed against the lower lip. Each target speaker was then asked to pronounce the following vowels which were presented in written and in the context of a French word: /i/ as in "pie"; /e/ as in "thé"; /ɛ/ as in "paix"; /a/ as in "patte"; /ɑ/ as in "pâte"; /ɔ/ as in "pote"; /o/ as in "pot"; /u/ as in "poux"; /y/ as in "pu"; /ø/ as in "peu"; /œ/ as in "peur"; /ə/ as in "te"; /ɛ̃/ as in "pin"; /ɔ̃/ as in "pont"; /ɑ̃/ as in "pan"; /œ̃/ as in "un". The spoken sounds were recorded on cassette tape. Then they were asked to pronounce the sound, to hold the vocal tract in that position while not phonating and then to pronounce the vowel again. While they were not phonating, the spectrometer measured the response of the vocal tract as described above. The second phonation was used to check that the position had not changed.

For each speaker, R1 and R2 were measured for each vowel and the mean values $\overline{R1}$ and $\overline{R2}$ calculated. We define d, a deviation weighted displacement in the resonance plane:

$$d = \sqrt{\left(\frac{R1 - \overline{R1}}{\sigma_1}\right)^2 + \left(\frac{R2 - \overline{R2}}{\sigma_2}\right)^2} \qquad (1)$$

where σ is the standard deviation over all speakers and all vowels. For each of the six vowels studied, we chose a target recording which was closest to ($\overline{R1}$, $\overline{R2}$), i.e. the recording of that vowel by the speaker whose (R1, R2) for that vowel had the smallest d.

Eleven female Australian native speakers of English acted as subjects. None of them spoke French. They were tested in two individual sessions of about one hour each. On their first visit, members of one group of five were taught to raise the velum and to use vocal tract feedback to imitate the six chosen target vowels. The other group of six subjects, on their first visit, were given auditory feedback only and used it to imitate the target vowel sounds. On their second visit, both groups were given both types of feedback. All subjects were told that the study aimed to test methods of feedback in teaching the pronunciation of vowel sounds. They were not told that the sounds were from a foreign language.

To use the vocal tract feedback, subjects were taught to raise the velum while not phonating and measurements were made in the same way as for target speakers. The impedance ratio spectrum was shown on a computer monitor, over which were superimposed two vertical lines at the target frequencies R1 and R2 for that vowel. They were asked to match the peaks in their spectra to the target values, and were told that opening and closing the mouth primarily affected R1 and that moving the tongue backwards and forwards primarily affected R2. They were asked to practise until they could match the resonances of the first target vowel. This took up to 15 minutes. When they were satisfied with the match, they threw a switch which simultaneously turned off the acoustic current and started a tape recorder using the same microphone, and then vocalised. This was done three times for each vowel. The next target was presented, and the procedure was repeated for each of the target vowels. Subjects were able to match second and subsequent vowels in less time than the first, although the vowel /y/, which is most different from the vowels of English, took several minutes.

To use auditory feedback, subjects listened to vowel sounds via headphones. A target sound was played to them, and they were asked to imitate the sound. They could hear their own sound as they attempted to match the target. When they were satisfied they could not improve the match with the target, their imitation of the target sound was recorded. In each case the subjects pronounced each vowel three times. For the third repetition, a spectrometer measurement of the vocal tract was made just as the velum was involuntarily lifted to begin to speak.

Sixteen adult native speakers of French acted as the listening panel. Each listened to a recording lasting 30 minutes of the the sounds of the target speakers pronouncing all 12 non-nasalized vowels and of the various imitations of the target vowels. The order was randomised, and each sound was identified by a number and played three times consecutively. They were given a printed form containing instructions and 214 numbered lines for replies. Each line had the twelve words *pie, thé, paix, patte, pâte, pote, pot, poux, pu, peu, peur* and *te*, and the instructions asked the panel members to circle the word whose vowel was closest to the sound on the tape.

RESULTS AND DISCUSSION

Imitation of the targets.

Figure 3 shows the resonant frequencies of the vocal tracts of subjects imitating the six target vowels. (a) shows the measurements taken during the impedance training session, i.e. measurements made by subjects using vocal tract feedback only, without hearing the vowels they were attempting to imitate. (b) are the results for the group using auditory feedback only. (c) and (d) are measurements made using both auditory and vocal tract feedback. The subjects whose results are shown in (c) are the members of the (b) group in their second visit. They used both types of feedback, but had had no previous training with vocal tract feedback (hereafter called the c treatment). The group whose results are shown in (d) is the group that had had one previous training with vocal tract feedback (hereafter called the d treatment).

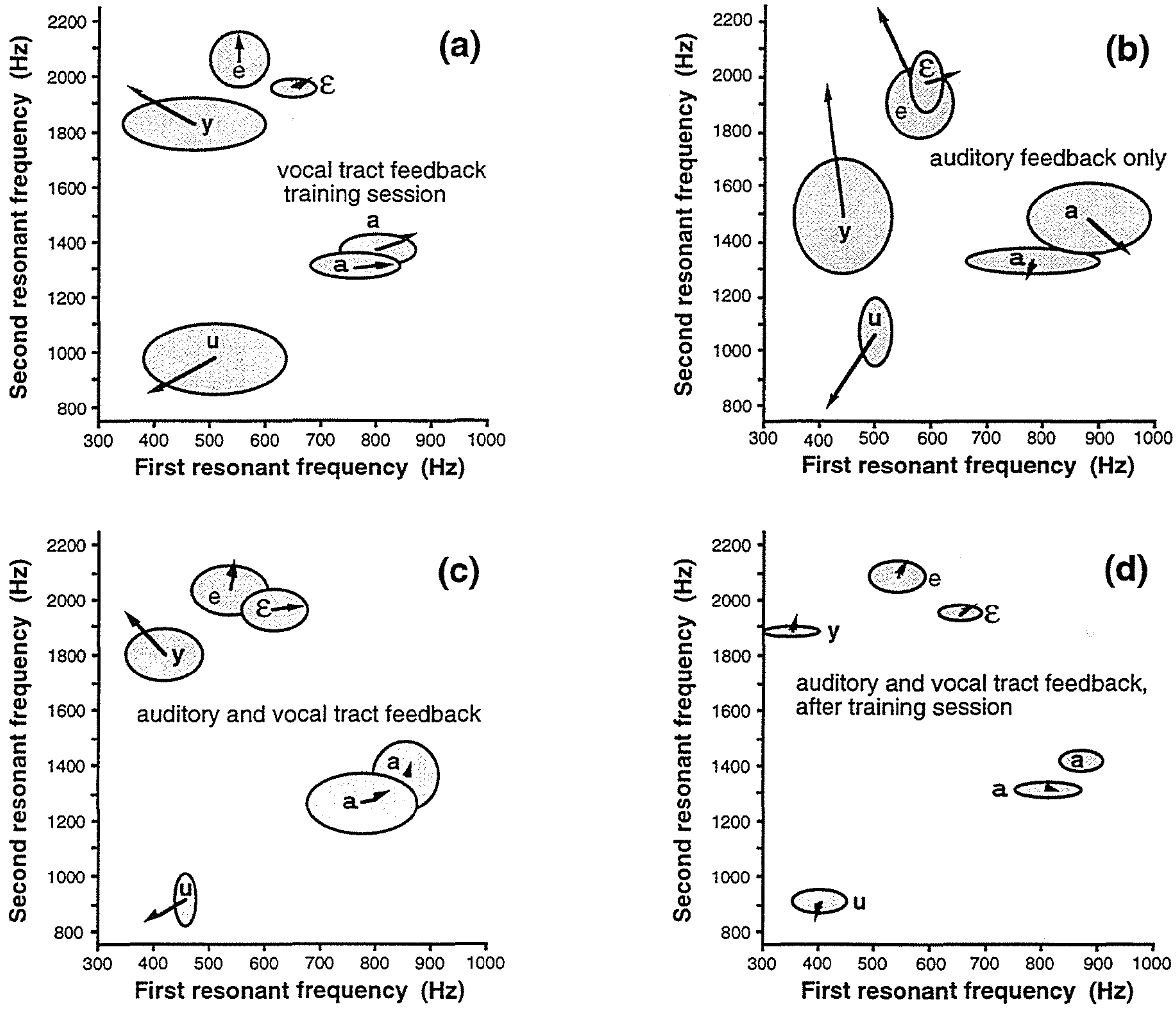

Figure 3. The frequencies (R1,R2) of the resonances of the vocal tracts of the target speakers and the subjects imitating the phonemes. The values for the target speakers are given by the heads of the arrows. The tails of the arrows are the means of R1 and R2 for the subjects. Thus short arrows represent a good match. The semi-axes of the ellipses are the standard deviations R1 and R2 for the subjects. The measurements (a) were made on subjects at their training session using vocal tract feedback: these subjects did not hear the sounds they were attempting to imitate. Subjects (b) used auditory feedback only, whereas subjects (c) and (d) used both auditory and vocal tract feedback. The (c) group had no previous training with vocal tract feedback, while the (d) group had had one previous training session of about one hour.

It is clear from these figures that the vocal tract feedback improves the match with the target resonance values (R1, R2). The lengths of the arrows are longest for the auditory feedback only, and shortest for the d group who had the most training using vocal tract feedback. There is improvement in the distinction of the pair /e/ and /ɛ/ and the pair /a/ and /ɑ/. These pairs are sufficiently similar that not all native speakers distinguish them reliably. The pair /u/ and /y/ provide an interesting case, which is well known to those teaching French or German (both of which languages use these vowels) to native speakers of English (which does not distinguish them). The vowels are acoustically very different - see the separation of the arrowheads, indicating very different tongue positions by the target speakers. Using auditory feedback alone, the English speakers do not separate these very far but with increasing use of vocal tract feedback they separate them further and improve the match substantially.

Note that, for the group with no vocal tract feedback, most of the vowels (tails of arrows) are displaced towards the middle of the resonance plane, and show a small range of R1. Australian speakers of English are said to move their lips little in speaking. The groups which used vocal tract feedback, especially those who had a training session with it, were more able to reproduce (R1,R2) of the target vowels. As these resonance frequencies are closely related to mouth opening and tongue positioning, the simplest explanation is that vocal tract feedback has taught the subjects, relatively quickly, how to articulate the target vowels with considerable accuracy.

On the one hand it may seem surprising that the subjects could learn to use the novel method of feedback well in only two hours, whereas they had been using auditory feedback for most of their lives. On the other hand, the quantities displayed in Figure 3 are exactly those which were minimised using vocal tract feedback, and they are not the only parameters involved in pronunciation. The proof of the vowel is in the hearing, and so we turn to the listing panel for their verdicts.

Listening panel results

Table 1 shows the percentages of correct identifications by the listening panel for each vowel for each treatment. Note that there is not perfect identification of the vowels spoken by the target speakers: /e/ was mainly confused with /ɛ/ and /a/ mainly with /ɑ/. /u/ and /y/ were confused with a range of other vowels, but never with each other. The average identification score for the target speakers was 81%. This may seem low, given that native speakers can usually identify words spoken by other native speakers with higher accuracy. Spoken language has relatively high redundancy (Fletcher, 1992), and the contextual information (both from the word containing the vowel and the sentence containing the word) is missing here. For example, a French speaker will not confuse e and ɛ in the context /ʒietɛ/ *vs* /ʒiete/, because *j'y étais* is grammatical and *j'y été* is not.

Percentage correctly identified by listening panel

vowel	target	auditory feedback only	both types of feedback	both types of feedback with training
e	94	20	27	40
ɛ	75	67	71	64
a	50	48	52	59
ɑ	93	69	64	75
u	88	31	45	55
y	87	2	20	79
average	80 ± 17	38 ± 36%	46 ± 29%	62 ± 28%
(number)	(6)	(36)	(36)	(30)

Table 1. The first column is the percentage of correct recognition of the vowels spoken by the target speakers themselves, the other columns are for treatments b, c and d described above.

Taking the average over all six target vowels, the target results are significantly higher than those of all other groups except the d group (using both types of feedback with training) with 95% confidence. The d group's results are significantly higher than those of both the b group (auditory feedback only) or those of the c group (both types of feedback, no training). The effects are rather different for the different vowels. For the vowels /ɛ/, /a/ and /ɑ/ there is little or no improvement using vocal tract feedback. These are the vowels for which auditory feedback alone gave its best scores, and two of them, /a/ and /ɛ/, are the vowels which were least well distinguished in the pronunciation of the target speakers. It is difficult for the subjects to do as well as, let alone better than the target speakers. For the vowels /e/, /u/ and /y/ the scores improved with the use of vocal tract feedback and with training in using it, especially in the vowel /y/ which is relatively unfamiliar to anglophones. Native speakers of English often find it difficult to distinguish /u/ and /y/ in both pronunciation and recognition. The subjects using auditory feedback only scored less than random in producing /y/: the listening panel identified their attempts as /u/, /ø/ or another vowel. For subjects using vocal tract feedback with training, the imitations of /u/ and /y/ were never confused with each other.

CONCLUSION

A sample of native speakers of English improved the articulation and recognition of a set of French vowels with a limited use of vocal tract feedback. This feedback requires novel mouth-eye and tongue-eye coordination and so one might expect continued improvement with further use. One might also expect improved results from a technological implementation of the vocal tract feedback technique which was easier to learn and to use. We are currently developing such a system.

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THE PROFESSIONAL CONFESSIONAL: TOWARDS A MORE PROFESSIONAL CULTURE FOR ACOUSTICS IN AUSTRALIA

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ABSTRACT

Some suggestions are made for improving the usefulness and professional standing of the Australian Acoustical Society and its members. These suggestions are based on personal experience and involvement in other professional societies, universities, conferences, consulting and research. It is concluded that one potentially useful way of creating a stronger profession is to involve its members more in the functioning of the society and to strengthen the educational function of it.

There should also be a culture of learning from mistakes and an attempt should be made to publicise these so that they are not repeated by others. Catastrophes are a basis for advancement (at least that was a popular 19th century idea) and although there are rarely catastrophic failures in acoustics there are still very significant ones; for example the Third Runway at Sydney Airport. This paper explores the possibilities and limitations of disclosing mistakes, misinterpretations, errors, fabrications and falsifications.

As there are strong parallels between professional and learned societies and religions and so it is also suggested that religious practices may be appropriate for maintaining and improving ethical standards within the society. The practice of "whistle-blowing" is also raised.

INTRODUCTION

I am concerned about a number of aspects of the acoustics profession in Australia. I see very little cooperation in the profession to exchange information. Papers at conferences are rarely given by practitioners and when they are the papers contain very limited information about "successful" projects: there is rarely objective analysis of successes or failures. There is little interest in improving standards of professional practice (It is common in other professions, for instance, that members have to participate in continuing professional development programs.). There is no united approach to issues of concern to members of the profession and therefore little notice is taken of the profession (For example, some of the acoustic provisions of the Building Code of Australia are unworkable in a number of respects and inadequate in others and yet the Society, as far as I am aware, has not made any representations to change the Code. A similar comment could be made about environmental legislation and helicopter and road traffic noise criteria.). There appears to be very little in the way of technical meetings, tutorials, workshops, working groups, publications (other than Acoustics Australia), review papers and other professional activities (The Acoustical Society of America combines these activities with its conferences and it would seem appropriate that the Australian Acoustical Society consider undertaking similar activities at the time of its annual conferences.). There has been little or no attempt to establish ethical standards since the Society adopted its Code of Ethics. There is no professional degree in acoustics in Australia and there appears to be no attempt to coordinate universities to provide such a program.

In this paper I would like to explore some of these issues using, as a basis, some recent experiences. The main experience which I think is worth documenting is a consulting one. The others come from experience as a member of other professional organisations and as an academic responsible for setting up an audio engineering degree at Sydney University. In doing this I would like to state clearly that I am not being critical of past or present committees of the society. Rather, I am putting a personal view of how the Society could be strengthened for the future and put on a more solid professional footing, for the benefit of members and Australian society in general.

A PROFESSIONAL QUALIFICATION IN ACOUSTICS

If Acoustics is to survive as a respected profession in Australia there is an need for a recognised degree here or overseas using distance learning. In the time since Anita Lawrence started the Master of Science (Acoustics) at the University of NSW the universities have changed significantly and it seems as though there are going to be even greater changes in the near future. There are several possible directions for higher education. There is likely to be a far greater emphasis on distance learning. If this happens then one possible reason for the demise of MSc(Acoustics) at UNSW, the difficulty of getting students to move to Sydney, would be overcome. Another likely development will be the cooperation between universities to present programs. (Senator Vanstone said on 9 August 1996, in presenting the government's policy on funding universities, that "We will welcome and encourage moves within and between institutions to rationalise staffing courses and infrastructure where that is educationally sensible and leads to long-term efficiencies.") This is already happening and the concept of doing a degree in one department or one university has been changing for several years and with it the university regulations stating that more than half of a program must be completed at one university. The main emphasis however has been on "articulation" (the ability to change from one program to another and to another or continue from one level of education to another, and one institution and another, in a logical and orderly way without having to repeat subjects or have gaps in one's education). It is not just articulation which is important, it is the cooperation between institutions to present programs. In acoustics this is unlikely to happen without the Society initiating, facilitating and intervening.

Probably the most important change will be in the post-graduate coursework programs. It seems almost certain that these programs will have to be self-funding and so the chances of setting up a new program, like the previous MSc(Acoustics) at UNSW, are slim unless employers are prepared to pay (Graduates will now be paying back their HECS contributions from the time that they get their first job and it is doubtful whether many will be able to afford to pay $4000 or $5000 a year for part-time study.). The answer seems to lie in undergraduate programs.

Until recently it has not been easy to establish specialist undergraduate programs and so specialist postgraduate degrees were established. Now the concept of "boutique" or "adjectival" degrees has been introduced at Sydney University in the Faculty of Science. There will be basic common subjects such as maths and physics and, in addition, subjects in the speciality. Acoustics and audio could be such specialities using existing courses at Sydney University or other universities. Such an arrangement is already being made between the University of Technology, Sydney and the University of Sydney in a B.Tech with specialisation in Building Services. A BSc(Audio) is being considered by the Faculty of Science at Sydney University. These two programs are drawing from a relatively large student group. A B.Tech(Acoustics) or a BSc(Acoustics) would not have the same attraction and another approach is necessary.

There are a number of universities which teach certain aspects of acoustics. Some universities, such as Sydney and ADFA, present short courses. It would seem to be within the bounds of possibility that, with distance learning and short courses, students throughout Australia could gain enough units to obtain a specialist degree in acoustics, if the universities cooperated. If each university department in Australia which had an involvement in acoustics could offer just one subject, in distance learning mode, which was articulated with other courses on offer from other universities, there could be more than adequate material for an Australian Masters of Acoustics program and the program could be presented relatively cheaply. I suggest that the Society should push for this cooperation and for the abolition of requirements by universities that at least 50% of the subjects taken by a student must be from the university from which the student wishes to obtain a degree.

AUDIO ENGINEERING AT SYDNEY

Over the last three years Sydney University has been planning and, this year presenting, an audio engineering degree. During this time discussions were held with many people and organisations, though I realise, to my chagrin, that there was never a formal discussion with the Australian Acoustical Society. Never the less there was informal contact and, I hope, ample opportunity for the Society to express an interest in what was being done. Although help was received from the Society in the distribution of information about a course on audio acoustics which was run last year, as a way of determining interest in the proposed program, there seems to have been very little interest from the Society in this or other education programs. Even giving programs such as ours some free publicity would be of enormous benefit.

When I suggested to the NSW Division of the Society that the Department of Architectural and Design Science organise an evening for the AAS and the AES to hear students give technical presentations about some of their work and at the same time to inform anyone interested in what we are doing, I got an enthusiastic response. The Acoustical Society, however, needs to be more proactive on such issues and offer assistance and request representation. Although the audio program at Sydney University is of more relevance to the Audio Engineering Society and the Institute of Radio and Electronic Engineers there has been a noticeable lack of interest from the acoustics profession, which is especially significant given the lack of a recognised university program in acoustics, the possibility of using the audio degree as a basis for an acoustics degree and moves by other professions to require evidence of continuing professional development of their members.

SUPPORT FOR POSTGRADUATES AND RESEARCH

The acoustics profession benefits from both undergraduate and postgraduate students but more from postgraduates who have undertaken research degrees in acoustics. Many such students find work with acoustical consultants and it would benefit both the students and their potential employers if the students were given some paid experience during their candidature. This happens already in a few cases but I would encourage consultants and other employers such as the EPA and the motor vehicle industry to help students and help themselves by giving students some work experience.

To use a religious analogy (or rather, to acknowledge that universities have not changed much since their monastic beginnings) students are the novices who will eventually be ordained into the professions, or priesthood. The priesthood (Order) needed the novitiate to maintain itself and so the Order fed and clothed the novices while they trained. The one thing that seems to have changed is that the Order (the Acoustical Society or University in this case) no longer supports the novices. In fact it is worse, the novices pay for the privilege of training in most cases. While it is out of the question for the Society to support all students who may join the acoustics profession, giving students the odd handout of work will help maintain them in their struggle through the education wilderness towards enlightenment.

Supporting research in universities and initiating research work is another area where both potential employers and the Acoustical Society could and should help. Admittedly the Acoustical Society of America is very much larger than its Australian counterpart and it can therefore afford to do much more. Nevertheless it should be possible for the Society, in conjunction with industry and government instrumentalities, to suggest research projects that they would like to see undertaken......and possibly even put up some money for a scholarship, supplementary scholarship or postdoctoral fellowship, as the Acoustical Society of America does.

FUNDING OF UNIVERSITIES

It is very satisfying to win awards for work and papers but in the end it is the money that is needed to carry on. We have been extremely fortunate in being the recipient of equipment from the EPA and the former NSW PWD. Much of this equipment is somewhat dated but without it we could not run the acoustics laboratory as funding, within the university and department, is so bad (Sydney University's support for acoustical research amounts to employment of one technician and little else.) that we are dependent on such loans, research grants, the hiring out of facilities and equipment, the odd consulting job, student fees, an exchange of space for access to facilities and donations in order to be able to carry on teaching and research programs. We buy second hand equipment to cannibalise it to keep existing equipment working. We borrow equipment and we solicit gifts in kind. It is demeaning, very time-consuming and begging is not our core business. (Having said that I would like to make it clear that there is a very vibrant and productive group of students in the lab at Sydney and its research output has never been higher.) The point here is that all universities would welcome donations of equipment and software and that these would be all the more appreciated if we didn't have to go begging for provisions like Hindu monks with their begging bowls.

There is a further way that the profession can help in the training of students in these difficult times for universities. Between 85 and 90% of university budgets are spent on salaries for staff. Contrary to popular belief, tenure of staff does not mean a job for life and many departments, including my own, have only about half their staff on tenure. The issue is that universities have to face is that they are committed to teaching programs because students take years to flow through and graduate. Even if a staff member resigns another member of staff must be hired as a replacement. Increasingly staff are required to undertake more teaching, research and administration while their relative pay is decreasing. It is difficult to encourage staff to leave but it is even more difficult to encourage good professionals to become academics. Most universities are now resorting to employing staff on short term contracts because Federal Government financial support is not guaranteed. This only makes the hiring of staff more difficult. Part-time employment is also becoming more common and with it the administrative load on full-time members of staff is becoming even greater. Adjunct appointments are also becoming common. These appointments are for practicing professionals, who are sometimes even paid a retainer, to help with teaching and research supervision. Retired practitioners and researchers can also hold adjunct positions and I would encourage any member interested to offer his/her services for such a position. The next move will doubtless be to employ more women on a casual basis: a well-tried solution elsewhere in the workplace.

A CONSULTANCY EXPERIENCE

Occasionally I am asked to undertake consulting jobs. Sometimes this is because of particular expertise that I have but more often than not it is because an ex-student of mine remembers enough of my lectures to call up the university switchboard and ask to speak to the guy who taught acoustics and more often than not the consulting jobs come to me because there is little or no money involved, so other consultants have turned down the opportunity to do a bit of community service.

One such case occurred last year. The following is a simplified version of events because it is a long and involved story. An ex-student had a friend who worked for a construction company. The company had taken on a job at a rock bottom price, in the building recession, in order to save its employees and keep afloat. The contract was disadvantageous to the construction company in other ways as well and the company was responsible for detailing the building so that it complied with relevant provisions of the Building Code and local government ordinances. My ex-student was being retained as the Building Code Consultant to ensure that the building complied with the BCA. Could I give them some advice? It would only take half an hour. When I put this down the way I have I wonder why I fell for it but then that is with hindsight.

What it turned out to be was that the building structure was almost complete and the interior was being fitted out. As is common in such buildings the plastic wastepipes on one level penetrated the structural slab into the apartment below. This is fine provided that the wastepipes are adequately sound isolated. The 10mm suspended ceiling which was planned was not adequate to comply with the spirit of the building code and there were other complications such as a lightweight flexible air exhaust duct beside the wastepipes which provided another transmission path. There was no money and sundry other reasons why nothing that I suggested could be done but eventually it was agreed that the pipes would be wrapped and that I would return when this had been carried out on several pipes so that I could test to see whether the noise reduction was sufficient as a certificate of compliance was required. Months passed and when I was called the pipes had not been wrapped in the way I had specified.

The details of what had happened are not clear to me but I presume that the supplier of the wrapping material was pressured into reducing his price and under this pressure had said something like, "I could supply some paint-on bituminous material which is similar to the wrapping material and has been used in other buildings. It would be much cheaper." It turned out that the paint-on material did nothing, acoustically. This was initially labelled as my mistake and there was considerable animosity even when I pointed out that I had documented what had to be done and that my recommendation had not been followed. It was left that if the company could supply me with the paint-on product data to show that it had the same properties as the wrapping I would accept responsibility. Neither the construction company nor the supplier could give me any relevant information on the properties of the material and so an independent test was arranged. Again it was found that there was no measurable reduction in the sound. At this stage there was an about-turn in the attitude of the building contractor as it was recognised that the supplier could be made to pay for the failed insulation of the wastepipes in the building and that is what happened.

There were several more visits to the building as work specified was either not carried out or carried out incorrectly. Each visit lasted several hours even though there was little to be said or done. As I was on a fixed fee there had to be a point when I complied with their wishes and supplied a compliance certificate or didn't get any fee at all. I was determined that this was not going to occur but in the end I softened my position a little because I was about to leave the country for a few months. And the work, or my part in it, was completed just days before my departure.

The most disturbing aspect of the whole situation was the continual suggestions that the work was close enough to what was required and that the easy way to fix it would be to go out for a good meal with a couple of girls. (I have heard of threats of physical violence in similar situations and although this was not used against me the tactics used suggested this possibility.) The pressure applied to go drinking, at the end of the day, at the contractor's expense, (or was it) was another disturbing issue. There was also the racist argument used that the owner of the building was from Hong Kong and all the units had been sold to Hong Kong Chinese and so it really didn't matter what the quality of the sound insulation was like and, by inference, that my concerns were unjustified.

The next most disturbing aspect was that the Building Code was useless as there was no way of testing compliance and that the Code was not specifying what was required and it was in fact discouraging designers, builders and contractors from using pipes which produced less noise. What is specified is that the STC rating of the enclosure/duct/partition between the pipe and the occupied space should be at least 30. What needs to be specified is that the sound level in the occupied space should not exceed a certain level when the wastepipe is used in a particular way.

The testing was done in an ad hoc way by pouring water down the wastepipes. (At one stage the contractor did not pour any water down the wastepipe and it appeared as though there had been a miraculous improvement in performance.) There is no indication whether the testing should be done in this way or with sound. There is no indication of how many situations should be tested. How is it possible to measure the STC rating of a wrapped pipe and ceiling system? Is the area correction based on the pipe circumferential area, the area of the wrapping, the ceiling or what? It seemed obvious to me what was intended by the Code and it was obvious that the system used did not comply with the intention. It was equally obvious that if the matter was dealt with by the legal profession my position would be unsustainable. During this time I realised that I was being retained to provide a certificate of compliance, not to ensure that the design of the building was adequate. As a result of this I wrote to the Building Inspector of the council concerned and, without mentioning the name of the company or building, informed him of the pressure that was being applied to get me to sign a certificate of compliance. Within a couple of days I had a call from the Code Consultant to say that some bastard had written to the Building Inspector about the state of the noise control in the building. I told him that I was that bastard (a fact I'm sure he knew) and that I had done it to ensure the spirit of the BCA was complied with and my backside was covered.

LEARNING FROM EXPERIENCE

The type of situation described in the previous section is probably experienced by every consultant on a regular basis and I'm sure there are infinite variations on a theme but where is this written-up or presented so that the technological, ethical, financial, and educational aspects can be made use of by others? Where is it stated that paint-on bituminous material does not reduce noise radiated by plastic wastepipes even though several buildings with this treatment have been certified as complying with the acoustic provisions of the building code? Where is it stated that acoustic absorbent lining in the ceiling of bathrooms is unsatisfactory when used with 10mm plasterboard? Who has raised the issue of whether STC 45 or STC 50 is required between apartments when, as is usually the case in most new apartments, the kitchen and living space are contiguous? How is the STC rating of a wrapped pipe and ceiling determined? Why are cast iron wastepipes not recognised in the BCA as being better, acoustically, than plastic pipes? Are people like me regularly being exploited by unscrupulous developers? Are consultants giving certificates of compliance for buildings which are unsatisfactory? If this is happening what effect will this have on the reputation of the Society and its members? Where too is the attempt to create culture that members of our profession can accept and whose standard of morality they can willingly adhere to?

In a contribution to the Noise Control Engineering Journal [1] I suggested that the acoustics profession should try to learn from its mistakes (as well as its successes which are usually well documented in the scientific and professional literature) and that an attempt should be made to publicise these so that they are not repeated. I would like to restate this concept and extend it to cover the issues of professional and ethical standards within the profession. In the NCEJ article I pointed out that there were a number of difficulties associated with the publicising of failures, not the least of which are the reputation of the person responsible and the possibility of legal action if the "mistake" is too closely identified. On the other hand if the anonymity of everyone concerned, is respected, as well as the details of where and when the mistake, omission or obfuscation occurred, then the material presented will lose a lot of its impact.

The only person who doesn't make a mistake is the person who doesn't do anything .... and that would be a mistake! To err might be human but to admit to erring could bring out the animal in others but to admit to mistakes should not be a big deal. In fact the person admitting mistakes would be in good company. Edison's journals are largely accounts of his failures. For instance in 1889 he said that, "There is no plea which will justify the use of high tension and alternating current, either in a scientific or in a commercial sense......". Even the great Einstein talked about his failures and described his "greatest blunder" as the concept of a "cosmological constant" though he also stated at one stage that, "There is not the slightest indication that energy will ever be obtainable from the atom". Even though any of us admitting mistakes would be in exalted company it would probably still be considered as committing professional suicide for a consultant to do so. The legal system should change to recognise that humans are human but that, I think, is too radical a concept so we need to explore how the publicising of mistakes may be handled without causing unnecessary embarrassment or adverse professional consequences.

There is a very famous case in the USA of a prominent structural engineer, William Le Messurier, who blew the whistle on himself and came out of the experience with his reputation enhanced. The case concerned the Citicorp Building in New York. LeMessurier was the structural engineer for the building which was a very unusual design. After the building was completed and occupied LeMessurier realised that he had not considered the worst case wind-load and that this, together with some assumptions and alterations to drawings which had been made without his knowledge, meant that structural members would fail in a 16 year return wind and result in the collapse of the building. As LeMessurier remarked [2], " You have a social obligation. In return for getting a license and being regarded with respect you're supposed to be self-sacrificing and look beyond the interests of yourself and your client to society as a whole. And the most wonderful part of my story is that when I did it nothing bad happened." (That is not entirely true as the modifications to the building cost about US$5,000,000 and LeMessurier's insurer paid a part of this but there was no loss of life, as there would have been if LeMessurier had kept quiet, and there was no litigation.

Those who write papers about their successes often inadvertently publicise their failures. The most notable example of that recently was the work of Fleishman and Pons, on cold fusion, but there are many other examples. This will continue to happen (see for example [3]) and I don't think that this aspect needs to be addressed.

One way of showing how mistakes can be made would be to give the same problem to a number of practitioners and ask them to come up with a solution. A similar "experiment" was recently carried out by Vorlander [4]. He sent out details of a room to over a dozen authors of software for calculating acoustical measures in rooms. The results of the predictions were compared with the measured values and the differences were found to be very significant. Other "Round Robins" of absorbers, walls and hearing protectors have also been carried out in the past. Like the previous class of mistakes, or less-than-accurate predictions, these types of "failures" are reported on in the scientific literature, even though they may not be labelled as such. Usually the details of which software or laboratories are performing below expectation is kept confidential. This type of comparison work should be encouraged and I would hope that consultants, researchers, standards authorities and others will continue to organise these. In particular there should be encouragement to undertake comparisons of designs and design approaches. This could be done in a number of ways such as a scientific investigation or a design competition and I recommend that the Society consider these approaches. Another way would be for readers to ask questions and pose problems and for other readers to respond to these. This would be like a printed version of an Electronic Bulletin Board and could easily be part of Acoustics Australia or the Acoustical Society Annual Conference.

CONCLUSIONS AND RECOMMENDATIONS

This is a very difficult subject to come to a conclusion about. If there is a conclusion it is that if the profession of acoustics is to survive in Australia then the Acoustical Society has to become more pro-active in many areas, several of which are mentioned in this paper. These needs are summarise below:

- Work towards the development of a recognised educational stream in acoustics.
- Provide support for teaching, research and research students.
- Encourage continuing professional development of members, including the presentation of review papers and workshops at conferences and competitions.
- Involvement of members in developing standards.
- Establish working groups to recommend changes to the Building Code of Australia and other codes and

legislation involving acoustics.

- Publish information on mistakes, omissions etc.

There are a few extra items which I would like to add for good measure:

Dubious practices by members of the Society should be exposed to scrutiny. One way of doing this is by encouraging "whistle-blowing". Such activity is usually discouraged by professional groups, governments and commercial organisations because it may bring these organisations into disrepute. The activity can also be seen as trying to improve the organisation and there should be brownie points for that. The Society should have a policy on this otherwise its whistle-blowing members will, like most other whistle-blowers, find themselves without a job and without the support of the organisation they were trying to serve.

Acceptable professional and ethical standards need to be determined and effort applied to maintain them. This is a very difficult issue which members must decide what to do with. There will never be a unanimous view but to ignore the issue will spell death to the Society.

The Society should consider instituting a prize similar to the annual "Ignoble" awards. For those not familiar with these they are the opposite to Nobel Awards. They are made in various categories such as the most useless piece of research and the most unreadable paper. There is a humour in the awards, and also a serious message.

Finally the Society should consider its roots and realise that it is similar to a religious sect: an off-shoot of science. If the experiences of individuals were written down as modern parables (or myths) and the collective wisdom from these was formed into commandments, if we had a priesthood (Fellows of the Society?) and we introduced professional confessionals where our acoustical sins could be admitted and penitence done (reading or reciting Lord Rayleigh or Leo Beranek for instance), then one day we may all experience heavenly peace.

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TEACHING THE SCIENCE OF AUDIO

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ABSTRACT

The formulation of an educational syllabus in audio engineering with minimal mathematics is described. The syllabus relies heavily on practical experiments which give the student an intuitive understanding of the acoustical and electrical concepts involved. A portable trainer kit has been developed which allows these experiments to be performed in any convenient location. The modules within the kit and the experiments which use them are outlined.

1. INTRODUCTION

The Graduate Audio Program at the University of Sydney commenced in 1996, accepting enrolments for Graduate Diploma in Design Science (Audio) and Master of Design Science (Audio). The Master's degree by coursework has a dissertation option, while the Graduate Diploma is by coursework only. A Master's degree or PhD may also be undertaken by research only.

The program is multidisciplinary, covering:

- the physiology and psychology of hearing
- physical acoustics
- musical acoustics
- architectural acoustics
- electroacoustics
- audio electronics
- signal processing
- music
- studio sound systems
- sound reinforcement systems
- broadcast sound systems
- digital audio systems
- principles of audio production

In short, the program aims to promote a greater understanding of both the science of audio and the art of audio production.

The understanding of the science of audio which an audio engineer needs is not the same understanding as that needed by an acoustical or electrical engineer. It is in one sense narrower, as an audio engineer does not need to know the entire field of acoustics or electrical engineering, and in another sense far more wide-ranging, as audio engineers need to also understand a number of other fields. Consequently, the audio engineer's knowledge cannot be in as much depth as the "specialist". In this sense, audio engineers have a great deal in common with architects. An architect needs to be a structural engineer, a graphic artist, a plastic artist, a sociologist, a psychologist, an environmental scientist and a number of other things. Similarly, an audio engineer needs to be an acoustician, an electronics engineer, a perceptual psychologist, a musician, a technical producer and a creative producer. Although audio students may never be expert in all of these fields, they can leave the course with an improved understanding of all of them which they can apply to their work to get better results, faster results and (hopefully) more creative results.

The core subjects in the Audio Program are:

- Audio Acoustics
- Audio Practice
- Electrics, Electronics and Electroacoustics 1 & 2
- Music 1
- Analogue and Digital Audio
- Audio Production

A number of elective subjects are also available, including:

- Loudspeaker Design
- Architectural Acoustics
- Music 2
- Multimedia in Design
- Project Management
- Facilities Management

The laboratory equipment and exercises described in this paper were prepared for the Audio Practice course, which is one of the first courses in the curriculum. Its approach is a good example of the overall approach of the Audio Program.

2. GOALS

One of the core values in our syllabus is the recognition that an understanding of both art and science are important for the audio practitioner. Aural skills and creative technique are just as important to the audio professional as an understanding of the physical and perceptual aspects of acoustics and sound systems. Part of the practical work is therefore done in a commercial recording studio, with exercises on signal processors to achieve various spatial and timbral effects.

Many of the students entering the Audio Program already have considerable experience and production skills, but often do not fully understand the physical or perceptual basis of the techniques they have learned. Others have not yet had the opportunity to develop these skills, and are looking to work in the audio industry during or after their studies. Hence all of the students need to learn the physical and perceptual principles of acoustics and sound systems, even though their motivations vary.

3. PREREQUISITES AND ASSUMED KNOWLEDGE

As a graduate program, the prerequisite for entry is a tertiary qualification, although this need not be in any specific field. In addition there is articulation with VETAB-accredited (Vocational Education and Training Accreditation Board) courses such as are offered by the JMC Audio Academy, School of Audio Engineering and Australian Institute of Music. Significant experience in the audio industry also qualifies. This range of entry qualifications means that the students have a wide range of backgrounds, from electrical engineering to music.

Much of the theory in the science of audio is mathematically based, but most of the practitioners in the audio industry are not. The assumed level of maths for students entering the Audio Program is Higher School

Certificate 2 unit (i.e. standard rather than advanced high school level). This poses a fundamental problem for educators: how to give students a sound understanding (pun intended) of theory and principles without first teaching a lengthy course in mathematics.

4. STRATEGIES FOR TEACHING QUANTITATIVE CONCEPTS

It is often forgotten that the symbolic manipulation which is the language of mathematics arose from the desire to analyse, prove or solve physically-observable phenomena. Since these origins, mathematics has grown in many areas into a purely abstract science.

While some more advanced analyses of physical phenomena may be awkward or even impossible without symbolic mathematical skills, the phenomena remain observable and quantifiable. It is this observation and quantification, along with interpretation and application of the results, which is a prime focus for our program.

The strategy for managing the maths has three elements:

- present principles in graphical form wherever possible. Provide analogies, graphical derivations, and demonstrations.

- present mathematical results wherever tractable but omit the mathematical derivation. References are given for students with the background and / or desire to pursue the mathematical aspects.

- wherever possible, allow students to explore phenomena for themselves through physical experiments.

The first two elements are essentially appropriate preparation and delivery of lecture material. The third element requires lab equipment.

5. LABORATORY FACILITIES

Just as there are many possible approaches to teaching theory, there are also many possible approaches to teaching in a laboratory. The most suitable approach is generally determined more by the available facilities than by the students, the teachers or the subject.

Two of the standard ways of providing equipment for student experiments are:

i. Provide specialised equipment for each experiment and rotate student groups between experiments

ii. Provide a common set of equipment to each group of students for all experiments.

The former has the advantage of optimising the equipment for each experiment, having a permanent installation which reduces setup time but increases familiarisation time, and possibly requires less equipment. The disadvantages are that it may be difficult to adapt the equipment for other experiments later, and the experiments may not follow the lectures in a combined theory / lab course.

The latter has the advantages of:
- allowing the experiments to follow the lecture program
- allowing all students to perform the experiments simultaneously, reducing the opportunity for pre-empting results while still giving the chance to compare notes after completing the exercise
- simplifying the supervision and administration
- reducing familiarisation time, as the same equipment is used for each exercise

while it has the disadvantages of:
- potentially needing more equipment and hence a bigger outlay
- requiring more time for the equipment to be reconfigured for each experiment.

If the equipment can be kept fairly simple and not too expensive, the second option is an attractive one for both educators and students.

As acousticians will know, there is no such thing as a universal set of lab equipment, although there are standard work-horses which will cover many needs. In the case of an electroacoustics course, even a modest student lab needs:

- a sound level meter
- a microphone preamplifier
- a range of commercial microphones
- a measuring microphone
- a wideband voltmeter
- a balancing transformer
- a sinewave generator
- a noise generator
- a monitor amplifier
- a phase shifter
- a phase reverser
- an oscilloscope
- a phase meter
- a monitor loudspeaker

A more advanced lab kit might also contain:

- a reverberation analyser
- a spectrum analyser
- a response curve tracer
- a narrow-band filter set
- a narrow-band equaliser
- a tape recorder
- a noise and distortion analyser
- a personal computer with appropriate software and hardware for simulation, data acquisition and analysis

The budget for even the basic kit will run to thousands of dollars. The more advanced kit will be well over $10,000. Multiply this by twenty-plus students, and the budget quickly becomes unrealistic.

Aside from budget considerations, the equipment needs to be portable, easily stored when not in use and easy to use.

Similar requirements have been met by educators in the past by building trainer kits. The most common example in electrical work is the digital logic trainer (figure 1). Other examples include the analogue computer (figure 2) and the wiring board used by electricians.

Other requirements for an audio trainer include:
- limited cost to allow a number to be made on a reasonable budget
- simple construction to allow non-specialist technical staff to assemble and test
- physically and electrically robust design
- a mains power supply for reliability and convenience
- ideally, balanced circuits: in practice, at least the microphone preamplifiers must be balanced

- easily-obtained parts
- shielding from electromagnetic interference

Figure 3 shows the trainer which was designed for our audio program. It contains all the modules of the basic lab kit listed above with the exception of the sinewave generator and the oscilloscope, which were readily available. The technical specification is shown in appendix 1.

Experiments designed for the unit include:

- observing sinewave addition and cancellation due to phase shift
- measuring proximity effect in microphones
- measuring microphone directional response
- measuring loudspeaker amplitude and polar response
- measuring binaural lateralisation for amplitude and phase differences
- measuring common-mode rejection and interference pickup from various cables and sources
- earth loop simulation
- measuring gain-before-feedback for different microphones and microphone - loudspeaker geometries
- constructing a simple telephone hybrid and measuring its performance

Student reaction to the trainer kits has been very good. There has been little confusion about how to use them, instruction and setup time is minimal, and they can be easily transported if acoustic isolation is required.

The prototype was designed and built in two weeks for a trial prior to commencement of the course, and technical staff made an outstanding effort to prepare another six chassis in time for the course commencement. The short design time and heavy reliance on point-to-point wiring mean that around five working days are needed to complete each unit.

6. ACOUSTICS OF LABORATORY SPACES

Most of the experiments in our course require acoustic measurements. If the students all perform experiments at once, some acoustic isolation is needed between workspaces. This can be achieved in a number of ways, depending on the class size and facilities available:

i. Use of a large common space with all measurements made in the near-field i.e. well below the "critical distance" where direct and reverberant sound fields are of equal intensity. Student groups need to be separated as far as possible. This is a compromise solution which simplifies supervision but creates acoustical interference problems.

ii. Use of separate rooms for each group of students. This solution makes life easier for the students, but requires a little more legwork from the supervisors. Spaces of a reasonable size are still needed for a workable critical distance. This is quite a practical solution, however, especially if the course is conducted after hours in an unoccupied institutional building with many available spaces (e.g. Physics Building at Sydney University)

iii. Use of an acoustically partitioned workspace. Most lecture theatres and laboratories in education buildings are acoustically "live" (long RT60 and low acoustic absorption coefficient) and even when acoustically isolated, are less than ideal for making free-field measurements. An acoustic laboratory area with absorbent partitions is a better solution for both students and educators, and is a long term goal for our program.

7. THE FUTURE

As with any educational program, the Audio Program aims not only to keep up to date with industry developments, but also to continually improve and refine its course material and facilities.

As well as the planned upgrade to the laboratory acoustics mentioned previously, virtual instrument facilities (computerised test and measurement) are being expanded. The flexibility and cost-effectiveness of virtual instruments make them particularly suited to teaching and research in audio. These will be an adjunct to the audio trainer kits rather than replacing them.

Future revisions for the audio trainer kit may include a redesign of the printed circuit boards to reduce the wiring and a chassis revision to allow automated hole punching (chassis fitting is currently done by hand). A third-octave filter option is also planned.

8. CONCLUSION

A quantitative understanding of audio and acoustic principles is a key element in any educational program in audio or acoustics. While higher mathematics may be the ultimate embodiment of this, it is not always a teaching option in practice. Demonstrations, graphical explanation and participatory laboratory work, however, are a viable alternative.

There are a number of possible approaches to outfitting an audio laboratory, depending on available facilities, budget and lead time. In the context of the Sydney University Audio Program, construction of a number of identical portable, modular audio trainer kits was an economical and functionally attractive option.

The acoustical measuring environment is an equally important consideration in laboratory work. While purpose-built acoustical environments are desirable, useful results can still be obtained in general-purpose laboratory spaces.

The modular laboratory trainer kit has proved to be a most useful tool for training in audio and acoustics in this context. It allows a wide range of experiments to be performed with flexible location and timetabling and a manageable budget, gives students an intuitive understanding of physical principles and provides an unintimidating introduction to laboratory techniques.

9. ACKNOWLEDGMENT

The author would like to thank Messrs Phil Grainger, Rick Moss, Ken Stewart and Peter Mumford of the Department of Architectural and Design Science for their invaluable efforts, above and beyond the call of duty, in helping to build and test the audio trainer units in time for this year's course.

APPENDIX 1: MODULAR AUDIO TRAINING UNIT

Specification

Dimensions:	385 x 210 x 305 mm
Weight:	approx 10 kg
Case:	enamelled steel with removable front and back panels.
Power Consumption	< 30 VA
Operating levels:	1V r.m.s. nominal, 10V r.m.s. maximum unbalanced except for microphone amplifier
Microphone Preamplifier:	
Input:	Transformer coupled, maximum level -5 dBu
Gain:	+20 to +70 dB in 10 dB steps
Equivalent input noise:	< −129 dBV, 20 kHz bandwidth
Bandwidth:	-3dB @ 7 Hz, 30 kHz independent of gain
Common-mode rejection:	> 105 dB @ 50 Hz > 80 dB @ 1 kHz
THD	< 0.1% @ −5 dBu in, 30 Hz - 20 kHz
Voltmeter	
Ranges	100 mV - 10 V r.m.s. f.s.d.
Amplitude response	-3dB @ 2 Hz, 200 kHz
Nonlinearity	< 2%
Accuracy	4% all ranges except 1V r.m.s. range 2% 1V r.m.s. range
Rectifier type:	averaging
Phase Meter	
range	-180° to $+180^{\circ}$
accuracy	+/− 5° for 100mV - 10V r.m.s.
VU Indicator	
Integration time	160 mS for -2 dB
Calibration	+4 dBu = 0VU
Input impedance	47 kohm
Mixer	
Type	2 input, 1 output.
Maximum output level	10V r.m.s.
Input impedance	47 kohm
Phase Shifter	2nd order all-pass, Q = 0.5, centre 200 Hz - 2 kHz
Phase Reversal	active, unbalanced, unity gain

Audio Transformer	
Primary winding	150 + 150 ohm
Secondary winding	150 + 150 ohm with electrostatic screen
Maximum input level	+24 dBu
Amplitude response	+0 / -0.5 dB, 20 Hz - 20 kHz
Noise Generator	white or pink noise, switched
Monitor Amplifier	
Number of channels	2
Maximum output level	20W / 8 ohm
Amplitude response	-3 dB @ 10 Hz, 80 kHz
Protection	protected from short-circuit and thermal-overload.
Accessories	1 x 10mm electret capacitor omnidirectional microphone
	1 x dynamic hypercardioid microphone
	1 x 2 way speaker with switched drivers
	125mm woofer in vented enclosure
	25mm dome tweeter
	1 x light dimmer board with simulated earth loop
	1 x tape head demagnetising wand
	Range of connecting leads

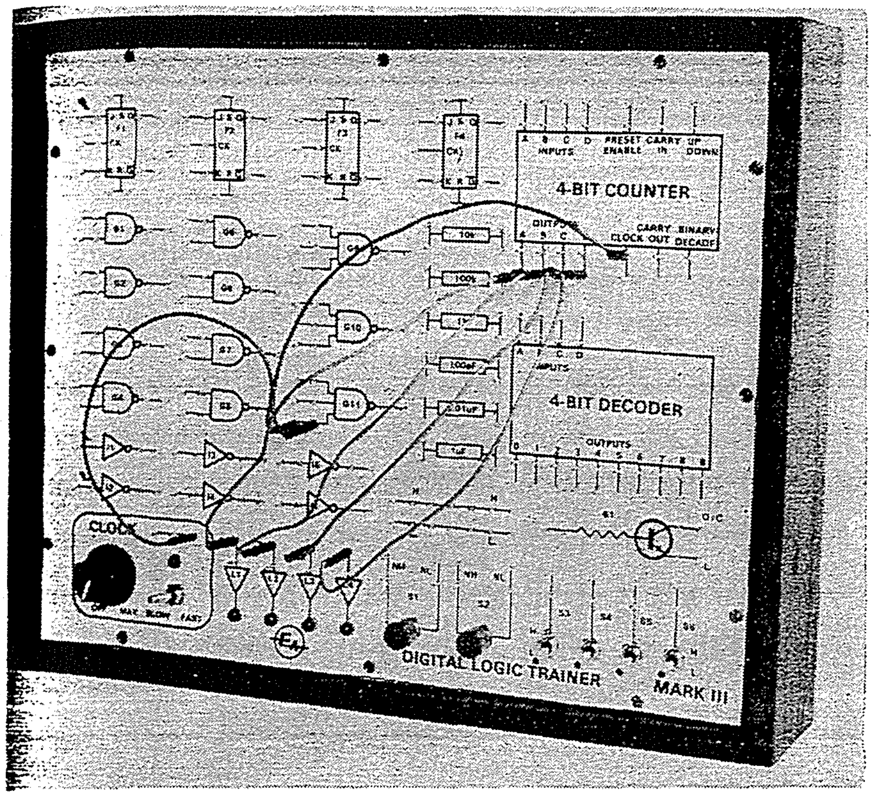

Figure 1: Logic Trainer

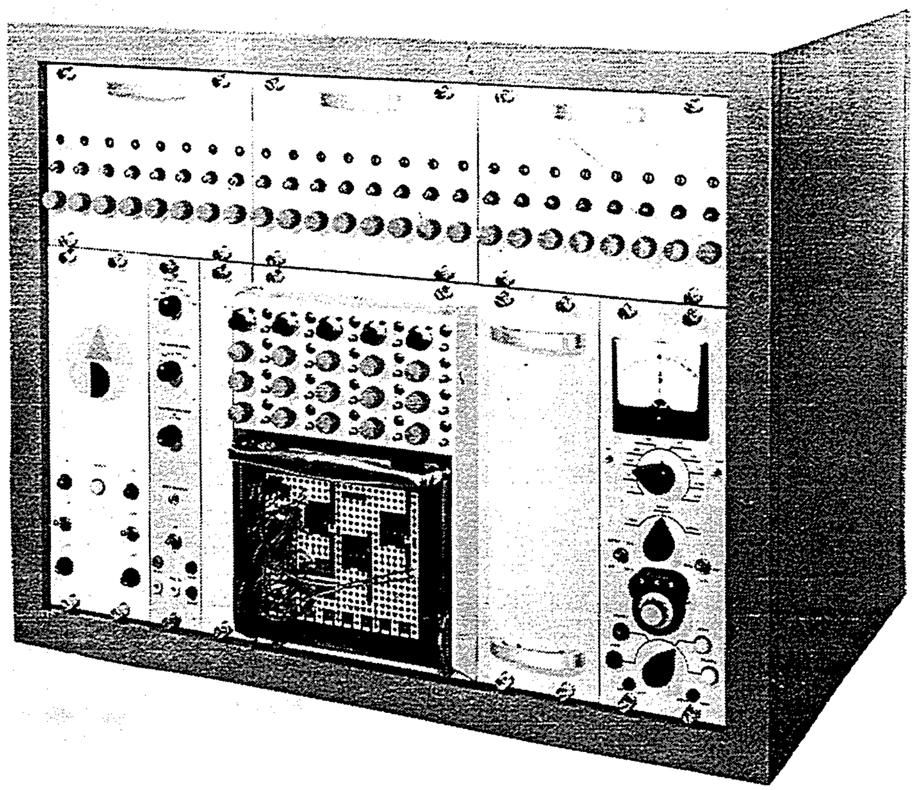

Figure 2: Analogue Computer

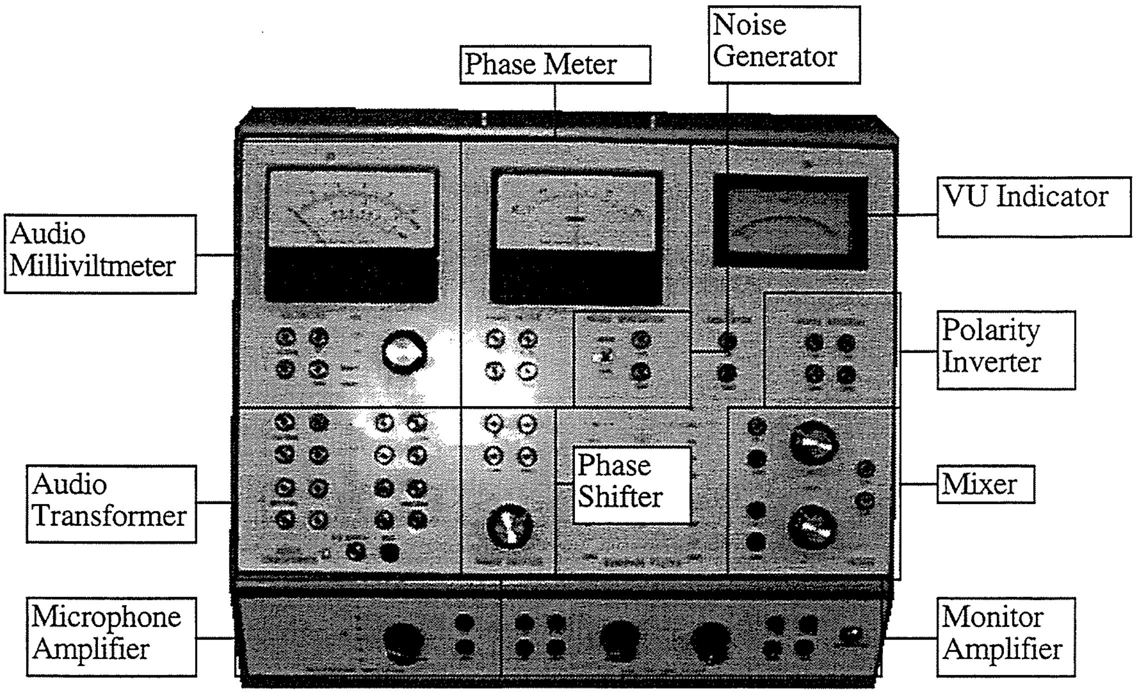

Figure 3: Audio Trainer

IMPLEMENTATING A STRATEGY FOR NOISE MANAGEMENT FOR THE BRISBANE RIVER AND CATCHMENT

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INTRODUCTION

This paper summarises the findings of a study commissioned by the Brisbane River Management Group (BRMG) to determine the most appropriate options for resolving the conflict between the noise generating uses (recreational, commercial and industrial) of the waterways within the Brisbane River Catchment and noise sensitive landuses.

The BRMG is responsible for the development of the Brisbane River Catchment Management Plan. The purpose of the noise management study was to assess the noise conflicts inherent in the uses of the catchment, and to identify a range of potential solutions. These solutions are to be incorporated into the final Brisbane River Management Plan, where appropriate.

ERM Mitchell McCotter was commissioned to perform the noise management study and to investigate and provide a range of management options for a series of issues related to noise. The findings of the analyses were presented to the Noise Pollution and Abatement Working Group (NPAWG) that was convened by the BRMG to review the issue of noise. The NPAWG was chaired by Associate Professor Lex Brown, joint author of this paper. The outcome of the study and deliberations of the NPAWG was a draft Implementation Strategy for Noise Management for the Brisbane River Catchment. The strategy remains in draft form as the recommendations and outcomes have yet to be adopted by BRMG.

Clearly, a study of this type contains a number of elements of interest in terms of acoustics. Issues such as the measurement of noise from moving vessels and technology options to reduce noise emissions from rowing coaching activities, for example, were considered in some detail during the study. Technical issues of this nature alone would provide sufficient material for an acoustics conference paper. However, at the outset of the study it was recognised that there was a need to analyse the issue of noise within the Brisbane River Catchment in a manner that encompassed landuse planning issues, environmental planning and rigorous acoustical analysis. The findings of the study reaffirm that noise management is not simply a technical issue, but has to be multi-dimensional. This paper considers each of the which were recognised in the study to be key elements that were essential to the final noise management solutions.

BACKGROUND TO THE STUDY

Historically the waterways of the Brisbane River Catchment have been an essential element of the social, economic, cultural and recreational aspects of everyday life for all that live and work within the boundaries of the catchment. Although economic aspects such as goods transport have lessened with time, the availability of the river in terms of a resource remains a vital feature of life within the catchment. Today, uses of the river tend to focus on tourism, recreation and passenger transport, although some industry remains. Noise is a feature of the majority of these uses, whether directly related to the source (eg, engine noise) or indirectly produced as a result of land side activities such as boat ramps or departing/arriving passengers at ferry terminals.

The continued use of the Brisbane River and other waterways within the catchment as a resource is currently subject to increasing pressure from noise sensitive development. This is due to the fact that the waterways of the Brisbane River Catchment, notably the lower reaches of the river that pass through the heart of Brisbane, are highly constrained in terms of noise sensitive landuse. Residential development is a dominant feature of riverside landuse and, towards the centre of the city, multi-storey and high rise development is the norm. Other potentially noise sensitive landuses also occur, such as educational and health establishments, and areas where the focus is passive recreation or cultural heritage uses.

At the outset of the noise management study it was recognised that compromises are necessary in order to achieve an acceptable balance between the two. Conflict is inherent in the two environmental values for noise management within the Brisbane River Catchment which have been proposed by the NPAWG.

- the right to live free of intrusive noise; and
- the right to use water areas in the catchment for a variety of purposes including recreational, transport, entertainment and industrial purposes.

In order to determine the most appropriate management solutions for the catchment, it was necessary to clearly establish the extent of the noise conflicts and opportunities in terms of potential for increased use of a waterway as a resource for noise generating activities. This is discussed below, followed by an overview of the development of a series of management solutions.

IDENTIFICATION OF CONFLICTS AND OPPORTUNITIES

In order to establish existing and potential future noise conflict in the Brisbane River Catchment, a study to identify noise sources and noise sensitive receptors had previously been commissioned by BRMG (John Butler Consulting Pty Ltd, 1996). The results of the study were drawn upon and further developed in terms of identifying planning zones and strategic planning designations in the lower catchment and upper catchment of the Brisbane River to assess the potential for further noise sensitive development in the short to medium term.

This landuse analysis enabled the identification of areas of the catchment that were highly constrained in terms of noise sensitive receptors (eg. high rise riverbank development) or offered opportunities for development of the river as a resource. The latter included areas that were already constrained by noise from transport infrastructure, stretches of riverbank where existing and future development was low density or had a setback sufficient to provide a noise buffer, and areas where industry was currently operating or had recently ceased operations (eg. former mineral workings).

Geographical Information Systems were accessed to locate number and location of the dwelling units in proximity to waterways, the numbers directly on the riverbanks, and the numbers likely to be exposed or not exposed depending on the topographic situation between water based sources and dwellings on the land. The results of the analysis concluded that approximately 2,150 households are directly exposed to noise from activities on the Brisbane River, estimated to be equivalent to 6,000 persons.

The analysis concluded that the majority of the lower catchment (ie, urban and suburban Brisbane) was highly constrained in terms of noise, whilst the upper catchment was largely undeveloped and was currently relatively unconstrained. Strategic planning designations indicated that in the future conflict between noise sources and receptors was likely to increase.

NOISE MANAGEMENT OPTIONS

Management Options

Having recognised the need to reduce existing and minimise future conflicts, a range of potential noise and landuse management tools were identified early in the study. The options identified for managing activities and landuse included:

- spatial controls;
- temporal controls;
- technology controls;
- legislative controls;
- planning policies;
- fiscal measures;
- voluntary controls;
- provision of alternatives; and
- education of users.

Each management measure was reviewed in the light of existing practice, feasibility, and likely effectiveness, and the following general approach was proposed for adoption in the noise management :

- *management of existing conflict:* spatial controls of uses, setting noise limits (which in turn lead to technology controls);

- *prevention of future conflict:* planning guidelines to ensure appropriate uses and protect environmental values, development of codes of practice and education of users, co-ordinated management of the Brisbane River Catchment in a sustainable and equitable manner.

Management Framework

To provide a framework in which to implement management measures for both existing and future uses, a noise zoning scheme was developed for the Brisbane River Catchment. These zones facilitate specification of differing levels of noise control according to localised sensitivities, and also serve to highlight those areas with high degrees of sensitivity or significant potential for increased intensity of use of the water resource.

Three noise zone categories were designated as follows:

- *Zone A:* areas with large numbers of exposed riverbank or waterfront properties, potential for considerable increases in such properties in the future or other cultural, recreation or environmental sensitivities. The primary environmental objective in *Zone A* designations is to minimise intrusive noise;

- *Zone B:* areas with few exposed riverbank or waterfront properties and generally where a significant buffer zone exists between sensitive land uses and noise sources, absence of other cultural, recreation or environmental sensitivities. These areas include locations that are currently the focus of water-based recreation or offer significant potential for use as a recreational resource. The primary environmental objective in *Zone B* designations is to maximise recreational amenity and opportunity;

- *Currently Unzoned:* areas that need more careful planning as they are currently undeveloped. The designation of zonings in these areas is left to the relevant planning or management authority on the basis of the guidelines prepared during the study.

The designation of these noise zonings involved consultation with local authorities and the community, recreation/sport, industry and commercial representatives on the NPAWG made representation on behalf of their respective interests. Compromises were negotiated between the various parties in terms of location and length of zones, and a high degree of consensus was finally achieved that recognised the need to protect the environmental values recognised for the Brisbane River Catchment.

Management Constraints

The development of the noise zoning system in itself served to highlight further constraints to management of the river. Of key importance was the lack of facilities such as boat ramps for the areas identified as suitable for increased recreational/sporting use, and the highly restrictive use policies adopted for certain dams in the upper catchment.

The study recognises that these issues need to be addressed to ensure the proposed management strategies that are consistent with the current and future use of the river. Consequently, the draft Implementation Strategy for Noise Management recommends further investigation and provision of additional infrastructure and facilities, where needed, to enable the river's resource potential to be realised.

RESOLUTION OF EXISTING CONFLICTS

Noise Sources

In terms of existing conflicts, the following issues were identified as being of primary concern:

- noise from the coaching of rowing crews in reaches of the river characterised by dense residential development;

- noise from power boats and personal watercraft.

A further nineteen potential sources of noise impacts were identified including:

- entertainment cruisers;
- ferries and ferry terminals;
- dredging;
- boat ramps;
- shipyards;
- barges;
- trawlers;
- fireworks and special events;
- canoes;
- construction;
- gravel crushing;
- industry;
- irrigation pumps;
- marinas;
- motor traffic;
- sewerage works;
- sailing;
- seaplanes;
- slipways;
- surf boats; and
- wharves (passenger and freight).

Draft noise limits for the majority of the noise sources were proposed in the draft Environmental Protection Policy (Noise) being prepared by the Department of Environment during the course of the study. A thorough review of the draft EPP, however, indicated that some of the limits proposed were inappropriate for a waterway as highly constrained as the Brisbane River. Further, the draft EPP did not provide a clearly defined control mechanism for dealing with noise nuisance from sources such as rowing.

Further detailed investigation of noise management options for power boats and rowing were therefore performed as discussed below.

Noise from Rowing

Rowing noise is a highly contentious issue in relation to the Brisbane River, and results in a significant number of complaints from local residents. The conflict relates to the use of the river for sports training and, on occasions, as a venue for sporting events. The key areas of concern relate to noise during the early morning as a result of noise due to coaching or the cox (particularly where megaphones are used) and noise from power boats used by coaches training the rowers.

Throughout the discussions of NPAWG during the study, rowing was recognised as a highly appropriate use of the Brisbane River. However, the conflict between early morning noise from rowing activities and large numbers of receptors in close proximity to reaches of the river heavily used for rowing activities was identified as a noise conflict situation that required resolution.

Given that the rowing noise problem is closely related to time of day, an obvious solution was considered to be limiting rowing activities to less sensitive times of the day. In the event that rowing activities could not be restricted to after 7 am in the morning on weekdays, or after 8 am at weekends, it was recommended that megaphones should not be allowed before these times, and that noise from other sources (eg, the power boat and cox) should be kept to an absolute minimum. A draft Code of Practice for Rowing was drafted on this basis, with the timing restrictions proposed for the sensitive *Zone A* designated areas of the catchment.

It was recognised that the draft Code of Practice had the following implications:

- either rowing activities involving coaching and/or a cox would not be possible during early morning periods; or
- communication technology solutions would be required.

Communication is vital to ensure the safety of the rowing crew, particularly on a waterway with different types of vessels such as the Brisbane River. In order to assess the viability and cost of communication technology options, a series of solutions were reviewed including off-the-shelf radio communication ranging through to proven purpose-built communication systems. Practical testing of these solutions was outside the scope of the study, however the draft Implementation Strategy for Noise Management recommended that field trials should be performed as soon as possible to enable adoption of the draft Code of Practice for Rowing.

An investigation of the cost of technology options indicated that relatively low cost solutions were available (FM radio transceivers) although effectiveness of this solution was unproven. To place the costs in context, it was estimated that costs for provision of communication technology ranged from approximately 2% of purchase price for the unproven low-cost FM radios to 5% of the purchase price for the purpose built proven technology option. A range of funding options were identified that could be implemented to assist the rowing clubs with purchase of the technology (the majority of clubs are amateur organisations, many associated with schools or universities).

The outcome of the investigations was a recommendation that the noise management measures incorporated in the draft Code of Practice for Rowing should be implemented as soon as possible, and no later than two years from adoption of the code. It was recognised that this recommendation would almost certainly necessitate use of communication technologies.

Noise from Power Boats and Personal Watercraft

Noise from power boats and personal watercraft was an issue in terms of level of noise and character of noise, and the fact that many complaints related to repeated use of a restricted stretch of waterway for activities such as water-skiing or use of a personal watercraft. The latter issue is best resolved through a code of practice for waterway users, and a code was drafted for power boat/personal watercraft users in the strategy to deal with this issue.

The issue of noise level and character was more problematic for a number of reasons. Firstly, although the draft Queensland Environmental Protection Policy (Noise) had been drafted, the proposed noise limit for recreational vessels of 78 dB(A) and 95 dB(A) for racing craft measured as a drive-by test at maximum vessel speed at 30 metres (in accordance with AS1949:1988) had not been validated for specific conditions such as the highly constrained urban and suburban sections of the Brisbane River. The constraint is in terms of the close proximity of noise sources and receptors on the river. Secondly, the noise test itself (AS1949:1988) was problematic as it lacks clarity in terms of defining the test conditions, and requires the vessel to operate at maximum throttle regardless of the local conditions (such as speed limit) of the waterway in question. Thirdly, the test relates to a drive by test at 30 metres. In the case of the Brisbane River, it is possible for vessels to operate at that distance from a noise sensitive receptor as there is currently no restriction on the minimum position a vessel must operate from the waterway bank except in the case of moorings or other features. On this basis, it was clear that noise sensitive receptors located along the banks of the Brisbane River could be affected by maximum noise levels of up to 78 dB(A) on a routine basis.

Given the concerns raised relating to the draft noise limit, limited testing of watercraft on the Brisbane River was performed. The objective of the testing was to assess the noise levels of different types of vessels, and to relate these noise levels to the experience of the NPAWG in terms of acceptability of noise levels. Testing of racing power boats, standard cruisers, small 'tinnies' and both stand-up and sit-down personal watercraft was performed. The data accumulated from the tests were supplemented with other sources of published data to provide a dataset for analysis of the potential for noise nuisance at riverfront dwellings as a result of power boats and personal watercraft.

The outcome of the analysis was to adopt a more stringent noise limit for highly constrained areas of the catchment, *Zone A* designation, of 75 dB(A) measured at the maximum speed of the waterway or the speed at maximum throttle (whichever is the lesser) for non-racing craft. Racing craft were recommended to be excluded from all *Zone A* designated areas. The adopted noise limit for non-racing craft was determined on the basis of the limited data available, and is intended to result in exclusion of all stand-up type personal watercraft and all racing type power boats or boats with engine modifications, from the *Zone A* areas.

A labelling scheme for watercraft was also developed as part of the study to enable policing of a system based on different noise emission levels for different zones within the catchment. It was proposed that the noise labelling scheme should be implemented by user groups such as power boat clubs, in conjunction with authorities involved in licencing of watercraft. Overall policing of the system would remain a responsibility of the central agency with responsibility for noise from waterway activities.

PREVENTING FUTURE CONFLICTS

Planning guidelines

In order to ensure that conflicts between sensitive land uses and waterway-related noise sources do not increase in the future, the noise management strategy defines a series of guidelines for planning authorities. These guidelines are aimed at maximising use of the river, which includes planning for waterway facilities such as boat ramps in appropriate areas, whilst ensuring that areas with constraints in terms of noise sensitivity are protected.

Planning guidelines are presented for the assessment of planning applications for land-side facilities, and these include assessment of the activities likely to result from the provision of the facilities. For example, where a boat ramp is to be provided, the effects of the watercraft accessing the facility and the use of the watercraft in nearby waters is to be considered, as is the effect of noise from vehicles using the ramp.

Further guidelines are provided for assessment of proposed noise sensitive development where there is a potential for noise impacts as a result of waterway related activities. These guidelines also recommend that the applicant should be informed that the proposed development is in an area that may be subject to noise from the waterway, and that some noise is an inherent feature of the use of such an important community resource. Appropriate acoustic protection measures could then be integrated into the design of the noise sensitive development - either by agreement or by regulation.

Finally, guidance is given to planning authorities in relation to specifying noise zones for those areas that are currently unzoned in the noise management plan. The guidelines recommend that careful consideration should be given to achieving a balance between the use of waterways as an active and passive resource.

IMPLEMENTATION

Implementation of the draft Noise Management Strategy will ultimately be the responsibility of the BRMG, however the study recognised that in reality whilst the BRMG may co-ordinate its implementation, other authorities (whether existing or new) will be responsible for the technical and planning aspects of implementation.

A review of the possible jurisdiction authorities revealed that no single organisation currently has both the skills and resources to implement all of the aspects of the noise management strategy.

Ultimately, the success of the strategy will also depend upon education and adoption of codes of practice by the various user groups utilising the waterways in the Brisbane River Catchment. These issues are strongly emphasised in the management plan. Careful land use planning, education of users and education of the local community to encourage acceptance of the waterways as a resource that should be integrated in the fabric of the social, economic and cultural life of an area are further management issues defined in the strategy that are essential to its success.

CONCLUSIONS

The draft Implementation Strategy for Noise Management provides a framework and series of management tools for facilitating use of the Brisbane River Catchment by noise generating activities in a rational and equitable manner. Both existing and potential future conflicts are addressed through noise management and planning measures in the strategy, and a jurisdictional framework for implementing the strategy is defined.

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ACCOUNTING FOR MOVING VEHICLE NOISE SOURCES IN PREDICTION OF ENVIRONMENTAL NOISE IMPACT FROM OPEN CUT MINE OPERATIONS USING A SIMPLIFIED STATISTICAL SOUND LEVEL APPROACH

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ABSTRACT

In assessing the potential impact of noise emissions from a new development, a typical methodology is to determine the sound power of the noise sources, locate them "in space" and calculate the resultant sound level at receiver locations. For fixed point sources of noise this is relatively simple. For open cut mines using truck and shovel loading methods, typical of Hunter Valley (NSW) coal mines, the major noise sources are mobile and move along haul roads between a pit and some other location.

For a recent noise impact study of a mine expansion, the variation in statistical sound level was used as a basis for prediction of received sound level from sources located along the haul route, by using a time weighting in the prediction of received sound level. This paper describes the method used and proposes its further use in assessing impact from open cut mine operations.

Keywords: Environmental noise, noise impact, impact assessment open cut mine noise

1 INTRODUCTION

The introduction of new developments or significant expansions to existing developments generally require approval from a consent authority before construction can commence. This is usually known as the environmental impact assessment process and the various levels of government are the consent authorities. In New South Wales for example, private developments generally require the consent of the local council, under the Environmental Planning and Assessment Act 1979; pollution control approval is also required from the

Environment Protection Authority. The reporting of the assessment to the consent authority is by the proponent, with advice also being provided by relevant government departments.

Part of the impact assessment studies for a mine development also include noise impact assessment. This paper deals with the environmental noise impact assessment of an open cut mining activity, where the major sources are mobile. This creates an added difficulty in the prediction process - how to account for sources which move around over a wide area and over a wide range of elevations. For the 100 to 230 tonne trucks involved in this type of activity, the haul road could be between a pit below the surrounding ground level and a reject emplacement well above the surrounding ground level. Locating the noise sources to allow prediction of the receiver sound levels is not a straightforward matter when the noise source is not at a location for very long. Road traffic noise calculation models do not work because the number of vehicles is too small.

In attempting to deal realistically with the problem for a mine expansion, the concept of including the statistical variation of sound levels at a receiver into the noise prediction process was considered and developed. This paper reports on the general aspects of noise impact assessment, the application of the statistical time variation approach to the predictions and compares the results for different approaches. The paper was developed from an actual case submitted to the authorities for a mine expansion in the Hunter Valley of NSW.

2 NOISE IMPACT ASSESSMENT

2.1 The General Approach to Predictions of Received Sound Levels

The general approach to assessment of environmental noise of a proposed development is to use some type of prediction modelling procedure and computer modelling is an accepted approach. In this procedure, sound power levels of noise sources are determined, usually from typical operations of the sources at another location (where possible), or from some other data source if not. The topography between the source location and the residential receiver locations is input to the model with the sound power levels and the model calculates the sound pressure levels at the receiver from the various sources used. Current models can account for meteorological effects of wind speed and direction, air temperature and humidity, and lapse rate (inversion) conditions.

For fixed sources, such as fans, factories, conveyors, crushers and so on, the calculations are relatively straightforward. For mobile sources, one needs to consider the locations where the noise emission occurs and for what period (the subject of this paper), where 230 tonne haul trucks travel along haul roads out of a mine pit to ground level, then to an elevated reject emplacement or coal stockpile. The intent of the method described was to seek a simple but statistically relevant approach.

2.2 The Subject Mine Expansion

The subject of the study was the proposed expansion of an existing open cut coal mine in the Hunter Valley. The objective was to assess the potential environmental noise impact of the proposed expansion on the residential receivers in the surrounding area.

The Mine Plan was reviewed and scenarios typical of operations at the 5, 10 and 20 year intervals (from commencement) were determined and agreed with the Project personnel. Machinery locations and elevations within the pit at the relevant year were identified - up to seventeen source locations within the mine area were selected, with up to 10 different sources operating at each of the source locations at any time. Four residential receiver locations (those where environmental noise measurements occurred) were selected to be the receiver locations for the calculations. Cross-sections prepared from each machine location to the four residential receiver areas from available mapping of the site and area.

For each source location, the typically expected equipment for that location was identified. For example, for the pit locations, almost all equipment were located, whilst for the overburden emplacement, the equipment was limited to a haul truck dumping, a haul truck passing by, a dozer and the water cart.

From these combinations of sources at the source locations, the sound level at the four receiver locations were calculated using the computer model "Program ENM". These calculated results could then give the total sound level for all equipment operating all the time at each of the (up to 17) locations.

From these calculations and predictions, isopleth contours (equal noise level) of LA10 sound levels were prepared for the three different mine year plans. The receiver locations and source locations for each of the Year 10 mine plan are shown on Figure 1.

Weather conditions used for the calculations were representative of mid-winter, with calm winds, $5^{o}C$ air temperature and 90% relative humidity. These correspond to minimum atmospheric attenuation of noise propagation. These climatic conditions are a conservative "worst case" condition for calm winds. The lapse rate used for the calculations was $0^{o}C$ per 100m. Normal lapse rate conditions are $-1^{o}C$ per 100m, again a conservative approach.

3 TIME-WEIGHTING THE CALCULATED SOUND LEVELS

At the completion of the calculations, the information available was the individual and combined sound levels for all equipment operating at each location for 100% of the time at full power. For a mine with moving sources, this is an unrealistic and over-prediction of the sound levels likely to occur.

3.1 Sources

In normal operations a **haul truck** is loaded in the pit, climbs out of the pit under a high load, coasts on the level, then may climb under load again to the top of an emplacement area. At the top of the emplacement area, it will dump the load at high power, then cruise back down to the pit to be loaded again and repeat the cycle. A **dozer** will move back and forward at one location moving rock, then sit idle for some minutes and move again. A drill will operate at constant load for several minutes, then be idle as it moves to the next drill-hole. A **shovel** will have a continuous noise emission, but the direction will swing up to 180° as it moves the loaded bucket from the pit to the truck loading point. These variable sources of noise were the only ones relevant to the mine expansion Fixed sources were not being changed. This was where the search for a method to account for the time and level variability commenced.

Using knowledge of the operating practices of the mine, an assessment of the percentage of time each item of mobile plant was expected to be emitting noise at a specific location was made. For example, a dozer was assumed to operate and emit noise for 90% of the time at a location. A haul truck was assumed to operate for 10% of the time at a location along a haul road. An expected percentage time of operation and a maximum expected time of operation were predicted for each source. (The percentage time of operation was set to not exceed 90% of the time.) These are given in Table 1.

Table 1: Source Operation Times

Source	Operating	Percentage
	Expected	**Maximum**
Dozer D11	90	90
Loader	50	90
Shovel	50	90
Drill	90	90
Haul truck Pass-by	5	10
Haul-truck high idle	5	10
Water Cart	10	50

3.2 Time Weighting

The next part of the study was to determine the effect of this variation in period of operation at a receiver location - which is the basic part of the paper. It was noted in the statistical sound levels measured at the receiver locations that there was a similar difference between the descriptors on each of the attended measurements taken.

Location C was subject to noise emissions from other mines surrounding the area. This made its recorded sound level results indicative of the variation in sound level from mining operations. The variation in levels noted was as given in Table 2 below. It is noted that there is a difference between the differences for daytime and night-time. To simplify the assessment procedure, one set of differences (or weightings) was used. The values of the weightings used in the assessment were as follows, with the Reference level being L99:= 0 dB.

L90 = +2 dB(A); L5 = + 5 dB(A) L10 = + 11 dB(A);

L05 = + 14 dB(A); L01 = + 15 dB(A)

These values under estimate the differences for daytime and overestimate them for night-time, as reference to Table 2 indicates. These nominated differences were then applied as a weighting to the calculated receiver sound levels to obtain the expected average (or maximum expected) sound level caused by the varying time operation of the noise source. These were used to account for the variation over a whole day.

Table 2: Variation between Statistical Descriptors : 13 to 17 February 1995
(372 measurements at 15 minute intervals)

Period	Variability	L1-L99	L5-L99	L10-L99	L50-L99	L90-L99
Overall	Mean	19.4	14.4	12.0	5.7	1.7
	SD	6.6	5.9	5.5	3.3	0.9
	AVEDEV	5.4	5.2	4.9	2.8	0.7
Night: 13 to 14 - 2200 to 0700	Mean	11.8	8.2	6.7	3.2	1.2
	SD	7.0	5.2	4.3	1.2	0.4
	AVEDEV	6.2	3.9	3.0	0.8	0.3
Day: 14 - 0700 to 2200	Mean	24.0	19.0	16.7	8.9	2.5
	SD	3.9	3.8	4.1	3.6	1.2
	AVEDEV	2.9	2.9	3.3	3.1	1.0
Night: - 14/15 2200 to 0700	Mean	15.0	10.9	8.8	3.6	1.3
	SD	6.4	5.4	4.6	1.3	0.4
	AVEDEV	5.5	4.6	3.9	1.0	0.3
Day: 15 - 0700 to 2200	Mean	20.5	15.6	13.1	5.9	1.8
	SD	4.9	4.9	4.7	2.6	0.7
	AVEDEV	3.6	4.2	4.1	2.1	0.5

4 RESULTS OF CALCULATIONS

Table 3 gives the result table for Year 10 for Receiver Location C. The table shows which equipment is at each of the 17 nominated source locations, its predicted sound level for continuous operation, and its expected percentile contribution to the sound levels at the receiver. Not all of the table is shown because of space limitations - there were a total of 89 sources used in this particular calculation.

For example in Table 3, Column 1 shows the source location. At source location 2 there are four sources. The "CAT passby" has a received sound level of 36.6 dB(A) at Receiver Location C. Its expected average time of noise emission from this source location is 5%, while its expected maximum time of noise emission is 10%. The effective sound level from this CAT passby for its expected average time of emission is given as L%Ave = 22.6 dB(A), and for its maximum time of emission is given as L%Max = 25.6 dB(A).

The source contributory sound levels, as calculated at the receiver locations, were then ranked, with the ranking also being given in Table 3.

Table 3: Noise Assessment of Mine Expansion - Received Sound Levels: Ranked on Average % Time Result

Source Location	Source	Received Sound Level - dB(A)				
		100%	Expected Time %	Maximum Time %	Effective Level L% Ave	Effective Level L% Max
2	Dozer D11	34.9	90	90	32.9	32.9
2	CAT pass	36.6	5	10	22.6	25.6
8	Dozer D11	21.7	90	90	19.7	19.7
2	Water Cart	29.9	10	50	18.9	24.9
1	Dozer D11	20.3	90	90	18.3	18.3
14	Dozer D11	20	90	90	18	18
2	CAT high	31.6	5	10	17.6	20.6
3	Water Cart	26.3	10	50	15.3	21.3
3	CAT high	28.4	5	10	14.4	20.4
15	Dozer D11	14.3	90	90	12.3	12.3
4	CAT pass	25	5	10	11	17
14	Euclid	24.2	5	10	10.2	16.2
5	Shovel	15.1	50	90	10.1	13.1
8	CAT pass	23.7	5	10	9.7	15.7
9	CAT pass	23.3	5	10	9.3	15.3
5	Loader 501	13.4	50	90	8.4	11.4
~	~	~	~	~	~	~
Total		42.1			34.5	35.8

5 ASSESSMENT OF YEAR 10 SCENARIO PREDICTED SOUND LEVELS

Table 3 shows that the total sound level for all sources operating all of the time would be 42 dB(A), although the maximum contributed sound level from any source is 37 dB(A) - from the Caterpillar 230 tonne haul truck at Location 2 - on the haul road to the top of the overburden emplacement. If all of the percentage times of operation were added for all sources, there would be more than the total number of mobile items operating.

Table 3 also shows that for the expected time allocation of sources at the nominated locations, the total sound level is 35 dB(A). For the maximum expected time of operation, the total was 36 dB(A).

By considering the ranking of Table 3, the dominant sources and locations are identified. The table shows that the main locations for noise emission are the upper benches of the New Pit and the main Mine overburden emplacement.

The analysis was repeated for the four different receiver locations, and then for different weather conditions. From the results, sound level contours were developed for both all equipment operating 100% of the time, and for equipment operating for the expected percentage of time. There was up to 10 dB(A) difference in predicted sound levels between the two cases for the same conditions. The analysis could therefore be taken on the basis of the total predicted noise emission, or on the basis of the proposed time-weighting method. In the case of this study, the predicted sound levels were considered acceptable for the types of receiver area. These were either rural residential or residential in an area adjoining an industrial area - as Locations C and D were surrounded by coal mines.

A summary of the total results at each of the four receiver locations is given in Table 4, below.

Table 4: Predicted Residential Receiver Location Sound Levels from Mine Expansion at Year 10

Receiver Location	**Contributed LA10 - dB(A)**	
	Average % Time Sound Level	**Full Time Sound Level**
A-Jerry's Plains(north)	23	29
B-Jerry's Plains	16	23
C-Lemington	35	42
D-Carrington Stud	29	37

Note: Contributed sound levels are given for both the expected average amount of time (Average % Time) that a source is at any source point in both the New Pit and the main Mine Pit, and for the case where all sources at each location are operating all of the time (Full time) The results are calculated for $0^{o}C$ per 100m rate and calm conditions.

These results showed that the predicted sound levels for the expected average percentage time sound levels were less than the existing measured sound levels at all locations, except for receiver location C where the predicted level was at the middle of the range of measured sound levels.

6 REVISION / DEVELOPMENT OF SUGGESTED METHOD

On review of the data given in Table 2, it may be more appropriate to either use two sets - one for day-time and one for night-time; or if one was only concerned with night-time assessment, which occurs most of the time, then the night-time figures need only be used.

The suggested numbers, in this case, for use in time-weighting would be:

Day	1%: 15 dB(A);	5%: 15;	10%: 12 dB(A);	50%: 6 dB(A);	90%: 2 dB(A)
Night	1%: 10 dB(A);	5%: 9;	10%: 7 dB(A);	50%: 3 dB(A);	90%: 1. dB(A)

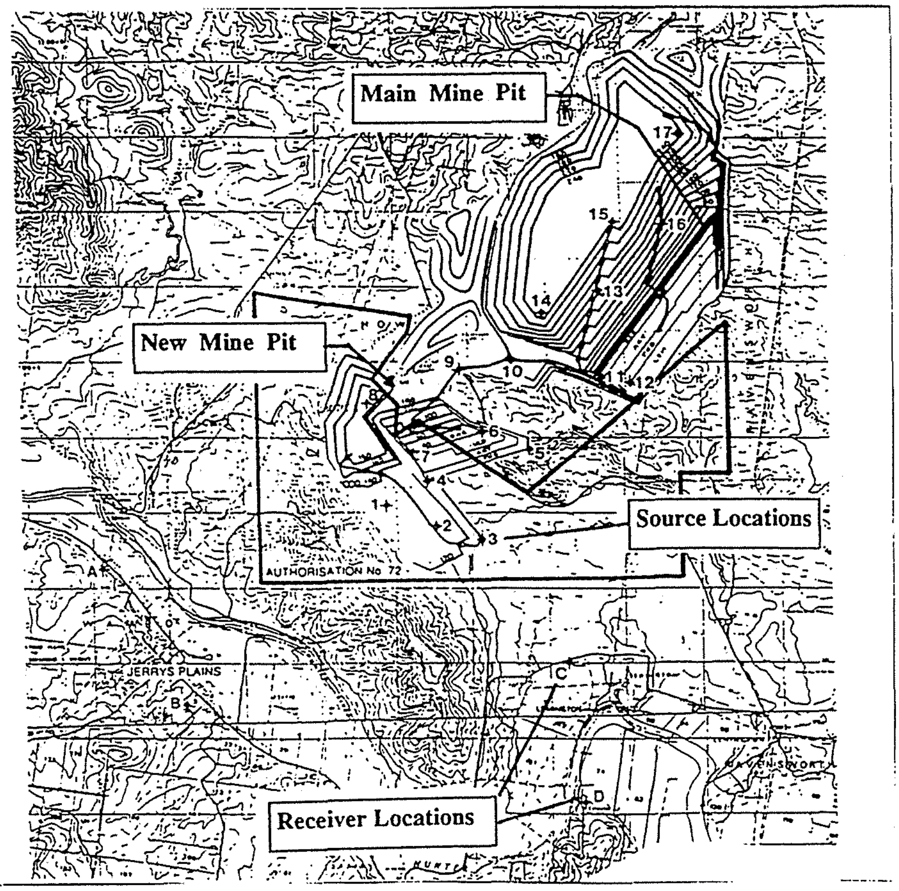
However, the basic premise of the method is that the time weighting used in the assessment of a noise source match the variability in statistical sound levels measured at the receiver locations. This is appropriate to the Hunter Valley of NSW where the types of sources and activities proposed are already occurring and can be readily measured.

In the long run, the intent has been to try to make a realistic assessment of the noise immission of residential receivers based on the existing variation from similar sources. The full-time method can also be used to provide an indication of the predicted results if the time weighting approach is not accepted, but a greater time would need to be spent assessing source locations in more detail, which negates the simplistic approach proposed.

ACKNOWLEDGMENT

The author would like to thank BHP for the time to prepare and present the paper. He would also like to thank Mr. Allan Lawson for his assistance in review and preparation of the paper.

Figure 1: Plan of Area showing Mine Layout, Receiver and Source Locations



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LINKING TRAFFIC NOISE ASSESSMENT WITH ROAD NETWORK PLANNING

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ABSTARCT

Transport planners use procedure known as Travel Demand Modelling to forecast future flows along all links in a defined transport network. This modelling occurs at both the city scale and also at the local scale, the latter particularly for local area traffic management schemes. The output from Transport Demand Modelling are the road network link flows, speeds and sometimes traffic mix, for future scenarios which are tested. These scenarios may be different land use patterns which are being evaluated or different planning time periods.

In fact the output form the travel forecasting procedures is almost half the input required for road traffic noise modelling This paper describes an ongoing project at Griffith university for linking the output from Transport forecasting modelling procedures for road systems with land use information through a simple GIS system. This can lead to automatic and routine evaluation of the road traffic noise condition along every individual link in the road network, and to state of the road noise environment reporting for the whole network under consideration.

LINKING TRAFFIC NOISE ASSESSMENT WITH ROAD NETWORK PLANNING

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INTRODUCTION

The efficiency of the transport network influences the level of mobility and access to facilities in an area. However the use of these networks by vehicular traffic has significant and well-known environmental consequences. These effects include noise, air pollution, as well as problems such as congestion, safety and lack of access. Transportation planning studies are commonly used within the various spheres of government: to estimate current and future transport needs; to distribute traffic flows along all links in the transport network; to determine deficiencies in the current transport system to accommodate future travel patterns; and to evaluate alternative transportation strategies. These studies generally use procedures known as travel forecasting, or travel demand modelling. In most cases, especially for large comprehensive transport planning, the activities involved in the travel demand modelling include: surveying the current travel behaviour of the people living and working in the study area; developing mathematical models which relate travel patterns to details of household structure, income, car ownership, etc.; predicting the distribution of travel patterns across the study area; and assigning these patterns to the transport network to predict flows on each link of the transport network.

Normally various scenarios utilising different land use patterns, different forecast periods, and/or different transport network configurations are tested. While the environmental consequences of each scenario are different, generally there is no environmental assessment at this stage of planning. If environmental evaluations were considered at this stage it would be possible to select a preferred scenario that satisfies pre-defined environmental criteria and policies. Current practice for tackling these environmental concerns usually involves an environmental impact assessment (EIA) of planned projects for specific links where changes will occur. These are necessary and EIA still has an important role to play in providing ameliorative measures for the worst environmental implications of any project. However, the problem with EIA of transport link proposals is that it is likely to be carried out many years after the transport network plans are in place, at which time any modification at the network level is impossible to contemplate. At present no systematic method exists for predicting these impacts at the network planning stage (Losee and Brown, 1996). A system for linking the output from travel demand modelling procedures for road systems with land use information through a simple Geographic Information System (GIS) is under development at Griffith University. This system is intended for use by transport planners in estimating the environmental consequences of transport network planning proposals. The eventual system will contain modules for estimating the environmental impacts of road traffic, namely, noise levels, near-field air pollution, energy and greenhouse emissions, stormwater and visual effects. This paper describes the development of the noise module and its application to travel demand modelling conducted in Canberra, Australia.

TRANSPORT MODELLING AND GIS

Studies have shown that traffic noise modelling can be incorporated with travel demand modelling (Brown and Patterson 1990, Taylor and Anderson 1988). In their study, Brown and Paterson (1990) demonstrated that the noise impact of a road network could be explicitly included in travel demand modelling at the time the network is being developed, modelled and tested. This study introduced a novel approach to the assessment of the impact of traffic noise by recognising that much of the data needed for estimating the noise emissions from road traffic are available from the output and part of the input data to the travel demand models. The rest of the data required includes detailed information on building setbacks, topography of the area, weather, etc. In some areas most of these additional data may reside in some form of database or can be obtained from field surveys and orthophoto maps. This recognition highlighted the potential to estimate noise emission levels from road traffic by integrating output data from transport planning models and land use information.

Since the earlier work of Brown and Patterson (1990), the advent of improved GIS technology and data availability have resulted in a change in the way spatial data is viewed and analysed. The capabilities of GIS (e.g. spatial analysis tools, graphical and data integration capabilities) have resulted in the rapid growth in usage of GIS technology in transport planning (Affum and Taylor 1996). These capabilities can provide an enhanced capacity for the management and analysis of spatial data and the creation and improvement in the way environmental information is produced. In particular, its display capabilities provide an improvement on the evaluation and quality control of data and results. Its spatial and data integration capabilities ensure more extensive access to geographic information throughout an organisation enabling greater responsiveness and ability to explore a wider range of alternatives. GIS also has features for working on different levels of data aggregation, although at the moment there is the need to develop a procedure for automating such work process (Nielsen and Jocabsen 1996). This is important especially when working with data provided on a detailed level of aggregation that need to be adjusted according to the planning context in order to make it possible to shift between different levels of aggregation. For example, with GIS it may be possible to estimate the environmental impacts of traffic at the micro level and aggregate it to obtain the effect at the macro level. The increasing trend in the use of GIS in transport planning will also make it possible for more of the zonal based data (e.g. land use and demographic data) to be available for modelling the environmental impacts of traffic. It is anticipated that GIS will form an integral part of transport planning modelling in the future (Jensen and Ferreira 1992), and hence in environmental modelling of transport plans.

MODELLING NOISE IMMISSIONS

In order to estimate the level of road traffic noise considerable effort has been expended over the past decades in developing noise prediction models. These models use traffic flow, mean vehicle speed, composition of traffic and road surface type to estimate the noise emitted along each link. A widely used model in Australia is the CoRTN'88 model (UK DoT 1988) and most of the input variables for this model can be directly obtained from the output of travel demand modelling. For example, in the studies by Brown and Patterson (1990) and Taylor and Anderson (1988), output from transport modelling was used to predict link-based noise emissions. The results from such studies, particularly when displayed in graphical format, readily depict those links in the network that produce high noise emissions.

However, noise emissions are only a problem if they affect a sensitive land use. In reality therefore it is the noise levels that impact on sensitive adjacent land uses such as schools, hospitals and dwellings, most appropriately termed noise immissions, that are of importance to planners (Brown and Patterson 1990). Predicting noise immissions, however, requires the collection of additional sets of data in greater detail at a level which is normally not present in the data used by transport models. The only way this additional data could be obtained in the past was by direct field surveys or use of aerial photographs. This entails additional cost which, for a large study area, may be quite substantial. Currently, the increasing availability of digital cadastral data in urban areas and the development of land information systems combined with GIS, the possibility exists for the automatic extraction of the additional land use data. The Griffith University study is investigating the use of land use data from available digital data to facilitate noise immission modelling for transport networks.

POSSIBILITIES FOR COLLECTING THE ADDITIONAL LAND USE DATA REQUIRED FOR NOISE IMMISSION MODELLING

As stated above, to compute noise exposure to sensitive land uses (noise immissions), additional data on land use is required. The additional data required include building setbacks, building type, land use type, roadway gradient, road surface condition, etc. There are several ways by which this land use information can be collected including: field survey and measurements, manual extraction from town plans and maps, use of existing digital land information data and extraction from raster images.

Manual field survey involves the direct measurement of the positions (especially the setbacks) of buildings and the type, nature and use of the building such as multiple unit dwellings, etc. Accurate results and measurements can be obtained from field surveys, however this method of data collection is expensive and time consuming. For large study areas the method may be impractical.

Current technology has led to the establishment by local authorities of land information systems (for example the Brisbane City Council's Brisbane Integrated Map of Assets and Property (BIMAP) system (BCC, 1995). These systems contain extensive digital information on land uses in an area. They were developed in order to computerise previously manually prepared paper maps and to automate the process of information retrieval. Normally, the land information contained therein is stored in themes or map layers. Each theme may contain one specific land feature or a combination of features. While some of these systems are in GIS format, most are not. However, in all cases, the information it contains can be easily converted and brought into a GIS. Once the information from each layer is converted into GIS format, data overlay and integration processes can be used to extract pertinent data sets. For example, the street centre-line layer may be overlayed on a cadastral base layer to compute the propagation distance from the street to the property boundary line. However the setback of dwellings from the property boundary line (required for computing noise immissions) will still be missing. To overcome this difficulty a model may have to be developed to predict the dwelling setbacks from the property line. This could perhaps be based on the location, nature and type of dwelling using manually collected field data. Predictions from the model could then be used in conjunction with the estimated distance to the property line to compute the propagation distance. Some land information systems such as BIMAP may also contain information on the general topography of the area in the form of contour maps. Again the street layer could be overlayed on this contour layer in order to estimate the slope of each link. During the overlay process the GIS spatial analysis capability could be used to capture the spot heights at each node. The computed spot heights and the link length are then used to compute the slope of the link. Thus with these systems and using GIS, detailed information on building setbacks, topography, etc. may be able to be automatically captured.

Another potential method of capturing the land use data involves the electronic reading of data from aerial photographs (aerial images) with a GIS. Images may be digitally scanned and registered so that they can be used as a basis for geo-referencing. This method can be expensive depending on the size of the study area. The attractiveness of this method depends on the availability and quality of the aerial photographs. Several authorities have already started the process of aerial photography of their areas, and this suggests the use of these images to capture the spatial components of land uses may be increasingly possible.

THE NETWORK ENVIRONMENTAL MODELLING SYSTEM

The current implementation of the transport network environmental modelling facility at Griffith University is that developed by Losee (Losee 1996). This model termed System for Modelling the Impacts of Roads on

Communities (SMIRC) is a system for estimating the environmental impacts of transport plans using MapInfo. It is intended that the implementation, when fully operational, will have options for the estimation and analysis of the following:

- road traffic noise;
- near-field air pollution;
- energy and greenhouse emissions;
- stormwater run-off; and
- visual effects.

Each of the above activities is carried out using a separate module. At present only the noise module is in place. The modules are embedded in a form of tool-box like structure controlled by an "organiser" menu under a shell (see Figure 1). It is developed based on the desktop mapping system, MapInfo, using the MapBasic programming language. It is developed as a tool for anyone involved in transport planning to be used in testing different scenarios of transport plans. It offers a simple interface and is completely mouse controlled against a windows background. The implemented system uses the standard MapInfo tables. It presents a menu from which the user manages the road network data, makes any assumptions required, executes the modules and manipulates the output displays.

Figure 1: Schematic diagram of the conceptual architecture of the current implementation of the program. Only the CoRTN'88 Noise module is implemented.
(Source: After Losee and Brown, 1996)

Design considerations

It has been shown that most transport planning and modelling activities can be accomplished within a GIS framework (Affum and Taylor 1996, Shaw 1989). However, its actual use and complete acceptance by transport planners will take some time due to the reluctance of planners to immediately abandon well-established travel demand models. At the same time, several methodologies and approaches have been adopted in integrating GIS into transport planning applications (Taylor 1995, Trinidad and Marquez 1994). In particular, to integrate GIS and transport planning for environmental evaluation purposes, two main methods are in use: the complete integration method and the add-on approach. The complete integrated approach involves development of new forecasting packages or modifying existing ones to run entirely within the GIS to produce outputs useful for environmental evaluation as well as transport planning. As a unified package, it might be more effective for transport planners and encourage them to undertake environmental planning. The main disadvantages of this method relate to the time and effort required in re-development or modification of the existing model. Another potential disadvantage is the extent to which transport planners may be committed to their existing forecasting models. The add-on approach aims to augment existing transport models. It merely provides an extension to the models already being used, and hence planners would only need to learn how to use the add-on program. Its main disadvantage involves the issue of compatibility and how to design a single program to fit on each of the many different transport models in use. The Griffith University study uses the add-on approach to develop the system to serve as an add-on module to existing transport models. This approach was adopted so as to enable its use by as many transport planners as possible irrespective of the transport model in use. The add-on method ensures that it can be used with any transport model. It eliminates the need to re-program existing transport demand models.

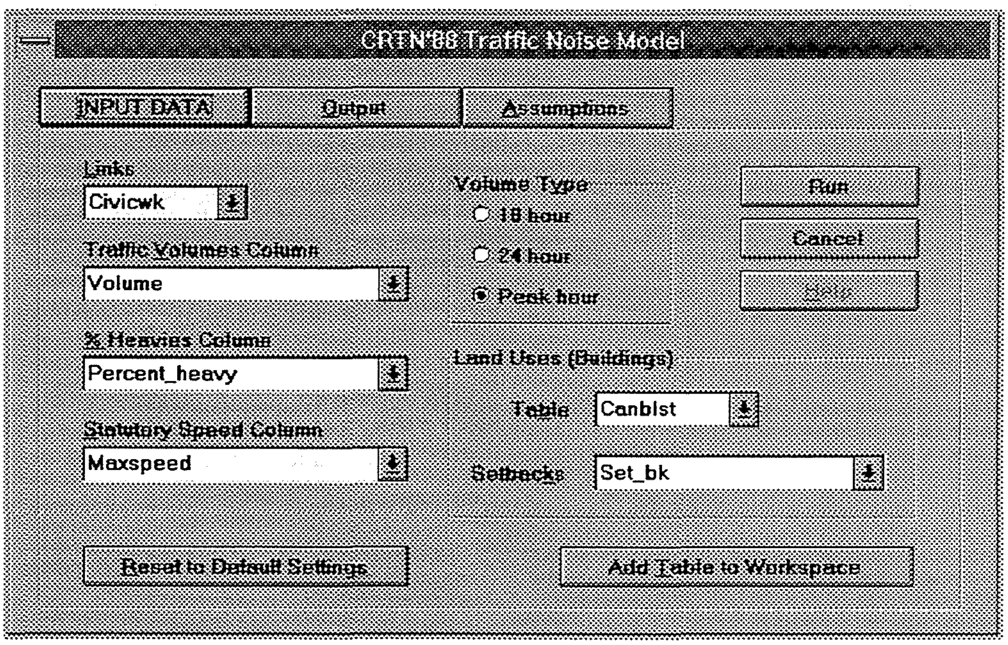
Another design consideration was the type of GIS software to use. In this regard a survey of transport planners was conducted to find out their knowledge of GIS, the type of GIS they use and its availability to them (Losee and Brown 1996). The survey results indicated that the most popular GIS among transport professionals were MapInfo and PC ARC/INFO. Since MapInfo is inexpensive, easy to use and could perform the problem at hand equally well as does PC ARC/INFO it was decided to initially developed the system as a MapInfo application, and later, perhaps, as a PC ARC/INFO application..

The Network Noise module

The noise module was implemented based on the UK CoRTN'88 method (UK DoT 1988). The current developed prototype version employed a simplified version of the above method using basic assumptions to predict $L_{10,\,18h}$ dB or $L_{10,\,1h}$. It is noted that only two data types need to be geo-referenced to shift the traditional noise modelling approaches to a GIS-based approach. These are a cadastral map containing vital information on the location and description of the various land use types and the road centre lines defined as interconnected links. Each link is defined by two end nodes. The remaining data sets do not need to be geo-referenced, but instead linked to the GIS using traditional database management approaches. The program makes use of the two main data bases namely the modelled street network and the additional building and land use data (BUILDINGS TABLE). The street data is generated as a MapInfo map layer using the x and y coordinates (termed LINKS TABLE). The attribute data associated with each link in this layer are the assignment results from the transport model. Any additional attribute may be added.

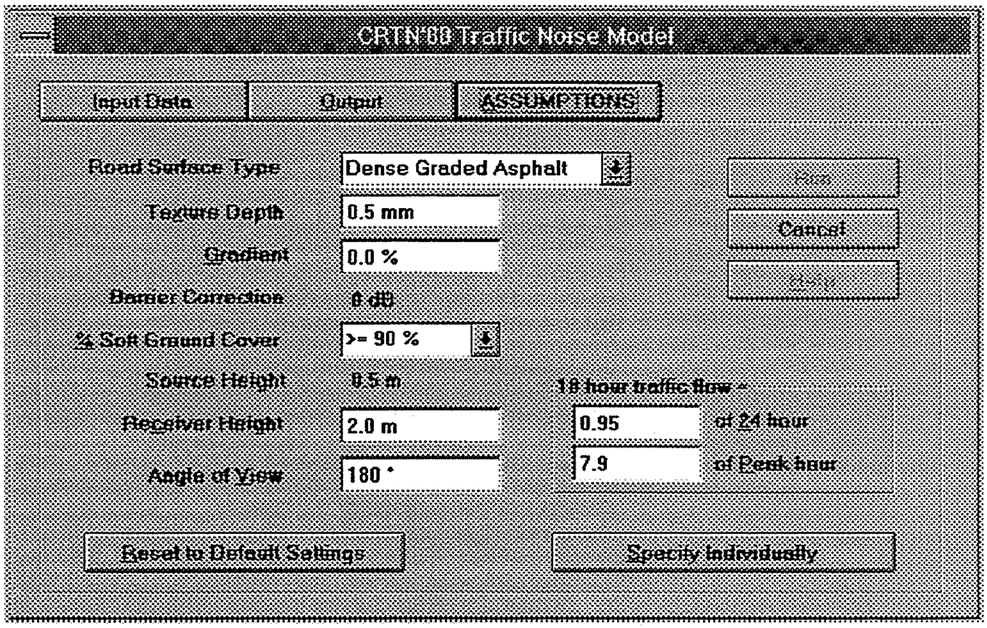
When activated from the "organiser", the noise model follows the pattern of loading, checking initial settings and assumptions, running the program and presenting progress reports and results. During loading, the program runs the MapBasic noise application and displays the pop-up menu shown in Figure 2. It contains three main options: "input data", "output" and "assumptions". The nature and parameters on the menu changes depending on which one of the three options is selected. The "input data" form (see Figure 2) is used to select the MapInfo TABLES and the relevant data parameters to use in the calculations. The "assumption" view provides a form for specifying the parameter values to use in the calculations (see Figure 3). The "output" options provide a form for specifying the types of output desired namely noise emissions, immissions, emission and exposure reports and whether the $L_{10, 18h}$ or $L_{10, 1h}$ noise level is required.

Figure 2: Input data form for the noise module



The model is designed in such a way that when loaded it compares field names in the LINKS and BUILDINGS MapInfo TABLES to a series of key words to guess and determine the locations of network and building parameters to use for calculations as shown in Figure 2. The user then confirms and/or make changes to "guessed" parameters shown in Figure 2. Changes to basic assumptions are also made at this stage. When satisfied, the user selects the "Run" button to execute the program. During the execution process, the program periodically presents progress reports on the execution process in the form of progress bars, percent completion and elapsed time. Once completed, the model indicates the displays that have been created and the user can access these through the organiser. The displays include the noise emissions, immissions and summary statistics of the number of dwellings exposed to various noise levels. Typical examples of such output displays are as shown later in this paper.

Figure 3: Assumptions input form for the noise module



CASE STUDY

The program was used to model noise levels for the Civic area in Canberra, the Australia Capital Territory. This area lies in the central heart of the city. While actual modelled transport data and land use data have been used in this case study, no attempt has been made to validate the outputs, and results presented here are intended to illustrate the modelling process rather than to provide accurate noise exposure data for Canberra. In particular, while future transport flow patterns have been used, only current (1994) land use data have been input to the model. Estimated traffic for the future year 2016 was provided from the then ACT strategic model of Canberra. The traffic was predicted using the Transtep travel demand model. Hence the noise levels estimated are for the future year 2016 based on the predicted travel volumes. The model produced AM peak flows, and these were converted to 18 hr flows by multiplying by a factor of 7.9 based on 1994 traffic flow estimates. The data supplied were in two separate ASCII files namely the node file capable of being used to generate the model network and the model output containing the details of the assignment results. The area contained 354 links. The proportion of heavy vehicles was unknown. Estimates of between two to ten percentage heavy vehicles were used depending on the type of road. A value of ten percent was used for freeways, five percent for arterial roads and distributor roads, and two percent for collector and other types of road. The road surface was assumed to be dense asphalt with zero percent gradient assumed for all links. Accurate values could be used if available.

Collection of land use information

The additional land use data, namely the location of sensitive land uses and the propagation distance between the source and receiver (ie. setback of buildings from the roadway) were measured manually from 1:2500 scaled orthophoto maps of Canberra. These maps contain detailed outlines of the road network and boundaries of dwellings. The distances (setbacks) from the road to the first row of residential buildings (dwellings) were measured from the maps. In all, setbacks of 1740 frontage dwellings were measured. Each of these buildings was associated with a link. The assumption here is that only traffic on the link for which the dwelling is associated contributes to the noise level at the dwelling. This assumption may result in underestimation of the noise levels at some dwellings especially for those close to two heavily trafficked roads such as those near the intersections of two major roads. No automated procedures have been used to obtain the additional land use data at this stage.

CASE STUDY RESULTS

Noise emissions

Figure 4 shows the noise levels generated by each link as obtained from the program grouped in four ranges. The noise emissions are computed at a distance of 10 m from the source. In the discussion that follows, due to the lack of universally adopted standard for road traffic noise in Australia, cut-off values of $L_{10,\ 18h}$ of 63 dB and 68 dB are adopted to define high and excessive levels of noise (ie. levels from 63 dB to 67 dB are referred to as high while 68 dB and above are classified as excessive). Predictions for the year 2016 showed that 144 of the links emitted excessive levels of noise while a further 150 links emitted high levels. These accounted for 83 percent of the links (194 links) in the study area. As expected, they are the roadway links that carry heavy traffic flows.

Noise immissions

Figure 5 on the other hand shows the noise distribution pattern when the impact of the noise on adjacent dwellings are taken into consideration. It shows where noise exposed dwellings are distributed across the network and represents what is termed noise immissions. The links are shaded according to the maximum noise immission at the facade of any dwelling located along the link. It should be noted that some dwellings along any particular link may experience a lower noise level than those depicted in the figure depending on the distance of the dwelling from the road centre line. Link labels show the number of dwellings exposed to excessive noise levels with the number exposed to high levels of noise in parentheses. The figure indicates that excessive noise problems are limited to dwellings on 40 links while those on an additional 51 links experienced high noise levels. Scrutiny of the characteristics of these links and dwellings indicates that high and excessive noise immissions occurred where the link carries high traffic flows and the adjacent residential land uses are located within short distance from the roadway.

Figure 4: Noise emissions from each link at a facade 10 m from road centre line. No facade correction is applied. (Number of links in each class intervals are given in brackets)

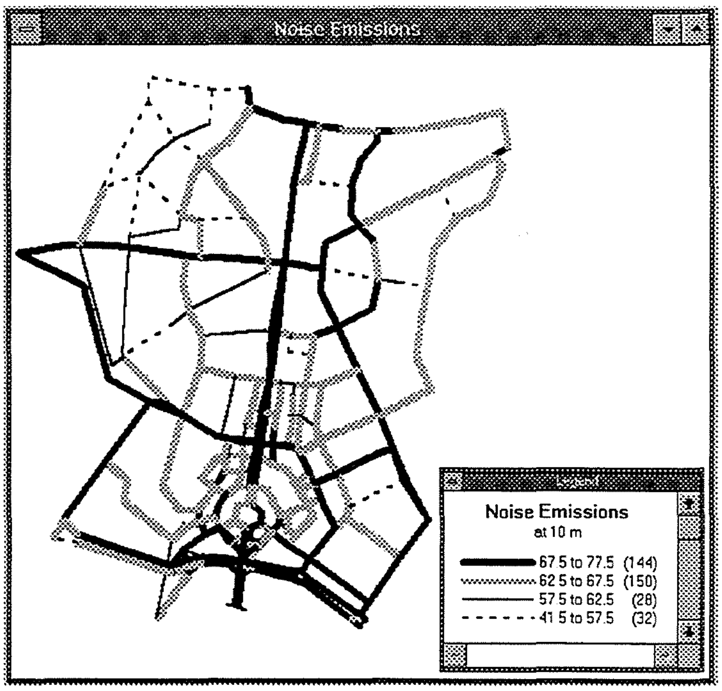
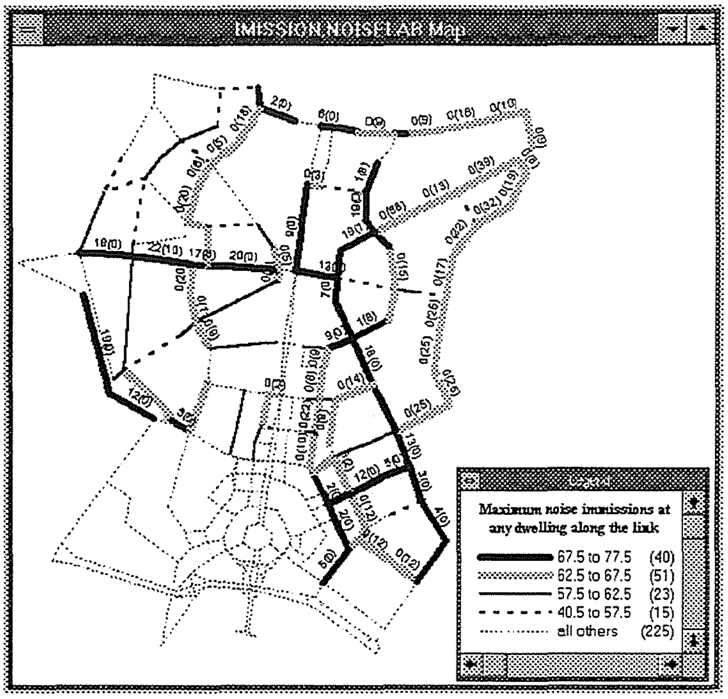


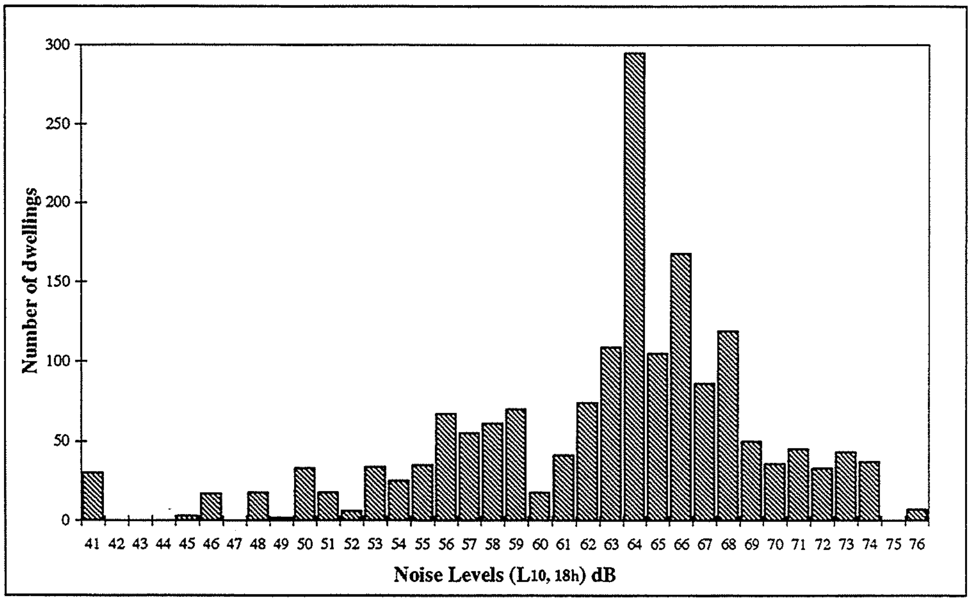
Figure 5: Noise immissions from each link to a facade of dwellings. Facade corrections applied. Link labels show the number of dwellings exposed to excessive noise levels with the number exposed to high levels of noise in brackets.



Comparing the results shown in Figures 4 and 5 indicate that most of the links generating high levels of noise emissions (as shown in Figure 4) did not result in any noise immissions to dwellings along the links. It is found that dwellings on only 91 of those links will be exposed to unacceptable levels of noise immissions. Of course, there may be other sensitive land uses not considered in this case study, such as schools, hospitals, non-air conditioned offices, etc., or even pedestrians, or passive open spaces. Developing the model to account for these would be a simple exercise.

Figure 6 shows the distribution of traffic noise levels at the front facade of dwellings in the study area. The number of dwellings in the area that would be exposed to high and severe noise levels will be 370 and 763 respectively. The results suggest that in 2016 about 65 percent of dwellings fronting the network used in the study will be exposed to high levels of noise. Using census data on the number of residents in each household, one could estimate the total number of people that will be exposed to high levels of noise.

Figure 6. Traffic noise distribution at facade of dwellings in Civic area (Canberra) for 2016



CONCLUSIONS

A GIS-based system for the estimation of noise emissions and immissions has been described. The system is simple to use and provides outputs in an easily understood maps and graphical displays. The system has demonstrated that, with GIS, it is possible to use output from transport forecasting models linked to additional land use data to estimate the noise impacts of road traffic. GIS is used due to the need for spatial analysis, improved display capacities and the need for data integration. The spatial analysis capability of the GIS ensures that the system can be used at any spatial scale levels. The output from the noise module of the Griffith University system gives the transport planner an entirely new insight into the noise effects of transport proposals

The noise modelling procedure can be used for the classification of roads, placing environmental flags for noise on land uses, and for State of Environment reporting. Traditionally road classification has been based on the amount of traffic carried. The ability of the program to readily estimate noise impacts on a link by link basis provides an excellent incentive for it to be used for road classification based on the noise immissions in the same way as traffic flow is used for classification today. The system may also be used to isolate and place special noise markers (flags) on parcels of land adjacent to links that have potential high levels of noise exposure. Equally important is the capability of the system to be used to provide a State of Environment reporting on road traffic noise levels in the community.

The system described is still underdevelopment. Efforts are currently under-way to develop models and protocols to automate the estimation and collection of the additional land use data using the Brisbane City Council BIMAP system as the source of land use information.

ACKNOWLEDGMENTS

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PRACTICAL ASPECTS OF THE INVESTIGATION OF ENVIRONMENTAL VIBRATION COMPLAINTS.

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ABSTRACT

Increasing environmental awareness has resulted in a large increase in the number of complaints about noise and vibration. Although many guidelines exist for noise and many excellent texts are available for vibration theory, few documents covering the practical aspects of investigating environmental vibration are readily assessable. This paper seeks to redress the balance by discussing the investigation of both, the fear of vibration building damage and the effects of whole body vibration from within buildings

1. INTRODUCTION

Vibration is a mechanical oscillation, the to-and-fro movement of any physical body. Oscillation is a word used in non-technical language, along with shake, tremor, wave, spring, swing, sway, reel, rock, roll and many others that describe this to-and-fro movement.

Vibration can be the cause of discomfort, pain and annoyance to humans and damage and destruction to sensitive equipment or buildings. We have all sensed vibration at some stage, whether it is when travelling over a bumpy road in a car or bus, meeting turbulence in an aircraft, the effect of heavy waves in a ship, using hand tools such as an electric drill or lawn mower or just walking over a springy bridge or floor. However, it is important to remember that not all vibration has a negative effect. In fact, without vibration we would cease to exist. We need the vibration of the beating of the heart and the to-and-fro motion of the lungs that pumps the air we breathe.

We use vibration as the basis for machines such as clocks and watches. In industry we use vibration for sorting, mixing, welding or moving components. Vibration is used for medical purposes and as a means of giving pleasure. For example, new born babies are comforted by being rocked in the arms of their parents, teenagers are attracted to oscillating fairground rides while the older generation prefer the gentle oscillation of a rocking chair.

The plant Earth is in a constant stage of vibration even though this is usually far below human sensitivity and can only be recorded with specialist equipment. Observation and analysis of the vibrations of the sun reported by Harvey (1995) allow us to probe its interior structure and dynamics to test and expand our understanding of physics and astrophysics.

Unwanted vibration can be the cause of dynamic loading on buildings that could be the result of damage, ranging from 'cosmetic', hairline cracks in plaster, to major impairment to the building superstructure. In very rare and extreme cases, vibration can be the cause of building collapse. Secondary effects of vibration, such as foundation settlement can also be the cause of major damage to buildings and should not be disregarded out of hand.

2. MAN MADE VIBRATIONS

Although earthquakes are a major cause of destruction and death, man made vibrations can be the cause of a high degree of concern in the built environment and this regularly results in complains. The vibrations are generated, for example, from:- Piling, building demolition, ground compaction, blasting, drop forging, trains, road traffic vehicles and rotating machinery.

People are very sensitive to vibration, when it is perceived in buildings their usual first reaction is (understandably) concern about the structural damage to the property. Often the magnitude is far below any reasonable risk of damage and on occasions, a vibration assessment and reassurance by an experienced and competent person, is all that is required to alleviate the concern - see Scannell (1993).

Before the damage risk can be evaluated it is useful to have an understanding of the basic physical principles involved.

Damage to buildings, such as cracking, can occur as a result of dynamic stresses (a variable force per unit area) and strains (a ratio of the change in length and the original length) of the structure. However, history has shown that one of the main causes of building collapse is due to poor design and man made error - see, for example, 'Why buildings fall down', Levy and Salvadori, (1992) .

Dynamic strains are directly related to the peak particle velocity of the vibration, and the latter is the parameter that is normally measured. The natural frequency (the frequency of free oscillation, where the vibration becomes amplified) and the degree of damping (which gives an indication of how quickly the vibration decays) of the building and building elements will also have an effect and should be taken into account where reasonably practicable. Accurate data are not easily obtained and for a first approximation the natural frequency is often close to 50 divided by the height of the building (in metres).

Unfortunately, damping is even more difficult to predict than the natural frequency, even as a first order approximation, as it is a function of the building construction and to some extent the intensity of the vibration. Damping can be found from the time history of the vibration response of the building but it is rare to obtain an opportunity to record such data.. Measurements reported by Medearis (1976) reveal a wide range of damping coefficients for residential structures with an average of 5% of critical. Where firm data are unavailable, this value is can be selected for initial estimates involving taller engineered structures.

3. STRUCTURAL DAMAGE CRITERIA

It is not possible to define universal criteria that could be used to predict structural damage to buildings because of the many variables that are involved.

These include:

(i) the condition, dimensions, foundation details and natural frequency and damping of the building;

(ii) the amplitude and frequency of the vibrating source;

(iii) the intervening geological strata (eg. the soil type).

The criterion, therefore, must be site specific, based on the knowledge of the many factors given above, current guidelines and the experience of previous case history.

No Australian Standards are available giving guidance to building damage from vibration. However, some useful guidelines to damage criteria from piling vibration are given in the British Standard BS 5228: part 4: (1992). Guideline values have also been issued by, German (1986), American (1980) and Swiss (1978) Standards. The British Standard BS 7385 : 1990 (ISO 4866) covers the evaluation and measurement for vibration in buildings, with part 1 of the Standard providing a guide for the measurement of vibration. In Annex A, the Standard also gives a useful method for the classification of buildings according to their probable reaction to mechanical vibration (unfortunately these cannot be linked to criteria). Part 2 (1993) does provide a simplified, but still useful, guide to criteria in terms of peak particle velocity. These are dependant on the type of structure and the frequency as shown in the table below.

TABLE 1 : VIBRATION GUIDE VALUES FOR COSMETIC DAMAGE (ADAPTED FROM BS 7385: PART 2: 1993).

<table>
<tr><th rowspan="2">Type of Building</th><th colspan="4">Peak Particle Velocity in Frequency Range of Predominant Pulse</th></tr>
<tr><th colspan="2">Continuous Vibration</th><th colspan="2">Transient Vibration</th></tr>
<tr><td>Reinforced or framed structures. Industrial and heavy commercial buildings.</td><td colspan="2">25 mm/s @ 4 Hz and above</td><td colspan="2">50 mm/s @ 4 Hz and above</td></tr>
<tr><td>Unreinforced or light framed structures. Residential or light commercial type buildings</td><td>7.5 mm/s @ 4 Hz increasing to 10 mm/s @ 15 Hz</td><td>10 mm/s @ 15 Hz increasing to 25 @ 40 Hz and above</td><td>15 mm/s @ 4 Hz increasing to 20 mm/s @ 15 Hz</td><td>20 mm/s @ 15 Hz increasing to 50 @ 40 Hz and above</td></tr>
</table>

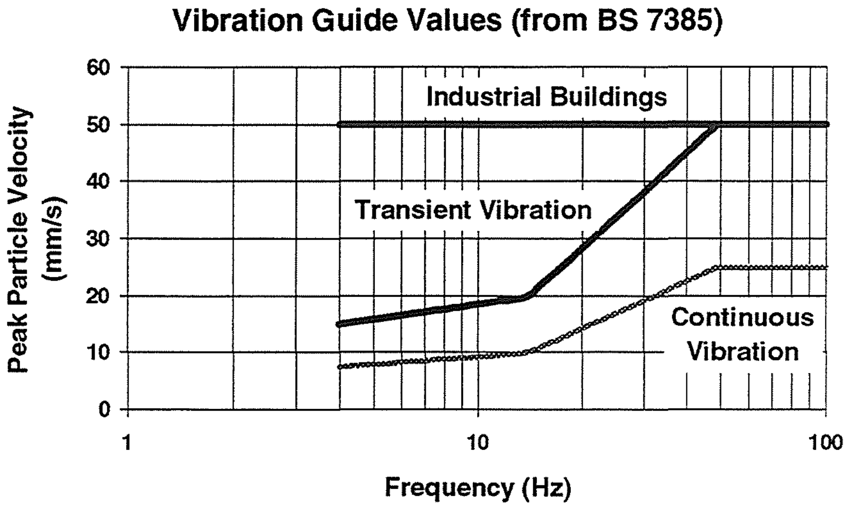

Figure 1. Vibration Guide Values for Cosmetic Damage as given in BS 7385 (1993). See table 1.

3.1 The Effects of Repeated Low Level Vibration.

The degradation of materials or structures due to repeated vibration is known as fatigue. Even if the name is not familiar to occupants of buildings subjected to regular vibration, the principle of fatigue is often raised as a cause of concern. Hence some knowledge of the risks involved is useful for the assessor.

It is not usually practicable to carry out fatigue tests in 'real' situations, as the time scale needed would be too long. Most of the evidence on the effects of fatigue comes from laboratory tests.

The U.S. Bureau of Mines, and the (then) Transport and Road Research Laboratory (TRRL), in the U.K. have conducted studies on full-size houses to determine the effects of fatigue.

The U.S. study by Stagg et al. (1984) involved a test house specifically constructed near a surface coal mine to investigate the effects of blasting). The test house was framed in wood with paper-backed gypsum board interior walls. It was initially shaken in torsion at an acceleration of 2.5 m/s^2 at the structures natural frequency of 7 to 8 Hz. This magnitude of excitation corresponds to an equivalent horizontal ground motion of approximately 12 mm/s. The first fatigue cracks occurred after 52,000 cycles. These 52,000 cycles correspond to approximately 10,000 blasts with five significant pulses each and would take over 30 years to accumulate at a rate of 300 blasts per year.

The study carried out by Watts, (1987) for the TRRL, involved an 80 year old pair of semi-detached houses that were subjected to typical heavy goods vehicle vibration with a 1 second pulse waveform. The vibration was generated on a small area of road pavement approximately 5.2 m from the cellar wall of the house. The maximum amplitude of the vertical component at the house foundation was 2.6 mm/s in the frequency range of 12 - 13 Hz. The structure was exposed to 888,000 pulses simulating the effect of over 3.5 million goods vehicle axles. Of the forty existing cracks that were monitored throughout the tests only five showed sustained change in width exceeding 0.1 mm and a small amount of additional cracking of plaster occurred.

4. HUMAN REACTION TO VIBRATION - BUILDING DAMAGE FEAR

Humans are extremely sensitive to whole body vibration, particularly in the low frequency range (1 Hz to 10 Hz). Although the perception magnitudes vary considerably, McKay (1972) found that values as low as approximately 0.01 m/s^2 r.m.s. acceleration or 0.2 mm/s peak particle velocity (at low frequencies) can be perceived by some individuals. Assessing and predicting annoyance from vibration is covered in the British Standard BS 6472 (1992) and the International Standard ISO 2631-2 (1989). In Australia, a simplified version is given in the Chapter 174 (1989) of the Environmental Noise Control Manual issued by the New South Wales Environment Protection Authority.

As stated earlier, practical experience indicates that, usually, the first complaints that arise are not from the direct effects of the magnitude of vibration on the human body, but from the fear of structural damage to the complainant's property. Not unexpectedly, this is particularly true where the dweller is also the property owner.

Complaints can be expected from 'building damage fear' if vibration is perceived at any magnitude over 0.2 mm/s peak particle velocity (New, 1986). This is true, even though the chance of damage (under almost any circumstances) from magnitudes up to ten times this value is negligible.

An assessment is usually required if a complaint is received about vibration, however, the assessment is quite different depending upon whether the complaint was made due to the direct effect of the vibration or building damage fear as shown below:

4.1 Direct Effect.

If the complaint is regarding the direct effect of vibration on the human body, then a procedure is followed as given in BS 6472. The root mean quad of the acceleration of the vibration is measured at the point of entry to the human body (or the centre floor position). The procedure is covered in more detail in Section 5 of this paper.

4.2 Building Damage Fear.

If, however, the complaint concerns a fear of structural damage then an assessment procedure should be followed as given in BS 7385. The measurements are taken in terms of peak particle velocity at foundation level and the plane of the uppermost full story. Further details are given in Section 6 of this paper.

The anxiety can be greatly reduced if the procedure given by Scannell (1993) is adopted. This involves the use of some of the psychological factors which can effect human reaction to stressful events, the most important of these are predicability and controllability.

4.2.1. Predicability

Psychological research over the past three decades has shown that a subject's negative reaction to vibration. noise or shock is reduced if they are predictable rather than unpredictable eg Pervin (1963) and Badia et al (1979). This result is perhaps not surprising since predicability gives the subject the opportunity to prepare for the event in a way that minimises its adverseness. However, many operators of plant or equipment, that cause perceivable environmental vibration, are under the impression that if local residents are informed about the activities they will complain more!

In most cases the opposite is true. Being able to predict when a stressful event is likely to occur and its duration, even if the individual cannot control it, usually reduces the severity of their anxiety. Katz and Wykes (1985) reported that with unpredictable, stressful events there is no safe period; with predictable events the individual can relax to some extent between events.

4.2.2. Controllability

Control of, or even simply the perception of control of, adverse events can also have an effect on reaction. In one study, by Glass et al, (1969) the effect on subjects who had the option of terminating a randomly presented aversive stimulus was much less than those who had no perceived control. This is true even though the subjects did not avail themselves of the control opportunity. In another experiment, Geer et al (1970) found that subjects who believed that they could control the duration showed lower autonomic (peripheral nervous system) reactivity to the shock than did subjects who did not perceive control over the same stimuli.

4.3 Monitoring Cracks During Vibration.

It is advisable to monitor the size of any cracks that have been recognised as having the potential for expansion. This can be done with the use of 'Demec' gauges or vernier callipers that required the adherence of small studs or screws on either side of the cracks. The distance between them can then be accurately measured with the gauge. Alternatively 'Avongard' (or similar) tell-tales can be fitted across the cracks that enable direct readings in vertical and horizontal directions to an accuracy of 1.0 mm.

4.4 Building failures.

It is important for complainant's and other interested parties to appreciate that there are many reasons for building failures are problems. These include damage from: solar radiation, moisture ingress, chemical attack, frost, vegetation movement, differential settlement due to extensions, etc. See Ransom (1987).

5. HUMAN REACTION TO VIBRATION - WHOLE BODY VIBRATION

It is accepted in the British Standard BS 6472 (1992) that human response to vibration in buildings can depend on many factors other than the magnitude of the vibration. These include:- the direction of vibration input to the body (ie. whether foot to head or back to chest, etc.), the frequency of the vibration, the place of occupation (ie. whether residential, office or workshop etc), the temporal structure of the vibration (ie. whether continuous, intermittent or impulsive).

5.1 Criteria.

For continuous vibration and blasting, criteria are given in table 2 below. This is an adaptation of the table given in BS 6472 and gives frequency weighted vibration magnitudes. For intermittent vibration the preferred method is to directly measure or to calculate the vibration dose value (VDV) or the estimated vibration dose value. Practical examples are given by Scannell (1990) .

5.2 Vibration Dose Values.

5.2.1 VDV

The vibration dose value (VDV) is based on the root-mean-quad principle, ie a fourth power time-dependency. This means that a doubling of vibration magnitude would require a sixteen fold reduction in duration to result in an equivalent dose. The VDV is a summation (dose) rather than an average and hence the value is cumulative and cannot reduce during a measurement period. The VDV is found from squaring the frequency weighted acceleration data twice, summing, multiplying by the exposure time period (in seconds) and then square rooting this value twice. This gives the rather usual units of $m/s^{1.75}$. *This originates from the acceleration in m/s^2 raised to the power of four giving m^4/s^8; multiplied by duration (s) giving m^4/s^7 and taking the fourth root giving $m/s^{7/4}$ or $m/s^{1.75}$.*

TABLE 2 SATISFACTORY MAGNITUDES OF BUILDING VIBRATION WITH RESPECT TO HUMAN RESPONSE.

Place	Time	Continuous Vibration (16 hr day, 8 hr night)		Blasting Vibration (up to 3 blasts per day)		Intermittent Vibration VDV
		m/s^2	mm/s	m/s^2	mm/s	$m/s^{1.75}$
Critical working area	Day and Night	0.005	0.141	0.005	0.141	0.09
Residential	Day	0.01 to 0.02	0.28 to 0.56	0.3 to 0.45	8.5 to 12.7	0.2 to 0.4
	Night	0.007	0.2	0.1	2.8	0.12
Office	Day and Night	0.02	0.56	0.64	18	0.3
Workshops	Day and Night	0.04	1.12	0.64	18	0.7

Note: All values are in the above table are frequency weighted - see BS 6472.

5.2.2 eVDV

Unfortunately, instrumentation for the measurement of VDVs is not widely available. However, a good estimation of the VDV can be made from multiplying the frequency weighted r.m.s. acceleration by a constant and the fourth root of the time period. This is known as the estimated vibration dose value (eVDV). The constant will largely depend on the crest factor (peak to r.m.s. ratio) and has been well documented for railway vehicle vibration (for example see Woodroof and Griffin, 1987) as 1.4. Hence:

$$\mathbf{eVDV = 1.4\ a_w\ (d)^{0.25}}$$

Where a_w is the frequency weighted rms acceleration and d is the exposure time period in seconds. For a pure sine wave the constant would be 1.1 and for a highly impulsive source the constant could be 2, 3 or even higher. However, the 1.4 constant can be applied to many sources other than rail vehicles, even blasting and drop hammer piling provided that there is relatively large distance between the sources and the assessment position. This is because the impulsiveness of the signal greatly reduces and approaches a sine wave with increasing distance.

When assessing close to shocks or other signals with high crest factors (over about 6) the VDV calculation will normally be required in preference to the eVDV. A procedure utilising a chart recorder for VDV calculations for shocks is given by Griffin (1990).

5.3 Peak Particle Velocity.

The current practice for blasting assessments and for structural damage assessments is to measure the peak particle velocity (ppv) with a velocity transducer (geophone). This method can also be used as an alternative to VDV (or eVDV) for whole body human reaction prediction provided that the magnitude/duration trade off in table 3 is used.

TABLE 3 : THE TRADE-OFF BETWEEN PPV AND DURATION

PPV (mm/s)	Low Probability of Adverse Comment	Adverse Comment Possible	Adverse Comment Probable
1	Up to 1 hr	1 hr to 16 hr	-
1.5	Up to 15 min	15 min to 3.5 hr	3.5 hr to 16 hr
2	Up to 5 min	5 min to 1 hr	1 hr to 16 hr
2.5	Up to 2 min	2 min to 30 min	30 min to 8 hr
3	Up to 1 min	1 min to 15 min	15 min to 3.5 hr
3.5		Up to 7 min	7 min to 2 hr
4		Up to 5 min	5 min to 1 hr
4.5		Up to 3 min	3 min to 45 min
5		Up to 2 min	2 min to 30 min
5.5		Up to 1 min	1 min to 20 min
6			Up to 15 min
6.5			Up to 10 min
7			Up to 7 min
7.5			Up to 5 min
8			Up to 4 min
8.5			Up to 3 min

NOTES

(i) All values are frequency weighted in accordance with BS 6472 (1992).

(ii) Values are calculated from eVDV's given in BS 6472 (1992).

(iii) All times are rounded to the nearest 'sensible' figure.

(iv) Times below 1 minute or above 16 hours are not shown.

5.4 Blasting Vibration

Where more than three blasting events occur in a 16 hour day, BS 6472 gives an alternative to the VDV approach may be used. This method also assumes a 'trade-off' between the magnitude of the vibration and the number of events multiplied by the duration of the events. To calculate satisfactory magnitudes ppv, use the formula shown below:

$$\text{ppv} = \frac{14.45}{\sqrt{N} x T^{d}}$$

for the z-axis and

$$\text{ppv} = \frac{41}{\sqrt{N x T^d}}$$

for the x/y-axis.

Where:

N is the number of blasts in a 16 hour day (and **N**>3);

T is the duration of the events in seconds.

d is zero for **T** less than 1 second (ie $T^d = 1$).

For **T** greater than 1 s,

d = 0.32 for wooden floors.

d = 1.22 for concrete floors.

The different magnitudes for wooden and concrete floors are thought to be due to the difference in the expected movements.

Despite the difference in origin between this method and the VDV method, the two produce remarkable similar results except at the extremes of duration or blast numbers.

6. VIBRATION MEASUREMENT TECHNIQUES

6.1 Instrumentation

There is a wide range of instruments that are suitable for the measurements of ground vibration, vibration in buildings and vibration of building elements, however, they all consist of at least:- a transducer (accelerometer or geophone), a signal processor, and a display or indicator.

6.2 Measurement Techniques - Structural Damage

To evaluate the effect of the vibration on the superstructure of a building, measurements of the peak particle velocity (in mm/s) as a function of time, must be obtained.

The measurements should be made at the foundation level and at the highest floor level (as shown in figure 2), in all three orthogonal directions (ie. in the vertical direction and horizontal directions in the plane of the building length and width).

Although of prime interest are: the vertical direction (z) at the foundation for compressive and tensile strain, the horizontal directions (x and y) for the shear strain.

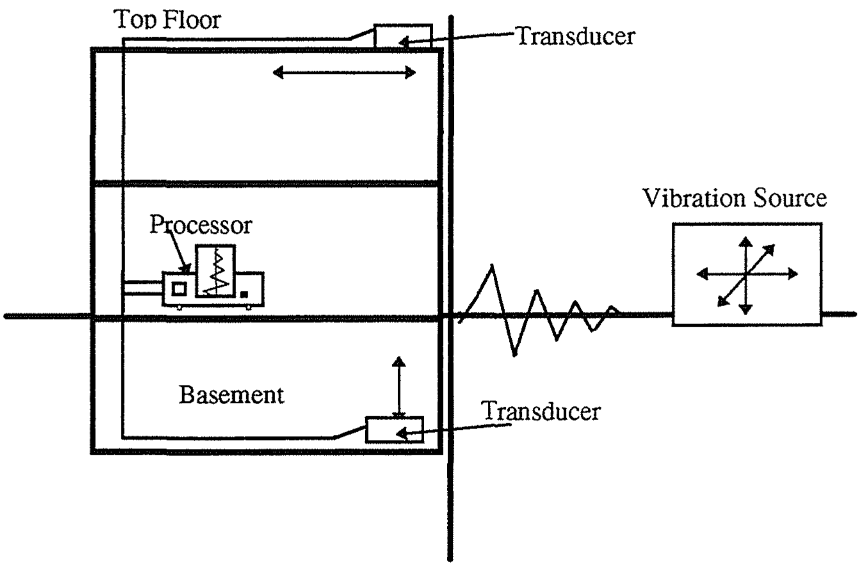

Figure 2. The measurement positions (as preferred by BS 7385 : 1990 and DIN 4150 : 1986)

When measuring vibration at the foundation, the transducers should be placed on the lowest storey of the building, close to an outer wall or in a recess of the outer wall. For buildings with no basement, the point of measurement should lie not more than 0.5 m above ground level. The measuring points should be located on the side of the building facing the source of the vibration.

When measuring the horizontal direction in the upper-most storey, the transducers should be placed close to the outer wall with one direction of measurement parallel to the side of the wall facing the source of the vibration.

If a multi-storey building is subjected to steady state harmonic vibration, simultaneous measurements should be made at all storeys where practicable and at least every 4 storeys (approximately every 12 m). This is because at higher 'modes' of vibration than the fundamental, much larger vibration magnitudes could occur at intermediate floor levels.

Where the building is more than 10 m long measurements should be taken at horizontal intervals of approximately every 10 m. All measured magnitudes should be reported and action should be taken with regard to the highest measured magnitude.

When potential damage to building elements (ie. floors, walls, ceilings, window panes) is to be monitored, lightweight transducers are mounted in a central position on the element. Although sometimes severe (the author has measured ppv's up to 100 mm/s without damage occurring) these vibrations are usually unrelated to structural integrity (Siskind et al 1980).

In the case of complaint, it is often useful to take measurements where the complainant states that the vibrations are strongest.

6.3 Measurement Techniques - Whole Body Vibration.

6.3.1. The position of transducers.

The vibration should be measured at the point of entry to the human body. If a complaint is made about vibration at a specific place (eg. when the complainant is seated) then the measurement should be carried out at the body-environment interface. The accelerometers must move with the interface, they must not alter the dynamic properties of the body or the seat and they must offer little impedance to movement over the frequency range of interest. The seat must be occupied by the complainant. The same rule would apply for complaints from people lying in bed. For many of these types of measurements a 'semi-rigid' interface device, defined by the Society of Automotive Engineers (SAE) (1974), will prove suitable. This device is shown in figure 3, see also ISO 7096 (1982).

However, it is often found that the complaint is not specific but more general, stating vibration occurs throughout the house. In this case the vibration should be measured at the point in the building where the highest vibration magnitude, that could be responsible for whole body vibration, is obtained. If the vibration source is restricted to the daytime, then the floor of the living areas would be the appropriate place to measure. In two storey buildings, the bedroom areas may provide the highest magnitudes but it is only reasonable to obtain measurements here, if the vibration is expected to be generated at night.

In principle, vibration should be measured in all three orthogonal directions at a number of points in a building and the measurement results from the point and the direction where the highest range of magnitudes is found should be acted upon. It is often possible, on the basis of a survey or guided by experience to select a few measurement positions and directions where the highest range of magnitudes occur.

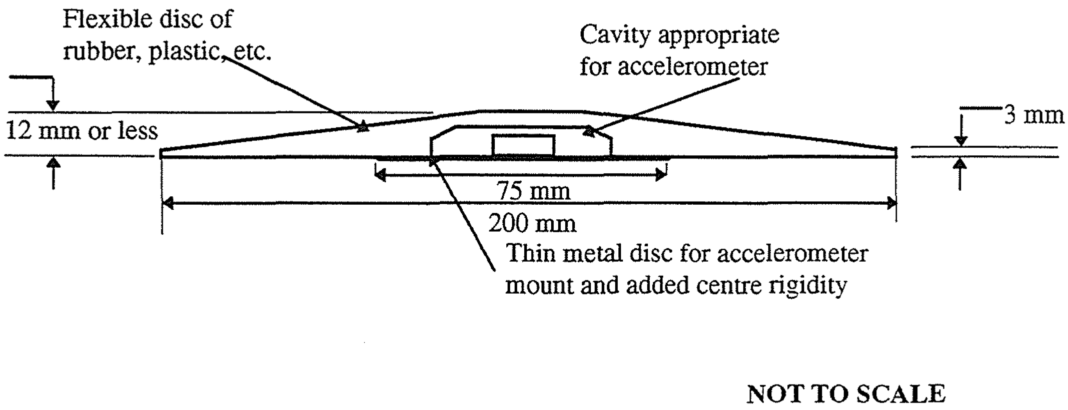

Figure 3. An example of a cross section of an accelerometer mount for measuring on soft seats from SAE (1974).

In many cases the magnitude of vertical vibration is higher than the horizontal vibration. The highest magnitude of vertical vibration is usually found in the unsupported centre of the beams with the longest span across the floor as shown in figure 4.

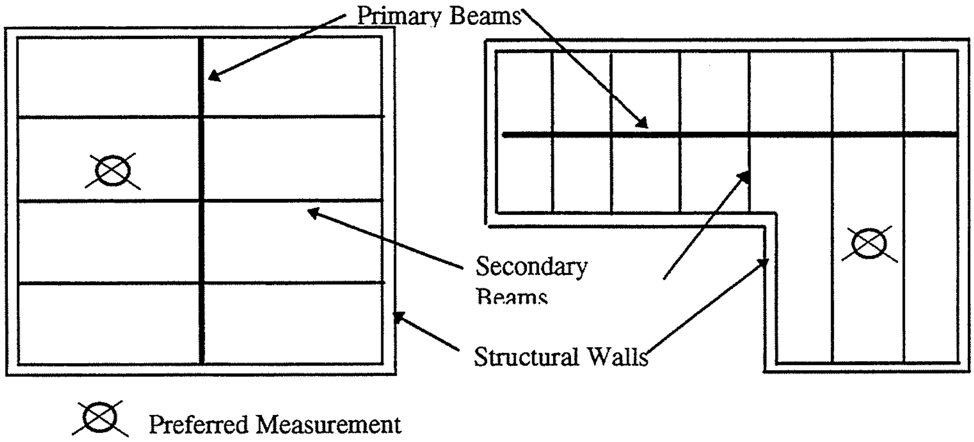

Figure 4. Examples of measurement points, where the highest magnitude of vertical vibration may be expected (from Danish Acoustical Institute 1987).

It is often practical to measure horizontal vibrations on structural walls as close to the floor as possible, rather than the floor itself. Care should be taken not to take measurements on lightweight or loose wall coverings or nonload-bearing partitions or columns.

6.3.2 Mounting the transducer.

For measurements on a solid and rigid floor or structural wall the transducer can be fastened with double sided adhesive tape or similar. If the floor is covered with a non-removable carpet or the like, the transducer shall be mounted with solid contact to the floor, ie. on a device as illustrated in figure 5. This mount is supported at three points. When placing the device, it must be ensured that the supporting pins fully penetrate the carpet and rest on the solid floor beneath. This will not cause damage to carpets of an open weave construction. It must be ensured that the mount cannot rock. Measurement of horizontal vibration must not be made with this device.

A transducer to measure horizontal vibration of a structural wall can be fixed with bees wax or double sided adhesive tape if the transducer is light enough to be held firmly in position in this way. Alternatively, a heavier transducer can be secured to a mounting stud that is glued to the wall or attached with a bolt fixed in the wall by an expanding plug.

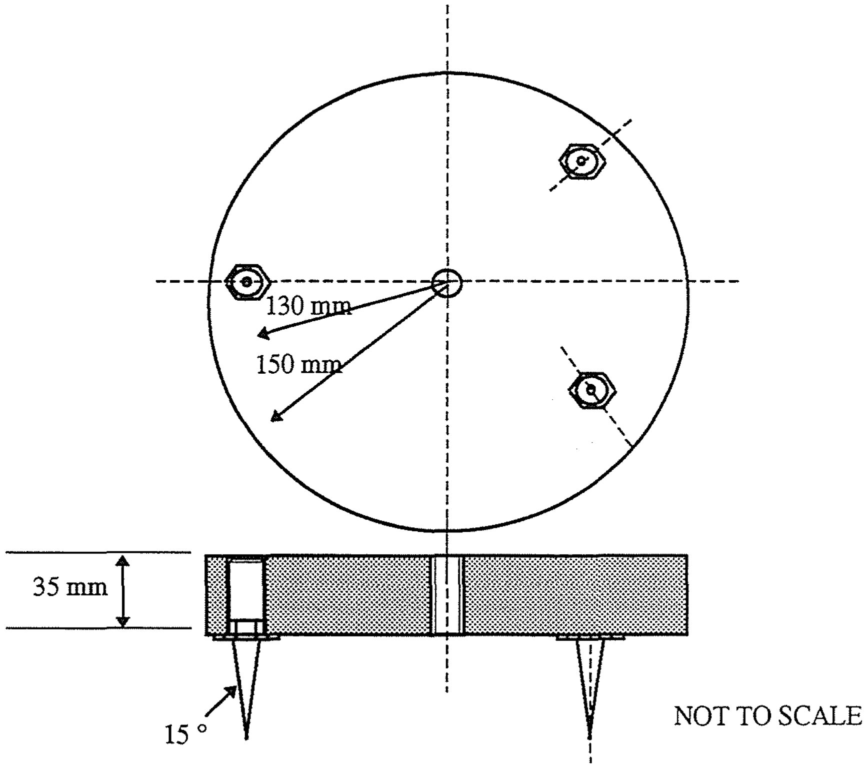

Figure 5. Mounting device for measurement of vertical vibration on carpet-covered floors. The combined weight of the device plus attached transducer should be about 1.5 kg.

7. SUMMARY

Although universal building damage criteria are not possible, many Countries have issued guideline vibration values which should be taken into account during any assessment. Limited studies that have also been carried out on the effects of fatigue, indicate that regular, low magnitude, vibration is not a major cause of building damage.

Complaints arising from vibration usually relate to either building damage fear or the effects of whole body vibration. It is important to establish which problem is involved as the assessment procedures are quite different (the complaint may, of course, involve both problems). In some cases (psychological) reassurance can significantly reduce anxiety.
Guideline criteria for adverse comment from whole body vibration are given in (preferred) frequency weighted acceleration and peak particle velocity. For intermittent and impulsive vibration, the vibration dose

value provides a reliable method of assessment. Where equipment is not available the estimated vibration dose value can be used.

Instrumentation usually consists of either a seismograph with a velocity transducer (geophone) for building damage assessments or a vibration meter and accelerometer for whole body vibration assessments. Geophones are usually positioned at the foundation level close to the wall facing the vibration source, while the accelerometer should be placed at the interface position between the vibrating building element and the human body. Where non-specific complaints arise, the accelerometer should be placed at the position where the highest vibration occurs. This is often found to be the centre floor position.

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RIVER DEEP, MOUNTAIN HIGH - QUIETENING THE SOUNDS OF THE SNOWY

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INTRODUCTION

The Snowy Mountains Hydro-electric Scheme is one of the engineering feats of the modern world. Completed in 1974 after a construction period of 25 years, the Scheme collects, stores and diverts water in the Snowy Mountains area through an interconnected network of dams, tunnels and aqueducts to supply water and electricity to much of South East Australia.

The Scheme generates more than 5000 gigawatts of electricity each year to meet peak power demand, provide contingency reserve and voltage/frequency control, and to act as the major interconnection point between NSW and Victorian grids. With a generating capacity of 3756 megawatts, the Scheme represents approximately 17% of the installed generating capacity of South Eastern Australia. To generate this power, there are seven power stations (Murray 1 and 2; Guthega; Tumut 1, 2 and 3; and Blowering) and one pumping station at Jindabyne. Tumut 1 and 2 are the only underground power stations in Australia. Tumut 1 is located 366 metres below ground level and is reached through a 410 metre access tunnel. Tumut 2 is 244 metres below ground level and is reached through a 1070 metre access tunnel.
The other power stations are multilevel constructions with at least half of the operations below ground level.

At the time of their completion and inauguration as a leading power generating facility, not a lot of thought was given to the levels of noise that would be generated, as an Occupational Health & Safety risk. The need to wear head protection at all times and to be cautious of the extremely high voltages that were ever present, was enforced from the start but the remoteness of each station and the acceptance that generators of this size would by default be noisy, was generally accepted as "par for the course".

THE HEALTH & SAFETY ETHIC

The Snowy Mountains Council which was enacted to oversee the operations of the Scheme began to actively pursue a carefully constructed regime of safety procedures influenced by various State and Federal regulations. the safety procedures of other energy authorities finally culminating in the NSW Electricity (Workers Safety) Regulations 1992.

With the appointment of an Executive Operations Safety Committee, the whole range of safety issues is now a striking feature of the Schemes excellent record in health & safety management. As the ever growing interest in work related OH&S issues began to gather impetus in the 80's, the question of noise reduction was considered along with all aspects of safe working conditions such as machine guarding and hazardous chemicals in what we now expect as standard in most situations - protection for head, eyes and ears.

The supply of standard styles of hearing protection formed the principal method of personal noise reduction and it was not until the early 90's that serious consideration was given to some form of permanent noise reduction.

Of prime concern were the noise levels at Tumut 2 and Murray 1 where the public were encouraged to visit and inspect. It is my belief that the Authority were concerned that in the course of time, someone would lodge a claim for hearing damage as a result of their having visited either station and that such a claim could seriously effect the excellent safety record of the Authority and could be prejudicial to the Authority's work. It is also my contention that this emphasis shifted to a concern for staff working in these stations rather than the public.

It was into this atmosphere of safety awareness that the idea of reducing the operational noise levels in the power stations, first gathered momentum, particularly with the advent of the draft National Standard on Occupational Noise in 1989 and other State regulations that invariably put into place, the 85dB LAeq 8hr noise limits as we now know them. With Murray 1 and Tumut 2, two of the largest power stations open for public inspection and utilising the most staff, it was felt that attention should be given to these stations with a view to reducing the noise levels to meet the requirements of those Regulations. The Authority sought to pursue the idea of using one the Murray 1 station as a pilot study to see just what could be done to reduce noise levels to below 85dBA.

In early 1995, tender's were called for a company to provide a complete consulting and installation package to reduce the noise levels in the Murray 1 power station, located 12 kilometres from Khancoban in NSW.

My company, *Shelburg Acoustics P/L*, was successful in winning that tender and preliminary measurements of operational noise levels and reverberation times were carried out. Briefly, the geography of the station is such that the incoming water flows into the turbines, three floors below ground level and after doing it's work, is then pushed out into a tailwater race to eventually become the primary water source for the Murray 2 station. The building is primarily constructed of poured reinforced concrete walls, ceilings and floors.

IDENTIFYING THE SOURCES

There are two major noise sources comprising firstly the direct sound and accompanying turbulence caused by the generators themselves and secondly the broad band sound of the water as it enters and moves through the turbines.

As a result we identified three major problems that needed to be addressed:

a) reduce the diffuse field and hence the overall noise levels to below 85dBA LAeq

b) reduce the reverberation times to about 2 seconds

c) reduce the incidents of standing waves at the predominant frequency of 160Hz.

The major source of direct sound was twofold:

a) the eight sided concrete enclosure that houses each generator, called understandably the "octagon", produced levels inside the enclosure of 110dBA!! It was decided to reduce the reverberant noise source by lining the walls with a 50mm absorbent polyester blanket in between 50mm top hat steel section. This was covered with an additional absorbent fabric and completely contained within steel wire mesh. This served to ensure that in the turbulent environment of the enclosure, no pieces of material could break free and create problems within the generators. All the steel sections, including the mesh had to be physically connected to the station earth plane

b) The bottom end of each turbine, two floors below, is accessed by a 2 metre long tunnel called a "draft tube". As a fully enclosed, circular concrete structure, this acted exactly like a reverb. chamber and at operational speed, produced noise levels in excess of 100dBA. It was therefore proposed to fit the entrance of the draft tube with a noise barrier. To satisfy maintenance and access requirements, the barrier had to be removable in an emergency, have an acoustic vent, an openable door and an inspection window.

While Appendix A of AS2107 does not extend to recommending reverberation times for rooms with a volume of over 150,000m^3 !!, it was felt that a 3 second RT would and should be reasonably achievable.

The measured RT at 125Hz was 6.5 seconds and with a calculated volume of the station (not an easy task) of approximately 150,000m^3, the absorption area derived from the familiar and simplified Sabine formula, RT = 0.161V/A, was about 3,700m^2. In order to reduce the RT to about 3 seconds, the new calculated absorption area needed to be about 8,000m^2.

The reality of the situation however precluded the addition of 4,000 m2 of absorption materials due to inaccessibility to much of the open concrete surface, extensive pipework, auxiliary plant such as compressors, storage tanks, switchgear and cooling systems. We therefore determined to place as much absorption material as possible on exposed and available walls and ceilings which finished up at approximately 2,000m^2 which gave a final RT of 4 seconds.

INSTALLING THE MATERIALS

Whilst reducing the diffuse field was not considered too difficult, the well focussed peak at 160Hz, due to the rotational speed of up to 10 generators operating at any one time presented the additional problem of a large number of standing waves throughout every level of the station.
An absorption panel was designed using a pine timber frame, suitably painted and water proofed, 2800mm by 1400mm, covered with an acoustically absorbent fabric with good low frequency absorption.

The panels were then infilled with 50mm of polyester blanket to improve this absorption. Since we know that maximum sound absorption will occur in a porous absorber when the particle velocity in the absorber is at a maximum, and that this will occur at the one quarter and three quarter wavelength distance from the walls, the calculations determined that in order to achieve maximum absorption, the quarter wavelength spacing from the wall should be about 1 metre!! As this was completely impractical, a space of 250mm reduced this absorption to around 75% at the dominant frequency. Where panels were able to be fixed to ceilings, this was achieved by using 10mm jack chain to achieve the 250mm spacing. A considerable amount of "shoe horning" was needed to place panels in some areas and while ,ideally, we needed to keep the panels down to one or two sizes, this was not always possible and several smaller panels were built on site to accommodate these awkward areas.

The acoustic barrier for the draft tubes comprised a 3mm layer of constraining vibration damping material sandwiched between two sheets of 3mm aluminium. One section was designed as a door with a 200mm^2 acoustic vent and a 300mm porthole inspection window. Because of the high moisture content at this level of the station, fittings such as hinges and handles, were required to be stainless steel.

SUMMARY

The final result was that, by default, most of the major standing waves in the station were damped and where standing waves persisted, additional panels were introduced into the wave to reduce them. The results of the overall work as shown in the appendices reveal considerable success in achieving the original OH&S goal of reducing overall station noise levels to 85 dBA or less.

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MURRAY 1 - FINAL NOISE MEASUREMENTS

	Octave Frequency Bands (Hz)								
	31.5	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
Outside Draft Tube Door, Unit 1	78	81	94	87	80	76	72	68	64
Outside Draft Tube Door, Unit 2	76	81	105	97	83	78	72	71	68
Inside Turbine portal, Unit 1	83	86	102	95	85	81	77	72	68
Inside Turbine portal, Unit 2	81	83	97	90	88	83	78	72	70
Outside U/S Octagon door, Unit 2	75	81	86	86	80	76	76	70	62
Machine Hall, U/S side, Unit 2	77	79	95	88	82	78	75	68	57
Visitors viewing area	74	75	87	82	77	74	71	63	49
Reception Area	70	78	85	79	70	63	60	51	36

MURRAY 1 - INSERTION LOSS OF DRAFT TUBE DOORS

Laeq (dB)

	Inside	Outside	Reduction
Unit 1	105	87	18
Unit 2	104	90	14
Unit 5	98	84	14
Unit 7	105	90	15
Unit 9	101	89	12
Unit 10	101	90	11

MURRAY 1 - FINAL NOISE REDUCTIONS

Octave Bands dBA

	31.5	63	125	250	500	1000	2000	4000	8000
Valve Fl., Unit 2, before treatment	94	98	108	106	96	92	89	81	74
Valve Fl., Unit 2, after treatment	76	81	105	97	83	78	72	71	68
Reduction	**18**	**17**	**3**	**9**	**13**	**14**	**17**	**10**	**6**
Turbine Fl., Unit 2, before treatment	95	97	105	100	92	88	86	79	70
Turbine Fl., Unit 2, after treatment	83	86	102	95	85	81	77	72	68
Reduction	**12**	**11**	**3**	**5**	**7**	**7**	**9**	**7**	**2**
Operating Fl., Unit 2, before treatment	86	88	99	94	89	87	82	78	69
Operating Fl., Unit 2, after treatment	77	79	95	88	82	78	75	68	57
Reduction	**9**	**9**	**4**	**6**	**7**	**9**	**7**	**10**	**12**
Visitors viewing area, before treatment	82	86	92	92	79	76	74	66	56
Visitors viewing area, after treatment	74	75	87	82	77	74	71	63	49
Reduction	**8**	**11**	**5**	**10**	**2**	**2**	**3**	**3**	**7**
Reception Area, before treatment	83	87	91	89	87	79	62	53	38
Reception Area, after treatment	70	78	85	79	70	63	60	51	36
Reduction	**13**	**9**	**6**	**10**	**17**	**16**	**2**	**2**	**2**

AN INTELLIGENT METHOD OF AIRCRAFT AND TRAFFIC NOISE MONITORING USING TIME-FREQUENCY DISTRIBUTIONS

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ABSTRACT

The main purpose of the present work is to develop an "intelligent" transportation noise monitoring system. With existing automatic transport noise monitoring systems there is the possibility that extraneous noise sources such as birds or animals, speech and music will influence the measurements. In more sophisticated systems noise monitoring is accompanied by radar detection of sources so that recordings are only made when the noise sources are in view of the radar. A system which can identify noise sources by their acoustical characteristics has advantages. This paper reports on methods for identifying aircraft and traffic noise automatically using the intensity of the sound and features of WVD and STFT distribution of acoustic signals.

INTRODUCTION

It is desirable to have a transportation noise monitoring system which can respond without using the radar and information from airports about approaching aircraft. A statistical noise monitoring system can be activated when the aircraft or traffic noise dominates the background noise. Such a system could be especially useful where temporary noise monitoring from commercial and residential premises exposed to the aircraft and traffic noise, is undertaken. Usually the noise monitoring in such places is interrupted by speech, music and animal sounds.

Existing intelligent noise monitoring systems are sensitive to the level of the noise rather than character of it. An intelligent noise monitoring software package, which is called "SmartNoise" has been developed in Labview for Windows (National Instrument, 1996), to identify and monitor only aircraft and traffic noise.

The main issues in the design of SmartNoise are the early detection and classification of the type of the noise. As the monitoring system is working in real-time, there is no chance for post-processing or using some time-consuming recognition methods (Mohajeri 1996). Therefore, short term feature extraction and classification methods have to be implemented (Robiner 1978). If the approaching sound source is transportation noise, then the statistical levels of the noise will be evaluated, displayed and saved in a file.

Five steps are involved in the "SmartNoise" software: a)data acquisition, b)event detection, c) event classification d)statistical data manipulation and e)display and monitoring the numerical values.

DETECTION OF THE NOISE SIGNAL

The main feature to detect the beginning of a noise event is the intensity of the acoustic signal. A noise event starts when the intensity level of the noise exceeds a set level or the level of the background noise. The intensity feature does not have any frequency information so that it cannot be used for classification of the noise. However some statistical characteristics of intensity can be considered in the knowledge based program. After setting the thresholds of background noise manually or automatically, by using the training mode of software, any rapid increase in the intensity level indicates the start of an acoustic event. Also the intensity of sound is a good measure to find the end point of a noise event.

TIME-FREQUENCY DISTRIBUTION FEATURES

Time-frequency distribution methods have been used to extract the acoustic features of a noise to identify the aircraft and traffic noise among other environmental noise sources. The main feature is the plot of the first moment of the Wigner-Ville distribution (WVD) plane with respect to the average of the first peak in the Sort Time Fourier Transform (STFT) plane. The Wigner-Ville distribution method provides a high resolution in time-frequency domains and it is a powerful tool for showing the characteristics of transient and non-stationary signals. WVD contains both time and frequency information in the same plot. For a given signal *s(t)*, it can be represented as (Boashash 1992):

$$W(t,f) = \int_{-\infty}^{+\infty} e^{-j2\pi f\tau} s^*(t - \frac{1}{2}\tau) s(t + \frac{1}{2}\tau) d\tau \qquad (1)$$

In this research, the first moment of the WVD has been used as a measure for classification of transportation noise. The first moment, or centroid of WVD in frequency direction, can be formulated as:

$$f_{centroid} = \frac{\sum_{j=0}^{m} t_j \sum_{i=0}^{n} f_i W_i(t,f)}{\sum_{j=0}^{m} t_j \sum_{i=0}^{n} W_i(t,f)} \tag{2}$$

where t and f are the time and frequency at each location of the WVD plane. As aircraft and traffic noise contains high energy at low frequencies, the centroid of WVD tends to be on the left side of the plane. In contrast, the centroid of WVD from speech, music and birds sounds is located around the centre of plane and tends to be at the right side of the plane. This occurs because of the variation and distribution of harmonics in different times, and the high frequency components in these sounds.

A Short Time Fourier Transform (STFT) is a sliding window FFT. The window function divides the signal into time intervals and the Fourier transform computes the frequency spectrum of each interval. It is found for transportation noise that the average of the first peak frequency in the STFT plane is always at the low frequencies. This is because of low frequency nature of transportation noise. Using the average values of first peak and centroid helps to reduce the false recognition of sounds because of interference of some loud tonal noises.

Figure 1: a) The Wigner-Ville distribution of aircraft noise, b) STFT of the same segment.

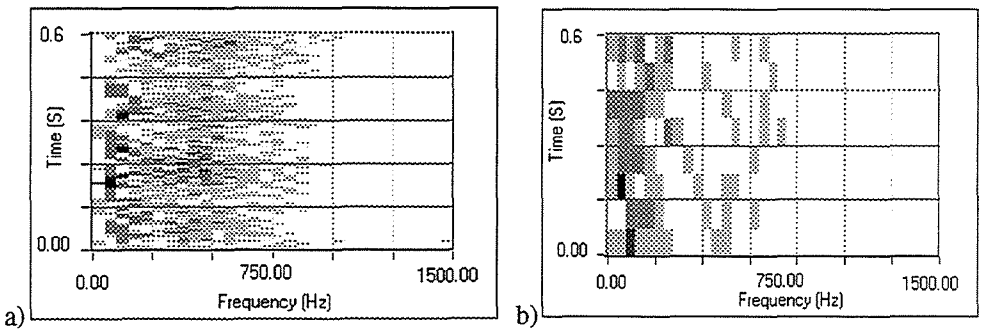


Figure 2 shows that the peaks of the STFT in each segment are located at low frequencies for the entire aircraft noise event.

In figure 3, the results of a classification using the combined features from WVD and STFT are presented. It shows a good separation between data from transportation noise , speech and music. This graph is comparable to the results of other statistical classifications presented previously (Mohajeri 1996). Figure 3 shows that there is a clear delineation between transportation noise and other sounds. The line drawn separates the two types of sources.

Figure 2: The STFT of a landing Boeing 747.

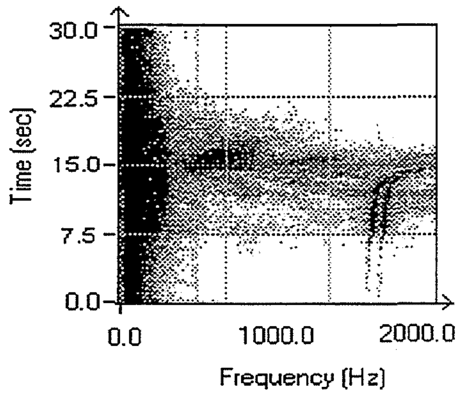
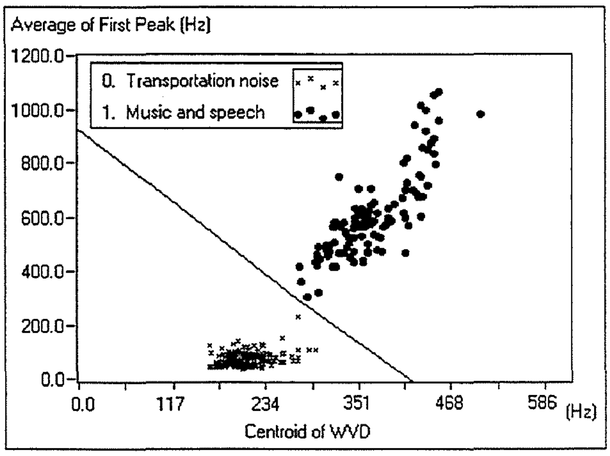


Figure 3: Results of using combined features from the centroid of the WVD and the average of the first peak in STFT.



ALGORITHMS FOR CLASSIFICATION OF NOISE SOURCES

A fuzzy classifier, based on the results of the time-frequency features of detected noise, was developed to identify whether the noise is from transportation or not. The algorithm shown in figure 4, gives the general strategy for detection and classification of the sound event and the triggering of the monitoring system. At the beginning the thresholds and criteria of detection and classification of noise sources are set manually or automatically from the training program. The computer starts sampling the acoustic data each 0.1 second. When the intensity level exceeds the threshold (in other words when a noise event starts) the program increases the sampling duration from 0.1 to 0.6 seconds. The longer duration samples go to the classification section of the program to confirm whether the starting noise event is transportation noise or not. If the noise event is transportation noise the statistical levels of the event are monitored for a 0.1 second sampling duration. While the monitoring is carried out the intensity level is checked to identify the end of the noise event. If the noise event is not transportation noise, the classification will be continued until the noise event finishes and nothing will be monitored.

Figure 4: General strategy for detection, classification and monitoring of aircraft and traffic noise.

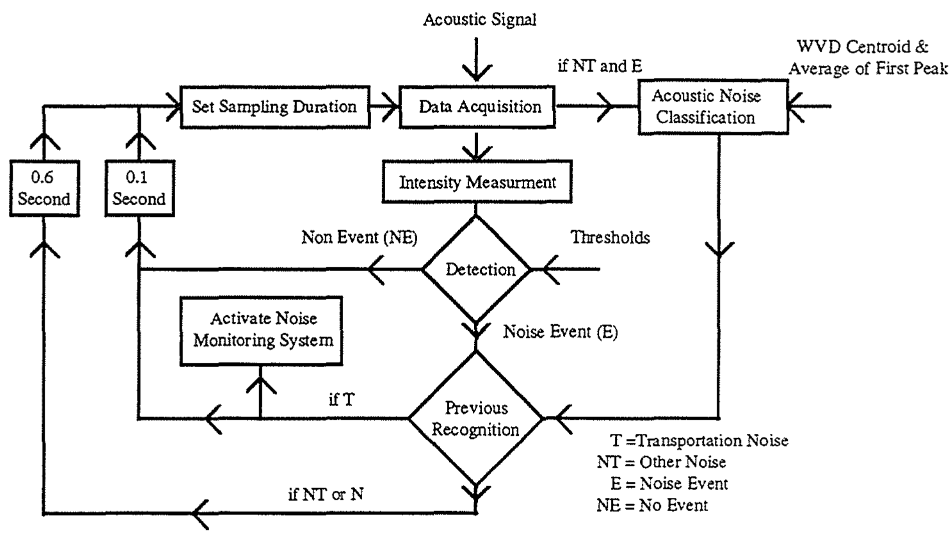
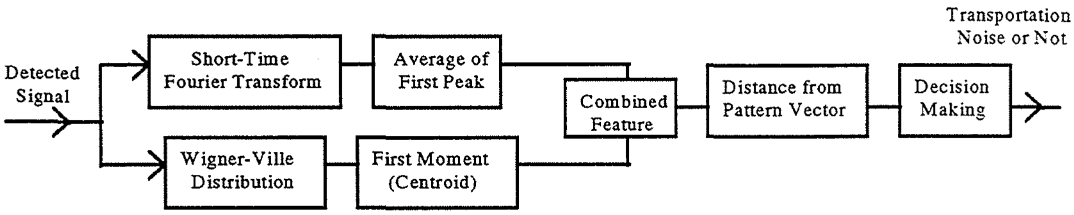


Figure 5: Steps of feature extraction and classification from a time-frequency distribution.



When the acoustic signal is detected, it is classified. Figure 5. shows the strategy for feature extraction and classification of a short duration signal in real-time.

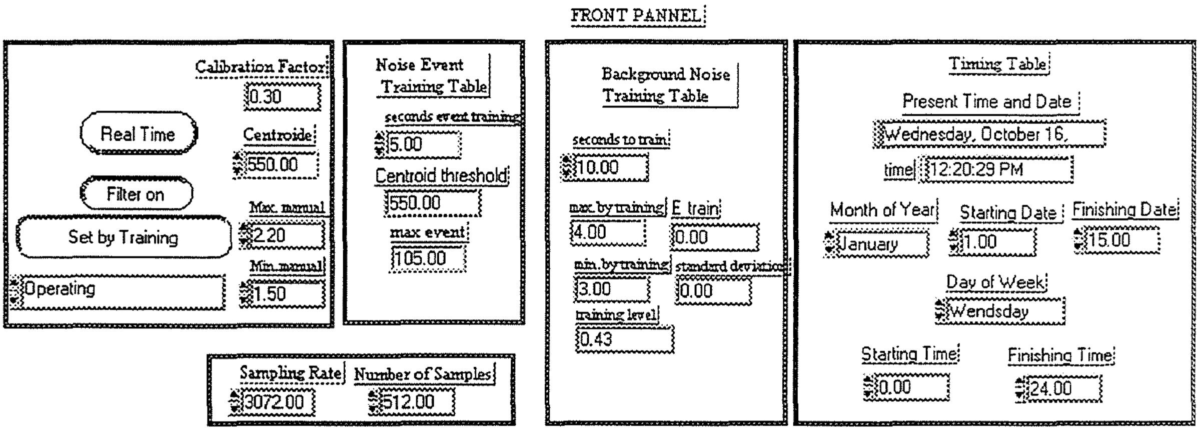
FEATURES OF THE SMARTNOISE SOFTWARE

SmartNoise is capable of undertaking the following tasks:

- Monitoring transportation noise which exceeds a given threshold.
- Setting the time and date that the user wishes to perform noise monitoring.
- Training to cope with new locations, conditions and instruments.
- Monitoring weather condition simultaneously with sound measurements.
- Stop monitoring in unreliable weather conditions.
- Saving the data every hour in a file and resetting the memories.
- Sending the statistical levels of noise with date and start and end time, to a spread sheet.
- Displaying the instantaneous frequency spectrum and the last noise event.

A view from the front panel of SmartNoise is shown in figure 6.

Figure 6: A part of front panel of SmartNoise.



Before starting the noise monitoring, the values of sampling rate, number of samples and the time and date of monitoring have to be set. If the user wants to set the thresholds and criteria for recognition by training, the training mode has to be run.

The time history of the sound pressure level due to the noise incidents can be automatically saved into a TXT file, every hour. The name of the file is a combination of date and time of monitoring. Leq, Lmax, L90, L10, SEL, start, end and duration of each events are indexed (N) in table 1. These values can be automatically saved as a spread sheet every hour.

TRAINING

A training mode is designed to modify the thresholds and criteria in different locations and weather conditions. The user has two options; either to set the detection and classification thresholds manually or leave them to be set by a training routine. The training software becomes more necessary when: a) the system is connected to a different microphone or pre-amplifier, b) the type or level of background noise is changed, c) the monitoring location is changed, d) the weather conditions change, e) the maximum and minimum range of the noise events are changed. In the training mode the old numerical values are replaced by the new values. This mode of the software consists of three steps, a) auto calibration, b) background noise training and c) noise event training.

Auto Calibration

The calibration factor is an important value to be set when different types of microphones and pre-amplifiers are used. The software can automatically set the calibration factor when a B&K calibrator 4230 is being used. The process is based on incrementing the correction factor of the input voltage until the sound pressure level becomes equal to 94 dB.

Background Noise

The level and the type of background noise is different at different times and places. The noise of wind, vegetation and the distant transportation are the usual outdoor background noises. In the training mode of the software the maximum, minimum and mean of the intensity level of background noise for a particular time, which can be set by user, will be determined. These levels will be realised as the thresholds of detection of a noise event in the operating mode.

Noise Events

It is important to consider that the pattern of a particular noise can be changed in different places and weather conditions. The numerical values of the centroid of WVD and the average of the first peak have to be evaluated for several noise events and then be considered as the criteria for classification.

Dealing with Wind

As wind noise has high energy levels at low frequencies it creates a similar pattern to transportation noise and can cause some false recognition in the system. A-weighted signals cannot be used for feature extraction and recognition tasks because the A-weighting filters out the low frequencies inherent in transportation noise which are useful for classification of this type of noise. A-weighting will affect both detection and recognition of transportation noise. In this case, the intensity detection of transportation noise events in a noisy background, particularly for the passenger cars, is difficult. Using A-weighted noise shifts the average of first peak in STFT to the higher frequencies and also the centroid of WVD will be shifted to the middle of the WVD plane rather than the left side of the plane. These two effects increase the rate of false detection and recognition.

To overcome the problem of wind interference in recognition, a knowledge based algorithm in the training mode has been developed. To detect the wind noise, the intensity level and its standard deviation for background noise will be compared with the same measures of the noise events in the training mode. If there is no transportation noise source and the intensity level and its standard deviation is as high as a transportation noise, the wind exists. When the wind is detected, a high pass filter of 50 Hz will be automatically activated to minimise the effect of wind. Also in the training part of software when the weather is constantly windy, the user will be notified that the results of noise monitoring is not reliable or cannot be performed.

RESULTS

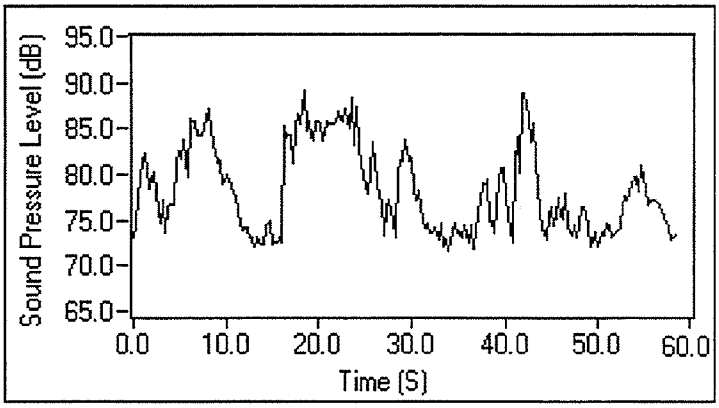
The results of 25 minutes of noise monitoring using SmartNoise, beside a classroom are shown in the figure 7 and table 1. The place where monitoring was undertaken, is exposed to single events of traffic and aircraft noise.

Figure 7 and table 1, are a part of the SmartNoise front panel and they will be updated by a new transportation noise event. The noise events which occurred during the monitoring were cars and aircraft, a mixture of car noise and loud music, three musical events, speech, whistling and strong wind. The background noise caused by wind and distant traffic was about 72 dB (A-weighted was not applied). Figure 7 and table 1, show that of the 25 minutes of recording only 58 seconds contains transportation noise which has the sound level of more than 72 dB.

The accuracy of the monitoring system is dependent on the background noise level. The quieter background noise, the better detection and recognition. Event 10 in table 1, is a false detection and recognition of transportation noise caused by an impulsive noise coincident with a wind. The judgement of true or false monitoring was based on the visual and aural observations.

Although the A-weighted noise data cannot be used for the detection and recognition tasks, the monitoring of the noise can be done in A-weighting after recognition has occurred.

Figure 7: The results of the intelligent noise monitoring of transportation over 25 minutes.



CONCLUSIONS

An intelligent aircraft and traffic noise monitoring software package has been developed in Labview. It can accurately determine statistical information about traffic noise and aircraft noise without the use of radar or visual observation.

The sound intensity, STFT and WVD methods provide a reliable method for detection and recognition. The detection and recognition rates increase when the background noise level decreases.

Development of SmartNoise is continuing and this work will include the development of a system for the monitoring of other selected environmental noises.

Table 1: Statistics of the single noise events with the time and duration of incidents are indexed in the software front panel and will be saved as a spread sheet every hour.

The Path to Save Data

c:\labview\ramin\data\

SAVE Spread Sheet or NOT

Table 1

Yes

N	Leq	L10	L90	Lmax	SEL	Start	End	Duration
1.00	81.20	85.52	73.42	87.23	93.54	3:09:48 PM	3:10:05 PM	17.14(s)***
2.00	73.56	74.47	72.60	74.94	78.06	3:11:16 PM	3:11:19 PM	2.82(s)***
3.00	84.44	87.32	74.42	89.03	95.85	3:13:58 PM	3:14:11 PM	13.83(s)***
4.00	77.38	82.40	72.71	83.87	87.38	3:18:52 PM	3:19:02 PM	10.01(s)***
5.00	76.93	78.39	73.93	79.50	82.69	3:22:22 PM	3:22:25 PM	3.77(s)***
6.00	78.63	78.78	73.35	80.81	81.75	3:23:51 PM	3:23:53 PM	2.05(s)***
7.00	84.49	87.10	74.43	89.00	90.70	3:24:19 PM	3:24:23 PM	4.18(s)***
8.00	75.59	77.19	73.19	77.94	80.79	3:24:26 PM	3:24:29 PM	3.31(s)***
9.00	75.10	75.59	73.38	76.42	74.97	3:27:30 PM	3:27:31 PM	0.97(s)***
10.00	72.17	72.17	0.00	72.17	67.82	3:30:37 PM	3:30:37 PM	0.37(s)***
11.00	76.50	80.10	72.74	81.13	86.59	3:30:39 PM	3:30:49 PM	10.19(s)***

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Hearing Protectors
- what are we trying to do?

A paper prepared for the
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Abstract

When exposed to excessive noise that is physically damaging to the ears it is tempting to assume that by the use of personal hearing protectors the problem can solved. There are many varieties of hearing protectors on the market and there are many different rating systems available to assist a user to findan appropriate device. However, there are also many reasons why so-called "hearing protection programs" don't work. Hearing protectors should be seen as a tool in the overall strategy of a noise management plan but not as the first, and often the only, attempt to reduce the risks of excessive noise exposure. This paper attempts to look at some of the pitfalls in the selection and use of personal hearing protectors.

The costs of noise

It is difficult to estimate how many Australians are exposed to high levels of noise in the workplace. However, the Bureau of Industry Economics (1991) estimated that in the Fabricated Metal Products industry 51% of the workforce were potentially exposed to noise levels above 85 dB(A). In the Basic Metal Products industry the corresponding figure was 21%. In rural communities, evidence gathered on 'field days' has shown that "of those at risk, over 87% had evidence of noise injury" (Forby-Atkinson and Meatheringham, 1994).

In NSW in 1995 there were 11,212 claims for 'industrial deafness' in the workplace. The cost of these claims was around $100m. This has risen from 5924 claims in 1992 at a cost of around $33m. This represents a significant economic burden on business and the community. However, not all of the costs of hearing loss are economic. The predominant costs to individuals with a hearing loss are personal and social and these must be considered.

One of the major ways that is regularly proposed for reducing the risk of noise exposure in the workplace is the use of personal hearing protectors. Any application of personal hearing protectors in the workplace needs to be carefully examined.

Hearing Protectors

Hearing Protectors, sometimes referred to as HPDs (Hearing Protector Devices), are devices worn to reduce the unwanted effects of sound[1]. They come in many varieties and forms.

Unfortunately it is all too easy to become trapped into believing that literature which claims that hearing protectors are the simple answer to noise exposure problems. This has been shown to definitely not be the case (Eakin: 1992, Berger and Lindgren: 1992, Scott: 1995, Waugh: 1993) and in practice, the reliance on hearing protectors in the workplace for the reduction of risk of excessive noise exposure can lead to many difficulties. Not only are there problems in implementing such programs but there have been shown to be poor correlations between the tested, laboratory attenuation of hearing protectors and the achievable, 'real world' attenuation (Murray, N M: 1988, Berger and Lindgren: 1992, Berger, Franks and Lindgren: 1994).

In Australia authorities are now following the lead of the *National Standard for Occupational Noise [NOHSC:1007(1993)]* and the associated *National Code of Practice for Noise Management and Protection of Hearing at Work [NOHSC:200991993)]* as published by the National Occupational Health and Safety Commission, WorkSafe Australia, in September 1993. The National Code of Practice states that

> *Personal hearing protectors should not be used when noise control by engineering or administrative noise control measures is practicable. They should normally be regarded as an interim measure while control of noise exposure is being achieved by these means.*

The *Occupational Health and Safety (Commonwealth Employment) (National Standards) Regulations* follow the WorkSafe guidelines and specifically require that personal hearing protectors not be seen as the solution to a noise exposure problem but only to be regarded as an interim solution until an engineering, administrative or similar control can be implemented.

All Australian States and Territories have adopted or are in the process of adopting, similar recommendations acknowledging that hearing protectors should be regarded as a last resort or interim solution and never accepted as the finalsolution to a noise exposure problem.

When calculating the potential noise exposure of an individual in a noisy workplace, any effects due to the presence of hearing protectors are by definition irrelevant and hence must be ignored. All jurisdictions emphasise this point. One point that is often over looked is that the individual is exposed to the same noise with or without the protectors, the difference in the case of wearing protectors is that the individual now experiences 'protected exposure'. We make the simplistic assumption that hearing protectors work and do reduce the risk of noise exposure to acceptable levels. Rarely does it occur to the provider that this may not be the case. Hearing protectors may not provide the necessary amount of noise reduction for many reasons, such as poor fit; inappropriate protection; lack of maintenance; lack of education; and workplace culture. (The details concerning these factors are presented below).

Types of hearing protectors

There is a vast array of hearing protectors. Their purpose is to attenuate the sound transmission path between the noise source and the inner ear. Most hearing protectors do this in one of three possible positions: over the whole ear, as in the case of ear muffs; by covering the ear canal entrance, as with canal caps; or by blocking the actual external ear canal, as with ear plugs.

There are also a range of non-linear devices that usually come either in the form of plugs or muffs. These respond in a non-linear manner to the amplitude of the sound in an attempt to reduce the incident sound on the ear drum. This can be done either mechanically - by forcing the sound to travel through a narrow aperture or 'shaped' pathway and thus restricting the passage of sudden, large amplitude noise; or electronically - by simply limiting the electronic signal delivered to the speaker; or by an electronic compression process followed by limiting high amplitude noise.

Each form of hearing protector can again come in several varieties. For example ear plugs come as pre-moulded, user formable or custom moulded; corded or uncorded; vented, un-vented or with mechanical or electronic filters. Similarly with ear muffs there are many varieties some intended for a specific purpose, for example communication muffs. There are also active noise cancellation devices amongst the many types of ear muffs

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[1] The terms sound and noise are used interchangeably in this paper.

currently on the market.

By virtue of the way hearing protectors work, through impeding the transmission pathway of the sound, there is an upper limit to the amount of attenuation they can provide. This is in the order of 35 to 40 dB. This limit is because other conduction paths which would normally be relatively minor, become the major pathways when the ear canal is blocked, for example bone conduction through the skull. It is doubtful that sound of sufficient intensity could be conducted through the skull to damage the workings of the inner ear (the cochlea) without causing other obvious physical damage.

The easiest way to achieve the maximum limit of personal hearing protection available is to use a combination of ear plugs and earmuffs. In this configuration it is the plug that plays the important role. For example, the use of a plug with a high attenuation rating in combination with a muff of relatively low rating will result in the same attenuation as using a highly rated muff in the combination. Testing found that "the choice of earmuff was relatively unimportant" (Berger; 1986, p 352).

Rating systems

There are a number of hearing protector performance rating systems in use throughout the world. Currently the only one recognised by any authority in Australia is the SLC_{80} (Sound Level Conversion)[2] system. Some rating systems are simpler to use than others but to claim that any one rating system is more accurate than another can be very misleading. Each has its advantages and disadvantages and these must be weighed up when selecting a particular system.

However, one should not be fooled into believing that because you can make a measurement of the noise and carry out what may be a complex calculation that you can predict the effective equivalent noise level to which an individual is exposed. It is all very approximate and only leads to a situation of 'protected exposure' as opposed to direct exposure. For reasons outlined below the assumption should never be made that every one can be protected all the time.

[2] You can protect 80% of the people all of the time and all of the people 80% of the time, but you can't protect all of the people all of the time. However, those most at risk are individuals who have not received sufficient education in the use and application of HPDs.

The easiest rating system to use is where the performance of the protector is matched against the A-weighted noise level measured in the environment. This system of matching hearing protectors to a noise range is applied in New Zealand. Here there are four classifications of protectors 1, 2, 3 & 4 that are considered to offer protection in levels up to 95, 100, 105 and 110 dB(A) respectively. If there are any tonal components present then use of this procedure is not recommended.

This system is easy to use and easy to implement in the workplace. It requires only one noise measurement. It does not take any specialised training and it approaches the problem on a level that most people will clearly understand.

The next simplest system to use is a single number rating system such as the Australian SLC_{80} and the American NRR Noise Reduction Rating). These systems take some training to use and practise competently. Two measurements are required in the workplace, both A- and C-weighting, and there is a calculation involved to estimate the workers protected exposure level. Again application of these systems is made on the assumption that the noise is broad band in nature with no tonal components. The current NRR system is not at all favoured in Australia as it seriously overestimates the attenuation provided by hearing protectors. This has been recognised in the US and is currently under revision. Test procedures for the rating of HPDs after the revision will be more along the lines of the Australian approach.

Both of the above systems are inherently open to inaccuracies which are easily overlooked and rather too complex to explain fully here. However, assumptions are made concerning the acoustic spectra of the offending noise. Their advantage in the workplace is that a single number is easy to use and apply. Research has shown that as the vast majority of noisy workplaces are below 100 dB(A), the classification system and a single number rating system can provide adequate protection. In any application of a safety measure it is always advisable to "keep it simple" as this is more likely to facilitate its use.

If the noise is extremely loud or complex in nature then it may be time to resort to a more complex system. There are several of these in existence. Two for example are the HML (High, Medium, Low) and HML Check methods. (The HML Check relies on a degree of guess work in application) These methods attempt to simplify a complex methodology and offer a more "accurate" method than a single number rating system. This is done by using a three number rating system that involves calculations and/or graphical prediction of the performance of the protectors. This can be more "accurate" than single number rating and the classification systems, however, caution must be applied as certain assumptions are made during the calculation phase and many scientists currently dispute their validity.

Other difficulties have also been shown to arise when implementing the HML method in that the introduction of a more complex method gives rise to mistakes in practical application. Hence any possible advantages in adopting a more complex system appear to be negated through the introduction of these errors (Thomas and Casali, 1995)

The most "precise" system currently available is the octave band method. The octave band spectrum of the offending noise is measured and this spectrm is then matched to the attenuation performance of a particular protector. So presumably one will have the exact protector for the job. This is complex to apply and as can be imagined, in practice has many drawbacks. It takes a lot of experience to apply and feel confident in using. The apparent complexity of this method frightens many potential users when in fact it is a relatively straight forward procedure.

Remember that "there is no known hearing protector that provides the upper limit [of attenuation] values for all test frequencies" (Camp, 1979). What ever you provide will be a compromise.

Some manufacturers have suggested that hearing protectors should be issued with a 'maximum performance' rating. This would, in reality, be a maximum achievable attenuation under ideal conditions. The plan would then be to issue a de-rating factor or warning stating that if the device is not used as prescribed then maximum performance should not be expected. This would be equivalent to marketing by the highest rating number. In practice there would be so many qualifications to a system such as this that it would seem to be unrealistic. Current rating systems have at least some relation to 'real world' performance.

All of the rating systems have their advantages and disadvantages and expert, independent advice should be sought before commitment is made to one particular system. There is no ideal system. It is all a matter of compromise. Be wary of anyone who offers the complete solution. By far the most important consideration when specifying hearing protectors is comfort and this currently, has no formal rating system.

The performance requirements of hearing protectors in Australia are specified by Australian Standard AS1270 *Acoustics - Hearing protectors* while their selection is detailed in Australian Standard AS1269 - *Acoustics - Hearing conservation*. Both of these Standards are currently under revision and will be combined Australian - New Zealand Standards. The latter will become *Acoustics - Occupational noise management*. There are a plethora of International and overseas Standards that can be referred to but none of them apply in Australia.

Why hearing protector programs don't work

For many years the "Hearing Conservation" approach for the solution of noise exposure in the workplace advocated fastidious wearing of hearing protectors followed by audiometric monitoring to ascertain who was not wearing their protectors. As discussed above, studies revealed that this was not an effective strategy. The failure of such programs was for many reasons.

For example as stated by Axelsson,

> *Much better ear protectors are available today and they are probably used more than previously. However, as has been demonstrated repeatedly, the fitting of ear protectors is frequently much less than ideal.*
> (Axelsson: 1996, p 127)

Lack of education: Too often hearing protectors are issued in the workplace and the education program consists of the insight "wear these or you'll go deaf". An advanced and serious approach to the problem by a determined manager may also include "if you don't wear these you'll be fired". Individuals utilising hearing protectors must understand why they are using the devices and how to use them most effectively. Maximum protection can only be achieved if the user wants to use the device, not because they are told that they have to use the device.

Lack of maintenance: Many individuals have been using the same earmuffs for years with no thought to the regular maintenance of the devices. For correct use checks must be regularly made on cushions, linings, ear cups, headband and any other part essential to best operation. Replacement of any damaged part must be carried out using the original manufacturer's recommendations. If the devices are in daily use this should be carried out on a daily basis.

Comfort: If hearing protection is not comfortable it won't be worn. Even the best hearing protector is of no use if it is not worn. Uncomfortable hearing protectors will be periodically removed by the wearer while still in the noisy environment and the overall effectiveness of protection is dramatically reduced even if it is not worn for only short periods while still exposed to the noise source. Perhaps comfort is the most important parameter to consider when choosing protection.

Culture: The culture of the workplace (and leisure setting) and acceptance of the use of hearing protectors greatly influences their use. Some 'macho' workplace cultures positively discourage the use of protectors. Typical comments from individuals in such workplaces are "you get used to the noise", "I'm already deaf", "I need to be able to hear what's going on", etc, etc.

Inappropriate protection: This can be either in the form of over protection or underprotection. Both present preventable problems in the workplace and should be avoided. Overprotection isolates individuals from their fellow workers and prevents the hearing of other, possibly essential, sounds. Underprotection can result in the individual still being exposed to excessive noise levels

Overprotection in particular, can result in protectors not being worn for the total time of exposure and hence an increased exposure to the noise.

Inappropriate protection may also include the issue of devices that are not suited to the particular work environment, eg the use of earmuffs in a hot or confined work space, or not suited to the individual, eg the issue of earplugs when the individual would prefer earmuffs. In these cases individuals may simply not use the devices or perhaps modify them in such a way that they are no longer effective.

Leaving it up to the workers: In most medium and large organisations there are often OH&S professionals, part of whose task is to disseminate information to workers and to ensure that they are educated in its use and purpose. Unfortunately in small organisations for various reasons OH&S is not seen as a pressing issue and management often 'leaves it up to the workers' as they are seen as being 'old enough to look after themselves' (Eakin, 1992). If, as is often the case, the workers are unaware of the potential hazards no preventative action is taken. The situation is summarised by the following quote by the Managing Director of a small engineering firm concerning a compensation claim by a former employee:

> *It is surely up to an engineering man with 40 years experience to care for his own health. It is not possible for us to force him to cover his ears*[3].

There are a miscellany of further reasons why individuals find it difficult to constantly wear hearing protectors, such as impairment of localisation or the loss of ability to detect directionality of any sound. This is much more pronounced with ear muffs than with ear plugs as the covering of the pinna has a significant effect. This appears to be an effect to which individuals cannot adjust. A similar effect is noticeable with the ability to judge the distance or depth perception of a sound source. Other perceived factors that deter individuals from wearing HPDs also include inability to clearly hear warning signals and lack of ability to discriminate speech when receiving instructions.

What do the manufacturers and distributors expect?

The manufacturers and distributors of hearing protectors expect you to buy and use their products. Unfortunately, contrary to what they will express at 'scientific' gatherings, most of the advertising and sales information rotates around the concept that a higher SLC_{80} or 'protection factor' is the best way to go. This is despite evidence to the contrary.

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[3]The source of this correspondence is not supplied for obvious reasons.

When issuing hearing protectors to your workforce remember that a duty of care is the responsibility of all employers and simply issuing hearing protectors with a high protection factor may not be exercising a satisfactory duty of care. The fundamental principal of a duty of care is to provide a safe working environment that does not require the use of protective equipment. If this cannot be accomplished then it is necessary to reduce the exposure risk to employees by issuing hearing protectors with the appropriate attenuation and providing an adequate level of instruction and education to ensure their proper use.

It is only realistic to reflect that most manufacturers and distributors are reliant on sales for survival and this must be taken into account when assessing their products.

What can employers expect?

It is a very general and over simplistic statement to say that employers expect a 'one stop' solution to all of their noise problems. Issuing hearing protectors to the workforce is not a one stop solution. Life is never that simple. Considering the factors previously discussed concerning why hearing protector programs don't work employers should be extremely cautious about relying on hearing protectors to reduce noise exposure.

For noise problems in the workplace the only approach that can hope to offer a total solution is 'Noise Management'. Noise management involves long term planning and objectives, and the setting of achievable goals for reducing, controlling and minimising noise in the workplace. Hearing protectors are only a small element of a noise management program. Hearing protectors can only minimise the risk of exposure to excessive noise they cannot minimise the actual exposure to the noise.

What should employeers and consumers expect?

When an individual uses a particular product he/she has the right to expect the device to perform as the supplier claims. There should be no conditions on their use except for reasonable care in their use and maintenance. There should be no necessary procedures to follow apart from the manufacturer's instructions provided with the device.

Careless use and lack of maintenance is the responsibility of the consumer not the supplier as it is with any other consumer device.

So what are we trying to do?

Through the use of hearing protectors some reduction in the risk of exposure to excessive noise can be expected. The mistake is to believe that hearing protectors are the long term solution and that they always perform 'as advertised'.

There is no doubt that hearing protectors can perform a useful function. They are a tool and just as with the use of any tool they are only effective when used for the appropriate job and in the appropriate manner. Misuse and over reliance on hearing protectors as the solution to noise problems will inevitably lead to disappointing results and the continuance of noise injury in the workplace.

A far more satisfactory solution is to use the overall approach of noise management.

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DEVELOPMENT AND USE OF THE NOISE EXPOSURE INDEX (NEI) IN OCCUPATIONAL NOISE MANAGEMENT

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ABSTRACT

Codes of practice for management of occupational noise and protection of hearing have been developed in most States of Australia. One method to assist in development of an occupational noise control strategy, as required by the Codes, is the ranking of noise exposure of employee groups and one objective method of this ranking is to use the Noise Exposure Indicator (NEI), a simple formula combining noise exposure level and numbers of employees.

The NEI concept was developed from a combination of a statistical approach to rating the effectiveness of noise control measures and determination of risk ratios for noise induced hearing loss - as the noise exposure level increases, there is a non-linear increase in the population suffering a hearing loss. This paper presents a description of the development of the NEI and its subsequent adaptation and use in occupational noise control strategies in industrial workplaces. NEI has been found to be an effective tool in assisting strategy development.

Keywords: Occupational noise, noise exposure, noise management strategies

1 Introduction

Codes of practice for management of occupational noise and protection of hearing have been developed in most States of Australia. The Codes are complementary to regulations defining limits to noise exposure at work. Where limits are exceeded in existing workplaces, the Codes include the need to determine a strategy for implementation of control systems, be they administrative or engineering.

The primary object of an occupational noise management strategy should be the prevention of noise induced hearing loss (NIHL) and associated disorders. Several factors are to be considered in developing a strategy, one of the main ones being the number of employees exposed and the level of exposure of groups of employees. A revision of AS1269-1989: Acoustics - Hearing Conservation, which provides a form of code of practice, is also soon to be published as Acoustics - Occupational Noise Management, and it also considers ranking of noise exposure areas in strategy development.

One method to assist in development of an occupational noise control strategy is the ranking of noise exposure of employee groups and one objective method of this ranking is to use the Noise Exposure Index (NEI), a simple formula combining noise exposure level and numbers of employees.

The success in the workplace in achieving the prevention of NIHL is difficult to measure in terms of the hearing ability of the employees and is much better assessed through observing progressive reductions in employee exposure to noise as the noise management process takes effect.

The NEI concept was developed to take into account the numbers of employees in the workplace within various exposure ranges. It includes a combination of a statistical approach to rating the effectiveness of noise control measures and determination of risk ratios for noise induced hearing loss - as the noise exposure level increases, there is a non-linear increase in the population suffering a hearing loss. It was intended for use with the revision of AS1269, but this aspect did not proceed.

The development of the NEI and its subsequent adaptation and use in occupational noise control strategies in industrial workplaces is described.

2 DEVELOPMENT OF THE NOISE EXPOSURE INDICATOR

2.1 Exposure Range Concept

The overall assessment of the risk of NIHL in a workplace needs to take into account both the numbers of employees affected and their noise exposure. For the calculation of the NEI, employee numbers are considered in 5 dB exposure ranges for convenience, with a separate category for exposures to peak sound levels above 140 dBLin. A 5 dB exposure range is common for noise exposure mapping, which is used with the method, as described in Section 3. The ranges are as follows:

N_{81-85}: the number of employees exposed in the range $L_{AEQ.8h}$ = 81 to 85 dB(A)

N_{86-90}: the no. of employees exposed in the range $L_{AEQ.8h}$ = 86 to 90 dB(A)

N_{91-95}: the no. of employees exposed in the range $L_{AEQ.8h}$ = 91 to 95 dB(A)

N_{96-100}: the no. of employees exposed in the range $L_{AEQ.8h}$ = 96 to 100 dB(A)

N_{101}: the no. of employees exposed in the range $L_{AEQ.8h}$ => 101 dB(A)

N_{140Pk}: the no. of employees exposed to peak levels of 140 dBLin or more

These numbers can easily be obtained from the data in the noise assessment report. The use of the 5 dB range is seen as a practical compromise. The introduction of noise controls causing a reduction of 5 dB(A) or more would indicate a number of employees moving from a higher range into the next lower range. Mehnert used this concept in a major study of the costs and benefits of noise reduction programs in Germany[i].

2.2 The Single Number NEI - Initial Development

A single number indicator can be readily developed from the above data by assigning weightings to each exposure range according to the likely levels of risk of NIHL. Robinson's 1988 data[ii] has been used to estimate relative risk of receiving a 30 dB Hearing Loss at age 60, for an employee exposed at the mid point of each range i.e.: 83, 88, 93, 98 dB(A). As Robinson's data only go up to 102 dB(A), this value is used for the highest exposure range. The Risk ratios (or exposure range coefficients) developed from the data, relative to an arbitrary value of 1.0 for 88 dB(A) are as follows:

Exposure level of	83 dB(A)	Risk ratio =	0.1
	88 dB(A)	Risk ratio =	1.0
	93 dB(A)	Risk ratio =	2
	98 dB(A)	Risk ratio =	4
	102 dB(A)	Risk ratio =	5

The risk ratios have been rounded off for simplicity. If the same analysis is done for unscreened population data in Robinson's report, the same ratios are obtained. In the absence of any other data, it would also seem appropriate to assign the weighting of 5, determined for the highest exposure range, to exposures above 140 dBLin Peak. In other words, an improvement in the NEI will not appear in respect of employees exposed above 100 dB(A) or 140 dBLin Peak, until their exposures are reduced below these levels.

Using the above ratios, the NEI can now be expressed as follows:

$$\mathbf{NEI = 0.1\,N_{81\text{-}85} + N_{86\text{-}90} + 2N_{91\text{-}95} + 4N_{96\text{-}100} + 5N_{101} + 5N_{140pk}} \qquad \textbf{(1)}$$

The equation can be used for a workplace at different stages in a noise management plan to compare the reduction in NEI, and therefore the effectiveness in the program in reducing the risk of NIHL.

The partial NEI values for employees in different ranges provide an indication as to where the most benefit is to be gained in the early stages of the noise management process. Partial NEI's can also be developed for areas of the workplace to facilitate prioritising noise control by work area.

3 USE AND DEVELOPMENT OF THE NEI

The initial development of the NEI concept was provided by Macpherson in April 1994. This was then taken for use in several occupational noise management projects by BHP Engineering. Initially, equation (1) was used as is and also in a modified version to give a greater influence to the actual sound level in the range of exposures.

The method of application of NEI was for both employee noise exposure results and for area noise exposure levels. These are described below.

3.1 Employee NEI with Employee Noise Exposure Results

As part of a noise management project, employee noise exposures were obtained for a factory. Employees were divided into different work areas of the factory, such as warehouse, mill stand area, cooling bed, shears and finishing. The NEI was used to rank the different work areas by a number of methods.

In the first method, the noise exposure results for an area were averaged to give a mean. This was used in the NEI equation with the number of employees being those normally in the area or with that job description. The different work groups were then ranked to show the highest ranking area. This would be the area with the highest combination of employee numbers and risk of NIHL. The calculation was repeated for the maximum noise exposure result obtained for the group to indicate the range of variability occurring from the mean and the potential maximum NEI expected for the group. The NEI was then used to rank the different groups. Table 1 shows the results for a typical factory survey.

Table 1: NEI for Employee Noise Exposure at Factory #1

Work Group	No. in Area	LAEQ Mean	NEI Mean	Rank Mean
Rollers	10	91.7	20	1
Cranes	5	94.9	10	4
Shears	8	92.5	16	2
Cooling Bed	6	94.3	12	3
Slingers	5	86.7	5	5
Maintenance	6	91.1	12	3
Total	40		75	

There was some minor change between ranking when using the maximum and mean results, but the higher ranked areas only changed by one level. The results in Table 1 indicate the first priority for treatment would be the group of Rollers. If their dose was reduced by 5 dB(A), the NEI for the group would reduce by 10, as would the Total NEI. To account for there being a change in employee numbers, there could also be a Normalised NEI, given by the total NEI divided by the number of employees. For the above this would be 1.875.

3.2 Area NEI with Area Sound Exposure Level Results

The second method used (in the same study) related to the sound level occurring in different areas of the plant. LAEQ sound levels were measured on a pseudo grid pattern over a whole plant. From these results, a noise exposure map was prepared having a 5 dB(A) contour interval. This is a common approach to occupational noise exposure management and allows a visual assessment of the areas of a plant where high noise exposure levels occur.

From the map, the average noise exposure of the employees in a particular area of the plant was estimated. This was then used with the number of employees and the NEI equation to identify the work areas of the factory where the highest risk to NIHL occurred. Table 2 shows a typical result for the first factory studied.

Table 2: Area NEI and Ranking for Factory #1

Area	**No. in Area**	**Area LAEQ**	**Risk Coeff.**	**NEI**	**Rank**
Stands	10	92	2	20	3
Repair	4	87	1	4	7
Compactor	6	90	1	6	6
Carrier	20	82	0.1	2	8
Shears	8	97	4	32	1
Bundler	6	98	4	24	2
Battery #2	6	92	2	12	4
Cranes	5	93	2	10	5
Warehouse	5	85	0.1	0.5	9
Total	70			110.5	

Using the area approach gives a better indication of where the highest risk occurs, and can be used with the employee noise dose NEI (measured later) to further refine the ranking, if required. A maximum level in the area could also be used to identify the variation in the ranking if all employees in an area were exposed to the maximum measured in that area. So there are many possible combinations. A noise control priority map can also be produced, separating the plant into high, medium and low priority based on the ranking of the area. Such maps assist in focusing management attention on the areas for noise control measures. As with the employee NEI, the normalised NEI could be calculated to account for changes in employee numbers, or to compare between plants.

Both methods of calculating NEI have been used in combination with other factors in developing priority lists for noise management plans. These other factors include costs to implement controls, ease of engineering, time to implement and down-time required, amongst other things.

3.2 NEI Equation Development and Refinement.

With the initial equation, difficulty was found when the LAEQ was close to the border of one range - for example, an LAEQ of 90.5 would have a risk coefficient of 1, but be almost equal to 2 because it was almost 91 dB(A), where the coefficient changed. Whilst this is not a major issue if rounding off results, where there are a large number of results and a spreadsheet calculation approach is used, manual rounding off is time consuming and it does not account for the variations in risk along the exposure range (for example: is 8 employees at 93 dB(A) a greater risk than 6 at 94 dB(A)?).

Use of Risk factor or Dose?

This then leads into whether to use an NEI based on the risk factors determined from Robinson's data, or based on dose, and whether to make the range smaller, say 3 dB rather than 5 dB. If dose is used instead of risk, a higher ranking is given to higher noise dose results. This is because dose is energy related, whereas the risk data does not rise at the same rate. This lower risk rate may be due to higher exposure workers generally using some form of protection, or removing themselves from the data, or some other factor.

If a lower range of groups is sought, the risk factors can be determined from Robinson's data. One could go to 1 dB steps, but this would create more work, and more uncertainty because of the variability between repeated measurements. In survey results, it is often relatively easy to group employee dose results into 5 dB ranges, and perhaps 3 dB would not be much more difficult - however this has yet to be tried. Development of this area is something for further discussion.

Variations to the equation were then tried to take account of this variability in the range of coefficients. and the rankings again compared for the different equations. The two new equations were:

$$\mathbf{NEI\#2 = C_i * Log\,\{N_i * 10\text{^}(L_i/10)\}} \qquad . \quad . \quad . \quad . \qquad (2)$$

$$\mathbf{NEI\#3 = N_i * C_i * 10\,Log\,\{L_i/10\}} \qquad . \quad . \quad . \quad . \qquad (3)$$

where: L_i is noise exposure L_{AEQ} measured or estimated, dB(A)
C_i is the NEI coefficient for the range of noise exposure L_i
N_i is the number of employees with exposure L_i

A coefficient of 0.05 was also given for the range 70 to 80 dB(A),.

Equation (2) was found to give a relatively higher risk to higher sound exposures than was considered appropriate when compared to other employee groups with large numbers of people at slightly lower sound level. For example, for one employee at 92.5 dB(A) and ten employees at 89 dB(A), the comparisons were as follows:

Exposure Group	NEI#1	NEI#2	NEI#3
1 at 92.5 dB(A)	2	18.5	19.3
10 at 89 dB(A)	10	9.9	94.9

The higher area risk given by NEI#2 was considered inappropriate for use in developing priority actions for a noise management plan. After a number of comparisons, NEI#2 was discarded and use of NEI#3 was continued.

3.3 NEI in Before and After Comparisons

As well as identifying the ranking of different work areas for use in developing a noise management plan, the NEI can also be used to indicate the success of implementation of noise control measures, one of the original objectives of developing the NEI. This has been done at a second factory, where a number of control measures were implemented and a significant reduction in sound level was measured.

In a follow-up survey after the implementation, the noise doses were not measured but area sound levels were as they were much easier to carry out. This then allowed a comparison of NEI's for the different areas, and a comparison of the total NEI for the whole plant before and after treatment. In this particular site, there were 21 different work areas. Treatment in 6 areas resulted in reduced sound levels in 13 areas for 32 out of 68 employees. For the areas treated, the reduction in total NEI was 56%. For the total factory, the reduction was 25%.

For management (and noise management workers) there is a greater understanding in being told that the noise control treatment has resulted in a reduction in risk of NIHL of 25% for the workforce as a whole, compared to being told the sound levels were reduced by 2 to 8 dB(A) for 45% of the workforce.

Table 3: NEI before & after Implementation of Noise Control at Factory #2

Work Area	No. in Area	Area Exposure LAEQ		NEI Coeff		NEI Area #3			Rank Area #3	
		Before	After	Before	After	Before	After	Ratio	Before	After
Single Hole	3	92	90	2	1	57.8	28.6	0.50	3	4
Welded Fabric	5	90	88	1	1	47.7	47.2	0.99	4	3
Extruder	1	92.5	85	2	0.1	19.3	0.9	0.05	5	11
Gal 2 Baths	2	90	84	1	0.1	19.1	1.8	0.10	5	10
Gal 1 Baths	2	90	84	1	0.1	19.1	1.8	0.10	5	10
Weaving	2	90	85	1	0.1	19.1	1.9	0.10	5	9
Wire Netting	2	86	84	1	0.1	18.7	1.8	0.10	5	10
Barb	1/3	90	83	1	0.1	9.5	2.7	0.28	6	11
Gal packing	4	80	78	0.1	0.05	3.6	1.8	0.49	7	10
Gal 2 winding	3	85	82	0.1	0.1	2.8	2.7	0.98	8	8
Gal 1 winding	3	85	82	0.1	0.1	2.8	2.7	0.98	8	8
Gal 2 Pay-off	1	85	80	0.1	0.05	0.9	0.5	0.49	9	12
Gal 1 Pay-off	1	85	83	0.1	0.1	0.9	0.9	0.99	9	11
Total	30/32					221.4	95.5			

Ratio After : Before 0.43

Equation 3 gives a higher risk than NEI#1 for exposures above 100 dB(A) - it was developed for the higher exposure levels to overcome the potential problem of the constant risk factor of 5 from Robinson's data for exposures above 100 dB(A). An alternative could be to use higher risk factors for the higher exposure ranges - say 5 for 101 to 105 dB(A), 8 for 106 to 110 dB(A), and so on (if the exposures go that high).

4 CONCLUSION

The Noise Exposure Indicator has been developed as a tool for use in occupational noise management. It relates an objective assessment of the risk of NIHL in a workplace to the number of employees and the sound level of exposure. It has been used in several workplace noise management strategies to assist in developing priorities for implementation of noise controls. It has also been used to compare the risk potential of a workplace before and after noise controls have been implemented - this can be used to demonstrate to the workforce and Occupational Health authorities that there has been a reduction in the risk and what degree of reduction has occurred. Such an approach is considered easier to understand when there has not been a uniform reduction in noise exposure in all areas.

Continued use of the NEI and its wider application is expected to lead to better noise management strategy development and result in greater acceptance of the benefits of noise control programs by all concerned. Its further use and development is recommended.

ACKNOWLEDGMENTS

The opinions expressed in this paper are those of the authors' alone. Our appreciation is expressed to the employers for the opportunity to present the paper, and other members of Standards Australia Technical Committee AV3/2 for their assistance in developing and using the concept.

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NOISE REDUCTION FOR FRICTION SAWS

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ABSTRACT

Friction saws are used extensively in industry to cut steel and aluminium pipes and structural sections. In some cases they are the only type of saw which is suitable for the task. However in many other cases they are used because of their speed even though friction sawing produces excessive noise. Little research has been done into noise reduction measures for this type of saw. The first step in achieving noise reduction is to have an understanding of the main factors contributing to the noise. The findings from noise measurements for a case study of a friction saw will be summarised. The effects of changing various parameters associated with the sawing process were investigated and the potential for noise reduction will be discussed.

1. INTRODUCTION

A friction saw is a fine toothed, circular saw which is operated at high rotational speed. The friction at the saw/metal interface produces a local softening of the metal which then leads to a separation at the saw line. The cut surface is generally rougher than for other metal sawing operations and may require grinding to achieve a smooth surface. These saws are used extensively in industry for cutting thin metal pipes and sections where other cutting saws are not suitable as they tend to "grab" the material. Because of their fast cutting action, friction saws are also used as cut off saws for light and heavy sections and for many other applications in industry.

Friction sawing produces excessive noise levels for the operators, and for others in the vicinity. While there has been considerable research into noise generation and subsequently noise reduction measures, for other types of saws, there has been little investigation of the noise from friction saws [1].

The first step in achieving noise reduction is to have an understanding of the main factors contributing to the noise. This means that the various parameters need to be identified and the effects of changes monitored. For friction sawing the major elements of the process which could have an effect on the noise are:

- the saw blade - the material of the blade, the shape of the teeth, blade diameter etc;
- the operation of the saw - rotational speed, tip speed, feed rate etc;
- the maintenance of the components; and
- the radiation from the product being cut - vibration transmitted into the product during the cutting process.

The ideal situation would be to have a friction saw under laboratory conditions so that one parameter could be changed at a time and the effect on the noise monitored. As such an arrangement was not available, investigations were made primarily using a friction saw in a production environment. A limited range of tests were undertaken in a laboratory environment. In this paper, the findings of a research project, funded by Worksafe Australia and supported by BHP Civil Products will be discussed.

2. CASE STUDY - HELCOR SAW

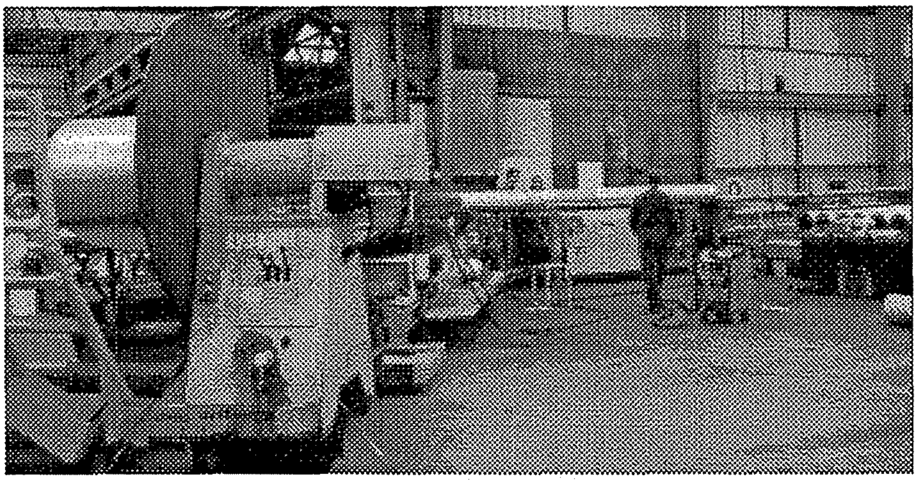

Figure 1 Helcor machine. Flat sheet on left enters formers then is spiral wound, seamed and the complete pipe emerges on right. An operator is standing near the saw assembly.

Helcor pipes are produced by roll forming sheet steel or aluminium and then forming a pipe by spiral winding and seaming. The Helcor production machine, shown in Figure 1, is used to produce a range of products with diameter varying from 300 to 3600 mm and wall thickness from 1.5 to 3.5 mm. To cut the pipe to the required length, a friction saw has been found to be the most suitable saw. The chrome-vanadium saw blade used is 400 mm diameter and 6 mm thick with 240 v-type teeth. The normal rotational speed is 4350 rpm. The pipe is cut while the production process continues. The saw assembly is mounted on rails parallel to the axis of the pipe and moves along with the product. The saw blade is below the pipe which continues to rotate until the cut is complete. At this point the saw retracts into the assembly and moves back, ready to cut the next pipe.

3. SUMMARY OF FINDINGS

As a series of noise measurements were to be made during production it was important to establish a fixed measurement position and a standard product so that the effects of changes could be monitored. A convenient position, near the operator location was selected and the majority of the noise measurements were made for steel pipe 1.6 mm thick, with 68 mm x 13 mm corrugations.

The noise level variation during a cut for 1200 mm diameter pipe is shown in Figure 2. As the noise level varied during the cut, possibly due to the corrugations and the seam, the noise was monitored in terms of the L_{Aeq} for the time of the cut using a Larson Davis Model 700 Sound Level Meter. The reproducibility was found to be of the order of ± 1 dB(A).

Within the limits of the machine and the production process, a number of parameters were changed and the effects of changes are shown in Table 1 [2,4,5]. In order to investigate more carefully certain aspects associated with the saw operation, measurements were also made in the laboratory where a friction saw bade was used on a milling machine to cut flat sheets of steel and aluminium. The findings from these tests [3] are also included in Table 1.

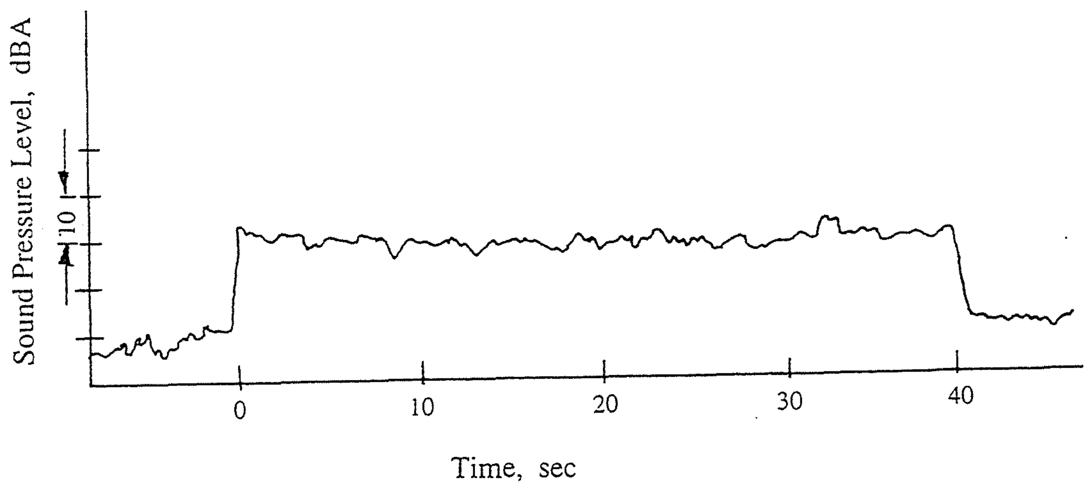

Figure 2 Noise level during cutting of 1200 mm diameter, 1.5m long, galvanised steel pipe, 1.6 mm thick with 68 mm x 13 mm corrugations

Table 1 Effect on noise level in terms of L_{Aeq} for changes in various parameters.

DESCRIPTION OF ACTION	EFFECT on CUTTING NOISE	
CHANGES TO SAW BLADE		
Metal spray one side: low carbon steel	**Decrease**	**4 dBA**
Metal spray both sides: low carbon steel	**Decrease**	**4 dBA**
Damping sheet on blade	**Decrease**	**2 dBA**
Damping collars on blade	**No change**	
Different tooth profiles	**Increase**	**3 to 7 dBA**
CHANGES RELATED TO SAW OPERATION		
Saw rotation clockwise vs anticlockwise	**Change**	**1 dBA**
Saw speed increase: 2060 to 4350 rpm with pulleys	**Decrease**	**7 dBA**
Tip speed increase: 375 to 400 mm diameter blade	**Increase**	**4 dBA**
Feed rate increase: 2.41 to 4.68 m/min, gear 2 to 3	**Increase**	**1 dBA**
Feed rate increase: 4.68 to 11.31m/min, gear 3 to 5	**Increase**	**3.5 dBA**
MAINTENANCE		
New bearing on saw shaft	**Decrease**	**3 dBA**
Sharpened saw teeth	**Decrease**	**3 dBA**
DAMPING OF PRODUCT		
Steel collars wrapped on pipe at 25 mm from saw	**Decrease**	**3.5 dBA**
Steel collars wrapped on pipe at 50 mm from saw	**Decrease**	**2 dBA**
Steel collars wrapped on pipe at 100 mm from saw	**Decrease**	**2 dBA**
Loaded vinyl over pipe at 200 mm	**Decrease**	**2 dBA**
Damping of guide rollers	**Decrease**	**1.5 dBA**
LABORATORY TESTS		
Saw speed increase: 1600 to 2500 r.p.m. cutting steel	**Decrease**	**5 dBA**
Feed rate increase: 0.25 to 0.5 m/min, cutting steel	**Decrease**	**3 dBA**
Feed rate increase 0.5 to 2.0 m/min, cutting steel	**Increase**	**10 dBA**
Feed rate increase: 0.5 to 2.0 m/min, cutting Al	**Decrease**	**1 dBA**
Material thickness increase: 1.6 to 3.5 mm steel	**Increase**	**8 dBA**

4. DISCUSSION

From the laboratory tests it was found that noise levels depended on both feed rate and saw speed in a non-linear manner. It would appear from the tests on the Helcor saw that the normal operating conditions are also close to an optimum as, with the exception of the replacement of the bearing, all the changes to the operation of the saw led to increased noise levels. The normal operating conditions for the saw have been established on the basis of experience for the quality of the product. The greatest potential for the reduction of the overall noise level appears to be from changes to the saw blade and with the application of damping to the product.

As can be seen from Table 1, the replacement of a worn bearing on the saw shaft and sharpening of the teeth of the blade both led to decreases in the noise level. This points to the importance of effective maintenance in achieving low noise in any machinery operation.

Treatments aimed at damping the vibrations of the blade such as metal sprayed onto the surface and damping collars, led to reductions of the order of 4 dB(A). The nature of this product makes the application of damping to the product in normal production, particularly difficult without a major redesign of the process. The trials of prototype damping of the product showed that a decrease of 2 to 4 dB(A) could be achieved.

On the basis of these investigations, the total noise reduction to be achieved with the optimum conditions for all the components would not produce sufficient reduction for the operators to dispense with personal hearing protection. However, the improvement will assist in reducing the noise exposure for the other workers elsewhere in the factory.

As a suitable alternative, low noise method for cutting the product was not found to be practical, it is necessary to consider the less acceptable option of an enclosure. For some uses of friction saws, where the nature and range of the product permits, an enclosure of the machine may be possible. However when the same machine is used for a varied product range, as is the case for Helcor, a machine enclosure would be impractical. The remaining option, without a complete change in the whole cutting process, is an enclosure for the operator. While this would require changes in work practice, such an enclosure would be practical.

5. CONCLUSION

The findings from noise measurements for a case study of a friction saw, have shown the parameters that have the most potential for noise reduction. Once the saw is operating at optimum conditions for the cutting process and maintenance is effective, damping of the saw blade and of the product offer the best potential for noise reduction. However the noise reduction achieved may still not be sufficient to reduce excessive noise exposure for the operators. Alternative cutting methods or enclosures for either the machine or the operator, may need to be considered.

6. ACKNOWLEDGMENT

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NOISE, THE CUSTOMER AND MULTI-UNIT DWELLINGS

ABSTRACT

Traditionally dwellings in New Zealand are detached houses of light timber frame construction, having low levels of sound insulation performance and the whole residential building industry has developed materials, skills and designs which have not been constrained by a need for high sound insulation.

However, the changing use of the space in the home and the relentless rise in customers' expectations of quality plus a move towards attached multi-unit dwellings has led to a requirement for better performance from building elements. Research has shown that residential noise has become a quality of life issue for the majority of New Zealanders.

Results are presented from work applying available passive technology - based on standard control techniques - to the challenge of providing privacy and acoustic comfort in dwellings of light timber frame construction.

Emphasis is given to the need to bridge the gap between the cognitive approach of acousticians and the pragmatic needs of the end user.

Noise, The Customer and Multi-unit Dwellings

Ken McGunnigle

Winstone Wallboards Limited

My Definitions

- Noise: Subjective sound, merely heard, usually unwanted.
- The Customer: The occupier of the space and user of the acoustic environment.
- Multi-unit Dwelling: A timber frame building up to 6 storeys.

Objective

- Addressing the Customer's Needs

Customer Requirements

- Ask Customers what they want, in their terms.
- Measure the acoustic performance of a space in customer terms.
- Provide "whole building" solutions.

Meeting the Needs

- Performance based building codes.
- The New Zealand Building Code Clause G6.

NZBC Clause G6 AIRBORNE AND IMPACT SOUND

This Clause is extracted from the New Zealand Building Code contained in the First Schedule of the Building Regulations 1992.

Provisions

OBJECTIVE

G6.1 The objective of this provision is to safeguard people from illness or loss of *amenity* as a result of undue noise being transmitted between abutting occupancies.

FUNCTIONAL REQUIREMENT

G6.2 *Building elements* which are common between occupancies, shall be constructed to prevent undue noise transmission from other occupancies or common spaces, to the *habitable spaces* of *household units.*

PERFORMANCE

G6.3.1 The *Sound Transmission Class* of walls, floors and ceilings, shall be no less than 55.

G6.3.2 The *Impact Insulation Class* of floors shall be no less than 55.

NZBC Limitations/Omissions

- External noise
- Airborne noise between spaces within a dwelling
- Impact noise for walls
- Horizontal impact noise for floors
- Banging doors
- Plumbing noise
- Appliance noise

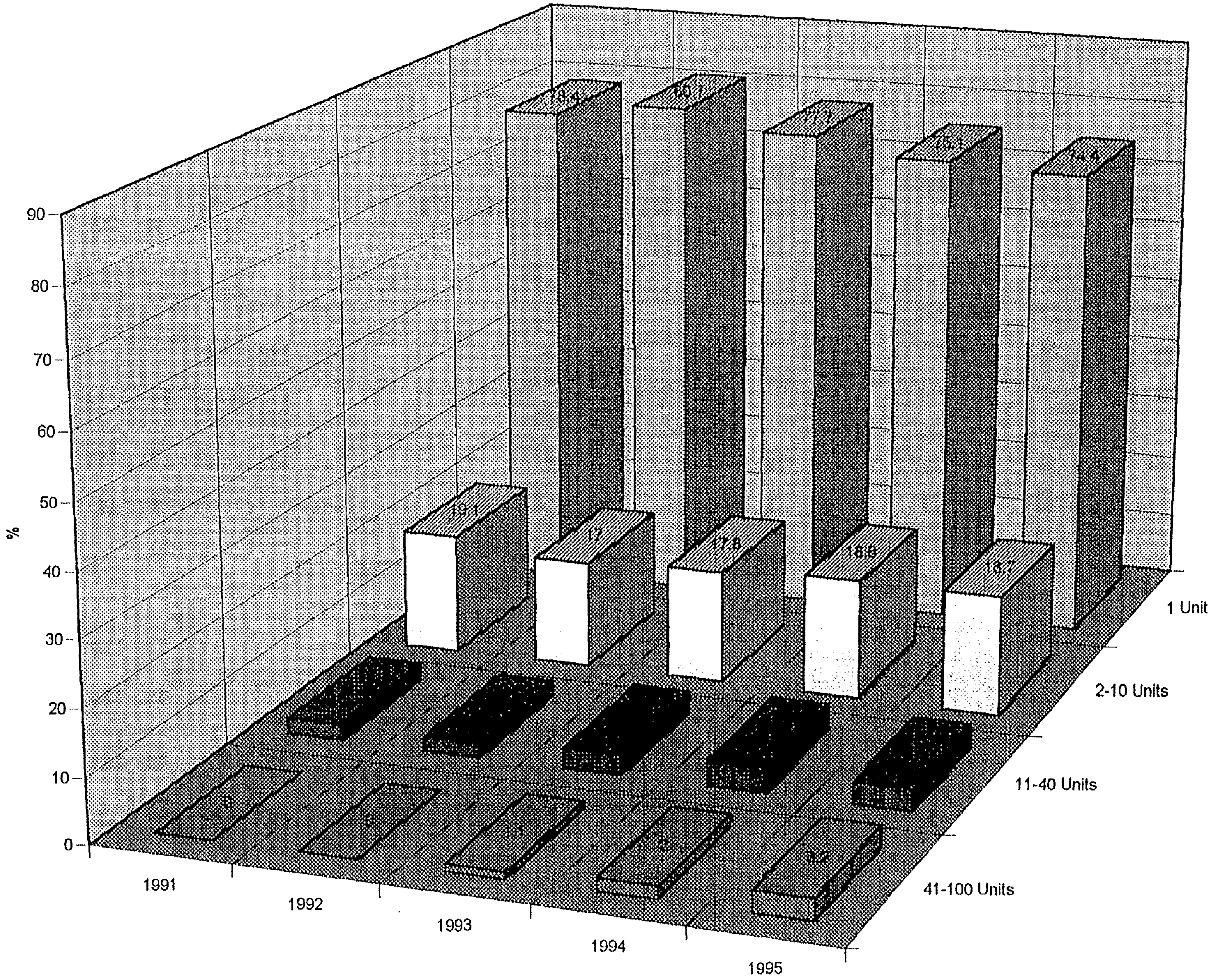
Market Research

- Interviews to establish customer needs in their terms
- Derive a market model
- Draw strategic conclusions

Market Trends

- Statistics New Zealand

Dwelling Type 1991 to 1995
Source; Statistics New Zealand



Marketed Systems

- Challenge; to use local traditional techniques, materials and skills.
- Double timber studs
- Single timber studs
- Direct fix clip floor/ceilings

Communications

- Simple message via TV to women in the home
- Simple benefits leaflet for customers
- Staff training; theory & practical
- Specialist subcontractor training
- Comprehensive technical literature
- Interactive computer software to all specifiers

Commercial Considerations

- Lower installed cost
- Simpler
- Thinner/narrower
- Least on-site labour hours
- Use local labour skills and methods
- Equal or better performance

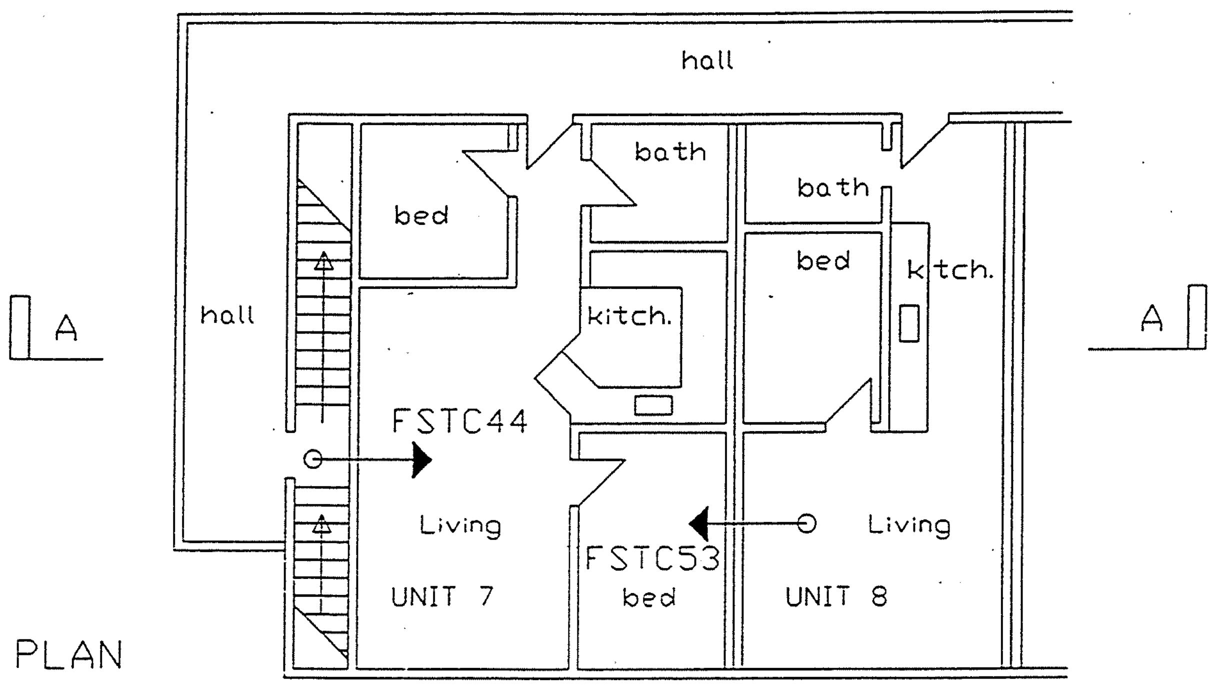
Field Tests

- Multi-storey apartments in Wellington
- Multi-storey apartments in Christchurch

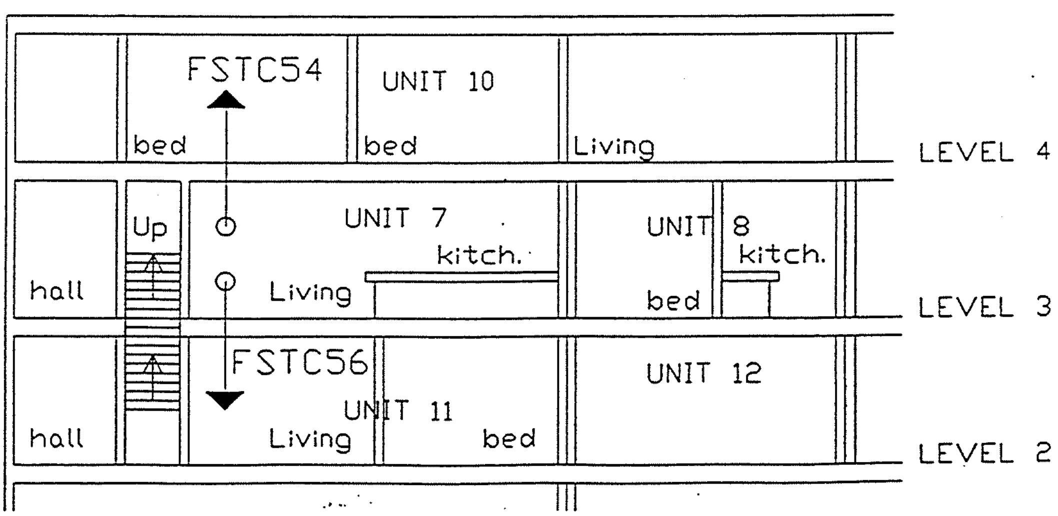
WINSTONE WALLBOARDS LTD

WELLINGTON APARTMENT NOISE STUDY

FIGURE 1: FSTC Test Results



PLAN
LEVEL 3



CROSS SECTION AA

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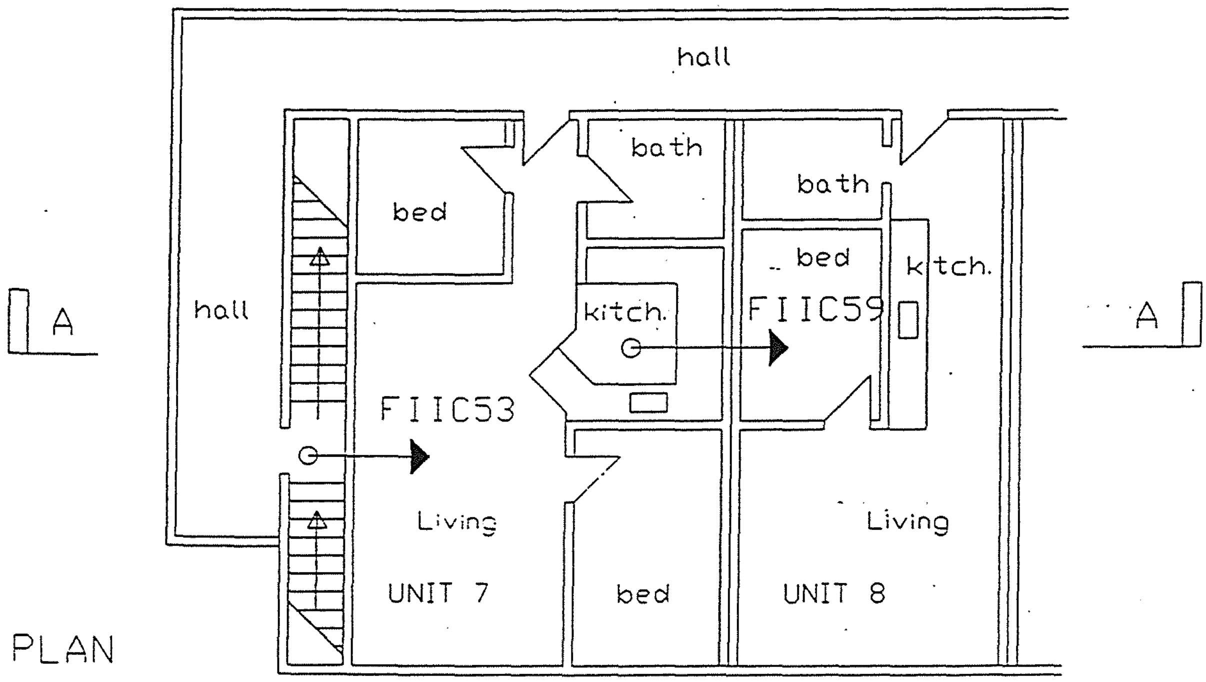
WELLINGTON APARTMENT NOISE STUDY

FIGURE 2: FIIC Test Results



PLAN
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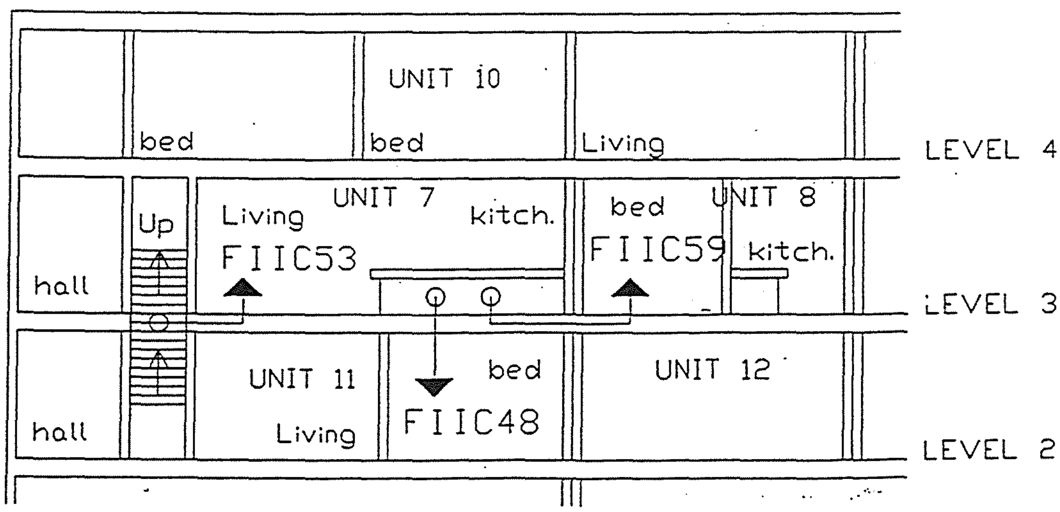

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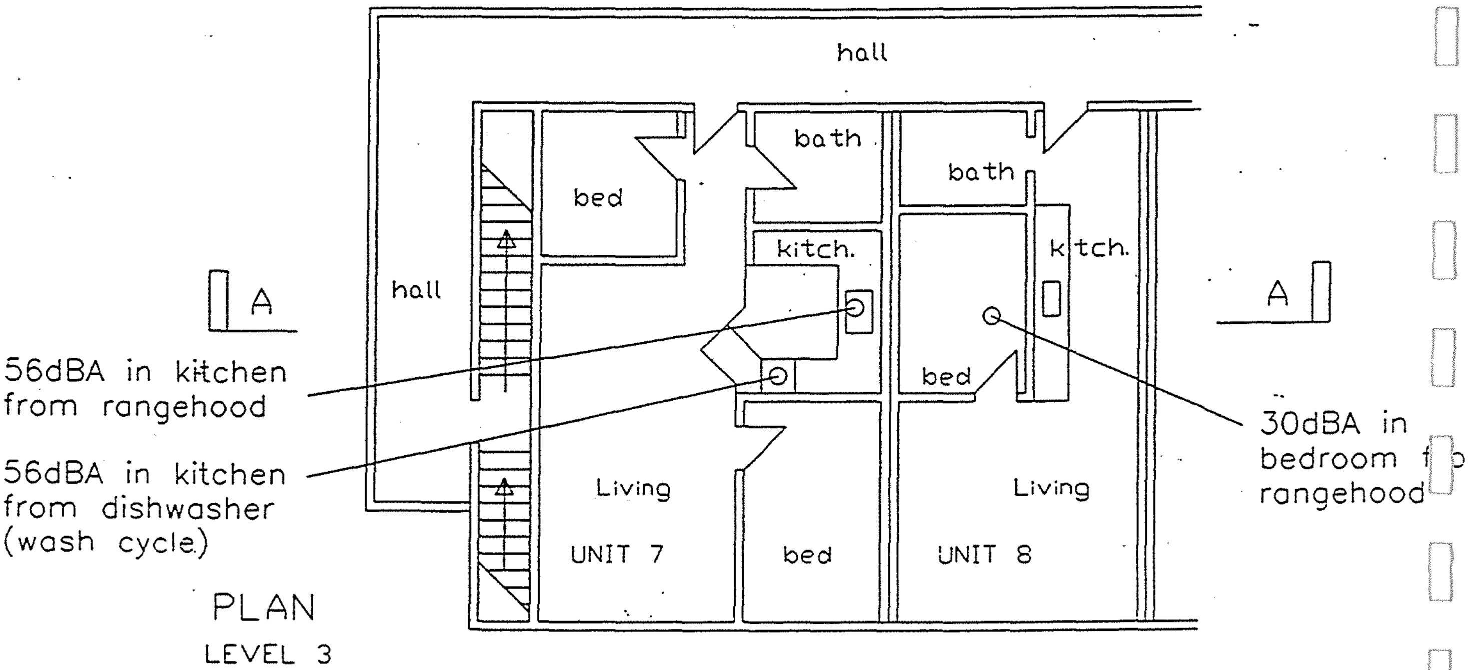
FIGURE 3: Appliance Noise Transmission



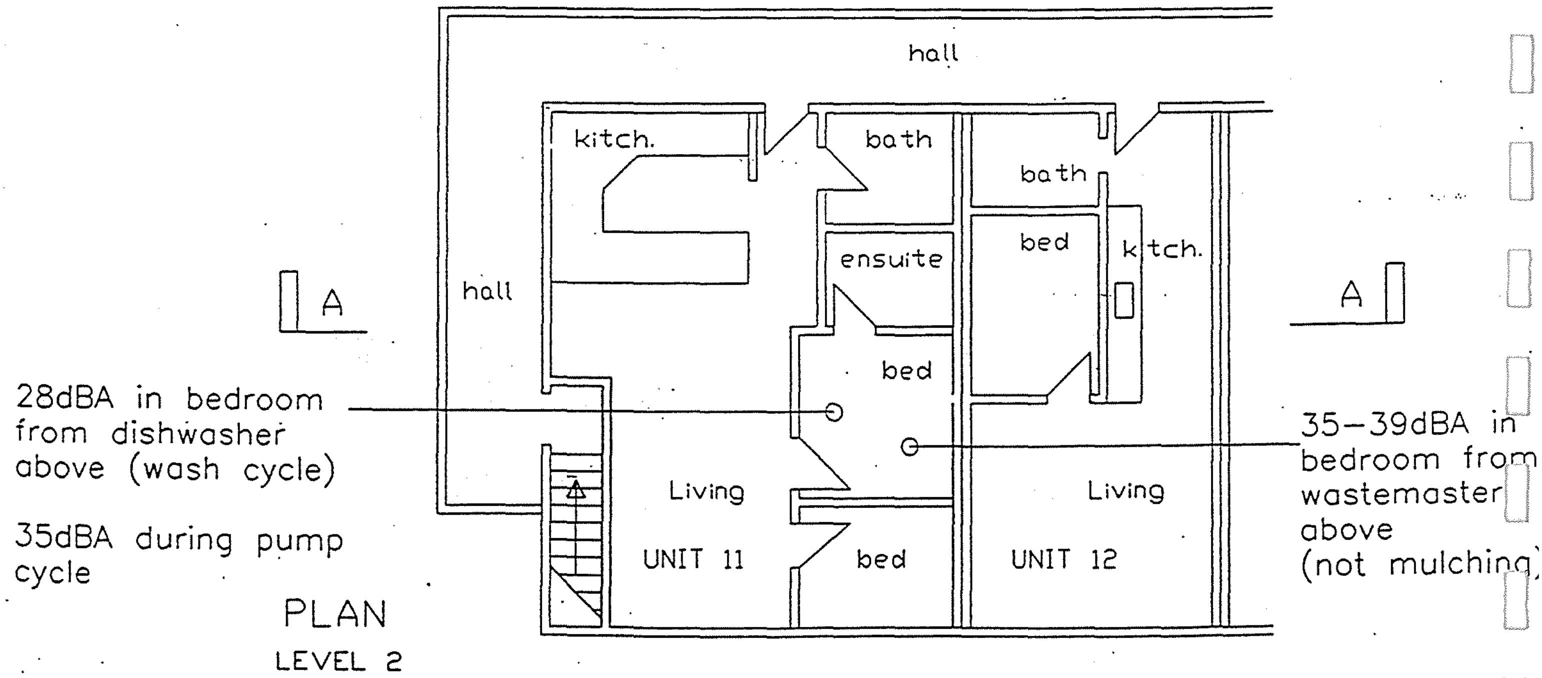


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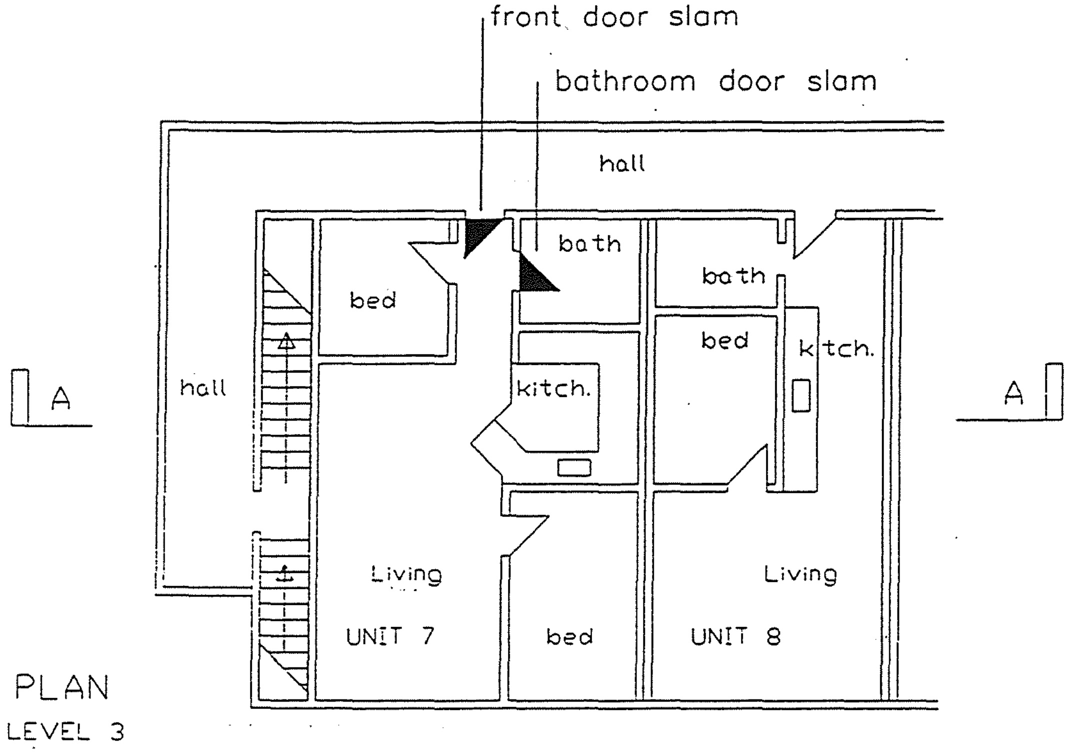
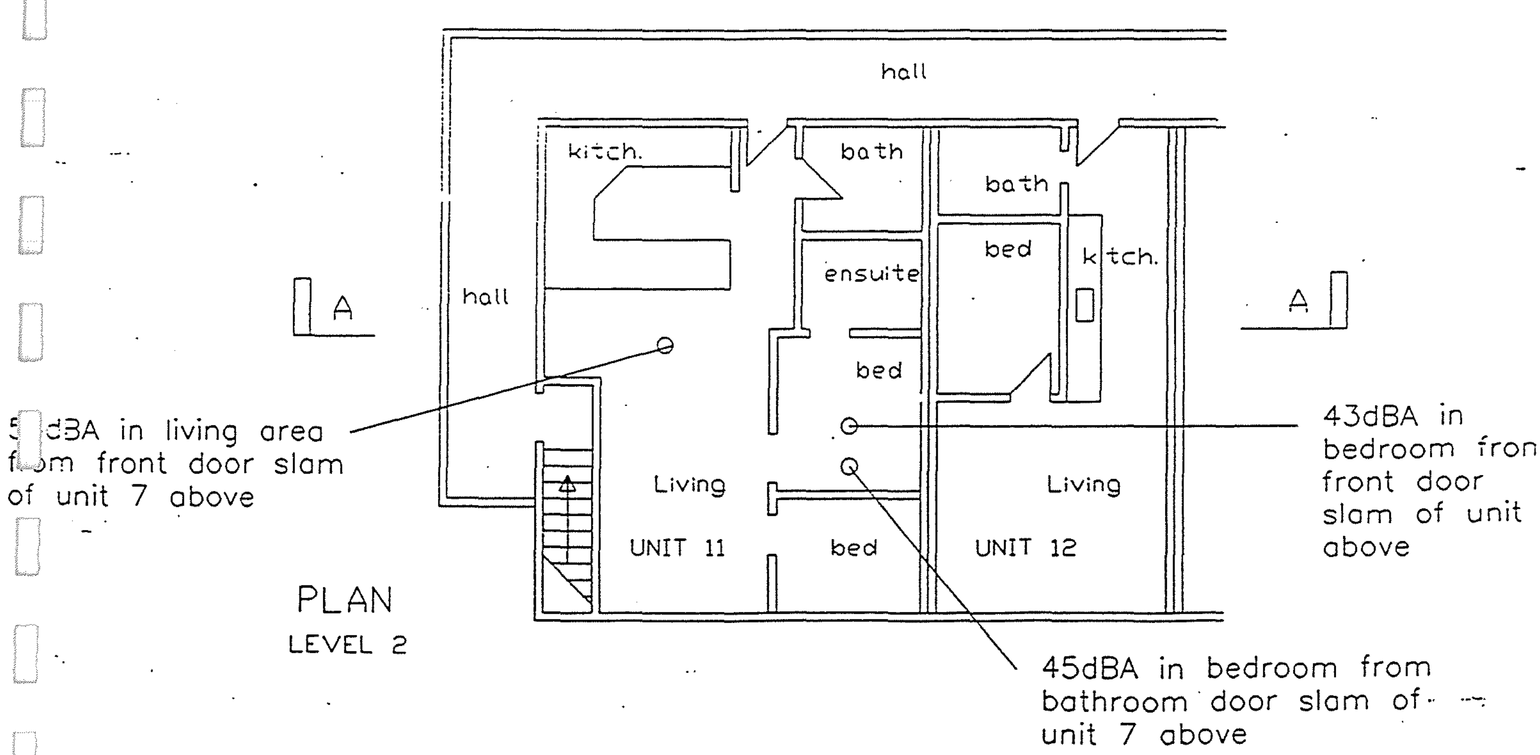
FIGURE 4: Door Slam Noise Transmission





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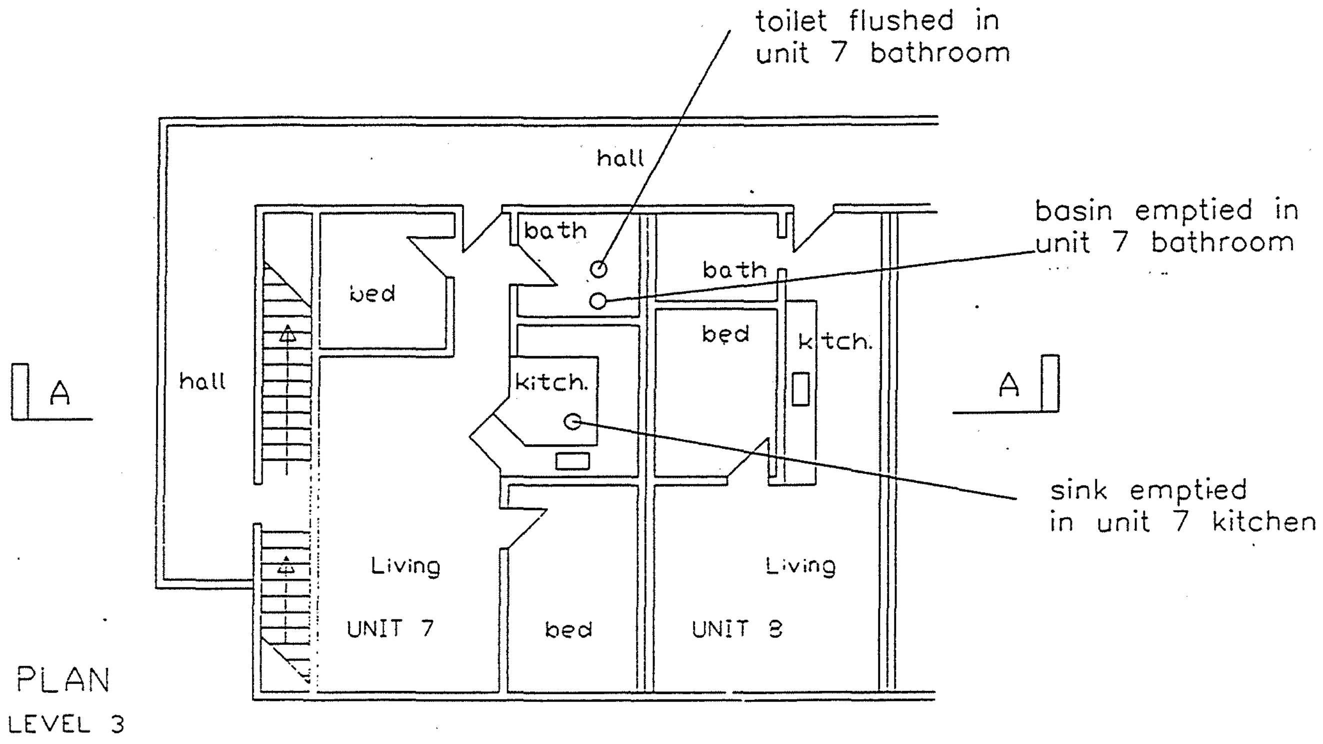
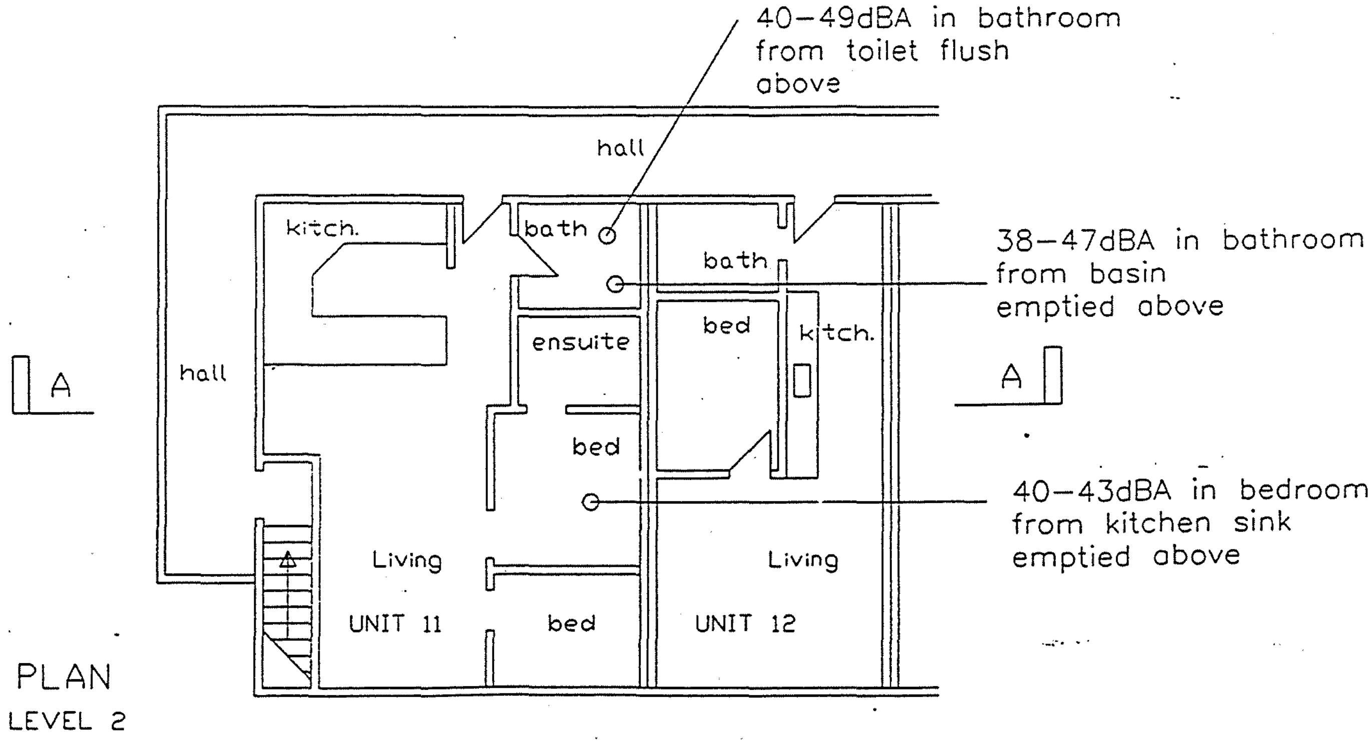
WELLINGTON APARTMENT NOISE STUDY

FIGURE 5: Waste Water Noise Transmission





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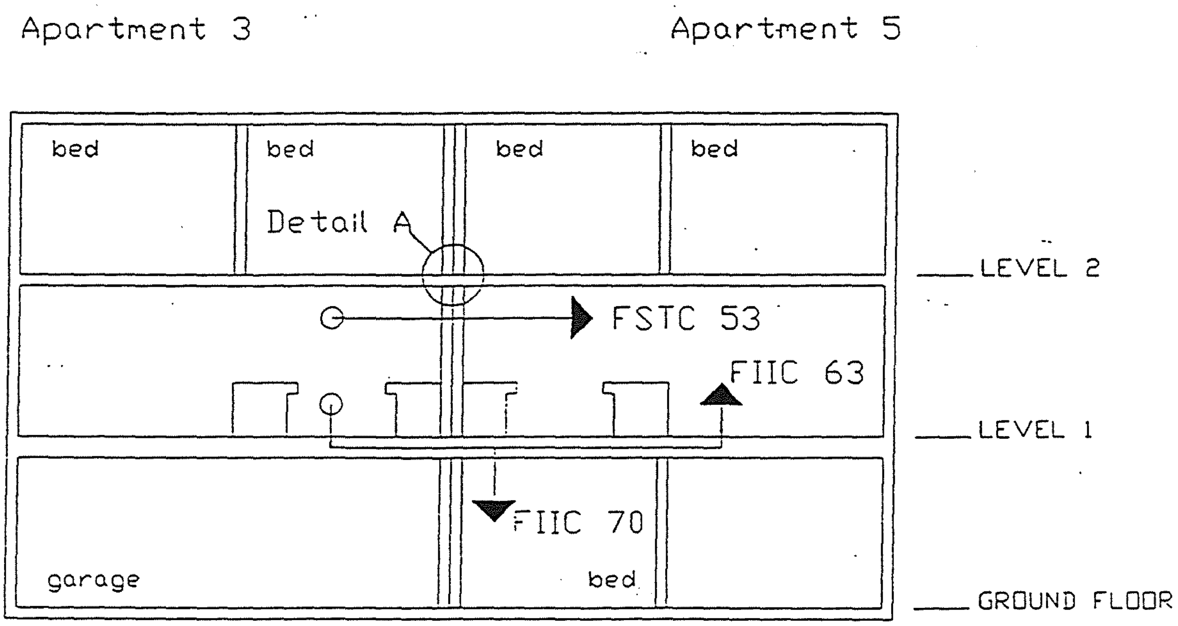
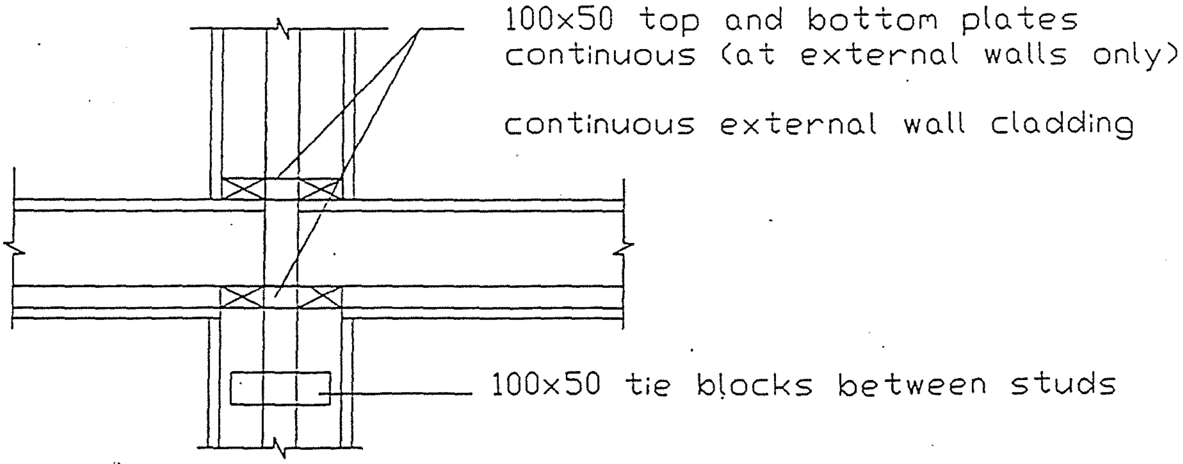
CHRISTCHURCH APARTMENT NOISE STUDY

Figure 1: FSTC and FIIC Test Results
Salisbury Apartments 3 and 5



CROSS SECTION



DETAIL A (at external wall)

Key Benefits

- Investigate what the customer wants from their perspective
- To provide the customers with what they need

Compare Europe & New Zealand

- Occupiers have similar needs
- The layout of the apartments are essentially the same
- NZ Intertenancy performance STC 55
- European aim STC 65+

The Challenge to Acousticians

- Investigate and understand end user needs in their terms
- Measure effectiveness of your actions in customer terms

SUBJECTIVE EVALUATION OF RESONATING ROOM SOUNDS: PRELUDE TO A STUDY OF RESONATING SOUND ART

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ABSTRACT

A reverberation room was set into resonance by repeatedly re-recording what began as white noise through a loudspeaker and microphone positioned within the room. Particular room modes were progressively stimulated by this process. Eight different loudspeaker/microphone positions were employed in this manner to produce experimental stimuli. Forty-eight 5s stimuli were evaluated by a small number of expert subjects in paired comparison and semantic differential tests. Generally, subjects positively evaluated rich timbres, diatonic-like sonorities and deep tones. This study was undertaken to explore the prospect of evaluating works of acoustic art in which extreme resonance is employed.

INTRODUCTION: SOUND ART AND RESONATING SOUND ART

While it would be problematic to assign a beginning to sound art, the diversification of the arts which modernism fostered, and the availability of electroacoustic technologies have seen a flourishing in this field sufficient to distinguish it from other artistic practice (Lander D., Lexier M.; Kahn D., Whitehead G.). Contemporary sound art is broad and disparate, influenced by a diverse range of aesthetic interests and philosophical currents, and involving distinctive genres. One of the most important of these is works produced specifically for radio broadcast. Such works attempt to transcend well-rehearsed radio formats, and to create an art idiomatic to radio. There is often an emphasis on the quality of the sound itself, and the medium may be used in an opaque manner. In sound sculpture acoustic phenomena can sometimes have a profound interaction with physical and visual forms. If music privileges temporal aspects of sound, sound sculpture privileges spatial aspects. Interactive installations might be third genre - where, for instance, a computer responds to a listener's movements by varying sound sources. A sometimes popular genre of sound art consists of making recordings of the environment, the acoustic equivalent to photography. Performance art is another disparate category, where sound may have a leading role. Thus sound art is a broad, fluid and developing medium. The study that the remainder of this paper represents is the beginning of an investigation into just one small area of the medium, which might loosely be called 'resonating sound art'. Such works employ extreme acoustic resonance, often over extended time periods. The contribution of a performer may be minimised, so that the emphasis is on the resonances rather than on performative interventions.

Alvin Lucier is one artist who has produced works falling into the category. In 'I am Sitting in a Room' (1969) a performer selects a room, the musical qualities of which they wish to evoke, and records the reading of a text, which is then repeatedly re-recorded and replayed in the room. What begins as intelligible text becomes a sequence of tones, which are room modes stimulated by speech formants. In Lucier's 'Music on a Long Thin Wire' (1977) a length of steel wire is stretched between two tables at opposite ends of a large room, and is stimulated into vibration by an electromagnet controlled by an electronic oscillator. The vibration of the wire is received at both ends by contact microphones, and is amplified into the room. Even though the frequency and amplitude of the oscillator is constant the resulting sound has much variation. Many artists have employed very long wires to generate rich and varied sounds (McLennan A.). Some play them directly, some use an electro-magnetic or electro-acoustic vibration source, while others employ the random movements of wind or water to cause vibration. Wires may be are constructed by the artist, or found in, for example, abandoned telegraph lines or a suspension bridge.

THE EXPERIMENT

A Reverberation Room as a Resonator

As it is not unusual for a composer or performer to have but a distant influence on the sound of a work of resonating sound art, it was speculated that the aesthetic value of such works is strongly related to the psychoacoustic properties of resonant sounds in general. Thus it may be possible to infer the dimensions of aesthetic response to works of art in the field using stimuli that have no artistic claims, but which are produced using a resonant process. This hypothesis is not evaluated in this study, as it only can be so by reconciling the present data with those obtained using stimuli with artistic standing. However the present data do reveal some of the dimensions of aesthetic response to a limited selection of resonant sounds.

Stimuli were generated following the 'I am Sitting in a Room' process, but with only white noise as the original sound source in a reverberation room. Eight loudspeaker/microphone positions (henceforth labelled LMP1, LMP2, etc) were selected, and the sequence followed to 31 generations in each case. The original white noise signal was of 60s duration. A small loudspeaker was used (JBL 4206), with a bandpass filter, set to 65Hz-2kHz @ 24dB/oct. A tripod-mounted microphone (B&K 4190) was connected to one DAT recorder, another DAT recorder used for replay. With the exception of the loudspeaker and microphone, the equipment was outside the reverberation room. Once the original noise was played and recorded, it was a matter of repeatedly exchanging tapes between the DAT recorders, optimising the recording level, and recording the next generation.

A question broached by this study is whether, in 'I am Sitting in a Room', there are arrangements of the electroacoustic components that produce results better than others. Lucier himself has indicated that some rooms are superior (Lucier A.). The present data suggest an affirmative answer, although as the sounds used in the experiment are so different to that work, the inference is not strong. On the other hand, the sounds have more similarity with works such as Lamb's or Lucier's long wire pieces, and therefore may infer a similar variation in the quality of results in such works. This question has the potential for controversy due to the abrogation of direct aesthetic control which is in some ways associated with artists in this field.

Properties of the Stimuli

As the generations progressed, what started out as close to white noise evolved towards a single tone. For some LMPs this evolution was swift, for others it was not completed within the 31 generations. The more prominent resonances were progressively emphasised, and there was a progressive bias towards lower frequencies. Audible beats were often present. For the purpose of subjective evaluation, six generations were selected to represent the variation in the process: generations 1, 3, 7, 11, 17 and 27 (henceforth labelled G1, G3, etc). With eight LMPs, this produced forty-eight stimuli. Even though smooth rhythms were an interesting aspect of the stimuli, testing stimuli of a duration sufficient to represent this phenomenon was not feasible in this initial study. Instead, the stimuli were reduced to 5s duration, including a 1s fade-in and 1s fade-out.

Properties of the stimuli were obtained using measurements through the headphones used by the subjects. Loudness was calculated following Stevens' method (Stevens S.S.), showing a 29 phon range. Most of the envelopes were steady, but some had an increasing or decreasing profile. Centroids were calculated based on Stevens' loudness index - it was felt that this came closer to a psychoacoustic centroid than the conventional power centroid. Pitches were calculated by applying Terhardt's pitch analysis program VPITCHu3 (Terhardt E., *et al.*), and were also determined by ear, along with approximate musical chords. The stimuli were categorised according to beat strength and frequency (in G3 especially, simultaneous beats created a 'rumble').

Subjective Evaluation Tests

Two programs were developed, the first being a paired comparison test where the subject would hear a pair of sounds in sequence, deciding which of the two was the more interesting. 'Interest' was considered more appropriate than 'preference', as the latter evaluation was thought to be more context-sensitive and contentious. With 48 stimuli, each 5s in duration, it was not feasible to test every combination: only pairs of different generations from the one LMP, or of different LMPs but the same generation were tested (a total of 288 pairs). The order of presentation and order within each pair was random and varied for each subject. The subject's response was limited to four values: "Sound 1 is definitely more interesting"; "Sound 1 is perhaps more interesting"; "Sound 2 is perhaps more interesting"; and "Sound 2 is definitely more interesting". There were three reasons for using four-alternative-forced-choice (4AFC) instead of 2AFC: it was much less frustrating for the subject; it provided a way of measuring the overall confidence of each subject; and it could be interpreted as both a three-alternative unforced choice or as a 2AFC, and the forced and unforced alternatives compared.

In second test, a stimulus was repeated every 8s while the subject typed a short free description, rated it on 33 bipolar semantic scales (Osgood C.E. *et al*), and perhaps typed final comments before evaluating another stimulus. Each subject evaluated 24 stimuli in this way. Stimuli were distributed between the subjects so that one sound from each of the eight LMPs was evaluated by every subject, and the remaining 40 stimuli were each evaluated by two of each multiple of five subjects tested. The stimuli were presented in random order for each subject. The scales were given six random predetermined orders, and the left-right orientation of each scale was individually randomly predetermined but constant for each subject. The scale itself was continuous but with the traditional seven markings, and the subject used a computer mouse to position their response. The tests were implemented on Psyscope computer software (Cohen J.D. *et al*), run on a Power Macintosh with Senheiser HD414 headphones, and were conducted in academic offices.

The antonyms were adapted from a survey of similar studies (such as Asmus E.P.; Bismark G. von; Björk E.A.; Kerrick J.S. *et al*; Nielzén S., Olsson O.; Solomon L.N.) in which sounds or music were rated. The antonyms might be divided into four loose and overlapping categories: aesthetic evaluation (eg dislike/like); literal descriptors (eg quiet/loud); more general descriptors (eg full/thin); and emotional judgements (eg sad/happy). Listeners familiar with the field of sound art who were involved in music, sound production or sound art were tested. As most of the eleven subjects were known to the author, and the sample was small, the results must be treated cautiously. Even so, some were very highly qualified, several being professionals or senior academics in a relevant field.

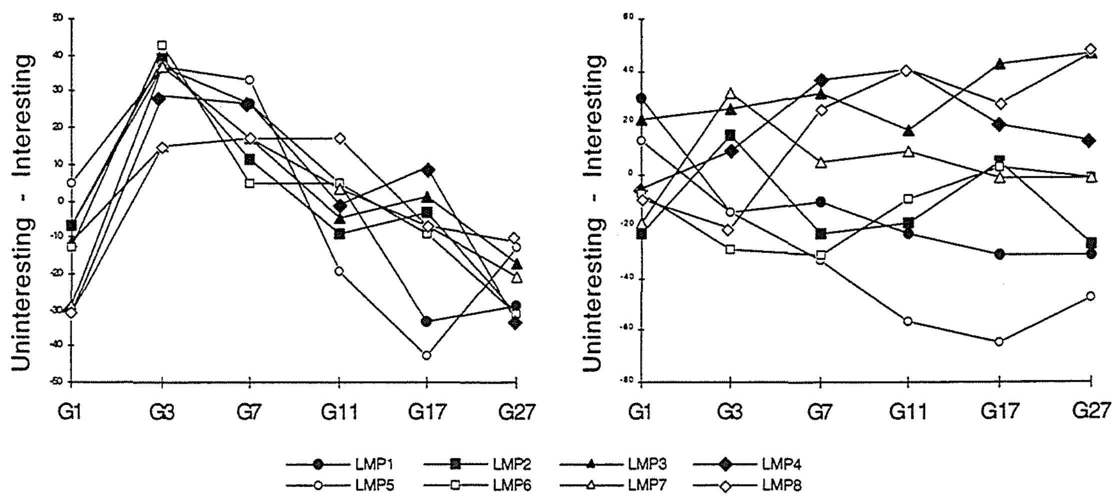
Results

Test 1

Test1 only ranked stimuli that either share an LMP or are of one generation making it of limited use in ranking all stimuli against each other. Nevertheless when the two types of comparison were combined into a single dataset, the results proved to be similar to those achieved in the uninteresting/interesting scale of Test2. The forced and unforced choice results in Test1 ranked the stimuli similarly, the forced choice results tending to be more extreme. Subjects' confidence ranged from 21% to 72%, with older or more experienced subjects tending to be less confident. For most subjects confidence was lower when comparing pairs of the one generation and different LMPs than when comparing different generations of the one LMP.

For pairs of stimuli sharing an LMP, G3 was ranked the most interesting, and G17 or G27 the least. G7 also held a fair amount of interest. Eight out of eleven subjects gave G1 ('coloured' white noise) a low or very low interest rank, but three subjects found it more interesting than G7-27. It is not difficult to understand that G27, where four of the eight stimuli were solitary tones, should have been ranked low on the interest scale. It is also easy to appreciate that the very rich tam-tam-like timbres of G3 should have held a high interest. The results can be represented as charts (Figure 1), with the degeneration of each LMP represented as a line from left to right. In the vertical scale represents the balance of 2AFC assessments for each stimulus, the mean assessment of all stimuli being zero.

Figure 1: Same LMP, Different G (left); and Same G, Different LMP (right)

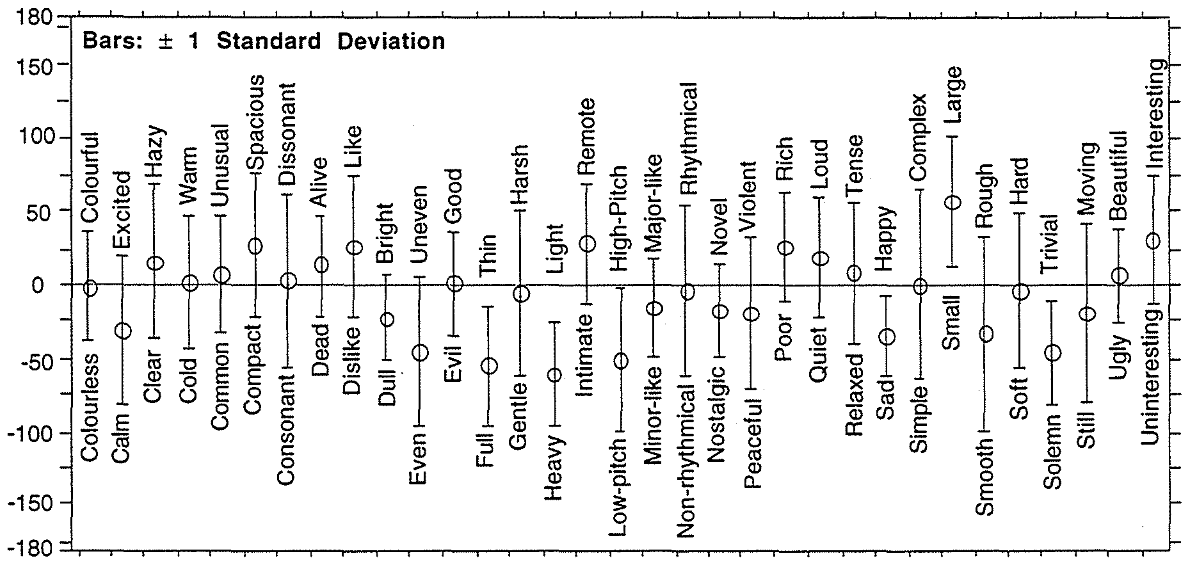


When comparing pairs of stimuli of one generation, it is notable that the four most interesting LMPs involve tone complexes similar to diatonic musical chords, whereas the four least interesting tend to either be dominated by a single tone or by a non-harmonic tone combination. The most highly rated one (LMP3) is close to a deep D major chord with a major ninth - a very consonant chord. Some subjects rated LMP4 higher than LMP3, the bulk of these being classical musicians. LMP4 is close to an A dominant 7th chord, with a major ninth (by coincidence the dominant of LMP3), and the tritone very prominent; these musicians may have found LMP3 bland by comparison. It is highly probable that the beating in LMP8, combined with its harmonic qualities, fostered interest, although beating in itself appears not to be sufficient, as the case of LMP5 indicates. The universal disinterest in LMP5 is probably related to the relatively high frequency (169Hz) of the single tone to which it quickly degenerates (LMP1, LMP2, and LMP7 degenerate to 97Hz, 128Hz and 86Hz respectively). The high pitch of LMP6's fundamental (157Hz) also appears to detract from its interest, and this instance shows that the evaluation is not based on loudness. In the cases of LMP1, LMP3 and LMP5, G1 tended to find relatively high interest, correlating with the lower centroid of these three G1 stimuli.

Test2

While it would be impossible to present the full responses to individual polar scales within the confines of this paper, their overall response over all stimuli can be illustrated. Most notably, the stimuli tended to be spacious, liked, dull, even, full, heavy, remote, low-pitched, rich, sad, large, solemn, and interesting.

Figure 2: Average Ratings of 48 Stimuli by 10 Subjects



The loudness measurements had only a small correlation (r=.318, P=.0273) with the subjects' assessments on the quiet/loud scale, but had stronger correlations with the solemn/trivial scale (r=.594, P<.0001) and with some of the factors, which are outlined below. To the extent that the measurements and results are reliable, the discrepancy between measured loudness and the quiet/loud scale may reflect a spatial element in the subjects' assessments. Envelopes most closely correlated with common/unusual (increasing envelopes associated with 'unusual'). Centroid measurements had strong correlations with a number of scales associated with Factor1 (see below). Dividing the centroid (of power) by the fundamental frequency has been used by others to characterise the brightness aspect of timbre (Krimphoff J. *et al*; Kendall R.A., Carterette E.C.); this was done for both the power and loudness centroids, but there were only small correlations.

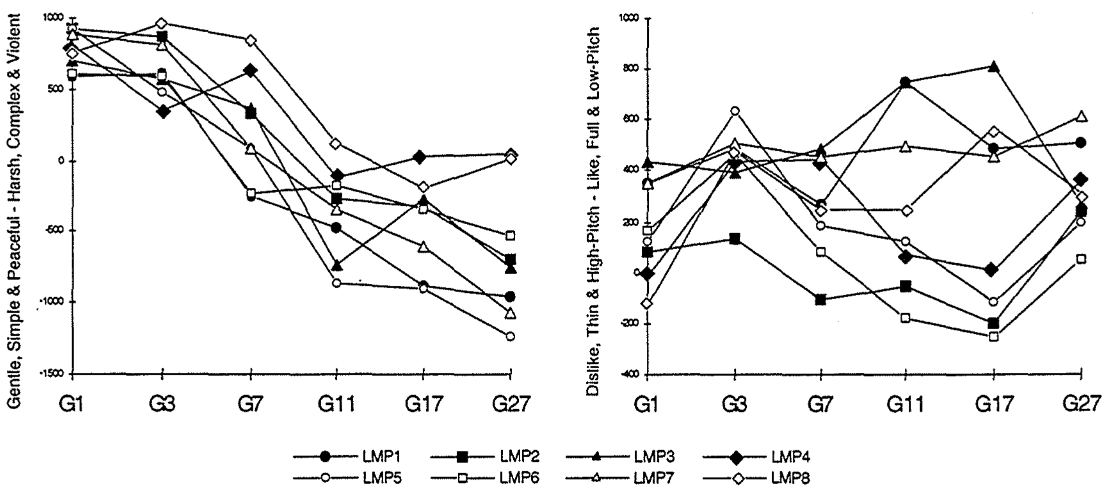
Factor analyses were conducted for individual subject results and for the mean results of ten subjects for the forty-eight stimuli. The factor analysis of ten subjects from Test2 produced six factors accounting for 82% of variance. It must be noted that factor analysis is an imprecise tool, producing results which might be best regarded as hypotheses. The variable loadings cited below are for unrotated factors.

Factor1 (38.9% of variance): gentle/harsh (.922) + simple/complex (.901) + smooth/rough (.900) + peaceful/violent (.899) + soft/hard (.892) + consonant/dissonant (.882) + relaxed/tense (.859) + calm/excited (.844). Versions of this factor were the primary factor for every individual subject. The general trend of Factor1 is predictable: later generations were progressively less harsh, complex, rough, etc. It is notable that LMP8 and LMP4 were consistently high - LMP8 had very strong beats, and from G7 onwards LMP4 had strong harmonic dissonance. LMP1 and LMP5 were consistently low after G3, these being the fastest to degenerate to a single tone. Indeed, at G27, the four single tone stimuli received low rankings, although the extreme harmonic consonance of LMP3 gave it a ranking lower than the tone of LMP2. Uninteresting/interesting made a small positive contribution to Factor1 (.422), but dislike/like made none. Factor1 also correlated with the loudness measurements (r=.678, P<.0001) and centroid (r=.737, P<.0001).

Factor2 (19.2% of variance): dislike/like (.870) + thin/full (.768) + high-pitch/low-pitch (.755) + poor/rich (.746) + ugly/beautiful (.692) + uninteresting/interesting (.659) + trivial/solemn (.637) + small/large (.616). Clearly this is the evaluative factor, dominated by dislike/like, but also involving ugly/beautiful and uninteresting/interesting. Positive evaluations were associated with fullness, low pitch, richness, solemnity and large size. In general the evaluation of the stimuli was positive. In a similar manner to Test1, G3 was generally positively evaluated, and in the subsequent generations the LMPs divide into two groups, which begin to converge again at G27. While it is difficult to measure precisely, LMP5.G3 has markedly less 'rumble' than the other G3 stimuli, and this may have contributed to its high assessment in Factor2. While

Test1 found G3 to be by far the most interesting generation, Factor2 and the response to the dislike/like scale gave higher and similar evaluations to certain later generation stimuli. It is interesting that ugly/beautiful should be associated with evaluation; while that would expected in a normal population, many of the subjects had extensive experience in art forms where beauty is not necessarily regarded positively.

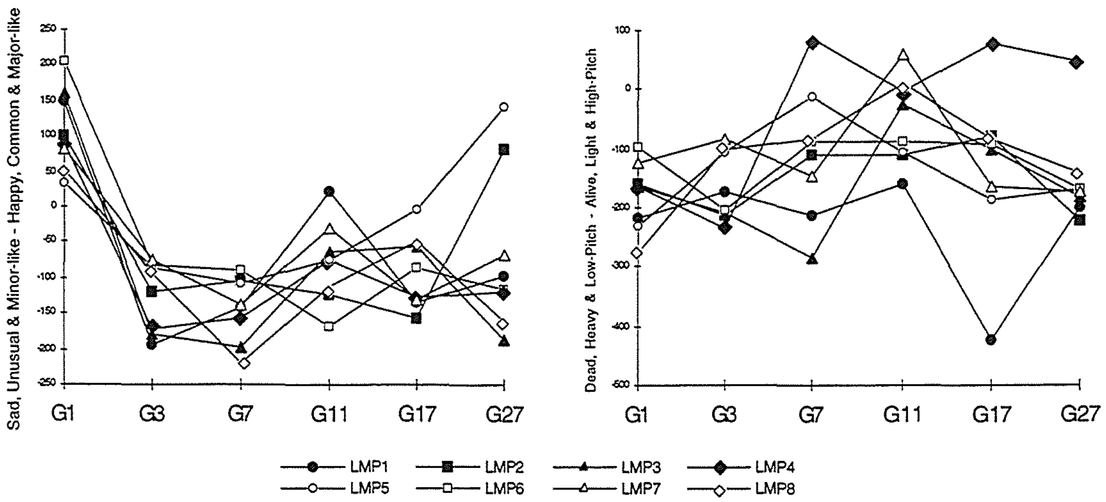
Figure 3: Factor1 (left) and Factor2 (right)



Factor3 (8.3% of variance): sad/happy (.685) + unusual/common (.649) + minor-like/major-like (.634) + quiet/loud (.583). An examination of responses to the minor-like/major-like scale itself shows it to be much more an evaluation of harmonic dissonance than the dissonant/consonant scale; in the former scale, tritones and ambiguous chords were rated as more minor-like than minor-like chords, pure tones and major-like chords. In Factor3, apart from G1, nearly all the stimuli were negative (ie sad + unusual + minor + quiet). Uninteresting/interesting made a small negative contribution to Factor3 (-.317), but dislike/like made no significant contribution. This factor correlated with measured loudness (r=.485, P=.0004) and had some negative correlation with beat strength.

Factor4 (7.4% of variance): dead/alive (.540) + heavy/light (.519) + low-pitch/high-pitch (.459) + large/small (.441) + ugly/beautiful (.425) + dull/bright (.419) + colourless/colourful (.411) + still/moving (.368). This factor contrasts massive stimuli with light and agile ones, most stimuli being massive. The seeming morbidness of the subjects (with their interest in harsh and sad stimuli in Factors 1&3) is countered to a small extent in this factor, which had small but positive contributions from uninteresting/interesting (.317) and dislike/like (.245). The presence of beats in the stimulus appears to be one element in positive assessments in Factor4.

Figure 4: Factor3 (left) and Factor4 (right)



Factor5 (5.3% of variance): non-rhythmical/rhythmical (.747) + bright/dull (.510) + solemn/trivial (.419) + novel/nostalgic (.359) + still/moving (.342). The best account of this factor may be that it reflects a rhythm in the auditory image stimulated by the sound, which in several cases was of a machine (which would begin to explain the otherwise odd connection of dull and rhythmical). This is confirmed to an extent by surveying the free responses given by the subjects: G1 was often thought of as rushing water or a jet engine (achieving some high ratings), G3 was frequently thought of as a gong (rated consistently low), G7-11 had the most machine references (receiving medium ratings), and the four pure tone stimuli of G27 were most commonly considered bland and featureless (receiving low ratings).

Factor6 (3.3% of variance): intimate/remote (.443) + poor/rich (.341) + hazy/clear (.335). This factor is primarily concerned with the apparent spatial properties of the stimulus. As a whole, the free written responses imply a greater and more complex role for spatiality than the results of the factor analysis suggest.

DISCUSSION

These preliminary results reveal an interestingly conservative approach to evaluation, considering the background of many of the subjects. There is interest in diatonic-like sonorities, and a strong correlation between liking a stimulus and its judged beauty. The high evaluation of G3, especially with regard to 'interest', shows that rich timbres also hold great appeal. In the case of late generation tones, there was a marked distaste for those around 150Hz, and a preference for deep tones below 100Hz. If the musical harmonic structure of these short stimuli was of interest to the subjects, influencing their preferences, we might speculate that for longer stimuli (such as long wire pieces), changes in harmony might be heard in a manner analogous to harmonic change in music, with corresponding affective implications. This speculation does not involve an explicitly progressive harmonic structure like that of tonal music, but a more concrete 'here and now' approach. This 'harmony' is more complex due to its intimate involvement with timbre - there is extended graduation rather than a boundary between pitch and timbre. There is also prospect for further study into analogies between musical rhythm and the smooth rhythms that characterise the sounds in question, and into the role of changing spatial imagery associated with complex resonant sounds of long duration.

Given the subjects' preference for full, low-pitch, rich and large sounds in Test1 and Factor2, it would be worth investigating whether such sounds are characteristic of the field of resonating sound art. We might expect artists to produce successful pieces that they and others liked and found interesting, and if these results have any relationship to the art form, they may represent some criteria for positive evaluations. Conversely, there may be artists or artworks subverting aesthetic norms, and which may therefore involve sounds that are thin, high-pitch, poor and small (the notion of size is complicated by its connection to a sense of scale). The present inclination of this author is that the general field is conservative rather than subversive, and that there may be many instances where works are dominated by sounds of the type that were positively evaluated in this study. There are many well known works across the entire range of Factor1, so the fact that it is not related to evaluation encourages the notion that the factors might have some relevance to real art works.

Another related issue is whether the present stimuli - which tended to be rated as calm, heavy, sad and solemn - parallel a lugubrious mood in resonating sound art in general. It would be fair to say that a meditative quality is common in the art form, and in that sense it may well be calm and solemn. The question of sadness is more difficult - emotions of this type are easily related to, for instance, Nineteenth Century art-music, but there is very little discussion of the role of such emotions in the type of pieces in question. They are heard as emanating from an acoustic system more than from a person. Nevertheless, investigations into natural sounds and noises have found emotional associations (Björk E.A.; Kerrick J.S. *et al*; Nielzén S., Olsson O.), so there is all the more reason to associate emotion with examples of resonating sound art. These and other questions are best addressed by using real artworks.

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PREDICTING THE ACOUSTIC QUALITY OF SMALL MUSIC ROOMS

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ABSTRACT

This paper reports on work to establish the critical elements of small music practice and teaching rooms that impact on the acoustic quality of these rooms. Monophonic recordings of musical pieces performed on various musical instruments in an anechoic environment were played back in the room under investigation, at normal listening level, and binaural recordings were made of the reproduced sound in the room. The binaural recordings were assessed and rated for their perceived acoustic quality by lay subjects and music professionals. Room surface diffusivity, room dimensions, reverberation times and background noise were measured and utilised as part of the input layer for a neural network while the subjective ratings of the binaural recordings were used for the output layer of the network. Results of the early experimental work indicate that neural networks can be successfully utilised to predict the acoustic quality of small music practice rooms and music teaching rooms.

INTRODUCTION

Although a large amount of research has been undertaken on the acoustics of concert halls and auditoria for the performance of live (non-electronically amplified) music, there has been very little research carried out on the acoustics of smaller rooms used as music practice rooms and music teaching rooms. Music students spend in the region of 40 hours per week practising. Their feeling about their practice rooms has a significant effect on how many hours they practise and, more importantly, how beneficial that practice is. The careful design of such rooms is therefore critically important to the success or otherwise of a school of music. (Lamberty. 1980)

The purpose of this study is to develop a method of predicting the acoustic quality of small music rooms by utilising a neural network trained with data collected and measured using the binaural recordings made in the music rooms. The amount of data required to create a viable neural network is substantial compared to previous traditional studies of music rooms. Hence the task involved in the making of quality recordings of sufficient resolution and the following listening tests involves a significant amount of planning and scheduling. The study consists of two sections. The first is making of the recordings, preparation of the recording for the listening tests and the conduct of the listening tests of the recordings. The second is the preparation of data from the listening tests and other data collected during the recording for the neural networks and the subsequent training of the neural networks.

TYPES OF ROOMS TO BE INVESTIGATED

In this group of experiments, the rooms being investigated are all basically rectangular in shape with the ceiling being parallel in plane with the floor as the majority of small music rooms fall into this category. In addition to music rooms used in schools, some of the rooms investigated in this work are office rooms of similar shape and construction to the music rooms. The volumes of the rooms investigated ranged from 24 to 427 cubic metres. It is anticipated that future studies will include small music rooms of more complex shapes.

EXPERIMENTAL DESIGN

The use of live musicians performing all the program material in the rooms under investigation was considered but was rejected due to the massive logistics involved. Further, precise repeatability of the performance of the program material by live musicians cannot be guaranteed. It was decided that the program material should be presented to the room under investigation from a monophonic recording replayed via a studio quality monitoring loudspeaker. The program material (recorded music) was then played through the loudspeaker and re-recorded stereophonically using a dummy head and digital audio tape (DAT) recorder. The re-recorded music was later presented through headphones (see section on The Listening Test) for assessment by subjects.

The front face of the loudspeaker was located 1.0 metre from the wall with the ears of the dummy head located 1.6 metres in front of the speaker and 1.1 metres from the floor. These dimensions represents the typical positions of instrument and listener in a music practice room. The dummy head and loudspeaker were located on the longer axis of the room. The loudspeaker level was set at 75dBA $\pm$ 1dB at the dummy head ears using a pink noise source. The recording was made with the room doors and windows closed and, as far as practicable, at a time with the least amount of background noise. All the recordings in the school music rooms were made after hours and during school holidays. In other rooms, the recordings were made after hours.

The duration of each set of recordings is approximately 10 minutes. A minimum of three sets of recordings were made for each room. The recordings with the least background noise of each program material were used for the listening test. Noise resulting from aircraft flyover, road traffic, adjacent playground, gardening machinery, and birds, especially magpies and kookaburras, were the main reasons for the rejection of some recordings.

SELECTION OF PROGRAM MATERIAL

Monophonic recordings commercially available were considered for this purpose but were rejected due to the poor quality of the monophonic recordings (original monophonic recordings were a rarity after the mid-sixties). Unknown levels of compression and equalisation, and the possible use of auxiliary microphones and other unknowns make these commercially available monophonic recordings unsuitable for this purpose. (Hansen & Munch, 1991). Selections of anechoic recordings from "Music for Archimedes" was preferred over self-made anechoic recordings as the recordings are well known to other practitioners in this field and hence the experiments can easily be repeated.

The program material selected from "Music for Archimedes" (Compact Disc No.CD B&O 101) is as follows:-
1) Female Speech in English spoken by Claire Clausen.
2) Male Speech in English spoken by Julian Isherwood.
3) Capriccio Arabe (by F.Tarrega) performed by I.Olsen on Classical Guitar.
4) Etude No.6 in E minor (by H.Villa Lobos) performed by I.Olsen on Classical Guitar.
5) Theme (by Weber) performed by A.S.Christiansen on Cello.
6) Latin American Rhythms performed by B.Lylloff on African Bongo.
7) Sabre Dance (by Khachaturian) performed by B.Lylloff on Xylophone.
8) Over the Rainbow (by A.Z.Mason) performed by O.Andersen on Bb Trumpet.
9) Trumpet Voluntary (by Purcell) performed by O.Andersen on Piccolo Trumpet.

In addition to the above material, test tones and other acoustic instruments (steel stringed guitar, flute and saxophone), not covered by Music for Archimedes, selected from Sound Check (The Professional Audio Test Disc) by Alan Parsons and Stephen Court (Mobile Fidelity Sound Lab Original Master Recording SPCD 15) were also utilised. Musical instruments recorded on this CD are close-miked recordings made in recording studio sound booths.

The length of each musical excerpt was between 15 and 45 seconds.

THE PROS AND CONS OF BINAURAL RECORDING

There are several compelling reasons for using binaural recordings or reproductions for psychoacoustical research. The primary advantage is that the binaural reproductions allow rapid and seamless subjective comparisons to be made, which in real time would be impractical or physically impossible without long pauses in between the comparisons. In view of subjects' limited acoustic memory, it is likely that the accuracy of the assessment of these variables diminishes as the gap in time between their presentation increases (Olive et al, 1994). The reliability and repeatability of data obtained using dummy heads as compared to real human heads has been demonstrated by Nakajima, Yoshida & Ando (1993), Hidaka, Okano & Beranek (1992) and Wightman & Kistler (1989).

PLAYBACK AND RECORDING EQUIPMENT

The selected program materials from "Music for Archimedes" and "Sound Check" compact disc (sampling rate 44.1kHz) were digitally transferred on to a digital audio tape (DAT). The program material was played back on a Tascam DA30Mk2 DAT recorder and utilising the DA30Mk2 internal digital/analogue converter. The analogue output of the DAT recorder was amplified via a Marantz PM64Mk2 amplifier operating in Class A mode below 25W r.m.s. The loudspeaker used for the sound reproduction is a Celestion SL600 studio monitor.

The binaural recordings were made using a Neumann Dummy Head (Model No.KU81) and a Tascam DA-P1 DAT recorder with its own 48volt phantom power source and microphone pre-amplifier. Like the source, the recordings were sampled at 44.1kHz as this will allow future binary comparison between the source and recording if necessary.

The recorded materials were digitally transferred to a hard disc on a P.C. via a Digital Audio Labs (DAL) 'Digital Audio Interface I/O Card'. The recorded materials were then regrouped under instrument category with the Digital Audio Labs Fast Eddie (an audio editing and fast selection program) and the loudness of the recordings were adjusted to a level similar to that of the source material. This program also allows the listener to select any of the recorded material with the P.C. mouse from a music catalogue graphically displayed on the screen. When a listener selects a recorded material with the mouse, the recorded material which is now stored in the P.C.'s hard disc is replayed and the digital S/PDIF (Sony/Philips Digital Interface) output from the DAL

I/O card is fed into the digital input of the Tascam DA30Mk2 DAT recorder. The listener hears the binaurally recorded material via a pair of Sennheiser HD250 closed type studio headphones plugged into the headphone jack of the Tascam DA30Mk2 DAT recorder.

THE LISTENING TEST

For this phase of the study, the following program materials were assessed by the listeners.
1) Capriccio Arabe (by F.Tarrega) performed by I.Olsen on Classical Guitar.
2) Theme (by Weber) performed by A.S.Christiansen on Cello.
3) Over the Rainbow (by A.Z.Mason) performed by O.Andersen on Bb Trumpet.
4) Saxophone excerpt from Sound Check CD by Alan Parsons & Stephen Court.

Two groups of listeners were used for the listening test. Each group consists of a full-time professional acoustic instrument musician, an advanced high school music student and a non-musician music listening enthusiast. The grouping of the listeners were based on the interest and familiarity of the listener with the instruments. One group listened to program materials 1 & 4 and the other to program materials 2 & 3. The listening tests were conducted individually without the other group members being present or discussions held about the results of their tests. The maximum time a listener was allowed to spend in any one listening test session was 2 hours with a 10 minute break during that period. In that period, a listener was able to choose and rank the 36 recordings of each of the 2 sets of program materials.

In the listening test, the listener was required to select the better of two recorded musical pieces presented to them using the two-alternative-forced-choice (TAFC) method as used by Hansen & Munch in the Archimedes Project (Hansen & Munch, 1991). The listeners can listen to the recorded material as many times as they wish by selecting the recorded material with a mouse. By the process of round-robin and elimination comparisons, all the recorded material can be ranked in order of preference. The resolution of the rankings is dependent on the amount of time that is available for the listening tests. As 36 samples were presented to the listener in each series of test, the highest ranked recording was given 36 point, the second 35 points, the third 34 points etc. The ranking of the 3 listeners in each group is averaged and percentage rated with 100 being a perfect score if a recording is ranked first by all three listeners in the group. Thus, as a result of a managed TAFC, the recordings were also simultaneously subjectively ranked by the listener. Although this method is more time consuming, the ranking of the recordings will be more accurate and reliable than the conventional subjective listening tests. The listening tests were supervised by the authors.

RESULTS OF THE LISTENING TESTS

A summary of the results of the listening tests and the measured signal-to-noise ratio of the recordings used for the listening tests is shown in Table 1. These results together with other physical aspects of the rooms and reverberation times were used as inputs and outputs in a neural network analysis.

The results indicate that a room that may be ideal for one instrument may be just acceptable for other instruments. The plucked guitar and the bowed cello represents the two extremes. It is anticipated that the results of a strummed guitar would be closer to that of the bowed cello.

ARTIFICIAL NEURAL NETWORKS

Artificial neural networks (commonly abbreviated as neural networks) are computer-based simulations of living nervous systems, which work quite differently from conventional computing. Neural networks have more in common with parallel processing than with sequential processing which is used in conventional processing. (Nelson & Illingworth, 1991). A neural network does not process instructions the way a conventional program does. No program is written to tell a neural network what steps to take to solve the problem. The neural network learns in a way that is specified by the training method, in most cases back-propagation, and the way the neurons are connected . (Lawrence, 1994). Most commercially available network programs take care of the training method and defining and connecting the neurons and hence they are relatively easy to use.

The relationship between room physical characteristics and the resulting sound quality produced from the room has been difficult to quantify. Neural network is a tool that has recently become available which has the potential of achieving this.

NETWORK TRAINING AND RESULTS

The neural network program used in this work is Brainmaker (Lawrence & Petterson, 1993), a back-propagation based program developed by California Scientific Software.

TABLE 1
Summary of Listening Test Results

ROOM	SAX S/N	SAX AQI	CGP S/N	CGP AQI	TR1 S/N	TR1 AQI	CEL S/N	CEL AQI	AVE S/N	AVE AQI
RM1	34	41	27	52	38	35	40	44	35	43
RM2	33	40	33	58	33	25	39	42	35	41
RM3	36	54	37	28	33	60	38	71	36	53
RM4	41	48	43	68	40	78	50	61	44	64
RM5	36	60	42	36	40	43	50	51	42	48
RM6	39	49	45	81	48	74	54	58	47	66
RM7	37	81	44	74	44	80	40	91	41	82
RM8	38	64	39	41	35	61	40	41	38	52
RM9	36	49	38	31	41	88	49	94	41	66
RM10	36	59	42	43	37	60	47	41	41	51
RM11	36	57	43	38	45	58	51	41	44	49
RM12	37	67	42	79	47	42	49	47	44	59
RM13	35	49	31	56	31	67	39	50	34	56
RM14	30	24	29	47	24	22	32	15	29	27
RM15	32	15	31	57	32	40	37	60	33	43
RM16	31	47	28	58	26	35	36	51	30	48
RM17	32	37	29	70	27	14	35	34	31	39
RM18	33	81	33	44	38	81	38	71	36	69
RM19	33	80	28	89	36	68	40	83	34	80
RM20	37	85	48	93	48	81	52	94	46	88
RM21	38	54	39	47	40	56	51	60	42	54
RM22	37	70	41	63	39	67	48	65	41	66
RM23	39	83	45	32	41	60	51	44	44	55
RM24	36	70	41	97	46	81	44	69	42	79
RM25	38	55	44	52	43	66	51	50	44	56
RM26	34	15	29	17	35	14	40	15	35	15
RM27	33	15	32	17	36	41	44	23	36	24
RM28	36	37	31	17	34	22	39	15	35	23
RM29	36	28	29	32	34	41	35	43	34	36
RM30	39	60	23	43	26	14	37	44	31	40
RM31	34	43	25	43	27	24	30	43	29	38
RM32	30	28	26	61	29	24	31	60	29	43
RM33	35	50	35	38	31	57	37	53	35	50
RM34	26	42	26	43	24	25	32	35	27	36
RM35	30	52	33	61	29	40	28	35	30	47
RM36	38	64	44	53	40	50	51	60	43	57

SAX S/N is Saxophone Signal to Noise Ratio
CGP S/N is Classical Guitar Signal to Noise Ratio
TR1 S/N is Trumpet Signal to Noise Ratio
CEL S/N is Trumpet Signal to Noise Ratio
AVE S/N is the Average S/N Ratio for all Instruments

SAX AQI is Acoustic Quality Index
CGP AQI is Classical Guitar Acoustic Quality Index
TR1 AQI is Trumpet Acoustic Quality Index
CEL AQI is Trumpet Acoustic Quality Index
AVE AQI is the Average AQI for all Instruments

The inputs used for the neural network training are as follows:-

a.Room Minimum Dimension in metres

b.Room Volume in cubic metres

c.Room Reverberation time in seconds.

d.A subjective measure of room diffusivity of 1 to 5 with 1 being very low diffusivity and 5 being very high diffusivity.

e.Recorded Signal to Background Noise ratio in dB.

The recorded signal to background noise ratio were used in favour of the measured room background noise as it better reflects what was being heard by the listener via the headphones and automatically compensates for the necessary gain changes made to achieve the same level as the source material.

The output pattern used for the neural network training will be a sound quality rating, 100 being a perfect score and 3 being the lowest possible score for group of 36 samples.

The network size was limited to 5 inputs and 1 output as more data will be required if the number of inputs or outputs was increased. A larger number of inputs and outputs or reduced training data can result in a poorly trained network and hence an unreliable one. Other combinations of inputs that can be utilised in future networks may include the following:-

a. Room Low Frequency Absorption Coefficient.

b. Room Mid Frequency Absorption Coefficient.

c. Room Low Frequency Absorption Coefficient.

d. Room Length in metres.

e. Room Height in metres.

f. Type of wall, floor and ceiling finishes.

g. Room Shape.

h. Instrument Type

I. Type of Music

j. Room temperature and humidity.

A neural network was set up for each of the four musical instruments and one for the averaged data of the four instruments using the data described above. All the neural network were successfully trained and tested. By using these trained networks, a music practice room can be optimised for a specific musical instrument by utilising the neural network generated for that instrument or a general purpose music practice room.

CONCLUSION AND FUTURE STUDIES

The repeatability and reliabilty of binaural recordings for listening tests and the use of a modified two-alternative-forced-choice method of ranking recordings in listening tests have provided us with a reliable method of remotely comparing and assessing the acoustic quality of music rooms. By collecting sufficient data in this manner, trained neural networks can be developed to indicate the most important parameters (and their interactions) for determining the acceptability of the acoustical conditions in small rooms. Early results indicate that the trained neural networks have the potential of predicting acoustic qualities of small music rooms. As more data is collected and the selection of inputs refined, the performance of the neural networks is expected to significantly improve.

An extensive programme of work is planned for further experimental work and the gathering of more data to create new neural networks with different combinations of inputs to search for the neural network that best accurately represent the acoustic quality of small music rooms. The effect of different positions of the dummy head and loudspeaker will also be investigated. The networks developed will be used to predict how good some rooms are and how to optimise the design of small rooms for music. Following the completion of this series of analyses, the results will be checked using musician opinions of playing in some of the rooms the qualities of which have been predicted.

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ON COFFINS AND SHOE-BOXES

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INTRODUCTION:

It is well known that classical musicians, by and large, if asked about a preferred shape for a performance space will nominate the so-called "shoe box" (SB) a reference to a rectangular parallelepiped approximating a double cube. That indeed was the first choice for the Performance Space in the Canterbury University Music School. This opinion is certainly based in the experience of both gratifying and difficult rooms for performance but also one suspects on a conservative position that one is safer with what one knows, and to some extent on conventional wisdom. In the event, and for non-acoustical reasons, the shape of the performance space became a "near shoe box" with appropriate height and width but actually a "coffin shaped" (CS) plan - an elongated hexagon, symmetrical about its long axis and asymmetrical about its short axis, and with a sloped or stepped floor. This form, according to acoustical theory should be able to perform correctly since - as I pointed out at the meeting, at which the preliminary design was revealed - if the height and width are correct a small degree of taper should not have as much effect on the sound as the details of the wall treatment.

Nevertheless, musicians' opinion is expert and I have undertaken the following study to see if there is a physical reason for the Shoe-box preference in which one might have confidence.

METHODS:

My first approach was to use a new computer programme - "Odeon" - which traces acoustical rays around enclosures and calculates the properties of the space from this process. Two rooms , one CS and one SB, of identical volume and the same seating area based on the design, with the same height, were entered. A single source position at stage centre and 1.5m back from the stage front and six receiver positions in one half of the seating plane were selected, and the results for Reverberation Time, Early Decay Time[1], Lateral Fraction[2],

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[1] Reverberation Time (RT) and Early Decay Time (EDT)are the two usual measures for the subjective sense of "Liveness" or "Reverberance"in a room.

[2] Lateral Fraction (Lf) is one of several measures which correlates reasonably well with the desired sense of envelopment in the sound in a room. It promotes an enhanced sense of involvement in the musical activity. Recently (1995) Bradley and Soulodre have shown very high correlations between the strength of the late lateral sound and the sense of envelopment. That is, late lateral reflections are exceptionally important.

Objective Clarity, and Centre Time[3] were calculated. The first intention was to compare the two room shapes. A range of desired numerical values for such rooms is known.

In addition the question of an optimum floor slope had been raised. It may be recalled that there was adverse comment on the Ilott recital hall which has a steeply stepped floor. The design called for a relatively steeply sloped floor (like the Ilott) to land at first floor level. Three floor slopes were introduced for each room: Horizontal, to half the storey height and as designed to the first floor level.

The results of this process were inconclusive and in some cases were counter-intuitive - trending against one's experience.

Because of my relative inexperience of ODEON I decided to repeat the study with the use of the physical models and the Acoustics Research Centre system MIDAS which has been proven over a period of 12 years. A high voltage spark causes ultrasonic waves in 1:50 'foam core' card models of the two spaces and with identical dimensions to those used in the Odeon study. This sound is picked up by microphone and the measures are taken as they are in the full-sized room. The only value not obtainable in MIDAS at this scale is the Lateral fraction (Lf).

Finally some geometrical analysis of the two shapes was undertaken, locating the 'image' locations in the walls to see how these are placed in successively later reflections for each room shape. To a first approximation such an analysis is useful. I start with this analysis.

GEOMETRICAL ANALYSIS

For the rectangle, it is obvious that the successive images lie on the line through the source perpendicular to the wall and that every point in the seating plane receives sound from every image, clearly lateral and from approximately the same direction. In fact the later the reflection the more lateral it becomes relative to the listeners' heads. See Fig. (1)

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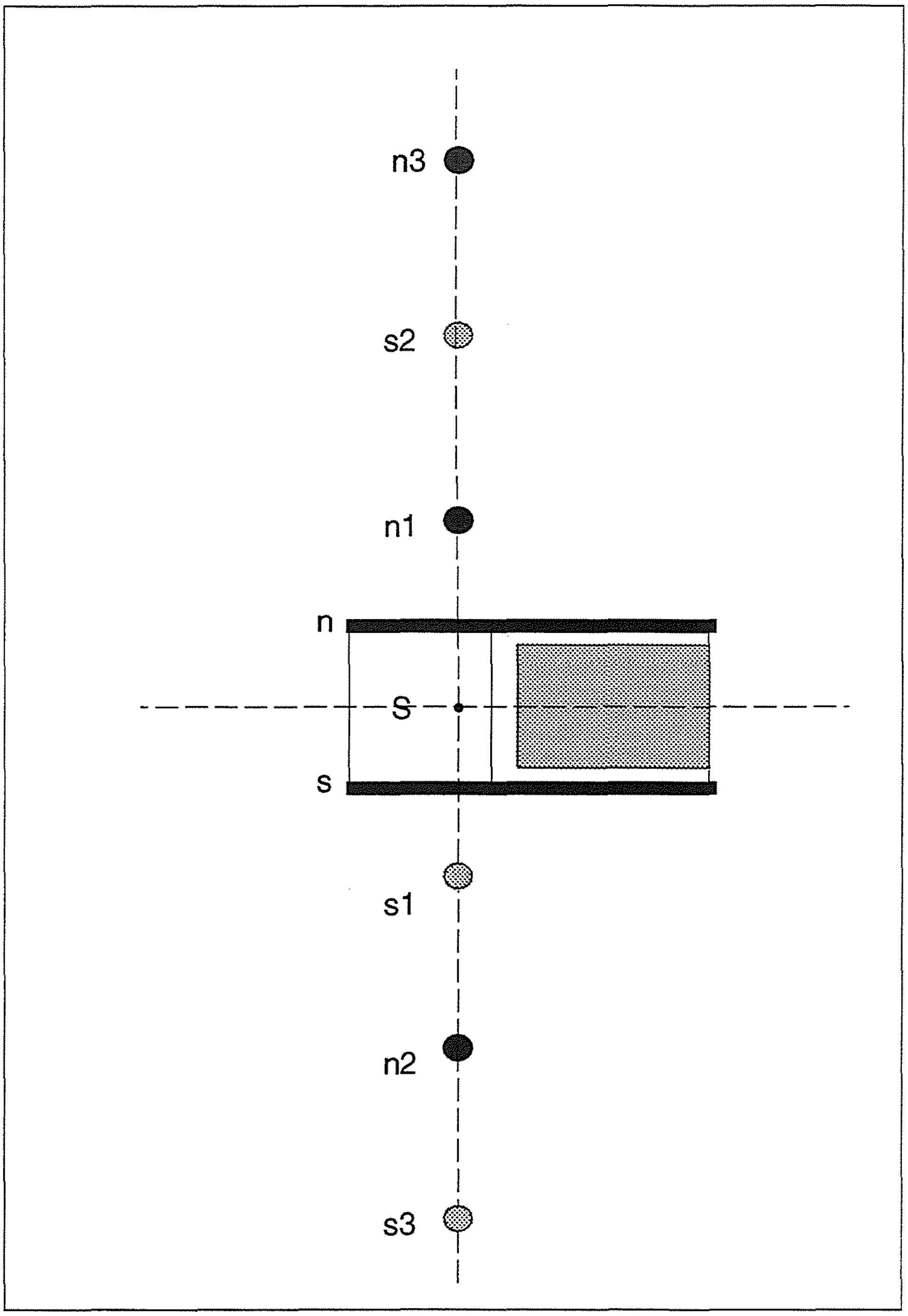
3 Objective clarity (C80) compares the early sound (earlier than 0.08 seconds after the arrival of the direct sound) with the later arriving sound. It depends both on the strength of the reverberant sound as measured by the two measures in Footnote (1), and on the presence of early reflections. Centre Time is the time delay at which there is as much sound earlier, as there is later. Both of these correlate well with the subjective impression of clarity.



How are the images distributed if the side walls are splayed? We start with the image in a single wall as seen in Figure 2. Then we add an oblique wall (ie not parallel to the first) at some arbitrary angle and project the image of the first image in it. It turns out that these images lie on a circle with its centre at the intersection of the two

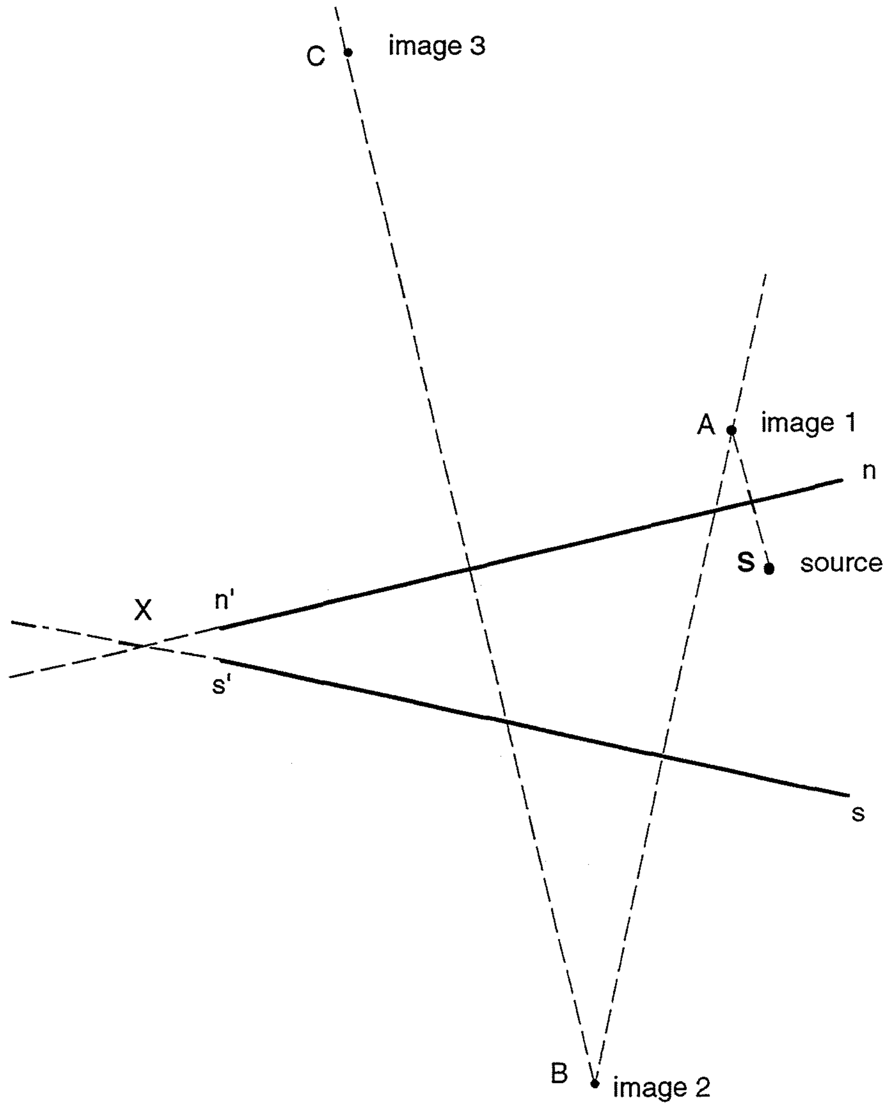


Figure 2
With a source **S** and a wall nn', the first image is at A. Add a second wall ss'not parallel to nn' and let their extensions intersect at X
The image of A in ss' is at B. By symmetry about nn', XA = XS and about ss' XA = XB = XS = XC. Hence the images lie on a circle, centre X and radius XS.

Now the successive images become <u>less lateral</u> - they move progressively toward the front of the audience. It follows that there is less **late** lateral energy in this case and as noted in Footnote 2 that means a lower sense of envelopment for the audience than is provided by the rectangular SB room. Figure 3 plots the images for the comparable stage position to Figure 1 for two walls about the inclination proposed in the Canterbury design.

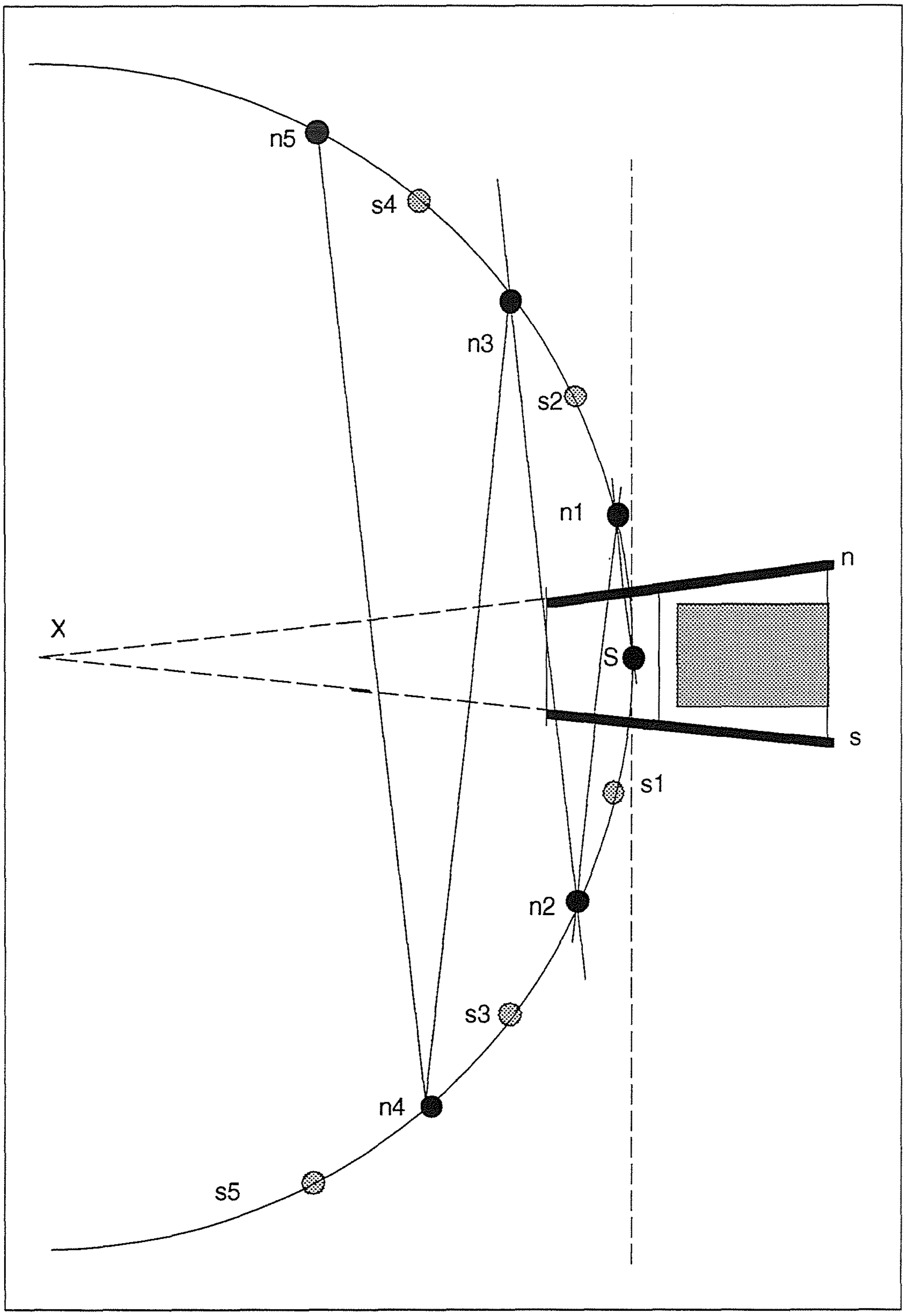


Figure 3: The images lie on a circle - approximate inclination of the Design walls.

We see then that there is a difference in the distribution of the images but that for a small inclination such as that proposed and for a relatively small room the difference in the direction of the images from a true rectangle is not great. This was confirmed in the Odeon calculation of the lateral fraction, Lf which had almost the same value for every situation calculated and was satisfactorily high in every case.[4] The other criterion Late Lateral Strength is too new to be included in the existing programmes but as already noted is likely to be lower in the CS room.

MIDAS STUDY:

Please see Figure 4.

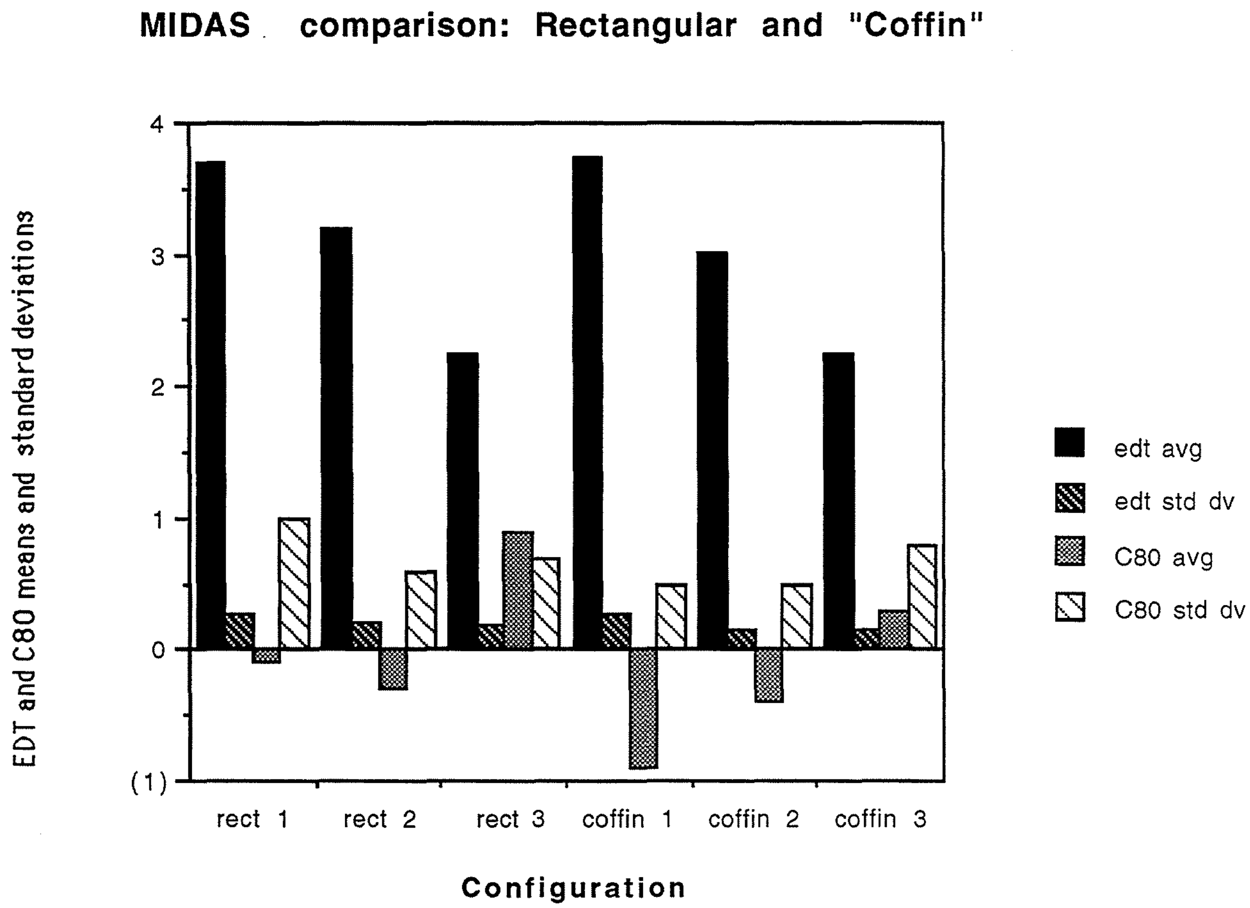


Figure 4: Midas Comparison - Rectangular & Coffin Room Shapes

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[4] Lf should be greater than about 0.15 for a satisfactory sense of envelopment. In all five situations calculated and averaged over 6 seat positions, in ODEON, the value varied only from 0.24 to 0.26.

The two rooms were compared. Three floor slopes were used in each space. The expected trend was clear -that increasing the floor slope decreases the reverberance (EDT) and increases clarity (C80).

This graph should be read comparatively, rather than as absolute values since the foam absorption used for the "audience" was not accurately matched to known audience absorption. It was however the same in each model. Comparison of the results for the intermediate slope shows that the two rooms are almost the same in reverberance and clarity. This is confirmed in the measure of "Centre Time" on Figure 5.

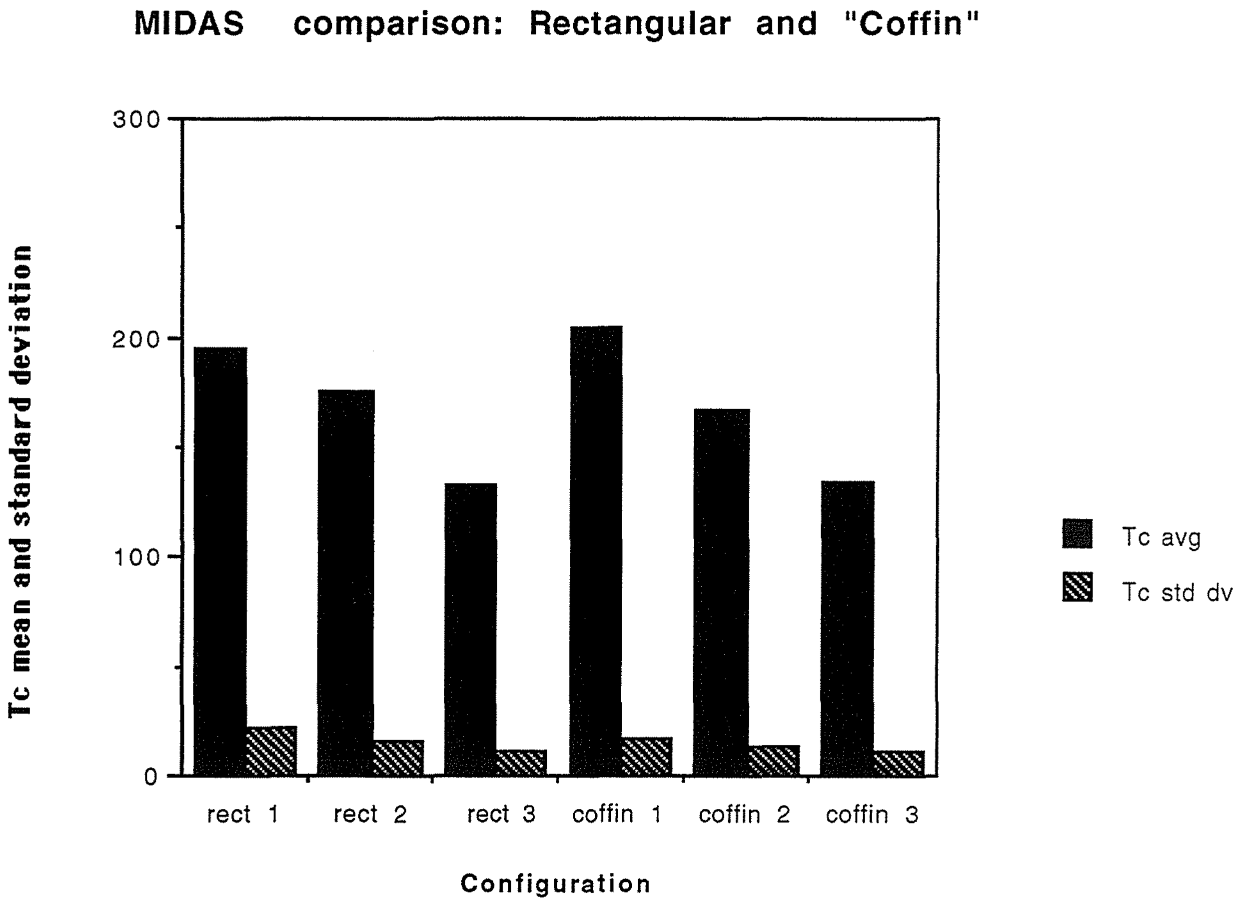

Figure 5: MIDAS Comparison - Rectangular & Coffin Shaped Rooms

CONCLUSIONS:

- There is no measurable acoustical index that indicates that the shoe-box form is preferable to the coffin-shape given a small degree of taper and with an intermediate floor slope.

- The steep floor slope is clearly disadvantageous in terms of reverberance in each case.

- The intermediate floor slope is the best of those tested in terms of Reverberance and Clarity.

- The choice between the CS and SB forms for a small angle of inclination must be made on other than acoustical grounds - for example the expressed preference of the users.

THE EFFECT OF SIGNALISATION ON ROAD TRAFFIC NOISE LEVELS - A CASE STUDY

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ABSTRACT

Signalised intersections produce interrupted flow that may be characterised by periods of relatively low noise levels followed by periods of higher noise levels as vehicles accelerate away from the traffic lights. As a result, it is to be expected that the introduction of traffic signals into an existing intersection would produce a general widening of the gap between the maximum noise levels and the average noise levels. This paper presents the trends evident in the results of two sets of detailed noise level measurements conducted at one such intersection - one set before and one after the introduction of traffic signals. It was determined that the L_{A10} and the L_{Aeq} road traffic noise levels decreased due to a change in the surface finish of the road pavement. At the same time, however, there was a general increase in L_{A01} noise levels. This paper proposes a possible explanation for this observation and questions whether, in fact, particular actions aimed at reducing community annoyance response may not actually achieve the reverse effect.

INTRODUCTION

In 1995/96, NSW RTA constructed various improvements to the Pacific Highway at the intersection with Terranora Road at Banora Point, northern NSW. The upgrading included traffic signals, road widening and a change to the road surface of the pavement.

Prior to the improvements being carried out, RTA commissioned an assessment of the likely impact of the road works on the nearby residential premises against the requirements of the "RTA Interim Traffic Noise Policy"(1). The assessment included a series of noise level measurements of existing road traffic noise levels, as well as predictions of the new road traffic noise levels after completion of the road works. Following the completion of the works, RTA again commissioned a series of noise level measurements to be conducted to verify the predictions. In each instance, the objective noise level limits were set in terms of $L_{Aeq,24hour}$ (2) and $L_{Aeq,8hour}$ (3).

The large volume of data that was collected during those measurements has been examined further to assess the likely effect of the change to the road surface and the trends in the values of various noise level parameters.

ORIGINAL AND NEW SITUATIONS

The intersection of the Pacific Highway with Terranora Road occurs at Banora Point, 102km north of Ballina and immediately to the south of Tweed Heads. There are some 20 residential premises located in close proximity to the intersection. All but two of these are located on the western side of the Pacific Highway.

Prior to the completion of the improvements, the Pacific Highway consisted of two lanes each of northbound and southbound traffic. Terranora Road was one lane each way. At that time traffic joining the southbound lanes of the Pacific Highway was required to give way to traffic on the Pacific Highway. Traffic turning left to join the northbound lanes was required to give way when merging.

The improvement to the intersection involved channelling of the intersection with a "seagull" central median arrangement controlled by traffic signals. The traffic signals control the northbound through highway lanes and the two southbound right turn lanes into Terranora Road. The southbound through highway lanes are not controlled by the traffic signals.

Prior to the improvement works, the road surface was a spray seal. The surface was changed to *Novachip* as part of the road works. In addition, the new situation resulted in a net widening of the Pacific Highway. The extent of the road widening was relatively minor, varying from less than 1m to 5m.

DATA GATHERING AND DATA REDUCTION

Overview

The level of road traffic noise existing prior to the improvements was measured at each of several representative residential locations close to the intersection. To quantify the expected effect of signalisation alone, a series of noise level measurements was conducted at the same time at a comparable site under both uninterrupted flow and interrupted flow conditions.

Upon completion of the improvement works, the road traffic noise levels were re-measured at the same representative locations. The comparative measurements method and general calculation methods of CRTN '88(4) have been used to quantify the likely effect of various changes operating conditions.

Measurement Locations at the Subject Site

The initial set of noise level measurements was conducted on-site over the 24-hour period 12:50pm (ESST) Thursday November 17, 1994 to 12:50pm (ESST) Friday November 18, 1994. These noise level measurements were conducted at each of seven residential premises (six on the western side of the Pacific Highway) and one free field location deemed to be representative of all of the residences fronting the Pacific Highway and located close to the intersection. Overall, the residences were located between 20m and 85m from the Pacific Highway.

The second set of measurements was conducted over the 24-hour period 11:40am (ESST) Wednesday March 27, 1996 to 11:40am (ESST) Thursday March 28, 1996. The seven residential assessment locations were retained for the second set of noise level measurements, but the free field location was discarded. Note: To preserve accuracy in the subsequent data analysis and data reduction below, the results obtained at one of the residential locations was also discarded owing to the influence of dog barking on the $L_{A01,T}$ results.

Comparable Signalised Intersection

Because one of the original objectives of the work commissioned by RTA was the prediction of the new road traffic noise levels after completion of the road works, it was necessary to quantify the likely individual effect of the signalisation on future $L_{Aeq,24\ hour}$ and $L_{Aeq,8hour}$ road traffic noise levels.

Historically, the likely effect of signalisation can be quantified by either measurement or prediction. Owing to the complexity of the topography, geometry and operation of the Terranora Road intersection, it was considered inappropriate to use predictive methods. Instead, a series of noise level measurements was conducted at another quite comparable intersection with the Pacific Highway. This site was situated some 13km north of Banora Point at Stewart Road, Currumbin, Queensland. Measurement locations situated 30-100m from the Pacific Highway were adopted at this site.

Even though this site is in another state, there are important similarities between the two sites that allow the results obtained at the Stewart Road site to be translated to the Terranora Road site. In the first instance, road traffic volumes are comparable. In addition, the road plans are alike (long curve followed by straight section) and, importantly, both roads are on a modest gradient.

Measurement Method - Terranora Road Intersection

For each test programme at the subject site, two types of measurements were conducted in general accordance with the comparative measurements method of CRTN '88:-

- Continuous logging of sound pressure levels at one reference location

  and

  Sample measurements of sound pressure levels at seven satellite locations

In each instance, measurements of the $L_{Aeq,T}$ value were conducted together with various short-term statistical noise level parameters, including $L_{A90,T}$, $L_{A50,T}$, $L_{A10,T}$ and $L_{A01,T}$. Using an extension of the comparative measurements method of CRTN '88, resultant values for various longer term noise level parameters were derived for each measurement location for each set of measurements.

Measurement Method - Signalised Intersection

The effect of signalisation on road traffic noise levels was assessed at the Stewart Road intersection by comparing the results of noise level measurements conducted under two conditions. Condition 1 was normal flow conditions (ie. alternating interrupted flow and free flow through the intersection). Condition 2 was free flow conditions only, ie. interrupted flow conditions were excluded from the measurements. In each case, the noise level measurements were conducted at each of the set of locations selected close to the intersection. Again, even though $L_{Aeq,T}$ was the prime parameter of interest, measurements of the $L_{A90,T}$, $L_{A50,T}$, $L_{A10,T}$ and $L_{A01,T}$ noise levels were also conducted. The differences between the results obtained for each condition were used to quantify the significance of the signalisation on each of the short-term noise level parameters.

Effect of Change to Road Surface

In addition to the signalisation, there are three other significant changes associated with the improvement works which will have an influence on the level of road traffic noise emitted to the nearby residences. These were identified to be the road widening, the increase in road traffic volumes over the intervening period and the change of road surface. The effect of each of these changes, except the change in road surface, were be determined by independent means. The effect of the road widening and increase in traffic volume on the $L_{10(18hour)}$ noise levels were each quantified by reference to the CRTN '88 algorithms. The effect of signalisation on $L_{10(18hour)}$ noise levels was quantified by reference to the results of the measurements at the Stewart Road intersection.

By applying the combined effects of these changes to the difference between the road traffic noise levels that were measured before and after the improvements, the effect of the change to road surface was derived. It should be noted, of course, that being fully rigorous, the accuracy of the resultant value should be viewed in the context of the accuracy of the determinations of the effect of each of the other changes to the road system. Consideration should also be given to the influence of changes in traffic flow conditions on the magnitude of the noise levels that were recorded and the subsequent differences that were derived.

RESULTS

Effect of Change in Road Surface

The change to the road surface from spray seal to *Novachip* was found to yield a 1.3dBA reduction in $L_{10(18hour)}$ road traffic noise levels on average at each of the residences located to the west of the Pacific Highway.

Effect of Signalisation at Stewart Road Intersection

The effect of signalisation at the Stewart Road intersection is summarised in Table 1 below.

TABLE 1

Effect of Signalisation at the Stewart Road Intersection
Measured as the Change in Noise Level Parameter Values
(Note: Negative values indicate lower noise levels under signalised conditions)

L_{A90}	L_{A50}	L_{A10}	L_{A01}	L_{Aeq}
-1.2	-1.0	0.2	0.1	-0.1

Relative Changes in Noise Level Parameter Values at the New Signalised Intersection

The relative differences in the changes to various noise level parameters at the Terranora Road intersection are presented in Tables 2 and 3. Table 2 presents a summary of the change in noise level parameter values at the reference location before and after the improvement works at the intersection. These values are derived directly from the output of the continuous logging of sound pressure levels at this location. Table 3 presents a summary of the change in noise level parameter values averaged across all measurement locations before and after the improvement works at the intersection.

TABLE 2

Change in Noise Level Parameter Values at Reference Location
After -v- Before Improvement Works at Terranora Road Intersection
(Note: Negative values indicate lower noise levels after road improvement works)

Noise Level Parameter Averaging Period	L_{A90}	L_{A50}	L_{A10}	L_{A01}	L_{Aeq}
18 hour average (06:00 - 24:00)	-3.7	-2.8	-0.7	-0.6	-1.5
24 hour average	-1.5	-1.6	-0.7	-0.4	-1.3
8 hour average (22:00 - 06:00)	2.7	-0.5	-1.1	-0.2	-0.4

TABLE 3

Change in Noise Level Parameter Values Averaged Across all Locations
After -v- Before Improvement Works at Terranora Road Intersection
(Note: Negative values indicate lower noise levels after road improvement works)

Noise Level Parameter Averaging Period	L_{A90}	L_{A50}	L_{A10}	L_{A01}	L_{Aeq}
18 hour average (06:00 - 24:00)	-0.8	-1.3	-0.5	0.5	-1.3
24 hour average	1.4	-0.2	-0.5	0.7	-1.2
8 hour average (22:00 - 06:00)	5.6	0.9	-0.9	0.9	-0.3

DISCUSSION

The 1.3dBA reduction in $L_{10(18hour)}$ road traffic noise levels due to the change to the *Novachip* surface is consistent with the extent of reduction found by others, notably Samuels (5). As noted above, because this result was derived by subtraction of the effects of each of several other influences, the accuracy of this result should be viewed in this context.

From Table 1, it can be seen that at the Stewart Road intersection, there is essentially no difference in the L_{A10}, L_{A01} and L_{Aeq} noise levels under signalisation compared to free flow conditions. From Table 2 and 3, it can be seen that there is a general reduction in L_{A10} and L_{Aeq} road traffic noise levels at the residences after completion of the improvement works. This is considered to be due primarily to the effect of the change in road surface.

Interestingly, however, the reduction is generally greater for the L_{Aeq} noise level parameters than it is for the either the L_{A10} or L_{A01} parameters. In fact, the values of all of the L_{A01} parameters when averaged across all locations increased slightly after the improvement works while the L_{Aeq} noise levels showed a consistent reduction.

It is recognised that the differences are relatively small values. Furthermore, the accuracy of these results should also be assessed in the context of the size of the measurement sample. Further, more extensive measurements at this and other intersections would be necessary to hone the accuracy of the absolute values. Nonetheless, the relative change is the issue. The results show that there is a general tendency for the gap between the maximum noise levels and the average noise levels to widen as result of the signalisation and change to the road surface.

If we examine the likely mechanisms for noise generation that are at work, we can see that there may be a reasonable explanation for the widening of the gap between the maximum noise levels and the average noise levels at this site. The L_{Aeq} noise level values result from the noise generated by all sources of noise in the traffic stream - notably, the noise from road/tyre interaction as well as engine noise. A change to the road surface would be expected to change the level of noise generated by the road/tyre interface, but have very little influence on the level of engine noise. All other factors constant, a change to a "quieter" road surface would be expected to yield lower L_{Aeq} values.

On the other hand, maximum noise levels measured as the L_{A01} values would be governed principally by the noise level peaks produced by engine noise, especially that generated by trucks. These peaks would remain largely unaffected by changes to the road surface. With the introduction of traffic signals on the northbound lanes of the Pacific Highway, it is to be expected that the peaks due to engine noise would occur more frequently and may be emitted at higher noise levels. This is because vehicles would now accelerate away from the traffic lights en mass where previously they would have passed through the intersection at a relatively constant speed, and in a stream rather than discrete groups. As a result, it is fully to be expected that there would be an increase in the value of the L_{A01} parameter.

Of course, signalisation by itself may also tend to change the L_{Aeq} noise level. The fact that a signalised intersection would be characterised by a period of relatively low noise levels while one traffic stream is stationary (ie. lower L_{Aeq} noise levels), followed by a period of higher noise levels as traffic flow re-starts (ie. higher L_{Aeq} noise levels), would tend to suggest that the overall effect on the L_{Aeq} noise level may be only fairly minor, especially when contrasted to the change in the L_{A01} values.

It is also to be expected that the differences in the noise levels will be a function of the traffic flow conditions at the particular situation at the time of the measurement. The Stewart Road measurements showed a negligible widening, the effect was more pronounced at Terranora Road.

These results raise the question as to whether the effect of signalisation on community annoyance can be adequately described by changes in L_{Aeq} values alone. What role does the widening of the gap between the maximum noise levels and the average noise levels play? What is the importance of the level of noise generated by the noisy events and by the number of noisy events? Are people attuned to the average noise level only or do they respond adversely to an increase in the number and intensity of high noise level events? Is it not possible that, even if the average noise level decreases, the increase in L_{A01} may offset the benefit of this reduction?

Perhaps the answers may be found in research by others. Brown (6) suggested that the number of noise events generated by a traffic stream may be a unifying measure for predicting annoyance under all traffic situations: free-flowing, congested, interrupted and low traffic volume. In support of this contention, Brown drew on work by others, notable the seminal work of Langdon (7). Simply put, a resident indoors would be aware of the passage of a heavy vehicle by the noise that it makes. It follows then that the vehicle's passage is identified by the excess of the vehicle's peak noise level above the noise of other sources, including that of the rest of the road stream. If the gap between the maximum noise levels and the average noise levels increases, might not the excess of the vehicle's peak noise level above the noise of other sources also increase. This would be perceived as an increase in both the number and intensity of high noise level occurrences which, in turn, would generate a greater number of interruptions - defined in the broadest sense - that the resident would experience. This would be expected to directly influence the degree of annoyance experienced by the resident.

Needless to say, more research would need to be conducted into this issue before any changes are contemplated to the objective criteria currently applied by the various State authorities. The objective of this work would be to develop a more complete understanding of the relationship between annoyance in these circumstances and changes to the gap between the maximum noise levels and the average noise levels.

It is curious to speculate, however, that by an interesting twist of circumstances, an action aimed at reducing annoyance by lowering L_{Aeq} noise levels - such as the application of a "quiet" road surface - when taken in conjunction with an action that has the potential to increase L_{A01} noise levels may, in fact, increase rather than reduce annoyance.

Acknowledgement: The assistance of RTA in permitting the use of data to develop Tables 1-3 is gratefully acknowledged.

NOTES

1. The "RTA Interim Traffic Noise Policy" applied at the time of the initial set of noise level measurements. This policy sets a daytime (24 hour) limit of $L_{Aeq,24hour} \leq 60dBA$ and a night time (22:00 - 06:00) limit of $L_{Aeq,8hour} \leq 55dBA$. These noise level limits are quire comparable with those currently used by other Australian State authorities.

2. $L_{Aeq,24hour}$ is the equivalent continuous A-weighted sound pressure level during any 24 hour period. It is the value of the sound pressure level that, within the 24 hour time interval, has the same mean square pressure as the sound under consideration whose level varies with time.

3. $L_{Aeq,8hour}$ is the equivalent continuous A-weighted sound pressure level over the 8 hour period 22:00 to 06:00.

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THE RELATIONSHIP BETWEEN TRAFFIC NOISE AND HOUSE PRICES

Warren D Renew

ABSTRACT

The paper begins with a discussion on the hedonic method of determining the relationship between traffic noise levels and house prices. Previous attempts in Australia and overseas to calculate this relationship are reviewed. The paper discusses recent research into this topic carried out in Brisbane which yielded values of the Noise Depreciation Index generally similar to those obtained overseas.

INTRODUCTION

Externalities in the transport market have been defined (Environment Protection Authority (Victoria), 1994) as "social and environmental impacts whose costs are not faced by individuals when making their transport decisions." Traffic noise is regarded as an externality because, although road users generate transport noise which lowers property values, the community in general pays for the noise effects. Other externalities of interest in transportation are vehicle emissions, accidents and congestion. The costing of externalities is useful for indicating the scale of the several impacts from transport, for assessing policies for controlling these impacts, and for evaluating projects (Environment Protection Authority (Victoria), 1994).

Cost-benefit analysis is one of a number of techniques which can be used to evaluate the environmental effects of proposed developments. Costs are determined for both environmental and construction factors, as well as for the environmental benefits the proposal is expected to produce. The usefulness of the technique depends upon the accuracy with which environmental factors can be evaluated. The term "benefit" has been defined (Alexandre A and Barde J P) as the damage avoided as a result of the measures incorporated in the proposal to control environmental factors. With regard to noise from a proposed road, the benefits may be lower levels of factors such as sleep disturbance, annoyance and interference with speech communication.

People exposed to intrusion from environmental factors such as noise, air pollution and vibration are assumed to be willing to undertake actions aimed at reducing the level of intrusion suffered. With regard to noise, possible actions include moving to a more suitable (ie quieter) locality and modifying the house by installing air conditioning or insulation

Cost-benefit theory assumes that a person will be willing to pay for improving the situation so long as the value of the benefit to be gained is lower than the cost incurred in providing it. Equilibrium is reached when the marginal benefit is equal to the marginal cost of providing it. The hedonic pricing technique discussed in this paper is one of several methods used in cost-benefit analysis.

It is widely accepted (Streeting M C) that road traffic has a negative effect on house prices. Other things being equal, a prospective buyer will tend to choose a house which is not located on a busy road because of considerations such as noise, air pollution, vibration and safety. Cost-benefit analysis can be used for assessing the advantages of a proposal, but a difficulty in its application lies in costing environmental effects. There are two ways of evaluating the cost of excessive noise (Stempler S):

- determine the cost of protection against noise
- determine the impact of noise on real-estate

The hedonic pricing approach has been the most widely used technique for evaluating noise effects.

THE HEDONIC PRICING APPROACH TO RESIDENTIAL HOUSING

Explanation of the Method

Several studies conducted overseas (Streeting M C) have employed the hedonic pricing technique to determine relationships between house prices and environmental emissions such as noise (from traffic and aircraft) and air pollution. This approach assumes that the value of a house to a buyer depends upon a number of attributes, each of which is awarded a price or value. The total value (V) of the house is the sum of the separate attribute values and can be expressed as:

$$\text{value } V = a_1C_1 + a_2C_2 + \ldots\ldots + a_nC_n$$

where the coefficient a_i (ie dV/dC_i) is the hedonic price or value of attribute C_i, and may be regarded as a measure of the marginal willingness to pay for that attribute.

The hedonic price function is not necessarily linear, although the linear form of the equation has been widely used to determine noise relationships because of its simplicity. The equation may also take a non-linear form, where attributes in, say, exponential or logarithmic form can be used. A non-linear form of the hedonic price function has been said to be the most favoured one for housing-market research (McLeod P B). The statement has been made (Alexandre A and Barde J P) that the unit percentage of depreciation (ie NDI, see below) increases with both the noise level and the value of the house. Another author came to a similar conclusion but thought that there was insufficient information to justify it (Weatherall F). Values of NDI subsequently estimated for two Victorian roads were shown to rise to 0.80 at 72 dB(A) from 0.50 in a 51-55 dB(A) interval, and from 0.44 in a 51- 58 dB(A) interval respectively (Environment Protection Authority (Victoria), 1994).

The calculation method consists in determining the values of the coefficients a_i by the use of multiple linear regression. The Noise Depreciation Index (NDI), defined in this paper as the percentage reduction in house value per decibel increase in traffic noise level, is then calculated by dividing the coefficient for noise effects by the average selling price of a house in the area surveyed.

Selection of Attributes

Attributes are characteristics of the house and its environs and may be related to the house itself, the location and the environment. When people are considering a house for purchase, they are assumed to take these attributes and their related prices into account, and are accordingly prepared to pay a higher price for a house with superior features. For instance, they would be willing to pay more for a house in a quiet suburb than an equivalent house in a noisier suburb. The choice of attributes is important, in that the accuracy of the determination will be affected by the number and nature of the chosen attributes. For road traffic it is most likely that, if only one environmental effect (eg noise) is considered, it will act as a surrogate for other related effects such as air pollution, vibration and intrusiveness.

Examples of attributes which have been used in analysis are shown below:

House related attributes are characteristics of the house and allotment, and include: number of rooms, garages and bedrooms; building material; roof material; house style (high-set or low-set); building age; allotment size; air conditioning; swimming pool.
Location related attributes are associated with: distances to schools, place of work, shopping centres, parks, the CBD; zoning of area.
Environment related variables commonly used include: traffic noise level; aircraft noise level; air pollution concentration (suspended particulates, oxidants, sulphation).

The selection of attributes is comparatively straightforward since most of them will be obvious choices. A problem may arise if one (or more) of the attributes is highly correlated with another. This situation can happen in the case of environment related attributes where the effect of noise, say, is difficult to separate from that of air pollution and vibration. The condition is known as multicollinearity and can be dealt with by the use of principal-component analysis (Nelson J P).

A REVIEW OF STUDIES

Values of NDI for traffic noise determined in numerous countries have been reported in the literature. A summary of results from Canada and the USA indicates a range from 0.08 to 0.88, with a mean of 0.4, on a basis of L_{eq} (Alexandre A and Barde J P). Another selection of results shows NDI values ranging from 0.4 to 1.26 (Environment Protection Authority (Victoria), 1994). Several investigations have been conducted in Australia into the relationship between traffic noise levels and house prices. The studies by Abelson and by Bradley and Holsman did not report values of NDI, but Modra (Environment Protection Authority (Victoria), 1984) estimated them by making certain assumptions.

Selected information from Australian and two North American studies is given in Table 1. The Australian NDI values are generally similar to those quoted above. However, the higher values from the Bradley and Holsman study, which are associated with selected main roads, are considerably greater. The lower NDI values were obtained for streets parallel to the main roads. There are also considerable differences between corresponding values of NDI for L_{eq} and L_{10}, a result contrary to that obtained by the present author (Renew W D).

The regression analysis indicates the statistical significance of the calculation for each attribute in the hedonic price function. The studies by Nelson, Taylor et al. and the author have shown that traffic noise is a significant variable at the 95% confidence level. The value of the variance (r^2) indicates the extent to which the selected attributes describe the factors affecting the price of a house. If a comparatively low value of variance is derived, the implication is that further attributes should be chosen for the regression analysis. Values of variance reported in studies are shown in Table 2.

Bradley and Holsman concluded that noise is statistically significant in determining house prices, but that its impact is relatively minor in importance. The other studies examined to date indicate that this conclusion is valid. It is interesting to note the conclusion from one study that the major determinants of house prices were house quality and size, land size, inflation and environmental factors (Abelson P W).

The Brisbane Study

The current study by the author (Renew W D) is part of a wider investigation into the effects of traffic on the community. House sales in 36 selected streets in Brisbane were examined for a three year period and a survey carried out to obtain relevant information about the ten chosen attributes. A twenty-four hour noise survey was conducted at a representative site in each street and noise descriptors calculated, resulting in L_{dn} values ranging from 58 to 74 dB(A). Linear regression was used to determine the coefficients of the attributes and the NDI values were calculated from inflation-adjusted selling prices. The next stage of the study is to gather data for an expanded number of attributes and then repeat the regression analysis.

CONCLUSIONS

There is general consistency in survey results from a number of countries for a variety of noise descriptors. This leads to the conclusion that the hedonic pricing technique is a reliable tool for evaluating the Noise Depreciation Index for existing and proposed roads. The question of whether the Noise Depreciation Index increases with traffic noise level is one which needs further study.

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TABLE 1

The Estimated Effect of Traffic Noise on House Prices

Year	Author	Houses	Attributes	NDI	Descriptor
1975	Nelson (USA)	456	13	0.8	L_{dn}
1982	Taylor, Breston and Hall (Canada)	2277	9	0.5	$L_{eq\,(4\,h)}$
1977	Abelson (Sydney)	592	30	0.5 *	L_{10}
1983	Bradley and Holsman (Sydney)	368	4	0.69; 1.8 * 0.53; 2.3 *	$L_{eq(5\,h)}$ $L_{10(5\,h)}$
1996	Renew (Brisbane)	350	10	1.0 1.0 1.1	$L_{eq\,(24\,h)}$ L_{dn} $L_{10\,(18\,h)}$

* Calculated by Modra (Environment Protection Authority (Victoria), 1984)

TABLE 2

Variance Values from Studies

Author	Variance (r^2)
Nelson	0.88
Taylor, Breston and Hall	0.65 (Arterials) 0.24 (Expressways)
Abelson	0.62 - 0.68
McLeod	0.78
Bradley and Holsman	0.17 (Main roads) 0.38 (Parallel streets)
Renew	0.32

THE ATTENUATION OF ROAD TRAFFIC NOISE ENTERING BUILDINGS USING QUARTER-WAVE RESONATORS INCLUDING THE MECHANISM OF ATTENUATION

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ABSTRACT

This paper continues research into the use of quarter-wave resonators for the attenuation of noise entering buildings. Previous work (Field 1995) demonstrated the effectiveness of a single cavity resonator, achieving over 6dB attenuation in the third octave band to which it was tuned. Further results with a multiple cavity system and a proposed attenuation mechanism were then presented, achieving 8dB and 6dB in the third octaves to which the cavities were tuned (Field 1996). Research has now been extended with further development of the mechanism for resonator attenuation. Further model results are also presented using a scaled DAT recording of road traffic noise as the noise source. This technique gives an indication of the results expected for a working prototype.

INTRODUCTION

Previous work into the use of quarter-wave resonators to attenuate noise entering buildings (Field 1995, Field 1996) has found that in addition to attenuation at the resonator frequencies, diffraction effects at high frequencies from the casing and inner partitions of the resonators is present. An attempt to minimise diffraction effects and to lower the Q factor of the resonators has been made by increasing the cross-sectional area of the resonator cavities and making the walls of the resonator cavities thinner, maintaining the same overall space taken up by the resonator system. The continuation of this work includes understanding the mechanism by which resonators attenuate noise. Experiments are also carried out with a traffic noise recording as the source of noise with the frequency content scaled to the scale of the model room.

REVISED MECHANISM OF ATTENUATION USING QUARTER-WAVE RESONATORS

A possible mechanism of resonator action has been proposed previously (Field 1996). This proposed mechanism implied an "active-passive" interference effect at the open mouth of the resonator due to the phase difference of 180° between waves incident on the resonator and those which had reflected off the rigid end of the resonator. The result would be a reduction in sound pressure level in the region around the mouth of the resonator. The phase difference of 180° between the incident and reflected waves was confirmed by experiment (Field 1996). It is now believed, however, that the mechanism of attenuation is more related to the impedance of the resonator at resonance.

When both incident and reflected waves are present in a quarter-wave resonator the varying phase relationship between them causes the acoustic impedance to vary along the length of the resonator cavity. The acoustic impedance is defined as:

$$\mathbf{Z} = \frac{\mathbf{p}}{\mathbf{U}} = \frac{\rho_o c}{S} \tag{1}$$

where $\mathbf{p}$ is acoustic pressure, $\mathbf{U}$ is volume velocity, $\rho_o c$ is the characteristic impedance of air and S is the cross-sectional area of the resonator cavity. When both the incident and reflected waves are present this expression becomes:

$$\mathbf{Z} = \frac{\mathbf{p}_i + \mathbf{p}_r}{\mathbf{U}_i + \mathbf{U}_r} = \frac{\rho_o c}{S} . \frac{\mathbf{p}_i + \mathbf{p}_r}{\mathbf{p}_i - \mathbf{p}_r} \tag{2}$$

where $\mathbf{p}_i = \mathbf{A}e^{j(\omega - kx)}$ is the incident pressure wave and $\mathbf{p}_r = \mathbf{A}e^{j(\omega + kx)}$ is the reflected pressure wave.

At resonance the reactive component of the impedance $\mathbf{Z}$, X_o, becomes zero corresponding to the minimum input impedance of the resonator. The condition of minimum impedance at the resonator opening implies that the impedance in the region around the resonator opening is very low relative to the rest of the surrounding medium. With X_o=0 at resonance, the impedance at the open end of the resonator reduces to the sum of the internal resistance and radiation resistance of the open end. Both the absorption and scattering characteristics can be optimised when the internal resistance and radiation resistance are matched. Hence for the resonator to scatter incident sound well, it must also be a strong absorber. The optimum scattering and absorption condition of the resonator at resonance can then be used as a mechanism for attenuation.

By inserting a quarter-wave resonator (or several) in the path between a source of undesirable noise and a receiver such devices can be useful for noise reduction. The scattering and absorption effects of a quarter-wave resonator can be illustrated by particle velocity vectors in the field of a plane wave normally incident on a Helmholtz resonator at resonance (Fahy 1989).

EXPERIMENTS WITH ROAD TRAFFIC NOISE AS SOURCE

Source of Noise Used

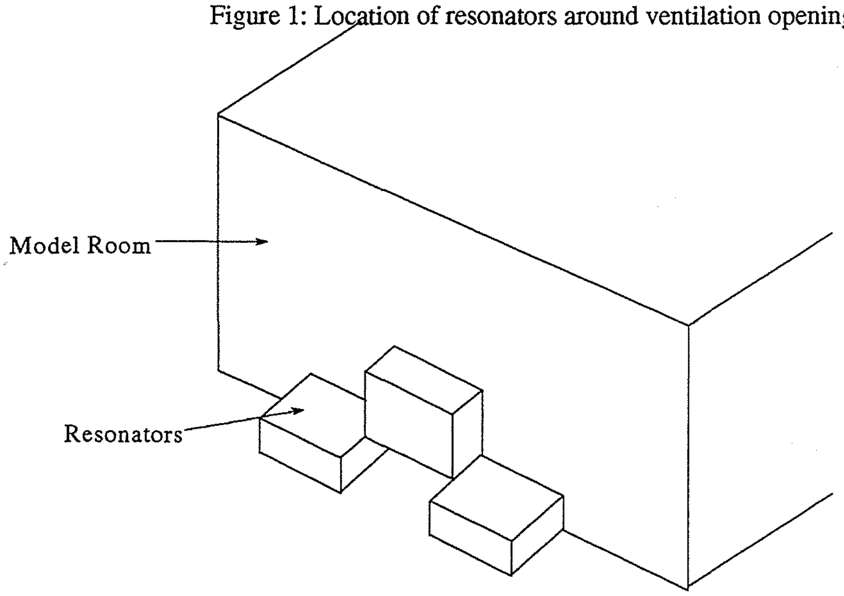
Experiments were carried out on a 6:1 scale model room(645mm×405mm×530mm) with a ventilation opening(60mm×35mm) located at the bottom centre of the facade exposed to noise (Field 1996). The source of noise was a Digital Audio Tape (DAT) recording of road traffic noise recorded on City Road in Sydney approximately 30m from the nearside lane.

Since the scale of the model room was 6:1, the frequency content and traffic flow rates had to be scaled up six times to conform with the principle of similitude. This process was performed by re-recording the DAT recording onto the hard disk of a Macintosh computer running SoundDesignerII software. SoundDesigner II could be used to shift the pitch of the traffic noise recording on hard disk by six times the pitch in the original recording. The pitch shifted traffic noise was then re-recorded from hard disk to DAT to be used as the noise source for the scale model experiments.

Resonators Used

With the ventilation opening located at the bottom centre of the model room, one resonator could be placed on either side of the opening and a third resonator could be placed on top or the other two. Figure 1 below illustrates the locations of the resonators.

Figure 1: Location of resonators around ventilation opening



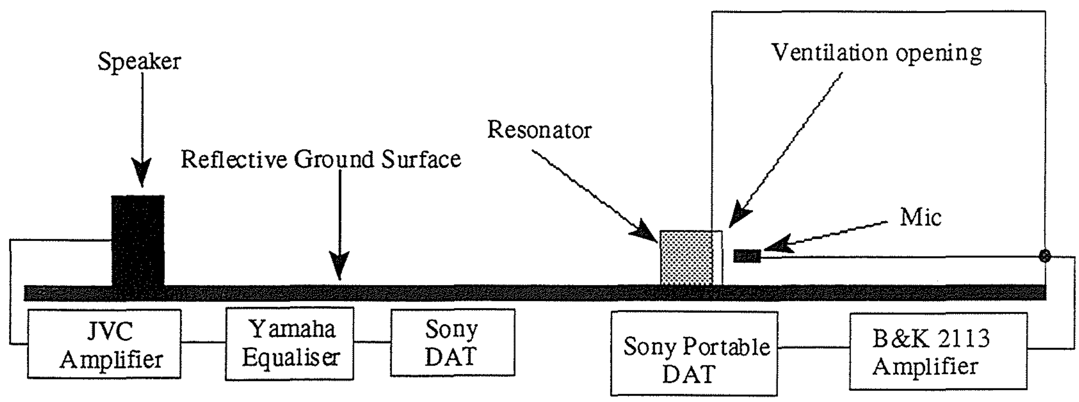
Since the effect of the resonators is local to the open end of the resonator cavities, it is important to have resonators positioned around the entire perimeter of the ventilation opening. Each of the two resonators on the sides of the ventilation opening consisted of two cavities tuned to 1.25kHz and 1.4kHz. The resonator along the top of the opening consisted of two 1.6kHz cavities. With a model scale of 6:1 these frequencies would become 200Hz, 230Hz and 270Hz respectively at full scale. Hence all the tuned cavities were concentrated within two 1/3 octave bands.

The cross-sectional areas of the resonators were increased and the thickness of the walls of the resonators was decreased compared to previous work (Field 1995, Field 1996) whilst maintaining the same overall space taken up by the resonator system. This was done as an attempt to lower the Q factor (extend the useful frequency range of the resonators) and to minimise diffraction from the resonator walls.

Experimental Set-up

The set-up for the experiments is shown in Figure 2. A B&K Type 4133 microphone was mounted at the ventilation opening in the room which was connected to a portable Sony Type TCD-D10 PRO DAT Recorder via a B&K 2113 microphone amplifier. A reflective ground surface was present between the source and receiver microphone. No attempt to model the ground impedance was made. The scaled recording of road traffic noise was then played through a four inch speaker via a Yamaha Type Q2031A Graphic Equaliser (to correct for the uneven frequency response of the speaker) from a Sony Type PCM-2700A DAT Recorder without any resonators located outside the ventilation opening. The noise detected at the microphone was recorded by the portable Sony DAT recorder. A multiple resonator system was then placed at the ventilation opening. The scaled recording of road traffic noise was played again, detected by the microphone and recorded by the portable Sony DAT recorder. The two recordings (without and with resonators at the ventilation opening) were then recorded onto hard disk using SoundDesignerII. The pitch of the two recordings was then shifted back to full scale.

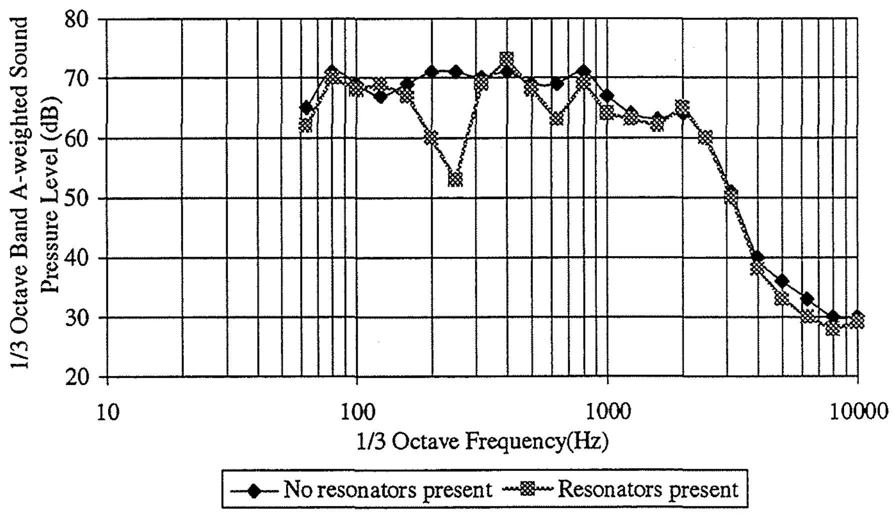
Figure 2: Experimental Set-up



Results

DAT recordings of traffic noise measured at the window of the model room without and with the resonators present were analysed using a B&K Type 2131 Spectrum Analyser. Sample results for a source height of 200mm and source-receiver distance of 1670mm (source directly in line with the ventilation opening) are presented in Figure 3.

Figure 3: Traffic noise spectra measured at the ventilation opening (without and with resonators in position) vs 1/3 octave frequency (sample results)



The highest level of attenuation of traffic noise with the resonators in position occurred within the two 1/3 octave bands to which all the cavities were tuned. In the 200Hz 1/3 octave band 11dB attenuation was achieved and in the 250Hz 1/3 octave band 18dB was achieved. Significant attenuation also occurred in the 630Hz 1/3 octave band (6dB) which is attributed to the second harmonics of the 200Hz and 230Hz resonators. The near identical nature of the two spectra at high frequencies (2kHz and above) indicates the elimination of diffraction effects from the resonator casing and inner partitions.

CONCLUSIONS

The mechanism by which quarter-wave resonators attenuate noise entering buildings has been revised. Experiments were carried out with re-designed resonators and a DAT recording of road traffic noise as the noise source. The frequency content of the recording was increased to the same scale as the model room on which experiments were performed. Sample results indicated 11dB and 18dB attenuation respectively after insertion at the ventilation opening in the 1/3 octave bands to which the resonators were tuned. The use of scaled DAT recordings of road traffic noise is a useful technique for giving an indication of the attenuation expected on a full scale prototype.

Future work will involve extending the frequency range of attenuation by a system of resonators and the construction and testing of a prototype to be installed at a ventilation opening in an existing building exposed to road traffic noise.

ACKNOWLEDGMENT

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SOURCES OF NOISE IN SHEET METAL SHEARS

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ABSTRACT:

The aim of this paper is to classify and investigate the noise sources in a high speed sheet metal shear machine. Three different design parameters, blade angle, blades clearance and speed of cutting, were considered. It is shown that the blade angle in shear cutting has the most significance for the reduction of the radiated noise. The effect of blade clearance on noise depends on the material. Under the conditions tested, steel sheets were not very sensitive to clearance, but aluminium sheets demonstrated a linear increase with clearance. Decreasing the speed of cutting also reduces the radiated noise. The effect of damping the sheet metal is also discussed.

INTRODUCTION:

The radiated noise from shearing, like impact noise, can be classified into two major components acceleration noise and ringing noise. Acceleration noise occurs immediately before cutting process, that is during the transferring of the force from the structure to the feedstock. It is called acceleration noise because the structure and the product have a sudden acceleration or deceleration during that time. Ringing noise occurs after the cutting process and is caused by local vibration of the structure and the feedstock. Ringing noise occurs when vibrational energy is transmitted through the structure and the feedstock after cutting.

THEORETICAL ANALYSIS:

Previous research on impact forming [1], has identified four sources of noise.

(1) Air ejection noise:
When two surfaces approach each other air is ejected from the space between them. When the gap is wide, the air flow is incompressible, but just prior to contact, the flow becomes compressible and sound is generated.

(2) Billet expansion noise:
This noise is a result of the impulsive lateral expansion of the product as it is compressed during the impact period.

(3) Acceleration noise:
The very large contact forces developed during impact (engaging of the tool with the workpiece) causes a rapid acceleration or deceleration of the impacting bodies. This produces a simple noise pulse from their exposed surfaces. The radiated noise level depends on the magnitude and duration of the impact.

(4) Ringing noise:
After impact or fracture, a longer term vibratory motion of the machine components or the product occurs as its vibrational energy is dissipated into ringing noise or heat. Ringing noise is the results of propagation of the vibrational energy through the structure and the feedstock.

It has been shown that the contribution of air ejection [1] and billet expansion [2] noise are relatively small compared acceleration or ringing noise to the overall energy radiated noise of an impact machine. Because of the similarity between an impact and shearing, it is expected that this dominance of acceleration and ringing noise will also occur in shearing.

Previous research on impact noise[3] has shown that acceleration noise contributes significantly to the peak noise, while ringing noise, which arises from structural vibrations, is the largest contributor to the total L_{eq}(A). For a typical impact noise, the L_{eq}(A) due to ringing noise is typically 20 - 30 dB higher than that due to acceleration noise. This strongly suggests that ringing noise is by far the most significant contributor to daily noise dose when impact machinery is used in industry.

Richards [4] using the energy accountancy concept, has developed the following expression for the ringing noise radiated from an impact machine or its feedstock.

$$L_{eq}(A,f,\Delta f)=$$

$$10\log\left|\dot{F}(f)\right|^2+10\log I_m[H(f)]+10\log\left[\frac{A\,\sigma_{rad}}{f}\right]-10\log\eta_s-10\log d+10\log\left[\frac{\rho_o c}{2\pi^2\rho_m}\frac{\Delta f}{f}\right] \qquad (1)$$

where:

$L_{eq}(A,f,\Delta f)$ is the equivalent A-weighted radiated noise for a single impact per second in a frequency band of width Δf with central frequency f;

$H(f)$ is the structural response at the point of impact defined as the ratio of the velocity in the direction of the force $V(f)$, to the force derivative $\dot{F}(f)$;

$\left|\dot{F}(f)\right|$ is the modulus of the force derivative spectrum ;

$A\,\sigma_{rad}$ is the A-weighted noise radiation efficiency of the vibrating structure;

ρ_m is the material density of the structure;

$\rho_o c$ is the acoustic impedance of the surrounding medium, air;

f is the central frequency of radiation;

Δf is the frequency band width;

η_s is the structural loss factor; and

d is the bulkiness or thickness of the structure.

In the above equation, the first term which contains $\dot{F}(f)$, relates to the shape of the force pulse induced when the tool interacts with the work piece. In a high speed shearing operation, this term will be influenced by tooling design parameters, such as blade clearance, blade angle and the speed of the blade prior to cutting.

The remaining terms in the above equation relate to the physical shape, size, material properties and mass of the shearing or impact machine, or the feedstock, under consideration. These terms cannot be readily changed without rebuilding or redesigning the machine.

Evensen [6] in another approach shown that the force pulse term in (1), $10\log\left|\dot{F}(f)\right|^2$, can be express in the time domain as a sum of maximum rates of change of the force ($10\log\sum(\dot{f}_{max}(t))^2$), formula (2). The constant C in this formula resembles the other terms in formula (1). This reflects the notion that a softer force pulse, will reduce noise from a impact.

$$L_{Aeq} = 10\log\sum(\dot{f}_{max}(t))^2 + C \qquad (2)$$

Cuschieri [7] has shown that by softening the impact force (reducing $\dot{f}_{max}(t)$), the maximum of its force spectrum will shift towards the lower frequencies and consequently gives a few force derivative spectrum, $\left|\dot{F}(f)\right|$ in formula (1). This softening can be achieved by employing different blade design parameters.

In the current study of high speed sheet metal shearing, we have concentrated on the effects of blade design parameters. These can more readily be changed in existing machines, and hence are more easily available as retrofit measures for the reduction of noise in factories producing sheet metal products.

EXPERIMENTAL PROCEDURE:

The hydraulically operated experimental shear, used for this set of experiments, was provided by BHP-Building Products, see Figure 1.

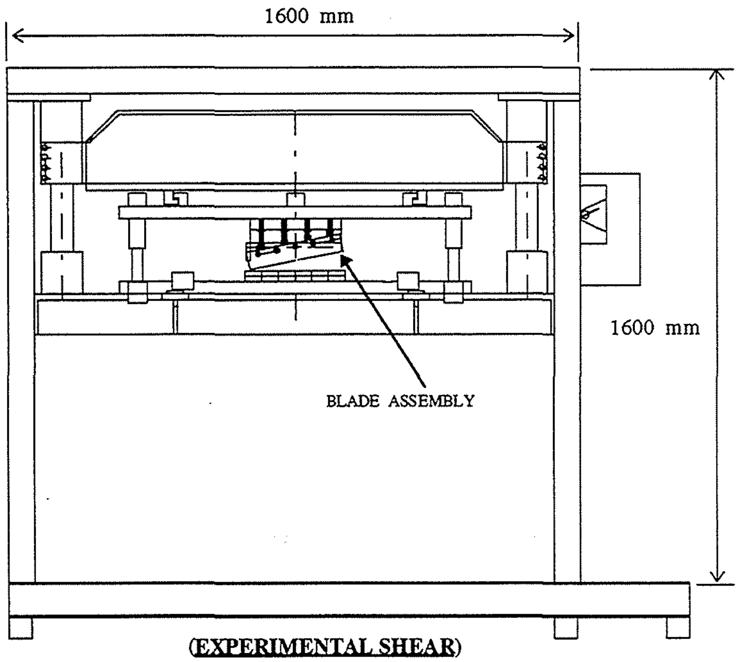

Figure 1 Experimental shear

In initial trials with this experimental shear machine, two extraneous noise sources were found. First, a high background noise $L_{eq}(A)=83$ dB(A) emanated from the hydraulic power pack, which was located next to the shear. To solve this problem, longer hydraulic lines were installed and the power pack was relocated outside the test room.

The other extraneous noise source was caused by a backlash in the upper blade carriage. It was noticed that the moving parts of the shear machine consisted of two relatively rigid parts (1 and 2 in Figure 2). These collided with each other at the start of each cut. The impact noise of that collision was considerable and had significant effect on the overall $L_{eq}(A)$. Three adjustable supports, Figure 2, were used to fix these moving parts together.

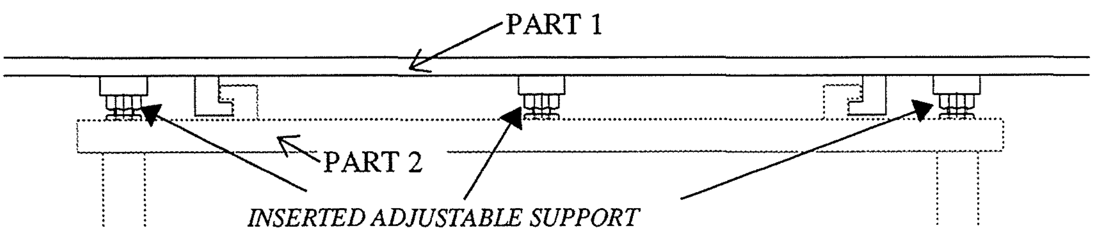

Figure 2 New inserted supports (adjustable spacer) for preventing relative movement

To investigate the fundamental noise sources in sheet metal shearing, a set of simple straight blades was designed and manufactured (Figure 3). The effect of two blade design parameters, shearing angle and blade clearance, were studied with this set of blades. The shearing angle of the blades could be varied from 0° to 11° degrees. The clearance between the blades could be varied from 0 to 3.0 mm.

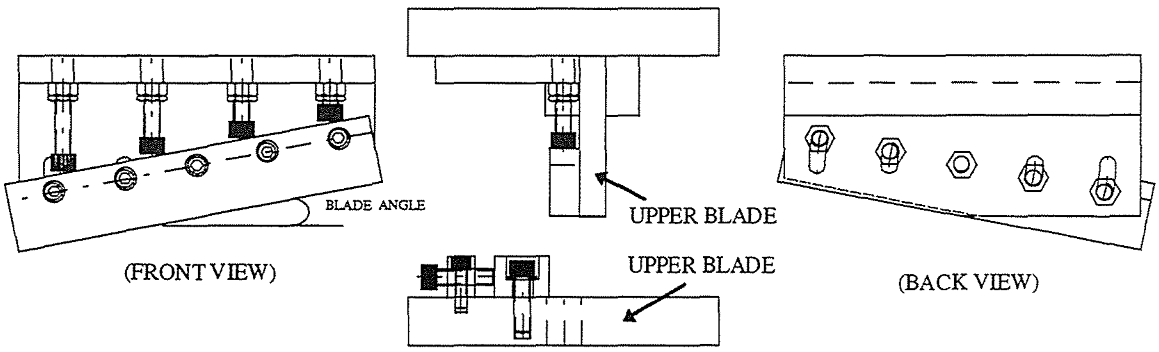

Figure 3 Views of the manufactured blade set

The effect of vibrational damping on the feedstock was also measured. A steel clamp which acts like a damper was used to keep the feedstock firmly in its position. Measurements were carried out with the damper in two different positions from the cutting edge of the blade namely 3.0 and 30.0 mm (Figure 4).

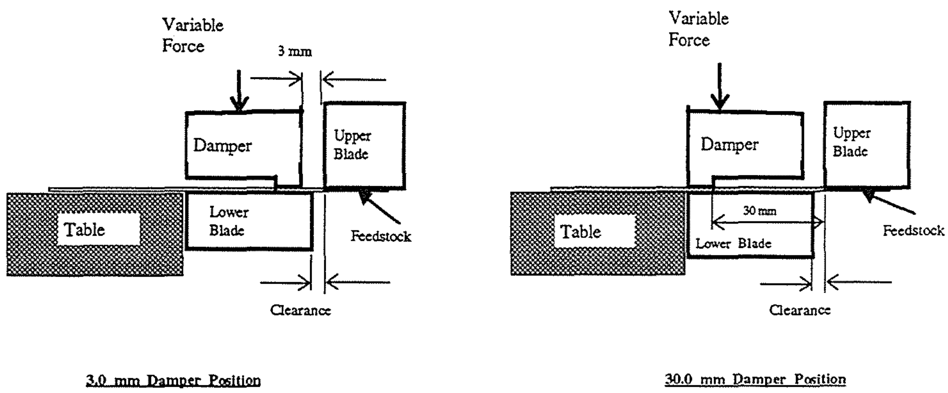

Figure 4 Position of damping on feedstock

One of the major parameters in shearing noise is the exposed area of vibrating feedstock. This effects the ringing noise, which produces the dominant noise in shearing process. Since all of our experiments were with small specimens, the effect of both size and damper position were minimal, however more experiments with larger specimens will form part of later experimental work.

INSTRUMENTATION AND MEASUREMENTS:

The noise was measured by using a Bruel & Kjaer (B&K) type 2231 Sound Level Meter (SLM) which was equipped with a $\frac{1}{2}$" B&K condenser type 4155. This noise has been recorded digitally using a Toshiba laptop computer through a Boston Technology PC30 data acquisition card. Using various signal processing software packages (SLM4B, HYPERSIGNAL, MATLAB), it was possible to extract information in the time and frequency domain from the saved data.

RESULTS AND DISCUSSION:

The variation of the induced force and its derivative in time domain in a cutting machine is typically [8] of the form shown in Figure 5. It can be divided into three parts, A, B and C, as shown.

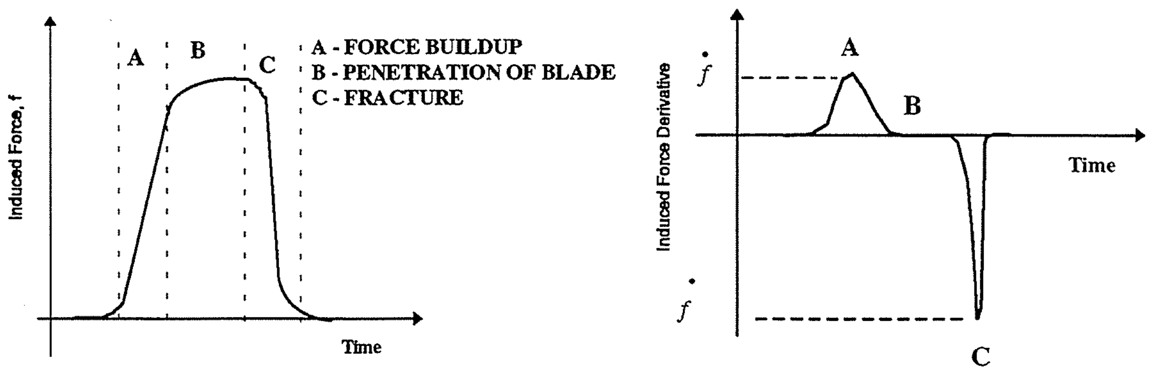

Figure 5 Variation of force induced in feedstock during the cutting process

In the Figure 5, part A corresponds to the rapid built up of force in the feedstock and the shear structure as the feedstock initially resists the movement of the blade. Part B corresponds to the penetration of the blade through the feedstock, creating the sheared surface (Figure 6) and reducing the effective thickness of the sheet. Finally part C corresponds to the rapid fracture of the feedstock, creating the fractured surface (Figure 6).

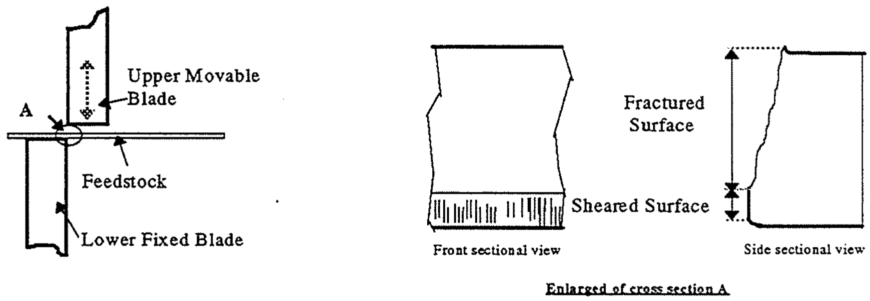

Figure 6 Creation of different surfaces during the shearing process

In the searching for techniques to reduce shearing noise, formula (2) provides useful guidance. In the experimental shear, it would have been difficult to change design parameters such as structural response, bulkiness, radiation efficiency, structural loss factor, or area and density of the structure. However altering the blade angle, clearance and speed of cutting allowed the shape of the force pulse to be controlled. The situation is similar to that which exists when one attempts to reduce the noise of an existing shear in a production facility. The results of experiments showed the following effects of the different design parameters.

1)- Blade angle:

As discussed above, in relation to formula (2), the radiated cutting noise will be affected by the rate of change of the induced force in the feedstock and the shear. At the design stage of a shear, by changing its tooling

factors, it is possible to make the shear such that it deliberately lengthens the fracture process. This can be done by making blades which have an angle with respect to each other so that the shearing will be a progressive fracture across the feedstock, see Figure 3.
Figure 7 shows the effect of blade angle on the radiated $L_{max}(A)$ noise. The specimens used for this experiment were zinc/aluminium alloy coated steel with the dimensions of (20.0 x 100.0 x 1.0) mm and the clearance between the blades was 0.12 mm fixed. The speed of cutting was 0.10 m/s.

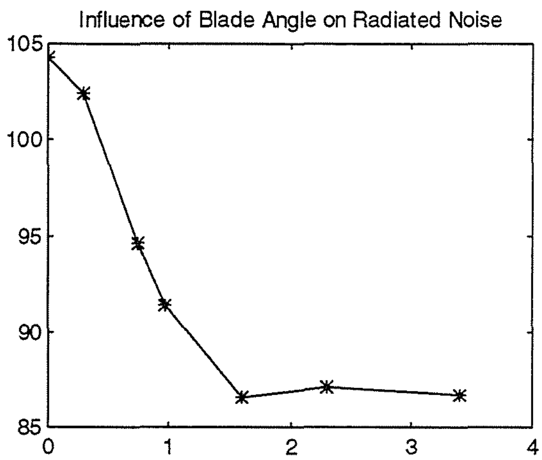

Figure 7 Effect of blade angle on radiated noise

2)- <u>Clearance of the blades</u>:

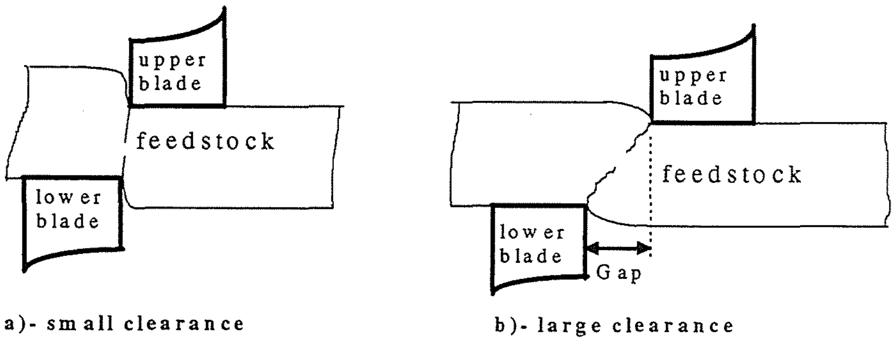

Figure 8 Effect of blade clearance in the shearing process

The variation of radiated shear noise due to changes in blades clearance (gap), Figure 8, is shown in Figure 9. For aluminium, the results suggest that a smaller clearance results in less noise, but for steel, in the range of tested thicknesses, it appears that clearance has little effect.
The specimens used for this experiment were zinc/aluminium alloy coated steel and aluminium alloy with the dimensions of (20.0x100.0) mm. The speed of cutting was 0.10 m/s in all cases.

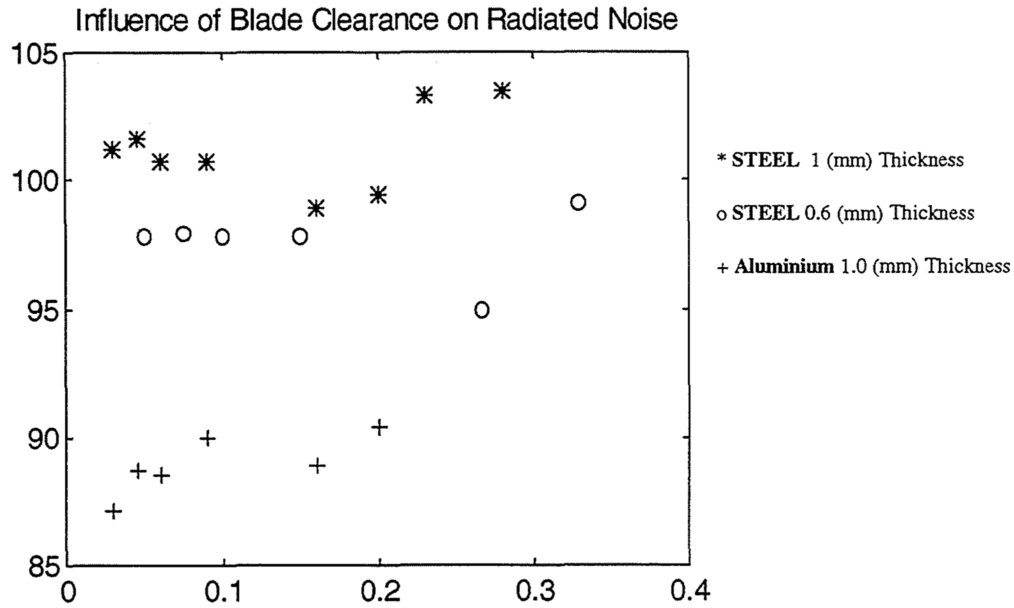

Figure 9 Effect of blade clearance on radiated noise

3)- Speed of cutting:

These results were obtained using a production hydraulic roll former shear in the Fyshwick Plant of BHP Building Products, Figure 10.

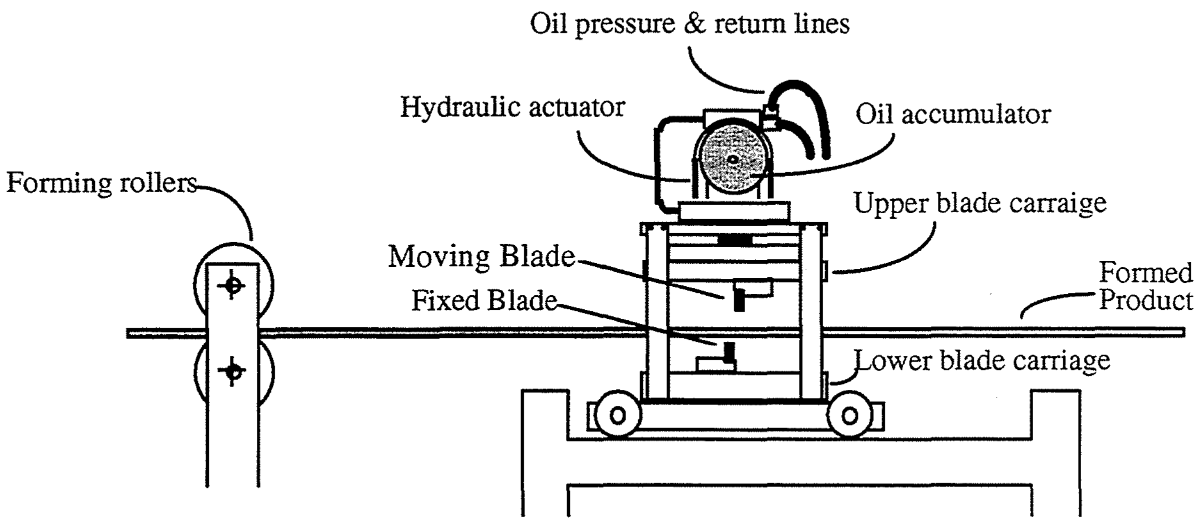

Figure 10 Hydraulic roll former shear in Fyshwick

The specimens used for this experiment, Figure 11, were zinc/aluminium alloy coated steel with the dimensions of (47x500) mm and the thickness of 0.45 mm.

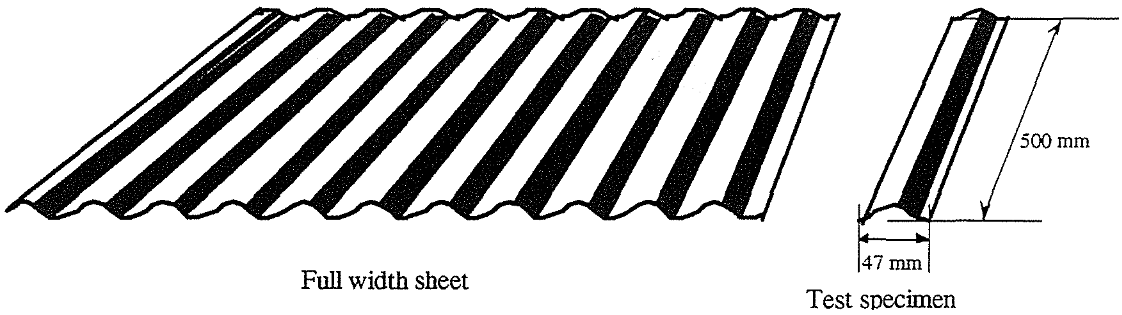

Figure 11 Specification of testing specimens

Reducing the speed of cutting tends to reduce the force derivative referred to in equation (2) by lengthening the time variation of induced force in part C of Figure 5. To monitor the speed effect on radiated noise, 8 specimens were cut at two different speeds namely 0.24 m/s and 0.17 m/s and the results (Figure 12) show an average of 2.6 dB(A) noise reduction at lower speed.

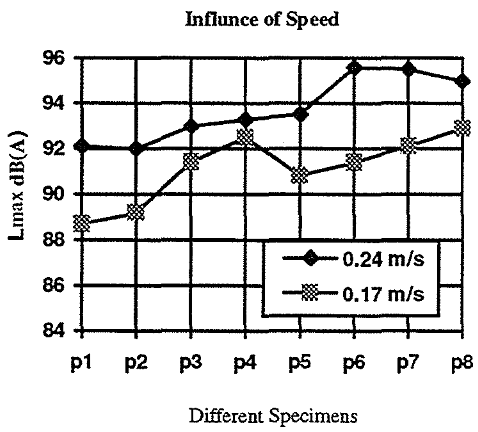

Figure 12 The influence of speed of cutting on noise radiation

CONCLUSIONS:

Experiments have been carried out into the affects of various blade design parameters on radiated noise from an experimental high speed shear. It has been found that blade angle has a very significant effect, and that shearing speed is also significant. Blade clearance has been found to have a much lesser effect in the thickness range of the materials tested. This is contrary to the findings of Burrows[8] for noise generated in punch presses using thicker materials, 2.09 mm hot-rolled steel.

The results are explained by the manner in which the above parameters affect the shape of the force pulse, by referring to theories developed by Burrows [8] and Evensen [6] for impact noise.

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Acoustics Calibration At The National Measurement Laboratory

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1 INTRODUCTION

The Australian standards of physical measurement are maintained at the National Measurement Laboratory (NML) in West Lindfield, Sydney. NML is part of the new division of Telecommunications and Industrial Physics of the CSIRO. The role of the CSIRO in the national measurement system is defined in the National Measurement Act (1948, amended 1960) in which the CSIRO is charged with the responsibility of developing and maintaining standards for physical measurement for which there are Commonwealth legal units of measurement. The Commonwealth legal units of measurement are listed in the Regulations to the Act. In an amendment to the Regulations in 1984 the acoustical quantities sound pressure level (dB) and sound intensity level were added to the list.

Other Australian measurement related organisations with which NML interacts closely are NATA (the National Association of Testing Authorities), SAA (Standards Association of Australia) and NSC (National Standards Commission). In most cases the NML role is as technical adviser on committees and assessment panels or as part of profiency trials. Since the tabling of the Kean Report[1] in 1995 this situation has become a little fluid and there have been some major changes but the roles of the NML, NATA and SAA will remain essentially the same as far as the acoustics community is concerned.

NML also interacts with the international standards setting organisations, ISO (International Standards Organisation) and IEC (International Electrotechnical Commission, origin of most electroacoustics standards) serving on many of their committees and working groups. In addition NML participates in the activities of the International Bureau of Weights and Measures (BIPM), the international equivalent of NML, on committees, projects and as the representative of Australia as a signatory to the International Metric Treaty.

The development and maintenance of the acoustics standards is seen as a national responsibility and the client for these services is the Australian Government. As such, this function is funded from government appropriation. Similarly, attendance of officers from NML on committees and assessment panels, both national and international, is seen as part of the same responsibility. NML also operates a first level calibration service which is conducted on a cost recovery basis. Calibration services currently offered in acoustics are

- Microphone Sensitivity calibration by Coupler reciprocity of 1 in and 1/2 in standard microphones at 250, 500 and 1000 Hz.
- Frequency response of 1 in, and 1/2 in capacitor microphones by electrostatic actuation in the range 0.1 Hz to 20 kHz.
- Comparison calibration against a primary standard microphone in an active acoustic coupler for most microphones for frequencies in the range 10 Hz to 2 kHz.
- Comparison calibration in the free field against standard microphones for frequencies in the range 200 Hz to 20 kHz.
- Comparison calibration against a low frequency standard for frequencies in the range 0.1 Hz to 250 Hz.
- Pistonphones and acoustic calibrators
- Sound level meters
- Data Loggers
- Filter sets
- Attenuators

Details of these services can be found in the publication Tests and Measurements 1995, National Measurement Laboratory. They are typical in most respects of those offered in many standards laboratories around the world but are enhanced where neccessary to accomodate Australian national requirements.

2 ACOUSTIC STANDARDS

At the foundation of a national measurement system must be a primary, absolutely derived standard. The internationally agreed primary standard for acoustical measurement is a 1 in capacitor microphone type IEC LS1P, as defined in IEC 1094-1, 1992, Measurement microphones- Part 1 - Specifications for laboratory standard microphones. These are calibrated using the principle of closed coupler reciprocity as described in IEC 1094-2, Measurement microphones- Part 2- Primary method for pressure calibration of laboratory standard microphones by the reciprocity technique. This method produces a primary standard set of 3 microphones, whose sensitivities are known in dB re 1V/Pa. Extensive discussion of the characteristics of these microphones and reciprocity calibration at major standards laboratories, including NML, is given in the AIP Condensor Microphone Handbook. At NML a modified method is used centred around a 3-port coupler rather than the IEC defined 2-port arrangement, allowing automation, amongst other advantages. The NML primary standard set has been calibrated over 700 times since 1980. The results are shown in Figure 1.

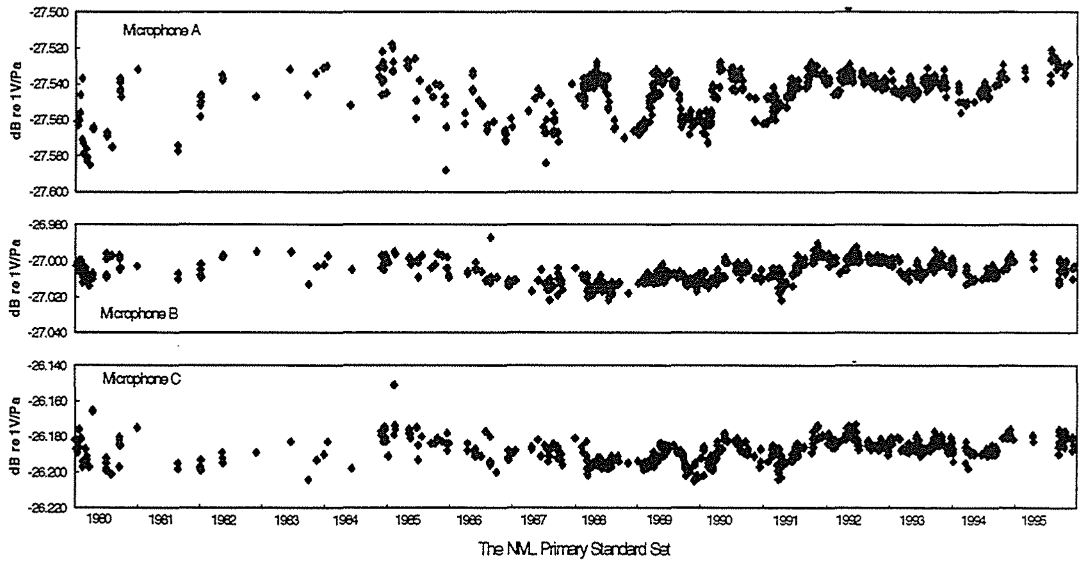

Figure 1 NML Primary Standard Microphone sensitivities: 1980-1995.

3 THE HIERARCHY OF STANDARDISATION

At every level of a physical measurement, system traceability through every step back to the standard needs to be demonstrable. In establishing the acoustic standards by the reciprocity method, measurements must be made of voltage, capacitance, resistance, frequency, volume, atmospheric pressure, temperature and humidity. Each of these must, in turn, be traceable to fundamental physical standards. If the chain of traceability is followed back through the instrumentation for each of these measurements, the physical standards required are found to be length, time interval, voltage and mass[2]. The units of the first three are defined in terms of constants of nature, and at NML are realised via a particular laser wavelength, a Caesium beam clock and a Josephson junction, respectively. Thus, the establishment of the acoustic standard becomes quite an involved process.

From the establishment of the sensitivity of primary standard microphones, a series of secondary steps occurs in the calibration of most instruments. For these tests, voltage traceability is achieved using a hp3458A digital voltmeter calibrated against the Josephson volt via AC/DC transfer standards at NML. The frequency reference is a 10 MHz signal piped to the acoustics laboratory from the NML Caesium beam clock. Other instruments such as barometers are calibrated at NML in the relevant section. At each step, and indeed at all the steps from the three fundamental units listed above, uncertainties in the measurements accrue; this is discussed later.

It is good calibration practice to make as few absolute measurements of quantities, eg. voltage, as possible. Instead, wherever practicable, restoration methods are used. Thus, extensive use is made of the inherent stability and accuracy of inductive ratio dividers to restore signals to initial reference levels rather than to simply measure them.

4 MICROPHONES

4.1 Sensitivity

The 3 microphones of the NML primary set are not used for routine calibration. Instead the laboratory maintains additional sets of 1 in and 1/2 in microphones, the auxiliary standard sets, and these are used as "transfer standards". At the start of any calibration requiring a transfer standard, the primary set is calibrated by recipocity and then several auxiliary microphones are calibrated against two of the primary standards. This is done by removing one standard microphone (microphone A) from the recipocity set and replacing it with the transfer standard. The remaining pair of microphones B and C remain undisturbed. The effect of even this minimal handling of microphone A is evident in Figure 1. This technique has the advantage that the equivalent volume correction for the replacement microphone need not be known. It is automatically corrected for using the apparent changes in the sensitivities of the remaining two primary standards.

At the top of the hierarchy are standard capacitor microphones, LS1P (1 in) and LS2P(1/2 in), as defined in (IEC 1094-1). All standard capacitor microphones such as are defined in IEC 1094-4, 1995 Measurement Microphones-Part 4 - Specifications for working standard microphones may be calibrated by coupler reciprocity exactly as the NML primary standards, either fitted with standards adaptors or with their protective grids in place. The repeatability of this measurement is < ±0.005 dB and the expanded uncertainty at the 95% confidence level is ±0.07 dB. The preferred frequencies are 250 Hz, 500 Hz and 1 kHz with 250 Hz most commonly used as it falls between the effects of thermodynamics of the air in the cavity at low frequencies and microphone resonance and other effects at high frequencies.

For most non-standard microphones, ie. any capacitor microphones of diameters other than 1 in and 1/2 in, and many other microphones, the reference sensitivity is established by comparison with a calibrated transfer standard in an active closed coupler using an insert voltage technique. The measurement uncertainty at the 95% confidence level is usually less than ±0.2 dB, depending on the microphone. Otherwise substitution in a free field must be used. These steps are illustrated in Figure 2.

		Standard Mics		
Primary	→	*Reciprocity*		
Standard Set		U_{95} = ±0.07 dB		
Reciprocity				
U_{95} = ±0.07 dB	→	Auxiliary Set		Non-Standard Mics
		Reciprocity	→	*Comparison-Closed coupler*: U_{95} < ±0.2 dB
		U_{95} = ±0.07 dB		*Substitution-Free field*

Figure 2 Steps in the transfer of microphone sensitivity from the primary standard.

4.2 Frequency Response

The frequency response of microphones is always established with respect to a reference sensitivity, usually at 250 Hz. On standard capacitor microphones where it is possible to gain access to the diaphragm by removal of the protective grid, frequency response is determined by electrostatic actuation. At NML special adaptors are used to also allow calibration of other types including the IEC LS1P and LS2P/F types and 1/4 in microphones.

The technique uses the electrostatic force between the diaphragm and a grid placed close to it to simulate the action of a sound field. The grid is charged to a polarising voltage of 500 to 800 volts and then a modulating signal superimposed. The resultant force on the diaphragm closely approximates to a pressure response such as may be used in a closed coupler. The calibrations are performed at 1/3 octave frequencies over the range 20 Hz to 20 kHz but may be extended above and below this. At NML, as the diaphragm is open to the environment during the test, it is performed in a pressure and vibration isolation vessel which considerably improves signal to noise. At low frequencies, <80 Hz, this is essential. The uncertainty is generally less than ±0.3 dB, but this depends on the microphone type.

For other microphones or acoustical devices such as SLMs, Noise Loggers etc., free field frequency response in the range 200 Hz to 20 kHz is determined by substitution with one or more NML free field standard microphones. This test is done in an anechoic chamber. In the range 16 Hz to 2 kHz the active closed coupler is used with an auxiliary standard whose pressure response is known. The cross-over between the two sets of measurements is generally taken to be 1 kHz. Below 1 kHz pressure and free field response are indistinguishable within typical uncertainties.

4.3 Low Frequency Microphones

Microphones such as are used in blast overpressure monitoring are routinely calibrated for frequencies down to 0.1 Hz using a pressure and vibration isolated vessel set up as a large volume pistonphone. Without the isolation vessel, at these very low frequencies even the effects of passing weather changes can interfere with the result!. Sensitivity and linearity are measured by direct comparision with a low frequency standard microphone. Since both microphones, including the back venting, are entirely within the vessel and sound field, the low frequency roll-off is measurable and a true low frequency response established. The low frequency standard is a 1/2 in capacitor microphone B & K type 4147 utilising an FM carrier detector system B & K type 2631. The system is calibrated by electrostatic actuation. The low frequency method is described in detail in the (AIP Condenser Microphone Handbook)

5 PISTONPHONES AND ACOUSTIC CALIBRATORS

The pistonphone may be distinguished from other calibrators on the grounds that it is a stable portable laboratory standard reference whereas calibrators are generally used for spot checks in the field. A pistonphone must be calibrated with the specific microphone type and adaptor, ie. 1 in, 1/2 in, high or low sensitivity, with which it is to be used. This is important as the determining parameters for output include the cavity volume which includes the frontal and effective volume of the test microphone. This effect varies not only with the type of microphone but to a lesser degree for samples within a given type.

Pistonphones and acoustic calibrators which use standard microphones are calibrated directly using auxiliary microphones recently calibrated by reciprocity against the primary standards. An insert voltage technique is used whereby the electrical output of the microphone, when used in the pistonphone, is compared to an accurately measured AC voltage applied in such a way as to simulate the microphone output. For pistonphones, this is done in a round robin of several pistonphones and microphones, made up with NML instruments and for the pistonphones oriented vertically upwards and downwards.

Pistonphones and calibrators which use other types of microphones are tested using the active closed coupler in conjunction with one or more reference microphones from the NML auxiliary sets and a so called "transfer standard" which is ideally the microphone to be used with the calibrator. An insert voltage technique is used with the reference microphone to set up a well known sound field in the coupler using its sound source. The transfer standard is used, along with ratio dividers or difference indicators to compare this level with the output of the calibrator.

6 OTHER INSTRUMENTS

Many of the instruments listed in Section 1 above other than microphones are best calibrated using electrical inputs and outputs as it is often not possible to accurately produce test waveforms in an acoustical environment or if used as an accessory or part of another instrument. For instruments with microphones, eg. Sound Level Meters and dosimeters, the signals are input to the microphone socket via an equivalent capacitance. The standard AS 1259.1-1990 for sound level meters specifies that an alternative input to the microphone be available.

6.1 Sound Level Meters

SLMs are calibrated according to the explicit instructions contained in Australian Standards AS 1259.1-1990 (IEC 651-1979) Sound level meters Part 1: Non-integrating and AS 1259.2-1990 (IEC 804-1985) Acoustics-Sound level meters Part 2: Integrating-averaging. The four types of SLMs, III, II, I and 0 have increasingly demanding tests and tolerances on the results. The reference sensitivity is determined using the closed coupler as described above and illustrated in Figure 2. The whole body frequency response, 200 Hz - 20 kHz, for each weighting network, is measured in the free field by substitution with a free field standard and the measurement continued down to the lower limiting frequency by comparison in the active closed coupler.

The electrical tests cover • Level range control accuracy, • Indicator level linearity • System noise • Frequency weighting characteristics • Time weighting characteristics, (Fast, Slow Impulse, Peak) • RMS performance • Overload indication, and where fitted • Integrating/averaging functions, (Leq, SEL etc.).

6.2 Filter Sets

Band pass octave and 1/3 octave filters intended for the analysis of sound and vibration have been calibrated according to the specifications in the Australian Standard AS Z41-1969 Octave, half Octave and One-Third Octave Band Pass Filters Intended for the analysis of Sound and Vibration; up to the time of writing. The AS Z41-1969 standard is being withdrawn and a new standard based on the IEC Standard 1260-1995 Electroacoustics - Octave-band and fractional - octave-band filters is under consideration by Standards Australia during October 1996. This new standard will be required to cover modern digital filter and fractional octave filter implementations which are now in common use.

6.3 Data Loggers

Many modern SLM's and portable data loggers incorporate the means for statistical analysis Lx of the time history of the measureand L_{Aeq}, SPL_A etc. To determine the performance of the statistical analyser, particularly the accuracy of the 'bin edges', a method based on a DIN procedure 45657 - Sound Level Meters; Additional Requirements for Special Measuring Tasks is used. In this procedure a sinusoidal signal is ramped up and down through accurately set level steps over the fixed period for the sample and the resultant analysis compared with the expected result.

6.4 Attenuators

The 600 ohm resistive attenuator commonly used for acoustic applications is calibrated at NML by comparison with an inductive voltage divider. The test is carried out at frequencies between 20 Hz and 12.5 kHz as these are the frequencies most used when the attenuator is used in the testing of SLM's.

7 UNCERTAINTIES

In conformity with international practice, uncertainties are quoted on all calibrations at the 95% confidence level. This means that the chance that any reported value differs from the "true" value by more the than quoted uncertainty is 5 in one hundred. Uncertainties are calculated in accordance with the ISO Guide to the Expression of Uncertainty in Measurement. In the calibration reports issued by the acoustics group a coverage factor, k = 2 is assumed unless otherwise stated.

8 THE ASIA-PACIFIC METROLOGY PROGRAM

The 22 member Asia-Pacific Metrology Program (APMP) is a collaboration between national/territorial measurement laboratories in the region. Some of its main objectives are to

- provide training, advice and consultancy on the development and establishment of new standards and calibration facilities.
- develop objective technical evidence of measurement traceability and competence as a basis for multi-lateral recognition
- support the objectives of the Asia-Pacific Economic Cooperation (APEC).

The APMP is run by an elected Regional Coordinator and the secretariat is run from the coordinator's institute. For the term 1994-1998 the APMP secretariat is at NML. Funding comes from a number of sources including APEC, member governments, the United Nations and the World Bank.

Two APMP activities in which the NML acoustics group is heavily involved at the moment are assessment of the needs of developing standards laboratories in the region and training at NML of officers from these laboratories. The group has hosted training visits from many South-East Asian countries over the last 6-7 years and the arrival of the APMP secretariat has accelerated this interest. Members of the group have visited and will visit new groups in Indonesia, Philippines and Vietnam during 1995/8.

Calibration intercomparisons between APMP members are on-going, coordinated by various member institutions. Concomitant with these activities is the stepping up of the membership and development of APLAC, the Asia-Pacific Laboratories Accreditation Cooperation, the regional equivalent of NATA. This has resulted in further regional intercomparisons. The NML acoustics group is coordinating a calibrator round robin for the APMP and a SLM intercomparison for APLAC.

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NOTES:

[1] The Report of the Committee of Inquiry into Australia's Standards and Conformance Infrastructure

[2] The standard of mass is a lump of platinum in Paris but there is a major international project to define the kilogram in terms of Avogadro's Number.

MULTI-IMPEDANCE SURFACES

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ABSTRACT

A simple technique for measuring ground impedance involves placing a pulse sound source and a microphone on the surface and from the change in waveform compared to the free space pulse, deducing the average impedance over the propagation path. This method is particularly useful for multi-impedance surfaces where theoretical calculations are very difficult. Measurements are presented which were taken over layered materials and strips of different impedance. The effect of changing the proportion of the media will also be considered. Examples to be considered include a sand to hard surface interface and across a road with grass edge and gutter.

INTRODUCTION

As noise pollution becomes an increasingly important issue, there is a greater demand for the accurate prediction of outdoor noise levels. The acoustic impedance of the surface between the noise source and the receiver is one of the parameters required in such calculations (van der Heijden L. A. M. et. al.). When there are two or more different types of impedance surface along the path exact theoretical calculations are virtually impossible to perform (Rasmussen K. B.), so approximate theory or measurements must be used. Two of the many methods of determining the acoustic impedance of materials in situ include continuous wave propagation in an open ended impedance tube placed on the ground (Dunlop J. I.) and using an impulse sound source with two microphones above the surface to be measured (Don C. G. et. al.). These methods determine the impedance of a relatively small area of the sample, so are not appropriate for non-uniform materials.

This paper investigates a method suitable for determining the average impedance over a large area (Rogers A. J. et. al.) using a impulse source and receiver on the ground, and compares the results with data obtained using a simplified single microphone impulse technique (Lawrence D. E. P. et. al.). Dug sand was chosen considered as an example of a material whose impedance varies from point to point, although this variation can be reduced by covering the sand surface with a homogeneous layer of fabric. The averaging method can also be used over an impedance discontinuity, such as a grass and concrete boundary although there are interesting consequences when it is applied to a non level surface, such as one with a gutter.

THEORY AND MEASUREMENT TECHNIQUE

The principal of measuring impedance by an impulse technique is to capture a pulse which has been modified by reflection from the impedance surface and compare its waveform with that of a direct or free field pulse which has travelled the same distance without interaction with any surface. By dividing the frequency components of the reflected pulse by the corresponding direct ones a quantity Q is calculated, which is given by

$$Q = R_p + (1 - R_p)F(w) \qquad \textbf{Eq. 1}$$

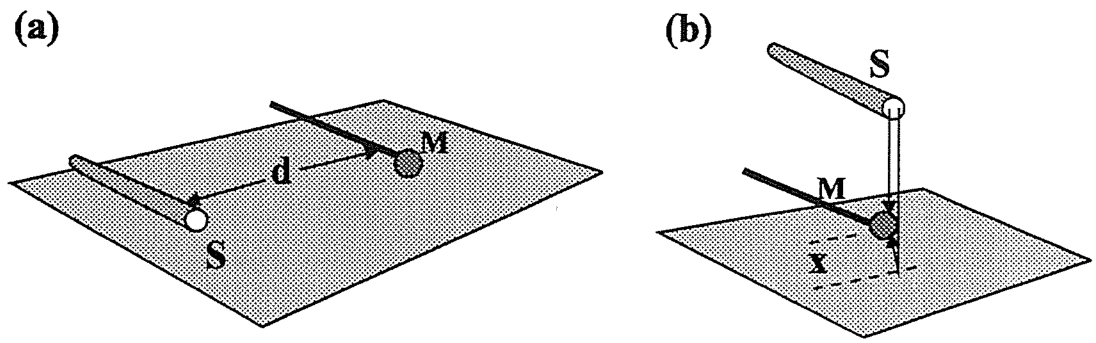
where the complex plane wave reflection co-efficient, R_p , and the boundary loss factor, $F(w)$, depend on the normalised surface impedance, Z, given by

$$Z = \frac{1 + R_p}{(1 - R_p)\sin\psi}, \qquad \textbf{Eq. 2}$$

and ψ is the angle between the incident sound ray and the surface. When an averaged impedance over a long propagation path is required, a grazing incidence geometry, method 1 as shown in Fig. 1(a), can be used. When the source and receiver are on the ground, ψ approaches zero and R_p tends to -1 for finite Z. Thus Eq.1 simplifies to $Q=1+2F(w)$. As $F(w)$ is a complicated function depending on frequency, geometry and impedance, it is necessary, at each frequency, to guess a value for Z and then calculate the corresponding value of $F(w)$and hence Q and compare this with the measured value. An iterative technique is then used to adjust Z to minimise the difference between the measured and calculated values of Q. It is more efficient to start at the highest frequency and proceed to the lower values, using the best estimate of Z from one frequency as the guess value for the next. In practice, the source and receiver are a finite distance above the surface so it is necessary to calculate R_p as well as $F(w)$ and use the full expression in Eq.1.

For the purpose of comparison, a normal incidence geometry, shown in Fig. 1(b), was also used. When ψ is 90°, $F(w)$ is sufficiently small that Q approximates R_p, and a simple substitution into Eq. 2 calculates Z. In this paper, method 2 is used for calculating the impedance of a small area of the sample.

Figure 1: Two Methods used to Determine Impedance; (a) method 1, (b) method 2.



In the impedance averaging technique, the microphone M and impulse source S are placed a distance d apart on the impedance surface, which means that the direct and reflected pulses arrive at the same time. Thus the direct pulse must always be recorded in a separate measurement.

In method 2, Fig. 1(b), the microphone senses the direct pulse and later, depending on the distance, x, of the microphone from the ground, records the reflected pulse. It may be necessary to record an isolated direct pulse in a separate experiment and then subtract it from the combined waveform. As most of the direct pulse is clearly distinguishable with the ground reflection sitting only on the tail, alignment and scaling can easily be achieved. Before processing, the direct pulse must be rescaled according to the inverse square law as it has travelled a shorter distance than the reflected pulse. In practice, the source was positioned 35.0 cm and the microphone 6.5 cm above the surface.

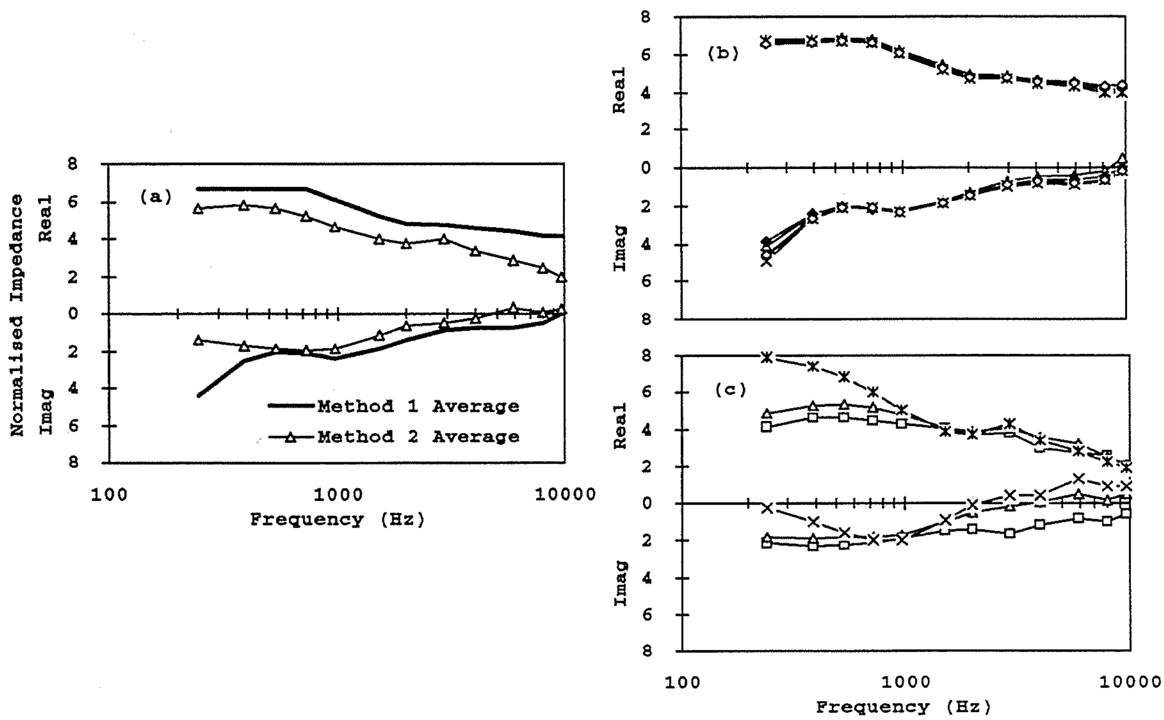
A Data Precision 2020 waveform synthesiser was used to produce the pulse which was then amplified and output through a JBL 2447H horn driver. This source gave a consistently reproducible impulse which was found to follow the inverse square law from the exit of the source tube. A Bruel & Kjaer 1/4 inch microphone and type 2218 sound level meter connected into a Data 6000 waveform analyser was used to record the signal.

EXPERIMENTAL RESULTS

To demonstrate the difference between the two methods, partially wet propagating sand was used as a test surface. The sand consisted of rock particles ranging from 0.5 to 4mm in diameter. The surface of moist sand was allowed to dry and then the sand was partially dug and smoothed, creating patches approximately 5 to 10 cm wide with differing moisture content. The sand was in a bin, approximately 1.0 m by 2.5 m by 0.3 m deep, which could be moved around under the source and receiver in method 2, so different areas could be examined without altering the measurement geometry. By removing the bin entirely the direct could be recorded free from any spurious reflections.

With d = 60 cm, method 1 was used to obtain a set of data, which consisted of an ensemble average of 12 individual waveforms recorded at the same location. Further data sets were obtained over a different area by moving both the source and receiver. The impedance deduced from the average of 5 data sets is shown in Fig. 2(a). Also shown is the impedance calculated from the average of three data sets using method 2. The impedances calculated from both techniques show similar trends, although method 1 results are somewhat higher in the real part. However, it should be noted that one measurement is at grazing and the other at normal incidence. Fig. 2(b) shows the impedance from five sets of data obtained using method 1 while Fig. 2(c) displays results from three data sets using method 2. Note that the change in impedance from point to point is only apparent in the method 2 data. The averaging method includes enough of the different impedance patches that the variation between sets is small.

Figure 2: The Impedance of Dug Sand; (a) Average using method 1 compared to Average using method 2, (b) Five Individual Sets using method 1 and (c) Three Sets using method 2.



To check that the variability in impedance observed in method 2 was due to surface effects rather than in the data processing, a uniform fabric 1 mm thick was placed over the sand bin and the impedances again measured both ways. Fig. 3(a) and (b) indicate that the cloth marginally lowers the impedance. The deviation between sets, Fig. 3(c), using method 1 is unaltered, however, the more uniform cloth has reduced the variations in method 2, especially in the imaginary part, as is apparent in Fig. 3(d).

Measurements were also obtained over a surface formed from two quite different impedances, in this case homogeneous sand and a hard board. The sand was the same as used previously after being uniformly dug. A smooth particle board, 1.2 cm thick, was placed so the interfaced was level. The propagation path was always perpendicular to the interface between the two materials and initially consisted of equal proportions of sand and board. Impedances obtained using different path lengths are shown in Fig. 4(a) and are reasonably consistent. Differences below about 1 kHz may be due to the wavelength being comparable to the distance to the diffracting edge formed by the impedance boundary. As expected the values are higher than those obtained for sand alone. The results were the same regardless of the direction of propagation, as demonstrated in Fig. 4(b).

Figure 3: The Impedance of Dug Sand with a Cloth Covering using; (a) method 1, (b) method 2, (c) Five Individual Sets using method 1 and (d) Three Sets using method 2.

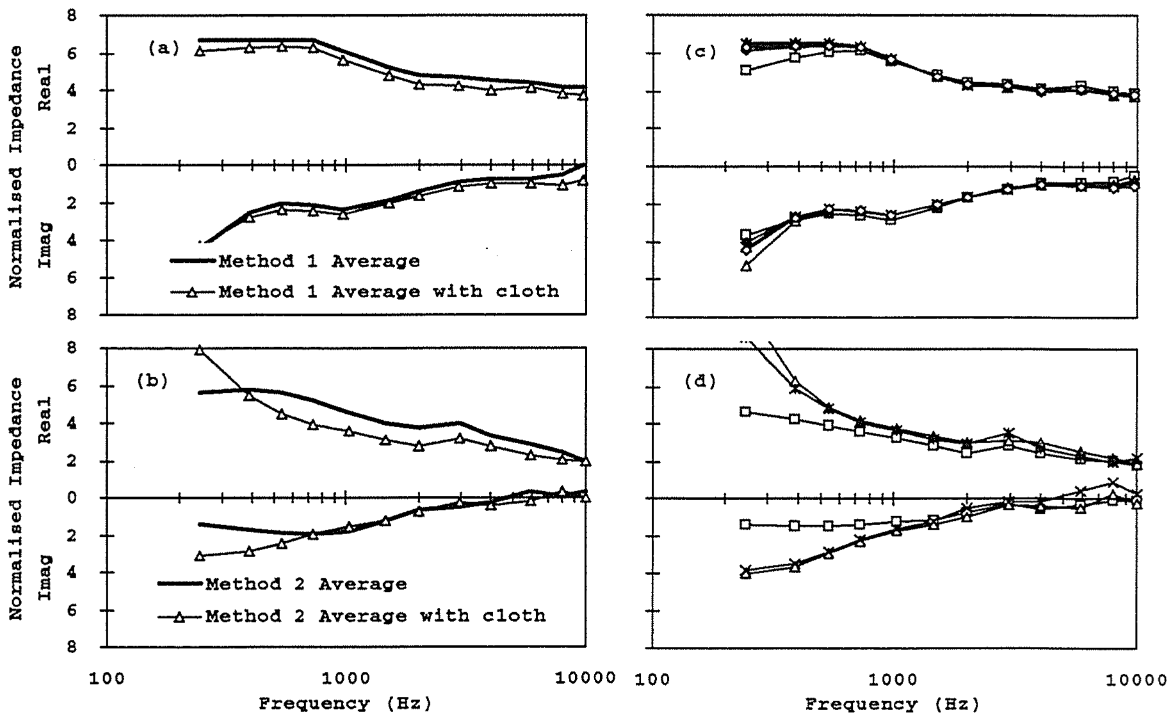
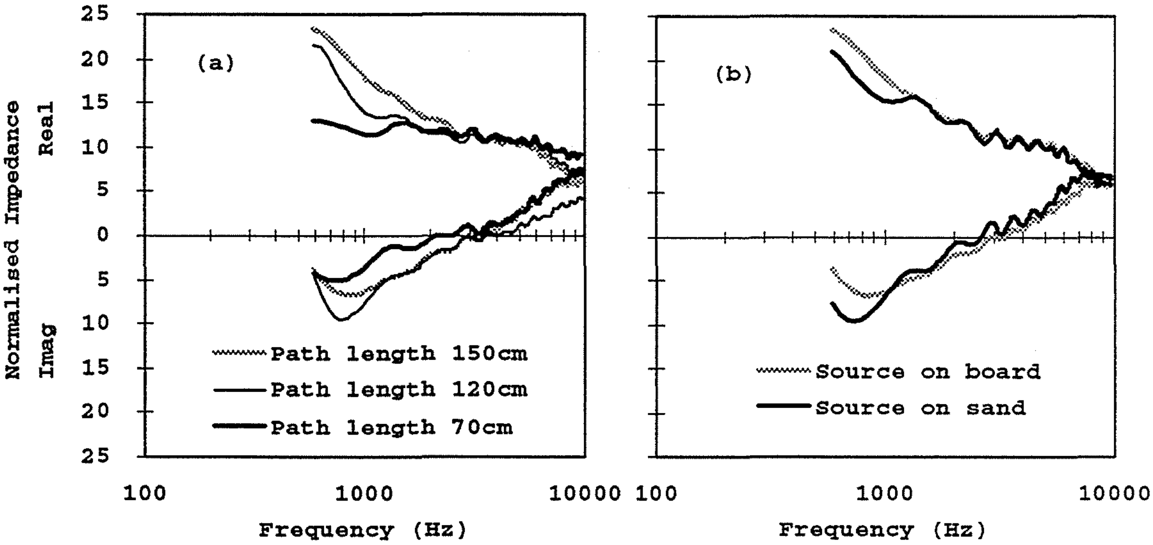


Figure 4: Impedance of Surface with Equal Proportions of Sand and Board, using method 1; (a) Varying the Path Length, (b) Interchanging Source and Receiver Positions at 150cm Path Length.



When the proportion of board along the propagation path is increased, the overall impedance increases, as shown in Fig. 5. These measurements are with the source placed on the higher impedance board, although when the source was on the sand the results were identical.

Concrete-and grass boundaries are often encountered at the edge of highways. Any noise propagation predictions based on the impedance of grass alone will be inaccurate as demonstrated in. Fig. 6. The measurements were taken with a source receiver separation of 3.18 m and with equal proportions of grass and concrete. In this instance the grass was about 2 cm high and the soil base was level with the concrete.

Figure 5: Results over a Two Impedance Surface with Varying Proportions of Sand and Board, using method 1.

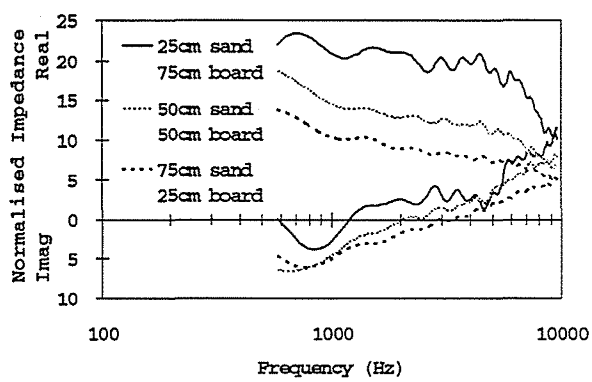
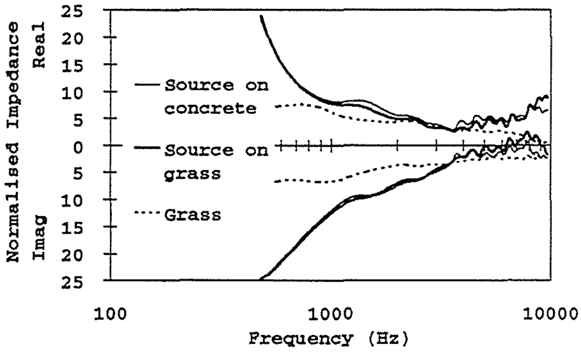
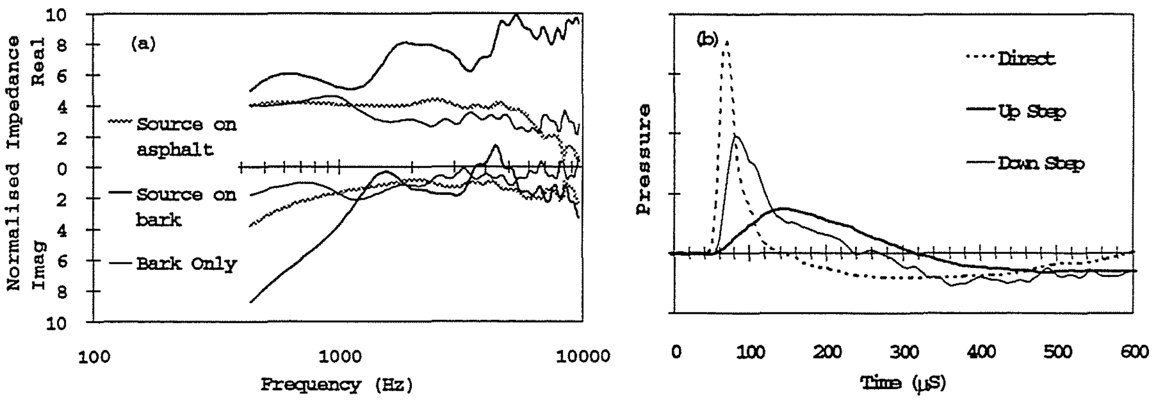


Figure 6: Results over a Two Impedance Surface with Equal Proportions of Grass and Concrete, Compared with a Grass Surface.



Finally, the effect of having a step in the surface level between source and receiver has been measured and the impedance results are presented as Fig. 7(a). In this case, the propagation path consisted of 60 cm of asphalt, a concrete gutter 10 cm high and then 60 cm of coarse tanbark. In this case the results are very dependant on direction apparently because the step acts as a diffraction barrier to an air-wave in one case and to a ground wave in the other. This effect can been seen in the accompanying pulse waveforms, Fig. 7(b), where the pulse diffracted up the step has a much reduced amplitude and longer period than that diffracted downwards. The impedances calculated in this example are of limited use as they apply only to the situation in which the measurement was taken. The situation of a step associated with a ground wave appears to be one where the results are not reversible.

Figure 7: (a) Impedance of Asphalt, Step and Tanbark Interface, (b) the Waveforms after Propagation over Step Compared to the Free Field Direct Waveform.



CONCLUSION

By having the source and receiver on the ground the average impedance along the propagation path can be determined. This method is less sensitive to local fluctuations in the impedance than a technique which measures the value over a small area. Measurements obtained over a flat two-impedance surface have shown that, generally. interchanging the source and receiver makes no difference, however when a step is present this does not appear to apply.

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