INTERIOR NOISE CLIMATES

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1989/90 NATIONAL CONFERENCE

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OUR NOISY FUTURE?

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Abstract

This paper is about the relevance of land use planning controls to environmental noise. It emphasises that to prevent environmental noise from industrial and commercial development continuing to be a major problem in the community consideration of noise aspects must be made at all levels of land use planning. This is true for the planning and development of small industrial and commercial properties as well as for major development. The principles in this paper would also apply to planning for noise from other sources such as traffic and aircraft.

If noise is not considered at all levels of planning we will indeed end up with a noisy future.

Introduction

In 1836 Colonel William Light was sent from England to establish a new colony in what is now the state of South Australia. He chose the site of Adelaide and then, as Surveyor General for the new colony, set about laying out the town.

Colonel Light was criticised very strongly for both his selection of the location of Adelaide and for the layout of the town(ref1). The principal objections against the site centered on the distance to the port (approximately 10km) and the poor quality of the port. Comments against the layout of the town included "you needed a horse to cross King William Street [the main street]". Another writer of the time "sneered" that the squares were bigger than those in London.

For those of us who live in Adelaide we are glad that Colonel Light stood up to his critics. We now have a beautiful city surrounded by parklands; a city with wide streets and one whose layout has stood the test of time. This is a good example that shows the benefits of planning decisions that look to the future.

Today it is equally as important to plan for the future as the decisions of today will be the decisions that our children and their children will have to live with. For example once an industrial area has been located near to a residential area it is likely to remain that way for many years regardless of the number of conflicts that arise between the two incompatible neighbours.

Despite this, some of those responsible for planning and development in our cities ignore the long term consequences of their decisions. Short term economic gain seems to be their principal concern. This is especially true in relation to development that is likely to have an adverse impact due to noise.

In many cases the planners and developers themselves have not considered environmental noise. In other cases they simply choose to ignore noise. In either case it is common that by the time the issue of noise is raised (if at all) developers will argue that the development has progressed too far to make significant changes to reduce noise.

It is therefore necessary that all those involved in acoustics work with relevant planning authorities and industry to ensure that noise is considered at all stages in the planning and development of areas. It is too late once a development has been completed to blame someone for not considering noise.

Recognition Of Need For Planning

During the 1970's most states of Australia established Environment Agencies. These agencies were initially set up to control pollution from existing premises. That is they were intended to be environmental "policemen". In the case of noise it was soon found that there were many premises that were making high levels of noise near to residential dwellings. The agencies therefore had a considerable amount of work to handle the number of complaints that were being received.

These situations arose principally because incompatible land uses were located near to each other. In many cases these premises shared near boundaries as a result of decisions that were made many years ago, some of them dating back to the early settlement of Australia. When these decisions were made different criteria were relevant to the selection of sites for residential and commercial activity than exist today.

For example, when selecting a site for residential development during the last century and early this century individuals would usually prefer to be within walking distance of their place of employment. Their expectations with respect to noise would therefore have been different from people today as they would have worked in the nearest factory or have known people that did. Factories also would have sited near to dense populations so that there would have been a ready source of labour.

As transportation improved workers found that they did not have to live close to the factory they worked in. They could therefore move into areas without the noise and smell. Industries also changed as new machines were developed. Machines such as angle grinders and other power tools were readily accessible even for small industries. Pressure on productivity and profitability increased. Small industries surrounded by housing expanded and brought in new machines. They may also have commenced working at night. Even local shops introduced noisy refrigeration equipment. This led to an increase in noise especially from small industrial and commercial premises.

As these changes occurred peoples expectations also changed. People from all walks of life felt that they had a right to a certain minimum quality of life. They should not have to put up with the noise and smell of the factory down the road. To this was added the trend that accelerated in the 70's for people to move back from the outer suburbs closer to the cities. As they returned they questioned why they should not have the same environment that they had in the outer suburbs.

At the same time that residents attitudes changed, the attitudes of those in the industries also changed. With more pressure to produce and less involvement with those living in the neighbourhood, management and staff of some industries began showing less concern for their neighbours. In my experience it is becoming increasingly common for the management and staff of some industries to adopt the attitude that they have a right to make as much noise as they want regardless of the effect on neighbours. This attitude is unfortunately becoming increasingly common right across society with many residents also saying the same about the effect of their actions on their neighbours. We are becoming a society that has little to do with our neighbours unless it is to complain about some of their actions.

When environmental agencies were first established their first task was to develop and enforce standards that would balance these conflicting attitudes particularly in mixed residential and industrial areas. This was done by developing standards that prescribed maximum permissible noise levels for various types of area. The type of area was either determined from a consideration of the types of premises around the point being assessed or from the existing ambient noise in the area.

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Where the levels were based on the type of area they were determined from a table of levels that itself was based on an estimate of the likely background noise level in areas of that type. Tables of this type can be traced back to the Wilson Report written in 1963(ref2). This report was produced for the the British Parliament and led to standards such as Australian Standard 1055-1973 Noise Assessment in Residential Areas. Figure 1 shows the tables from that standard on which most legislative standards in Australia were originally based.

Setting levels in this way or basing them directly on the existing measured background noise levels made no comment on whether the situations that lead to the background levels were appropriate. It simply tried to achieve a reasonable balance between the existing conflicting land uses. It also did not allow for change in the background noise level.

In many of the cases examined by environment agencies the levels prescribed were not ideal for either the residential or industrial/commercial neighbours. Residents found the noise still a nuisance or one that disturbed sleep. Industry was restricted in its use of machines, its hours of operating or was required to spend considerable amounts of money in non-productive areas to reduce their noise.

These standards therefore only aimed at achieving a compromise in existing situations. They did not always achieve a desirable solution and did not act to prevent the need for further compromise in new situations.

Environmental Impact Statements

In an effort to reduce the need for compromise many environment agencies in the late 70s and 80s, increased their involvement in providing advice to those responsible for approving development. his was intended to increase the awareness of environmental factors during the assessment of major development. Various acts of parliament were written or amended to allow governments to require environmental impact statements to be prepared. These statements are intended to fully describe the likely environmental impact of development proposals thus ensuring that decisions in relation to major development are made with a full knowledge of its likely impact.

There has been a lot criticism of the way EISs are prepared. For instance some have been accused of containing insufficient facts on which to base a decision. Others are accused of showing a bias in favour of the development as it is the developer who must pay for and supervise the preparation of the EIS. In the assessment of the Environmental Impact Statement for the proposed Sydney Harbour Tunnel the Department of Environment described the document as being "...in reality a "sales" document..."(ref3).

Despite this criticism the EIS system provides an opportunity for environmental factors to be considered for major projects. Unfortunately those responsible for calling for EIS's are not always aware of the potential causes and effects of noise. Environmental noise is therefore not always fully discussed in EIS's and in others is only mentioned briefly. It is therefore essential that all parties involved in environmental noise make it clear to governments, planners and developers, the effects of noise and the high community cost of allowing developments to proceed without adequate consideration of noise.

In particular it must be made clear to governments that noise is a major environmental issue that has a significant effect on the health and well-being of individuals in the community. The Australian Environment Council's National Noise Study (ref4) and the recent Brisbane Noise Survey (ref5) are examples of the mounting evidence of the large number of Australians seriously affected by noise.

Scheduled Premises

Environmental Impact Statements are only done for major developments and therefore do not enable the environmental impact of all the other development that occurs to be considered. Many states therefore have adopted in their environment and planning legislation provisions to enable them to consider the impact from a range of development. These mechanisms usually identify, in schedules, types of development that are known to have a high pollution potential. Applications for the development of scheduled premises must then be submitted for either comment or approval from the relevant environment agency.

This mechanism should ensure that issues such as environmental noise are taken into account. It allows for experts in the various areas such as air and water pollution and noise to apply their expertise to the development. It also allows bodies such as the State Pollution Control Commission in NSW to identify common problem areas and develop appropriate design criteria such as that in their Noise Manual. In this way developers are able to proceed with confidence that they will not be required to undertake expensive modifications to the design or operation of the development and the approving authority such as the relevant municipal council can with coonfidence approve or reject a development application. The community should therefore not be exposed to undesirable noise and future operators of the development will not be restricted.

However, the approach has three faults.

Firstly, in many cases the first that an agency hears about a development proposal is when an application for comment or approval arrives in the agencies office. By this stage the planning for the development has progressed to a stage that makes the developers reluctant to make any significant changes, especially when the appropriate changes would be to select another site or to totally redesign major components of the development.

Secondly, because almost any industrial or commercial activity or transportation can cause noise, the list of scheduled premises in relation to noise cannot hope to include any more than a small proportion of potentially noisy activities. Even a small shop with an refrigeration motor in the rear yard can prevent a residential neighbour from sleeping. Further, the complexity and number of premises covered by the scheduled premises in those states that use this system is already large and taxing the resources available.

Thirdly, in those cases where the environmental agency is only able to comment on a proposal, those charged with the responsibility for making the final decisions, usually local municipal councils, may be swayed by the short term financial gain brought about by the development instead of the long term effects it may have.

At this point it is worth making some comparisons between environmental noise and other forms of pollution such as air and water. As a general rule the larger the company, the larger is the potential amount of air or water pollution that can be produced and discharged into the atmosphere. However, in the case of noise it is often the reverse. Small companies are often located in or near to residential areas. They therefore have only short distances over which to attenuate noise and often the structure of their buildings are poor acoustically. Even ordinary domestic premises can cause high levels of noise on neighbouring properties through the use of a poorly sited airconditioners or swimming pool filter pump. Therefore, in the case of noise it can be the small development that causes the most disturbance to individuals in the community and yet it this development has the least attention paid to it by environmental agencies.

To effectively prevent future noise in the community it is therefore necessary to not only use the EIS and the schedule premises systems but also to tackle noise at all levels of planning.

Zoning of Land

In most states there is a system of land use planning. These systems zone land into categories that describe the appropriate uses for land. For example some land may be zoned as being intended for residential use while other may described as being intended for extractive industry.

In South Australia most of the populated areas are zoned under the Planning Act. These zones are shown in a large volume known as the The Development Plan. Figure 2 shows an example of a page from the Development Plan. Zones are determined by local municipal councils for areas under their control. In most states land use planning is primarily the responsibility of local government although, as in S.A., state governments also play a role by ensuring that local government's abide by the states planning principals.

Planning decisions are particularly relevant to environmental noise control. A buffer of commercial land use between residential and industrial land would usually ensure that occupiers of the two seperated land would not be restricted by the activities of the other. However, where planning allows an industry to share a common boundary with residences it is almost certaint that at some time in the future the operations of the industry will cause a noise that will interfere with the residents enjoyment of their property. It is also likely that this would lead to restrictions being placed on the companies operations.

Despite this many planning decisions do not take into account the compatibility of land uses to neighbouring land. Instead, decisions appear to be based on what presently exists on a site. Many planning maps show boundaries between zoning categories as clearly following the existing land use boundary regardless of the suitability of the existing situation. A clear example of this is shown in Fig. 2.

Another example is shown in Fig. 3. In this example a section of land is zoned as being suitable for general industry yet on three sides it has land zoned as residential. Also, part of that residential land protudes into the general industrial land. This has led to a situation where a heavy industrial activity is now causing high noise levels on nearby residences and at a school. Also, expansion of the industrial activities or the further development on the residential land is not able to be controlled as both developments are permitted on their respective sections of land.

These examples show that those responsible did not use the planning controls for any long term strategy for the orderly development of an area. Instead the controls were used to maintain or justify existing situations. These decisions were made either because those responsible considered that there was no use in trying to change what already existed in the areas or because they supported economic growth and development regardless of the social cost to neighbouring areas.

It is only by applying planning decisions that support orderly development that we can hope to improve existing situations which were developed before planning controls were effectively applied. It may be that it will take many years to alter an area in accordance with the desirable plan. However, if the plan for an area reflects the desirable long term goals for that area there is at least a chance to encourage appropriate change. Further, it is only by appropriate planning decisions that we can hope to prevent the mistakes of the past from being continued into new areas.

In S.A. the land use zoning of the old areas of Bowden and Brompton was reviewed and the zoning was altered to reflect the long term intent of areas. For example areas that were a mixture of housing and industry are now generally zoned as either one or the other. Industries are therefore encouraged out of areas zoned as residential and into those areas zoned as industrial. Similarly residences are encouraged into residential areas. These changes have been encouraged by the government taking an active role in purchasing and selling land. However, even without this action being taken the zoning in the area would still encourage more orderly development.

The zoning of land is therefore critical as the first step in ensuring that development is appropriate with respect to environmental noise. As zoning is usually done by local municipal councils it is essential that councils are advised of the consequences of their decisions so that long term orderly development will be the result.

Correct Use Of Land Use Zoning

Even when zoning is correct with respect to noise there is still a need to ensure that development complies with the intent of the zoning. A common buffer zone between industrial and residential land is "light industry". Part of the definition of light industry usually states that it is industry that does not affect the ammenity of an area by virtue of its noise, smell, soot, smoke, etc. Where development complies with this definition it is an effective buffer. However, it is common for developers and planners to consider any small industry as being light. This has led to a number of conflict situations where, for example, a panel

beating workshop is located next to housing.

In other situations where the proposed industry is large some developers and planners have still argued that it is light. It appears that in these cases there is a desire to encourage development for the economic benefits it brings regardless of the long term social and health costs to near neighbours. In some cases this also leads to significant economic cost to those who will eventually use the development when they are required to pay for compliance with legislated environmental standards or are prevented from expansion due to those requirements.

Recently, in a suburb of Adelaide a developer proposed to place an engineering workshop on vacant land near to a school and to a residential area. The developer argued that his activities were "light" as they would usually comply with the maximum permissible noise levels specified by the Noise Control Act. Initially the Council in that area strongly supported the development despite comments from the S.A. Noise Abatement Branch that the noise would distract students at the school, annoy nearby residents and would ultimately lead to restrictions on the industry, particularly with regard to future expansion. It appeared that the council could only see the economic advantages of using currently vacant land. In this case, following considerable opposition from a residents action group, Council did change its mind and found an alternative site for the industry next to existing general industry. This confirmed that there are solutions even in the most difficult cases.

These cases are not confined to industry. For example in another suburb of Adelaide the council is supporting a bowling centre that would be open until midnight. Many of the vehicles leaving this centre would drive through otherwise quiet residential streets. Again, the council appears more interested in the development of currently vacant land than the long term impact on residents.

In both of the above cases the developments proposed were not permitted under the Planning Act and therefore to obtain approval required that the proposals had to be fully considered. Had the relevant zoning for the area permitted the development there would have been no control over the siting of the developments. Also it would have been unlikely that residents and the Noise Abatement Branch would have been aware of the developments until they were built and complaints were received. This emphasises the need for appropriate land use zoning and the importance of correctly applying the requirements of the zones to development.

Appropriate zoning of land is therefore only the first step in controlling development to prevent excessive noise. Zoning must be followed by the correct interpretation of zones.

Development decisions

In South Australia, as in most states, the primary responsibility for land use planning is the Local Municipal Council. However, there are many other individuals and organisations involved in planning and development. They include the state government, appeals courts, builders and developers and their advisors (such as acoustical consultants), and of course the residents likely to be affected. Thus, to ensure that environmental noise is considered at the planning stage requires more than ensuring that the land use zones shown on a map have considered noise. Instead each of the parties must be aware of the need to consider noise at all stages of planning and development and be aware of how best to take noise into account.

One of the best means of ensuring that all parties are aware of noise at the planning stage of development is to develop environmental noise standards that can be related back to the relevant land use zoning for the area. If this were done planners and developers would be more likely to be aware that there are noise standards. It would also enable them to identify the consequences of their decisions in terms of environmental noise. Further, it would give developers and those likely to be affected by noise clear information that would enable them to plan with confidence at the very early stages of proposed development.

Such a system is used by the Victorian Environment Protection Authority and is being considered by the S.A. Department of Environment and Planning. This system uses a method that enables the maximum permissible noise level at any point to be determined by assessing the types of zoned land within 200 metres of that point (Fig.4, ref6). The maximum permissible noise levels at a point are high where the area contains a large proportion of industrially zoned land and low where the land is principally zoned residential. Where the land is commercial or light industry the levels are mid way between the industrial and residential levels.

In S.A. it is intended that this system will be adopted in such a way that it will enable a computer program to produce maximum permissible noise level contours around any land. In this way planners, developers and builders will be able to quickly recognise the environmental noise levels that must be met.

Conclusions

The work of environmental agencies in the 70s and early 80s to achieve compromise between industry and residents in existing complaint situations have not yet resulted in a resolution of all environmental noise problems. However, there is a more important task that confronts all involved in environmental noise and that is to ensure that the planning and development decisions being made today do not duplicate the mistakes of the past. The following are some of the methods available to ensure that noise is considered.

environmental impact statements

selection of land use planning zones

enforcement of the objectives of zones particularly for new development

assessment of development applications within zones and the application of appropriate and enforceable conditions where approval is granted

Given that any industrial or commercial development can cause noise the input into planning and development decisions should not be restricted to only large development. It should be at all levels and therefore all those involved in making planning and development decisions should be made aware of noise as a factor. This should be achieved through training at all levels and can be significantly improved if the noise level standards used by environment agencies support the relevant land use planning scheme.

Recommendations

1. That all those involved in acoustics work with relevant planning authorities and industry to ensure that noise is considered in all stages in the planning and development of areas.

2. That it be made clear to all those involved in planning and development decisions that noise is a major environmental issue that has as significant effect on the health and well-being of individuals in the community.

3. That the appropriate zoning of land must be recognised as the first step in ensuring that development is appropriate with respect to environmental noise.

4. That appropriate zoning of land must be followed by the correct interpretation of zones and by consideration of noise at all stages of development.

TABLE 2
ADJUSTMENTS TO BASE LEVEL FOR DIFFERENT TIMES OF DAY WITH RESPECT TO RESIDENTIAL SITES
(See Rule 6.2.3)

Time of day	Adjustment to base level dB(A)
Daytime, Monday to Friday (7 a.m. to 6 p.m.)	+ 5
Daytime, Saturdays and Sundays and Public Holidays (7 a.m. to 6 p.m.)	0
Evening (6 p.m. to 10 p.m.)	— 5
Nighttime, Monday to Friday (10 p.m. to 6 a.m.)	—10
Nighttime, Saturdays, Sundays and Public Holidays (10 p.m. to 7 a.m.)	—10
Early morning, Monday to Friday (6 a.m. to 7 a.m.)	— 5

NOTE: Time periods and levels are arbitrary and may be varied by Regulatory Authorities to suit local conditions.

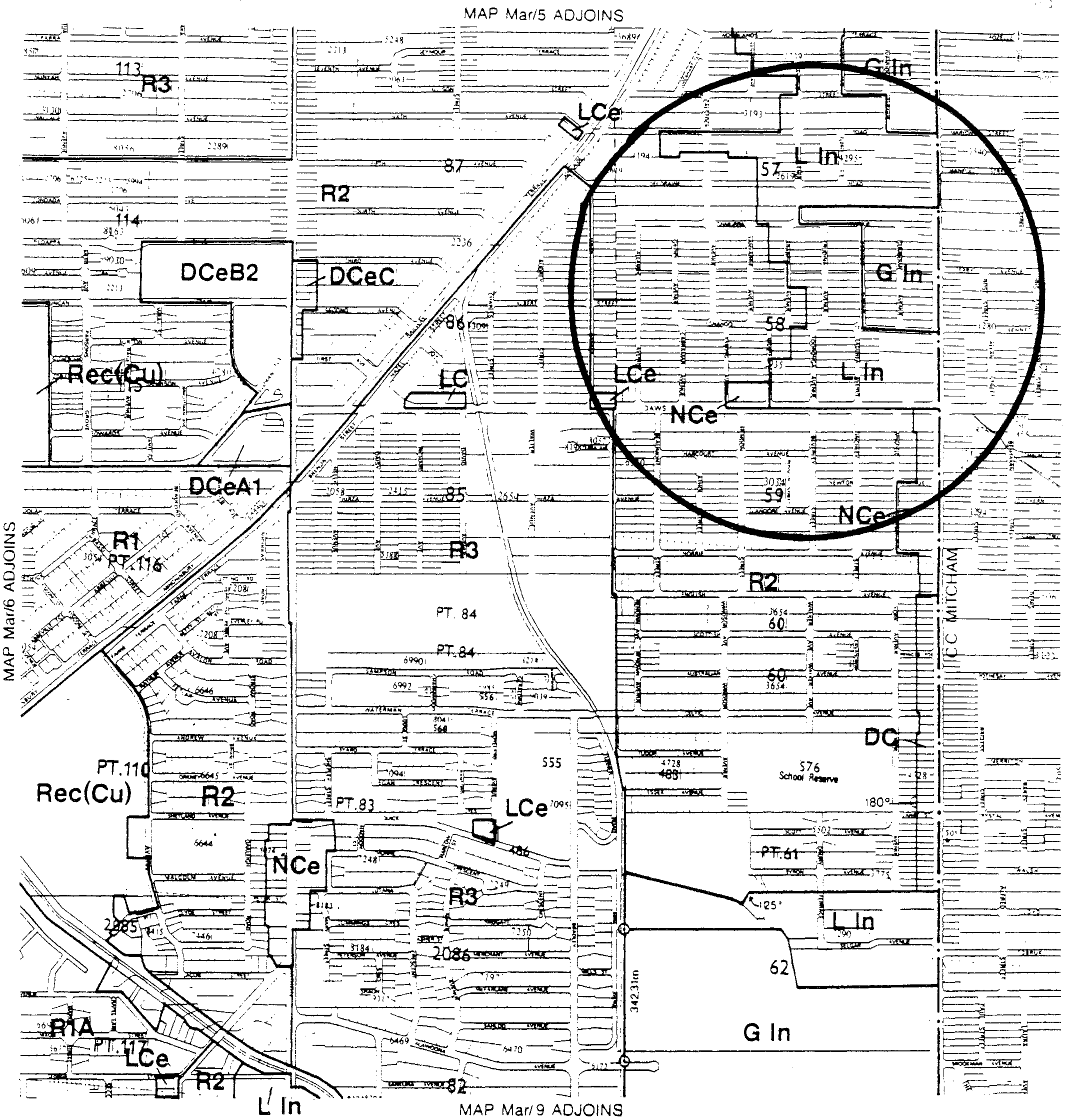
TABLE 3
ADJUSTMENTS TO BASE LEVEL FOR DIFFERING AREAS CONTAINING RESIDENCES
(For application to Rule 6.2.4)

Noise area	Description of neighbourhood	Adjustment to base level dB(A)
R1	In rural and outer suburban areas with negligible transportation	0
R2	In general suburban areas with infrequent transportation	+ 5
R3	In general suburban areas with medium density transportation	+10
R4	In suburban areas with some commerce or industry or adjacent to dense transportation	+15
R5	In city or commercial areas or residences bordering industrial areas	+20
R6	Within predominantly industrial areas	+25

FIGURE 1. AUSTRALIAN STANDARD 1055 - 1973

NOISE ASSESSMENT IN RESIDENTIAL AREAS

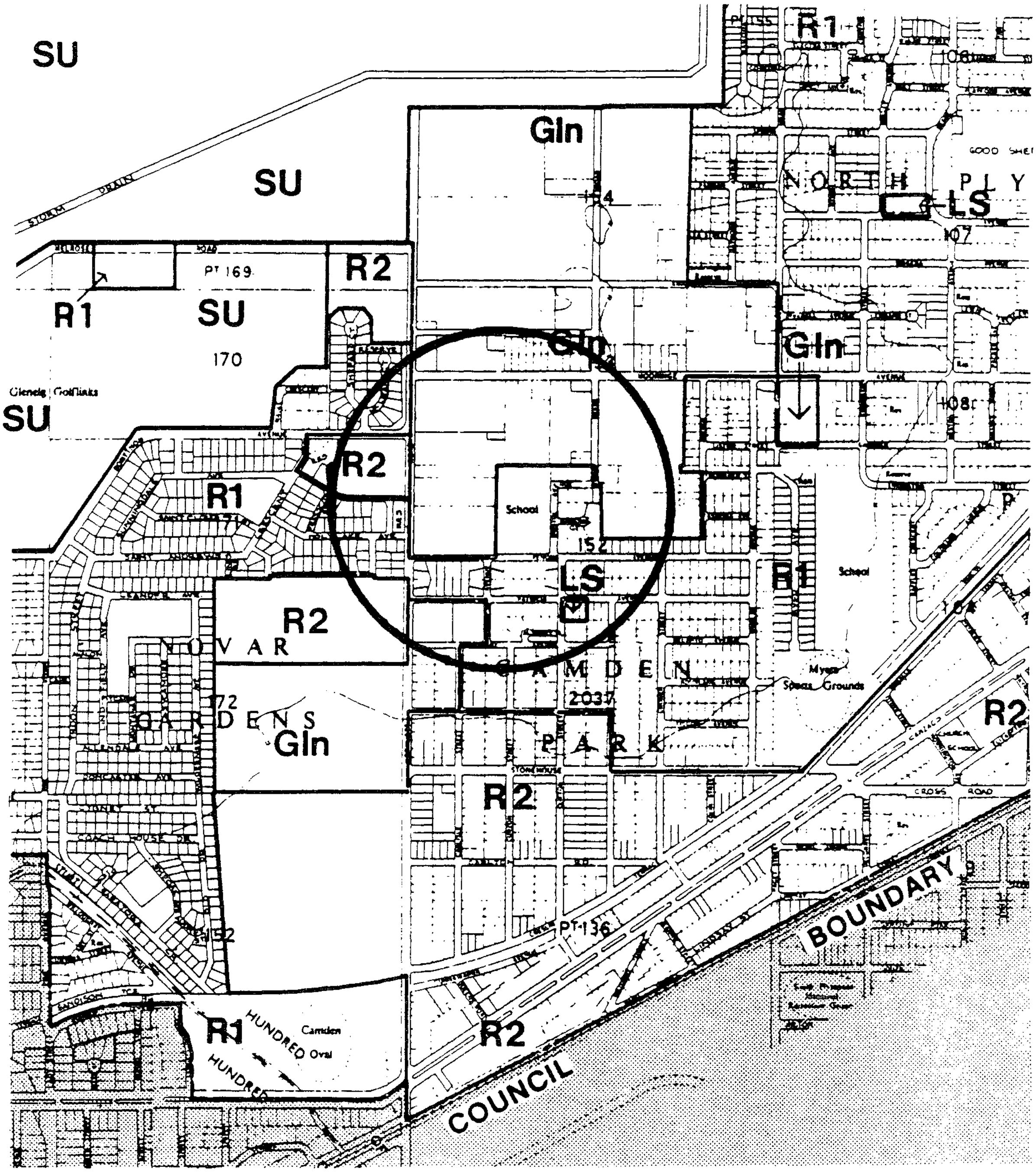
(NOTE. THE BASE LEVEL IS 40 DB(A))



R1	Residential 1
R1A	Residential 1A
R2	Residential 2
R3	Residential 3
DCeA1	District Centre A (Park Holme)
DCeB2	District Centre B (Park Holme)
DCeC	District Centre C
NCe	Neighbourhood Centre
LCe	Local Centre
LC	Local Commercial
L In	Light Industry
G In	General Industry
Rec(Cu)	Recreation (Community)
———	Zone Boundary
—·—	Local Government Area Boundary

FIGURE 2. EXAMPLE OF THE S.A. DEVELOPMENT PLAN

NOTE THE CIRCLED AREA WHICH SHOWS ZONING WHICH FOLLOWS EXISTING PROPERTY BOUNDARIES



SCALE 1:15 000

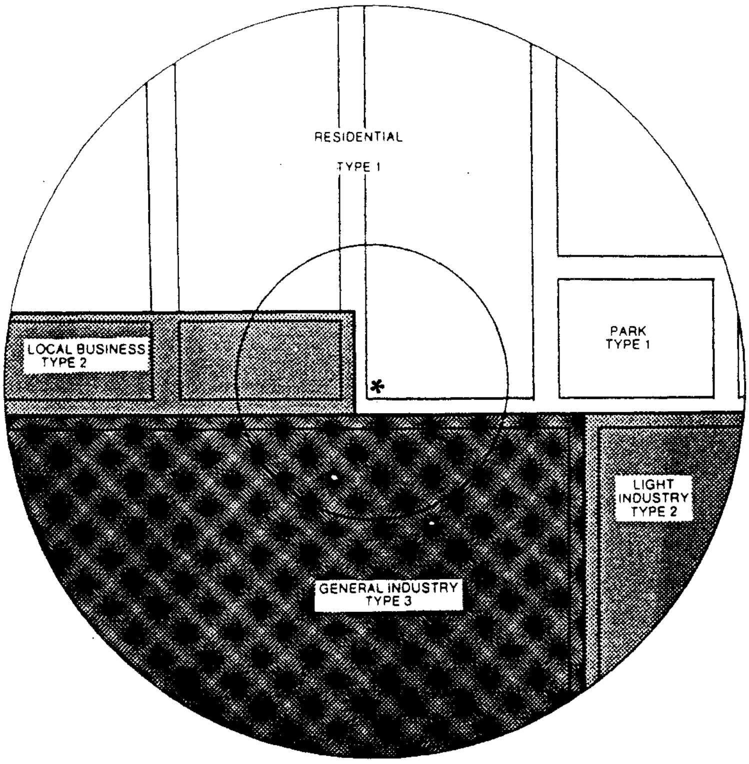
Residential 1	R1	Local Shopping	L.S.	District Commercial	D.C.
Residential 2	R2	Local Commercial	L.C.	Light Industry	L.In.
Residential 2A	R2A	Local Office	L.O.	General Industry	G.In.
Residential 3	R3	District Shopping	D.S.	Special Uses	S.U.

FIGURE 3. EXAMPLE OF THE S.A. DEVELOPMENT PLAN

NOTE THE CIRCLED AREA WHICH SHOWS A SCHOOL AND HOUSING NEXT TO GENERAL INDUSTRY

FIGURE 4 EXAMPLE OF MAXIMUM PERMISSIBLE NOISE LEVEL CALCULATION USING THE VICTORIAN STATE ENVIRONMENT PROTECTION POLICY NO. N-1

1. DETERMINATION OF THE AREA TYPES WITHIN 200 METRES OF THE ASSESSMENT POINT



	OUTER CIRCLE	INNER CIRCLE
AREA OF TYPE 1 (EG. RESIDENTIAL)	726	88
AREA OF TYPE 2 (EG. COMMERCIAL LIGHT INDUSTRY)	237	38
AREA OF TYPE 3 (EG. INDUSTRY)	584	75

*NOTE. THE UNITS USED ARE ARBITRARY AS IT IS THE RELATIVE AREAS THAT ARE USED. RELATIVE AREAS MAY BE DETERMINED WITH A PLANIMETER OR WITH A COMPUTER

2. CALCULATION OF INFLUENCING FACTOR

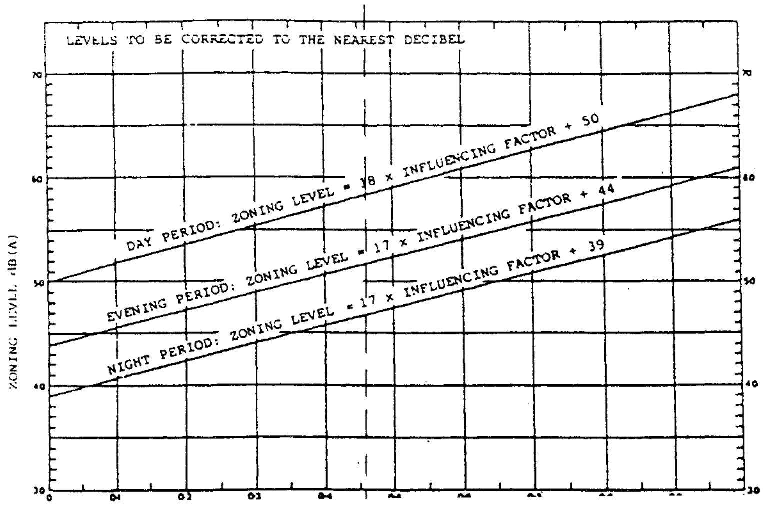
INFLUENCING FACTOR =

$$\frac{1}{2}\ \frac{\text{(AREA TYPE 3)} + 1/2\text{(AREA TYPE 2)}}{\text{AREA TYPE 1 + AREA TYPE 2 + AREA TYPE 3}}\quad \text{OF 140m DIA CIRCLE}$$

$$+\ \frac{1}{2}\ \frac{\text{(AREA TYPE 3)} + 1/2\text{(AREA TYPE 2)}}{\text{AREA TYPE 1 + AREA TYPE 2 + AREA TYPE 3}}\quad \text{OF 200m DIA CIRCLE}$$

$= \underline{0.46}$

3. DETERMINATION OF MAXIMUM PERMISSIBLE NOISE LEVELS



DAY TIME 58 dB(A)

EVENING 52 dB(A)

NIGHT 47 dB(A)

References

1. Elder, David _William Light,s Brief Journal and Australian Diaries_, (Wakfield Press, 1984)

2. Committee on The Problem of Noise _Noise Final Report_ (HMSO London, 1963) Commonly refered to as the Wilson Report

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6. Victorian Government _State Environment Protection Policy (Control of Noise From Commerce, Industry And Trade) No. N-1_ Victorian Government Gazette No. S 31 Thursday 15 June 1989

INTERIOR NOISE CLIMATES

PROCEEDINGS OF THE 1989/90 NATIONAL CONFERENCE OF THE AUSTRALIAN ACOUSTICAL SOCIETY

(INCORPORATED IN NSW)

19 AND 20 APRIL, 1990

COTTESLOE BEACH RESORT,

PERTH, WESTERN AUSTRALIA

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The following papers were accepted for presentation at the planned 1989 Conference which was deferred as a result of an airlines dispute. The authors were unable to attend the combined 1989/90 Conference, nor were there alternative presenters available. Given the circumstances, these papers are included in a separate section of the proceedings.

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KATEIFIDES, M	8
LAI, J	11,14
LAMBERT, J	20
LAWRENCE, A	21
LIU, J	7
MAC ALPINE, S	12
MAC LACHLAN, J	12
MACPHERSON, J	16
MARSHALL, H	1
MIDDLETON, W	2
MILES, A	2
NORTON, M	10
PAN, J	15
PAPADOPOULOS, A	22
RENEW, W	4
RUMBLE, R	4
SMITH, K	2
SORIA, J	10
WEATHERALL, F	13
WILLIAMS, W	3

PROGRAMME

WEDNESDAY 18 APRIL 1990

3.00 pm	Site visit - The Orbital Engine Company, Balcatta. (Bus leaves conference venue 2.30 pm).
6.00 pm	Informal registration period and Sundowner at the Cottesloe Beach Resort.

THURSDAY 19 APRIL 1990

8.30 am	Registration		
9.00 am	Official Opening		
9.15 am	Keynote address - Prof. Harold Marshall "Recent Developments in Acoustic Design Methods"		
10.15 am	Morning tea		
10.45 am	"RASTI Determination of Speech Intelligibility of Public Address Systems at Air and Rail Terminals", W. Middleton et al, Queensland University of Technology.		
11.15 am	"Classroom Noise in an Aboriginal School", W. Williams, National Acoustic Laboratories, Sydney.		
11.45 am	"The Design, Testing and Use of a Reverberation Chamber", W. Renew, Division of Environment, Qld, and R. Rumble, R. Rumble Pty. Ltd., Brisbane.		
12.15 pm	"Sound Transmission Loss of Fluted and Corrugated Panels - Damped and Undamped", C. Hansen, University of Adelaide.		
12.45 pm	Lunch		
2.00 pm	"A Catalogue of Acoustic Shortcomings in the School Music Department" G. Barnes, Acoustic Design Pty Ltd, Melbourne.		
2.30 pm	"A Study of Noise Criteria for Clean Workshops", Q. Chen, CSIRO, Melbourne and J. Liu, the Ministry of Chemical Industry of China.		
3.00 pm	"Speech Privacy Design Guide", M. Kateifides, Public Works, NSW.		
3.30 pm	Afternoon tea		
4.00 pm	Focus Group 1 Open Plan Offices	Focus Group 2 Industrial Building Interiors	Focus Group 3 Acoustics in Schools
5.00 pm	Optional additional focus groups		
7.00 pm	Conference Dinner (Royal Freshwater Bay Yacht Club).		

FRIDAY 29 APRIL 1990

9.00 am	Keynote address - Mr. L. Challis "New Challenges and Solutions for Interior Noise Climates".
10.00 am	"The Evaluation of Sound Intensity from Tape Recorded Signals Via the Quarter - Square Multiplier Principle", M. Norton and J. Soria, University of Western Australia.
10.30 am	Morning tea
11.00 am	"Noise Source Identification in Building Ventilation System: An Application of the Sound Intensity Technique", J. Lai, Australian Defence Force Academy, Canberra.
11.30 am	"Road Traffic and Interior Noise - A Survey of Noise Levels in Houses Exposed to Traffic Noise", S. MacAlpine and S. MacLachlan, State Pollution Control Commission, NSW.

12.00 pm	Stream 1	Stream 2
	"Measures by the RTA in NSW, to Reduce Traffic Noise, in Particular along the Wahroonga-Berowra Freeway", F. Weatherall Roads and Traffic Authority, NSW.	"Field Measurement of Transmission Loss Using the Sound Intensity Technique", J. Lai and M. Burgess Australian Defence Force Academy, Canberra.
12.30 pm	Lunch	

1.30 pm	Stream 1		Stream 2
	"A New Look at the Description of Acoustic Enclosures", J. Pan and D. Bies, University of Adelaide.		"Towards a Code of Practice for Occupational Noise in the Entertainment Industry in WA", J. Macpherson, Dept. of Occupational Health, Safety and Welfare of WA.
2.00 pm	"Predicting the Sound Transmission Loss of Cavity Walls", J. Davy, CSIRO,		"Labelling: A Useful Strategy in Environmental Protection", I. Eddington, University College of Southern Queensland and N. Eddington, Division of Environment, Brisbane.
2.30 pm	"An Automated Impedance Tube", R. Hooker, University of Queensland.		"Our Noisy Future", J. Lambert, Dept. of Environment and Planning, South Australia.
3.00 pm	Afternoon tea.		
3.30 pm	Focus Group 4 Noise & Urban Planning	Focus Group 5 Noise Labelling	Focus Group 6 Sound Intensity
4.30 pm	Summary and Close - Prof. Harold Marshall.		

RECENT DEVELOPMENTS IN ACOUSTICAL DESIGN PROCESS

Professor A. Harold Marshall

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Abstract

A discussion of the context of concert hall design process which defines and limits the role of physical science and related linear activity, is followed by a review of the state of knowledge of audience and performer preference. A strategy for design is illustrated by the Orange County Performing Arts Centre which opened in 1986. Alternative strategies are discussed and recent developments in acoustical modelling as an interactive design aid are described. An application of the new technique in the halls of the Hong Kong Cultural Centre concludes the paper..

RECENT DEVELOPMENTS IN ACOUSTICAL DESIGN PROCESS

Dr A Harold Marshall, Head of the Acoustics Research Centre and Professor of Architecture, University of Auckland, New Zealand.

Partner, Marshall Day Associates, Consultants in Acoustics, Auckland (NZ) and Melbourne (Aust).

Overview

To make sense of comment on concert hall acoustics in all its diversity, and to put the efforts and techniques discussed here in perspective, some general observations may be useful. Concert hall design - the process by which a concert hall - most expensive Arts hardware of the 20th Century at $20-60,000 per seat - is decided in all its detail, is, I believe, unique in its complexity.

That 20th Century communities are prepared to spend such vast sums, loads the project with a significance far beyond that of almost any other urban structure. Perhaps museums or art galleries are comparable, or perhaps the accommodation of the National Legislature, but such edifices are not then subject to critical comment, week in , week out for years after their opening ceremonies.

One suggests that concert halls have come to hold a place in modern society akin to the Cathedrals of Europe in their day - that music almost alone addresses spiritual values in a world dominated by rampant materialism of whatever colour. And as in the 15th Century, spiritual values were overlaid and corrupted by temporal power, so power plays a major role in determining the use and abuse of new concert halls today.

Be that as it may, it is certain that **concert halls are built to enable musical communication between performer and audiences as their primary function.** That is the function which acoustical research on these rooms must be directed towards.

Hierarchies of Complexity

Recent discussions on Cosmology and Chaos, popularised notably by physicist Paul Davies at Newcastle University in the UK, emphasise the role of holism in physics (2). The thrust of his argument is that thresholds of complexity exist above which complex systems can only be meaningfully discussed as 'wholes'. Physics, by and large, has addressed itself to the properties of the lowest members of this hierarchy. While their properties 'laws of physics' are incapable of predicting those of the systems above their horizon, nothing in the superior or higher order organisation is permitted to contradict the fundamental properties discovered in the lower. This makes sense in terms of the observed levels of activity in concert acoustics.

Discernible 'thresholds' or 'horizons' exist in the process of concert hall design, particularly between those activities which are linear, and those which are non-linear. Four levels are suggested:

Power
Craft./Art
Engineering
Physical Science

Principal examples of each are shown in Table 1.

Non-Linear	POWER	politics, criticism, public opinion, media, social dynamics, owner aspirations, management.
	ART/CRAFT	composition, performance, perception, design skill, craft.
Linear	ENGINEERING	engineering, psychophysics, contract, construction process, professional
	PHYSICAL SCIENCE	physics and mathematics

Higher members **must not contradict** knowledge collected in the lower but **neither are they explicable in terms of it**

TABLE 1: FOUR LEVELS OF ACTIVITY IN RELATION TO CONCERT HALLS

Physical Science/Engineering

These levels of activity are familiar to most of us because of our education and subsequent professional careers. Acoustics is of course a branch of physical science, and the body of knowledge is derived from this branch, supported by engineering practice. During most of this century physicists and engineers have sought to apply the methods of physics to concert hall design. The result has not been a striking success to the perplexity of scientists and often to the scorn of musicians. It is clear that physics is a **necessary but insufficient** dimension of concert hall design.

The **linear / non-linear** horizon lies above both physics and engineering.

Craft/Art

The violin is offered as an illustration of the strength of craft processes. This instrument was perfected before the integral calculus was invented without any analytical technique. Today we can use holographic interferometry, strobes and transforms to understand the resonances of strings, bridge and body but if we ask how were the **f** holes shaped, the double curvature and thickness of the plates designed, the asymmetry of the structure devised, we see that it has taken 300 years for analysis to catch up with craft skill.

In the same way musical composition, performance, design skill, and creative listening in a concert hall call on capacities of mind which are at this time inaccessible to Physics.

Power

It is a mistake to think that the exercise of even an outstanding skill at the art / craft level is enough to ensure a satisfactory outcome to a concert hall project. The structure of power is not explicable in terms of **skill** any more than skill is explicable in terms of physics. Critics recognise this power and exercise it with an astounding ferocity and they together with owners, management and the media directly affect the 'success' of a concert hall project irrespective of the qualities of the design. (Figure 1).

Multi-dimensional Task

As if those levels of complexity were not enough, the task of providing a space to enable musical communication is itself multi-dimensional. Research during the past 15 years or so has been directed toward preference as a measurable response to concert hall sound (3-20). At once one is confronted with a series of branches in the measured population. See table 2.

Performers

The requirements of performers as a whole are quite different from the preferences of audiences, and differ further even amongst performers. This is probably due to the masking effects relevant to each group (21-23). Certainly the strong preference amongst vocal groups for reverberant conditions is not matched in instrumental groups, while for solo vocalists the situation is more, (and at this time, inexplicably) complex still. There are still open questions.

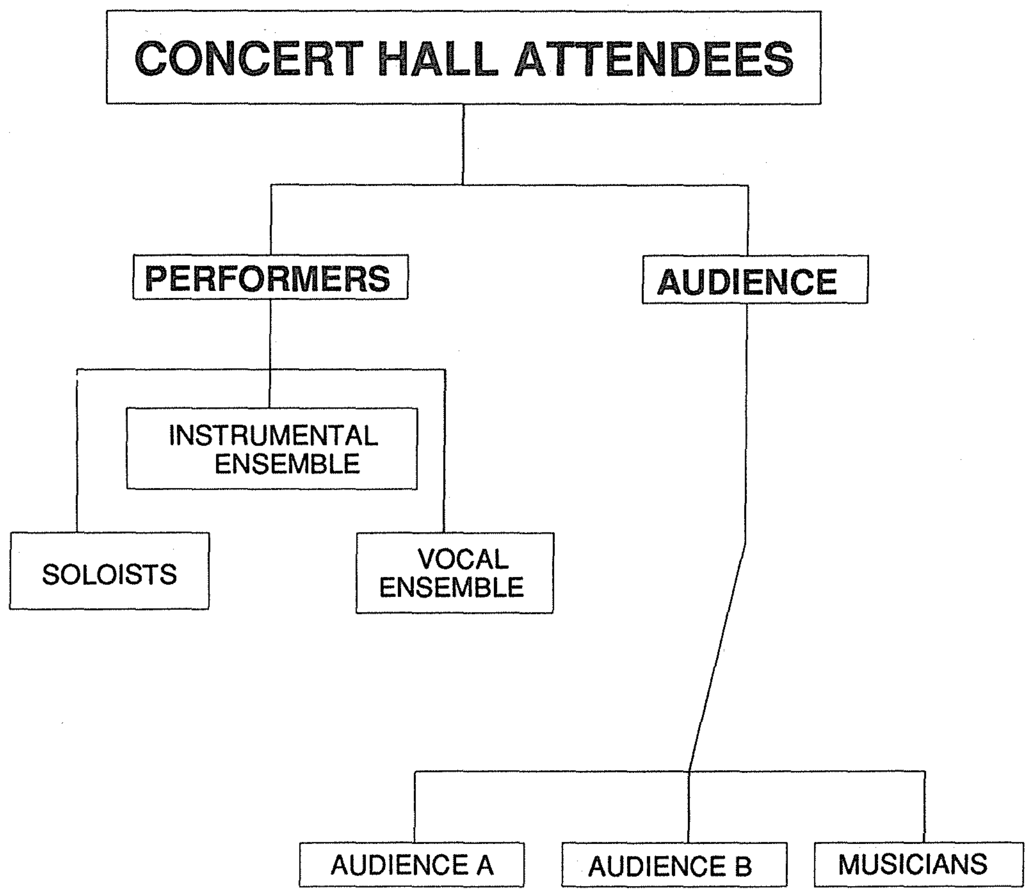

TABLE 2: THE POPULATION OF CONCERT HALL ATTENDEES

One of the perplexing features of the research on vocal ensemble Professor Jürgen Meyer and I conducted at PTB Braunschweig following the 1983 ICA, is shown in the slides which follow. In a series of presentations of synthetic sound fields to a group of singers, those with reverberant sound were preferred over those without, for all levels of reflection energy, except in the region corresponding to 40ms delay after the direct sound. There the preference disappeared. The result was striking in all the trials performed. On return to New Zealand I was confronted with complaints from choral groups performing on the stage in the most recent hall there - the Michael Fowler Centre in Wellington. Such complaints did not occur in the Christchurch Town Hall, which was rather similar The over-stage reflector was somewhat lower in Christchurch. The calculated reflection strength corresponding to the two situations seems to confirm the experimental result. No-one knows why.

Audiences

As Strøm (24) has pointed out, a translation system is necessary to interpret subjective impressions into physical quantities which can be measured. Research during these two decades has, as most will be aware, revealed four preference factors, about three of which there appears to be a consensus - **loudness, reverberance and spatial impression** (note that I am using the subjective properties rather than any of the competing physical measures available). In 1983 I offered clarity as the fourth member, but clarity is not an independent variable - it depends on loudness, the early reflected energy, and inversely on reverberance. That year, Ando had proposed his initial delay time factor as determining preference, relating it to the auto-correlation function of the music. The implications of this idea took some time to sink in, and has not yet been fully assimilated at the design level. Furthermore, Ando's measurement of the preferred delay of major reflections matched to the audio correlation function of the **music** is not ascribed a subjective impression ('What does it sound like'), and so does not fit easily into the scheme I have outlined here. It is nevertheless an intriguing notion which may have to be realised electroacoustically - perhaps in the home audio reproduction system, rather than in concert halls themselves - for maximum preference. There can be little doubt Ando's results are correct, but what does it mean for design? (25,26).

So far so good, but Barron (27) has recently published the results of a subjective review of British concert halls to supplement his earlier work on their objective properties. This revealed inhomogeneity in concert audience preferences, and so confirmed a similar finding some years earlier in Germany. Not only was preference divided between those who liked **reverberance,** and those who liked **acoustic intimacy,** but the interpretation of these and other subjective terms clearly had a range of meanings to the participants in his tests.

In this light, inhomogeneity in concert hall sound may be seen as advantageous, contrary to conventional wisdom. Audiences could be invited to have their **preference profile** measured individually and/or collectively. The collective value could be matched to the properties of particular areas in an inhomogenous hall.

The problem of reflection delays matched to the varying ACF of the music during a performance, however, remains.

Strategy for Design

In each of the halls referred to, a common pattern of design process has been followed. At the heart of it is communication between members of the design team - the architect, certainly, but also lighting and interior designers whose work always impacts the acoustician directly, as do indirectly structural and mechanical services engineers (28-30). (See Table 3).

STRATEGY FOR DESIGN

Communication between members of the design team is essential

A systematic process:

from **briefing** which defines acoustical objectives,

through **concept formation,**

design development,

engineering the **model studies**, 1:50 and 1:10,

refinement during **contract documentation,**

supervision during construction and **evaluation** prior to opening on completion.

TABLE 3: STRATEGY FOR DESIGN

ORANGE COUNTY PROCESS

All these steps occurred during the design for the Orange County Centre for Performing Arts

Architects: Caudill Rowlett Scott, (CRS)
Houston
the Blurock Partnership
Newport Beach
Acousticians: Joint Venture Acoustical
Consultants:

Paoletti/Lewitz/Associates,
San Francisco Ca
Jerald R Hyde, St Helena Ca
Marshall Day Associates, NZ

The emphasis was on **communication**, particularly during the concept formation stage - the **design squatters session**.

TABLE 4: ORANGE COUNTY PROCESS

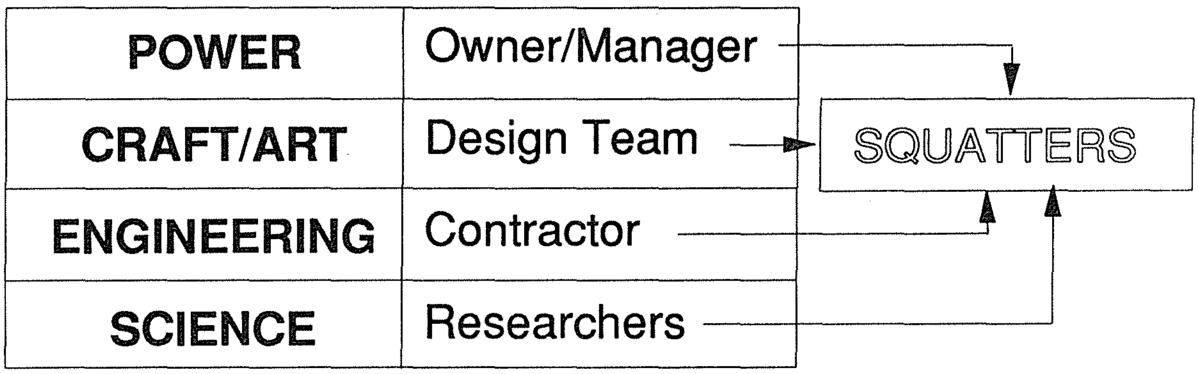

TABLE 5: DESIGN SQUATTERS PARTICIPANTS.

Orange County Process

In many ways the Orange County project represented an ideal process for the design, construction and evaluation of a concert hall - a process which recent developments in design process would enhance without replacing its essential characteristics. (See Tables 4 & 5).

Design Squatters

In terms of the hierarchical model suggested earlier, the **conceptual design phase** during the design squatter session is seen as an active bringing together of all four levels into the level three activity - design. This was possible only because of a willingness of all parties to a participate and the readiness of the principal architectural designers, Caudill Rowlett Scott of Houston, Texas, to allow them to do so. That CRS had particular team design skills was a tremendous advantage here (31).

The 'design squatters' is so called because the entire team 'squatted' in a temporary office space for two weeks. (Figure 8). During this time, management, architect, acousticians, planners, structural and lighting engineers prepared a design for submission to the Client which included every major concept which was finally realised five years later in the finished building group. The theatre / concert hall had the asymmetrical design which originated with the acoustical team to provide high lateral reflected energy in a broad fan dictated by the sightline requirements of 3000 viewers. The idea is most clearly seen in the oblique 'orchestra' or 'stalls' seating plan which provides a cliff and seating layout responsive to the asymmetry of the orchestra radiation pattern.

Successful though the "design squatters" was (and in only two weeks every major design feature of this project was already included in the scheme), it was still almost two years before reliable objective acoustic data was available to confirm the validity of the design. By then, of course, it was too late to alter anything major (eg. room volume), though minor corrections were made, eg. the areas to be treated by QRD diffusers. The building opened on time, on schedule and on budget.

Alternative Strategy

While acknowledging the validity of the hierarchy of complexity involved in concert halls as outlined earlier, there will be those who at the predictive level will seek to write a mathematical expression which accounts for the transmission of sound throughout an auditorium, and accounts for the resultant sound field at any position and time after the emission of the sound. That is surely the starting point for my distinguished colleagues Dr Yoshimasa Sakurai and Dr Yoichi Ando, though the latter bravely ventures even further to predict audience preference numerically as well. We moved away from it in favour of our physical models. Our reasoning finds an echo in Paul Davies'. After the first reflections even in a hall of only moderately complex shape, the transmission rapidly becomes intractable to precise solution.

In any case, analytical procedures are not helpful in an essentially synthetic procedure - the conceptualisation of a building form. They would be, if available, to check the resulting sound fields. I have dealt with the origin of the form-concept elsewhere (34). The advantages of physical modelling to the designer are considerable even in this age of accessible computer graphics.

That is why we have persisted with the 1:50 scale modelling during the past several years. Previous limitations have now been substantially overcome, and the model technique is now available throughout the designing process, and may even be useful in training the designer, because of the immediacy of its feedback to the designer's mind.

IMPROVED DESIGN PROCESS - 'MIDAS'
joint project between the Universite du Maine (Le Mans)
and the University of Auckland, NZ

OBJECTIVES :

1:50 (or larger) **scale modeling without the need for dry air**

hybrid system using the advantages of a physical architectural model and digital data acquisition

interactive with the design process

availability on **personal computers**

user friendliness

TABLE 6: MIDAS OBJECTIVES

Improved techniques - The Midas Package

Lack of **immediacy** certainly characterised our experience on the Orange County project. Even though the concept models shown on the slides were available after the first week's work, reliable acoustic data from the project were first **measured** nearly 2 years later as already noted.

That process came by way of a study 1:50 model at developed design stage, graphical studies, optical model studies at 1:10, the construction of the 1:10 model, which itself took four months, and finally the construction and operation of an air drying system. The plant for that operation alone cost in the order of $10,000.

In 1984, CNRS, through the Universite du Maine at le Mans, and the University of Auckland in its Acoustics Research Centre, initiated a joint project to improve this performance. (See Table 6). Dr Jean-Dominique Polack, the first author of reference (35) came to Auckland with the explicit intention of superseding the air drying necessity by numerical compensation after digital acquisition on our HP 1000F computer. Subsequently, a post doctoral fellowship for Dr Xavier Meynial from the same institution enabled him to join our group. We have recently replaced our HP 1000F with Mac II computers, and the suite of programmes is now implemented on that system, together with a number of improvements and extensions.

The programme works at all scales from 1:50 to full size, as recounted in ref (35).

What Midas Can Do

Midas performs room acoustic measurement based on FFT techniques -

Reverberation Time and EDT
Clarity and Definition
Amplification
Speech Transmission Index
Lateral Energy Fraction

at full scale or at model scale and using impulsive sources, pseudorandom noise, steady or interrupted noise as source signals.
Numerical compensation for real air conditions is included so that there is no need for air-drying or the use of dry nitrogen.
It makes measurements of absolute sound level, impulse response, and spectrum over user-defined time intervals.
Acquisition can be made in real time or from taped records, and operation is in single or dual channel modes. Statistical analysis of results is always available.

The process is menu driven and is so accessible and quick that with very little training, it is useful to architects at the conceptual phase of a project. In fact of course, the term 'architects' here is shorthand for the 'design team' in the design squatters scenario I've outlined. We have found simple and easily worked materials to construct this first model and represent the design in terms of 'hard surfaces' and the audience. So-called 'foam-core' board, lacquered for the hard surfaces, and stick-on foam strip for the audience, provide an adequate first order approximation for the conceptual phase.

With this technique, the results which took nearly two years in Orange County could have been produced to a first approximation in the time taken to build a foam-core model of the hall - one to two days. Table 7 gives typical output.

Date 19/ 7/89	MIDAS	T:

Source file:

Hall: TST CONC HALL UNOCCUPIED MAY 1989

Source position: CONDUCTOR

Microphone position: N118 Balcony

Column# Freq	1 Noise(dB)#	2 EDT(s)	3 RT30(s)	4 C_50(dB)	5 C_80(dB)	6 TI
Full band						
707 f	-30.0	1.99	2.12	-5.1	-1.2	.49
Octave bands						
62 o	-21.4	2.98	3.13	-4.5	-3.7	.42
125 o	-30.7	1.91	2.14	-2.9	-.2	.53
250 o	-33.1	1.99	1.85	-6.8	-2.1	.46
500 o	-33.7	1.92	2.03	-4.9	-.1	.52
1000 o	-33.6	2.19	2.11	-4.2	-.9	.48
2000 o	-32.6	1.96	2.03	-6.4	-1.4	.48
4000 o	-31.0	1.75	1.81	-6.4	-1.8	.48
8000 o	-26.1	1.50	1.52	-6.0	-1.5	.51
Third Octave Bands						
200 t	-35.0	1.80	2.07	-6.9	-.5	.53
250 t	-28.6	2.11	1.70	-6.7	-4.2	.42
315 t	-31.2	2.17	1.57	-6.7	-4.4	.44
400 t	-30.5	1.99	1.88	-7.7	-3.0	.47
500 t	-32.5	1.90	1.94	-8.3	-3.8	.45
630 t	-35.0	1.90	2.04	-4.0	1.0	.54
800 t	-32.4	2.27	2.03	-7.2	-3.4	.46
1000 t	-34.7	2.12	2.12	-2.5	.3	.49
1250 t	-33.1	2.23	2.19	-5.5	-1.1	.48
1600 t	-34.9	2.03	2.05	-5.5	-.8	.51
2000 t	-31.0	1.97	2.02	-4.5	-.8	.46
2500 t	-32.5	1.95	2.02	-9.6	-2.5	.49
3150 t	-30.8	1.71	1.92	-6.9	-1.9	.50
4000 t	-31.3	1.89	1.75	-7.1	-2.2	.47
5000 t	-31.5	1.63	1.75	-5.8	-1.4	.48
6300 t	-30.0	1.55	1.53	-6.0	-1.7	.51
8000 t	-23.1	1.30	1.29	-5.9	-.3	.56

TABLE 7: TABULATED RESULT OUTPUT FROM MIDAS

As the design firms and develops, more accurate and elaborate models become justified. We use machined acrylic sheet for QRD surfaces and reflectors, as we see in the Tsim Sha Tsui models.

Application at Tsim Sha Tsui - Hong Kong Cultural Centre

The final illustration of the use of MIDAS is in the evaluation of the new Tsim Sha Tsui Cultural Centre in Hong Kong. This is the first full-scale use of the programme at the evaluative phase of a concert hall design process, through the earlier version of MIDAS was used in the model studies at 1:50 scale.

The Centre has two large halls and the acoustical brief called for them to have distinct properties. The 2200 Seat Concert hall was to be appropriate for the symphony and for organ; the 'Grand Theatre' 2000 seats was to be excellent for speech and opera. The slides show the contrasting form of these rooms in their 1:50 scale models and their contrasting impulse responses respectively.

Recent measurements using Midas have confirmed that the sought for result has indeed occurred in these rooms,as the C80 energy fractions in the two halls illustrate. Measurements of RT & EDT, full and empty, lateral energy fractions, loudness distribution and spectrum were also made.

Conclusion

In this paper I have attempted to bring some order into the complex and perplexing field of concert hall acoustics by proposing an organisation of activity into **'linear'** and **'non- linear'** and discussing their relative positions. Some limitations of physical science have been outlined but as our own work shows this is not intended to disparage scientific endeavour. It is only as these limitations are recognised that scientific endeavour and knowledge can take their proper place with design-skill and power. Recent developments in modelling techniques improve access at all levels of design process.

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RASTI DETERMINATION OF SPEECH INTELLIGIBILITY OF PUBLIC ADDRESS SYSTEMS AT AIR AND RAIL TERMINALS

Warren C. Middleton
Lecturer in Physics
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and

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Measurements were made of the speech transmission index according to the rapid speech transmission index (RASTI) method at Brisbane air and rail terminals in order to assess and compare the speech intelligibility properties of the public address systems in use. The equipment used was the standard Brüel and Kjaer Speech Transmission Meter, Type 3361, consisting of separate transmitter and receiver instruments.

A preliminary study of the speech transmission meter was performed within the Anechoic Chamber at the University of Queensland and, subsequent to the overall study, a brief appraisal was made of the RASTI method of assessing public address systems.

It was found that RASTI measures varied predictability with speaker distribution, observer location, background noise and reverberation. Average RASTI values varied between 0.46 and 0.55 for the locations investigated at the two railway stations but were closer to 0.70 in the case of the Brisbane Domestic Airport Terminal which was the more recently constructed and the less reverberant.

The major restriction in the method is that the equipment gives meaningful results only under very stable conditions of background noise.

1.0 INTRODUCTION

There is obvious need in air, rail and coach terminals for announcements to be both audible and intelligible to those to whom such announcements are directed. The common expediency is to use a public address system (PA). Such systems vary widely in their overall capability of communicating effectively the announcer's verbal message to the ears of the commuters and visitors in the passenger terminal. It is clearly of great value to have a method of measuring objectively and rapidly the speech intelligibility properties of the acoustic channel between the announcer and the intended listener.

The rapid speech transmission index (RASTI) method of assessing speech intelligibility is such a method. It is based on the measurement of the reduction in signal modulation between the speaker and listener positions which, in this case, are linked by a PA system.

This paper outlines a study, using the RASTI method, which was made into the assessment and comparison of the speech intelligibility properties of the PA systems at two types of terminals and into the suitability of the RASTI system for obtaining such assessments and comparisons.

The investigation was conducted in Brisbane during early Spring 1989 at Brisbane Airport and at Central and Roma Street Railway Stations, Brisbane.

2.0 METHOD

2.1 The RASTI Method of Rating Speech Intelligibility

The RASTI method of rating speech intelligibility (Houtgast and Steeneken, 1984 & 1985) is an abridgement of the method of determining the speech transmission index (STI). It is fully documented in the IEC Publication 286-16, First Edition 1988, on Sound System Equipment, Part 16: "The Objective Rating of Speech Intelligibility in Auditoria by the 'RASTI' Method."

The STI method is based on the definition of the speech transmission index which is a function of the modulation transfer function (MTF) (Houtgast and Steeneken, 1973, 1980, 1983 and 1985) whereby "the speech transmission path between an enclosure is characterised by the modulation transfer function m(F), quantifying the degree of preservation of the original intensity modulations as a function of modulation frequency. Typically, the carrier is octave-band-filtered noise."

The STI method is based on the determination of the function m(F) for 98 data points, obtained for 14 modulation frequencies at 1/3 octave intervals ranging from 0.63 Hz to 12.5 Hz and for seven octave bands with centre frequencies ranging from 125 Hz to 8 kHz. The RASTI method is restricted to nine data points, obtained from four modulation frequencies at the octave band centred at 500 Hz and five modulation frequencies centred at 2 kHz.

The RASTI method is applicable to the evaluation of a PA system (Steeneken and Houtgast, 1985) and to the overall assessment of the speech intelligibility properties of a speech communication channel which includes a PA system. This latter application is the subject of this paper.

2.2 Components of a Speech Communication Channel which includes a PA System

The general case where transmitting and receiving spaces are separate is depicted in Figure 1.

The announcer, microphone and PA console are in one "room" and the loudspeaker and listener are in another "room". The room(s) may each be enclosed spaces or open or semi-open spaces. In the limiting situation, the transmitter and receiver spaces merge into the same space with announcer, microphone and PA console together with loudspeaker(s) and listener.

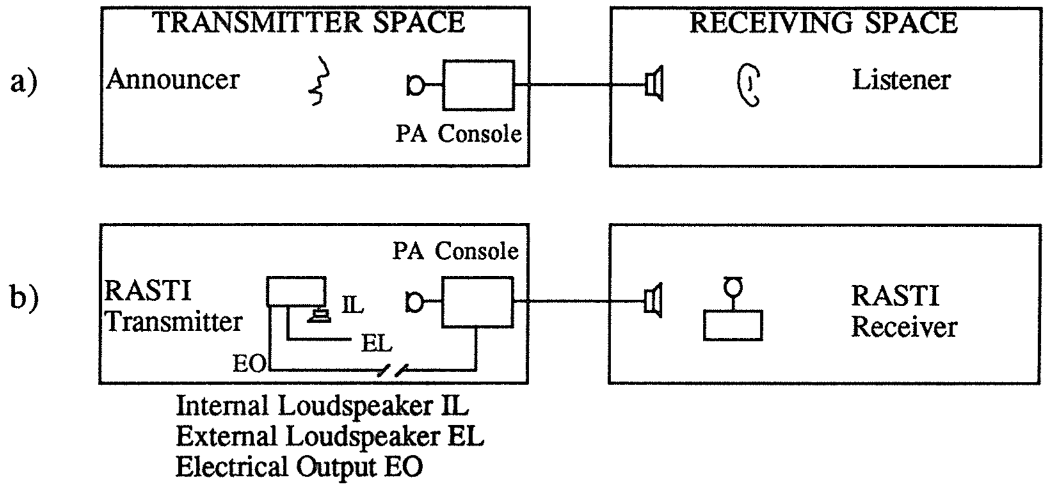


Figure 1: Speech Communication System Diagrammatic Only
(a) In Normal use (b) For RASTI Tests

2.3 Factors Affecting the Speech Intelligibility of the Overall Communication Channel

(a) **Response of the Electrodynamic System:** This depends on the steady state and transient frequency responses of microphone, amplifier system and loudspeaker.

(b) **Acoustic Coupling to the Electrodynamic System:** There is the coupling between the announcer and the microphone in the transmitting space and the coupling between the loudspeaker(s) and the air in the receiving space.

(c) **Acoustical Properties of the Transmitting and Receiving Rooms Per Se:** These depend on:

   (i) the nature and geometry of the reflecting surfaces of each and the way in which these surfaces contribute to the sound distribution and reverberation time, and

   (ii) the sources of background noise which exist within each room, eg. air-conditioner or computer equipment noise in the transmitting room or commuter noise in the receiving room.

(d) **The degree of acoustical isolation:** There may be complete or only partial acoustical isolation between the rooms and either or both rooms may be subject to noise intrusion from the passenger carrier itself be it aeroplane, train or whatever.

2.4 Test Options Provided by the RASTI Method

The following test options are provided by the Brüel and Kjaer Speech Transmission Meter Type 3361 consisting of separate transmitter and receiver instruments.

(a) **Speaker Option:** There is provision for using either the internal speaker or using the external speaker output from the transmitter to drive either an external loudspeaker or a head and torso simulator with a built-in loudspeaker.

(b) **Electrical Output Option:** This is provided for direct connection of the RASTI signal to the input stage of a sound amplifying system thus, excluding the acoustic characteristics of the microphone and transmitting room from the test.

(c) **Measurement Period Option:** Values of 8, 16 or 32 seconds may be selected to cater for different background conditions at the receiver microphone location.

(d) **Output Level Option:** The "Ref" output level corresponds to the reference level stated in IEC 268-16 (1988). There is the option of "Ref. + 10 dB" which is useful if simulating louder than average voices or investigating the influence of background noise on the RASTI value.

(e) **Noise Floor Option:** There is provision for entering a noise floor into the receiver in order to simulate the effects of local background noise.

There is no option for allowing for difference in quality of speech enunciation. This, however, raises other considerations beyond the present scope.

2.9 Scope of the Investigation

After a preliminary study of the performance of the RASTI equipment in an anechoic chamber at the Department of Mechanical Engineering, University of Queensland, studies were made of the public address system at Brisbane Airport for two different modes of operation corresponding respectively to the Ansett Airlines and the Australian Airlines usages and for both acoustic and electrical coupling to the sound amplification system, and of the public address systems at Roma Street and Central Railway Stations for the particular loudspeaker types and configurations in use and for both acoustic and electrical coupling.

The airport study was made during the daytime in the area of the terminal previously set aside for independent airline operation. The ambient noise was suitably low because there was little aircraft and commuter mobility at the time due to the particular industrial dispute then existing. The railway investigation of necessity was conducted overnight during the hours when no trains were scheduled.

Procedures were in accordance with IEC 268-16 (1988) and the equipment manufacturer's instructions.

Subsequent to the individual studies, a comparison was made between the two terminals and an appraisal was made of the RASTI method.

3.0 RESULTS

3.1 Performance of the RASTI Equipment in an Anechoic Chamber

Tests of the RASTI equipment using an acoustic coupling distance of 1 m between the internal speaker of the transmitter and the microphone of the receiver gave STI (500 Hz), STI (2 kHz) and RASTI mean values as shown in Table I.

TABLE I. STI and RASTI Values for Anechoic Chamber

Sampling Time s	STI (500 Hz)	STI (2kHz)	RASTI
8	0.93	0.98	0.96
16	0.97	0.98	0.98
32	0.99	1.00	0.99

STI and RASTI values approaching unity are to be expected in a space with little reflection and little background noise. Values of standard deviations for replicate readings were in accordance with the manufacturer's specifications (Brüel and Kjaer, 1986).

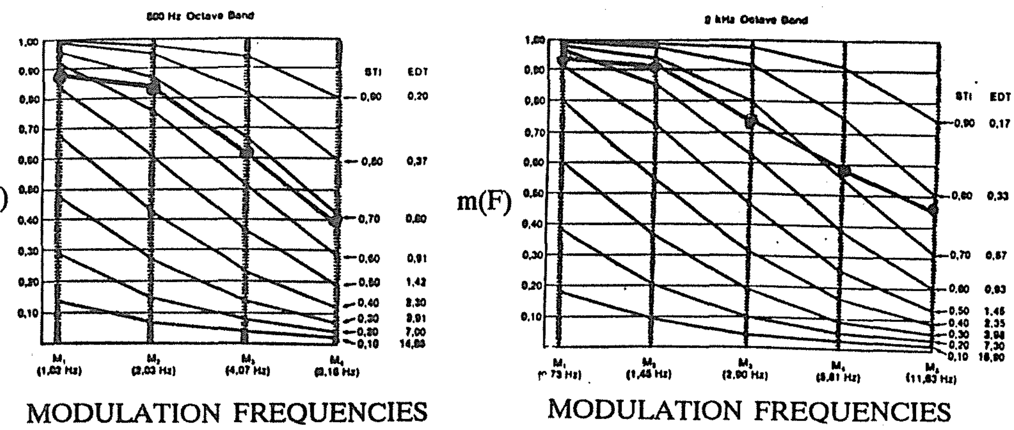
3.2 Brisbane Domestic Airport

The PA system for the terminal building is computer controlled and there are options, within each airline space, for central or local transmission of messages and for different configurations of loudspeakers to be activated for particular message broadcasts.

Australian Airlines (TN) transmit their messages from the relatively open local departure desk near the particular gate and use an appropriate loudspeaker configuration. Ansett Airlines (AN), on the other hand, transmit from a central control room which is enclosed and use loudspeaker configurations which may extend over greater distances. Different microphone types are used in the two systems.

For typical listener positions in the lounge near a gate, an average RASTI value of 0.63 ± 0.02 was obtained for the Tn mode and 0.70 ± 0.02 for the AN mode both values rating "GOOD" on the subjective intelligibility scale. In each case, the m(F) values follow fairly closely the theoretical values for pure exponential reverberation decay as shown in Figure 2 and the STI and RASTI indexes have magnitudes appropriate to the particular early decay time (EDT) values.

(a)



(b)

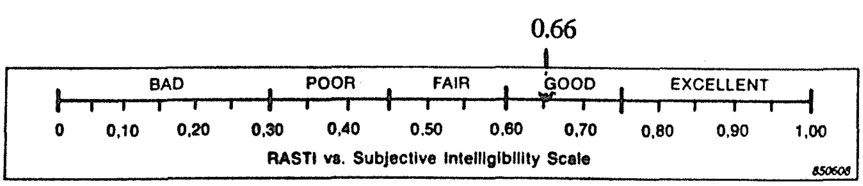


Figure 2 (a) Measured m(F) Values Superimposed on Theoretical MTF Curves
(b) RASTI as Subjective Intelligibility Scale

This trend for the m(F) values is to be expected as the tests were performed in the absence of any significant background noise. If the sound absorption of the receiving space were to be increased, eg., by replacing the glass walls with acoustically softer walls, then the RASTI values would be enhanced.

The similarity in RASTI values for the TN and AN mode is interesting in view of the different conditions alluded to earlier. One further difference is the fact that the RASTI transmitter was located at distances from the PA microphone of 0.8 m in the TN mode and 0.2 m in the AN mode. However, the results of a later test show that the RASTI value remains approximately constant to within ± 0.01 of 0.69 for distances from 0.8 m to 0.2 m, but then become invalid.

Tests using electrical rather than acoustical coupling between the RASTI transmitter and the PA system yielded an average RASTI value of 0.66 ± 0.02 which is comparable with the other two values for similar conditions in the TN and AN modes. Further, the MTF curves showed nothing extraordinary and were similar to those obtained with acoustic coupling. It was concluded that the PA microphone and transmitting spaces, both TN and AN, had negligible effect on the speech intelligibility.

The tests up to this point were done in quiet background conditions. In order to assess the effect of background noise in the listener space during normal airport operation electronically simulated floors were inserted into the receiver. The tests were performed using electrical coupling for convenience and with the receiver in the same listener position in the lounge (as before). The results are shown in Figure 3.

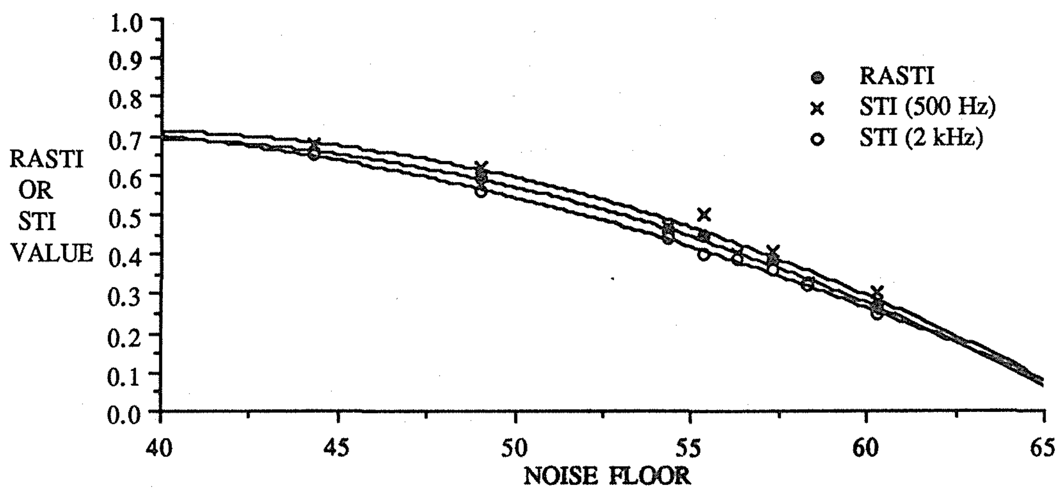


Figure 3. Variation of STI and RASTI Values with Noise Floor

RASTI values are seen to decrease from 0.66 to 0.27 bringing the subjective intelligibility rating down from "GOOD" to "BAD". The MTF curves become flatter than before in keeping with decreased signal to noise S/N ratio. In a test where the simulated floor had reduced the RASTI value to 0.35 corresponding to a rating of "POOR" increasing the output level of the RASTI transmitter by +10 dB raised the RASTI value to the "GOOD" 0.67 again.

Later tests involved different receiver locations. In the toilets the average RASTI value was 0.56 ± 0.02 corresponding to "FAIR" and the MTF graphs show the curves follow closely the theoretical curves for pure exponential reverberation decay due to the highly reflective surfaces and low background. Measurements in the canteen and bistro areas were low averaging 0.26 ± 0.02 corresponding to "BAD". The MTF curves indicate low S/N ratio values averaging at about 7.0 dB and resulting from a relative lack of speakers. Additionally, this location has the potential for higher background levels.

3.3 Roma Street and Central Railway Stations

Roma Street is of a semi-open style of construction. It was surveyed in order to compare the two types of PA systems present within the station complex, one with a point distribution configuration of suspended loudspeakers and the other with a spaced system of loudspeakers mounted flush with the roof. One was used for suburban commuters and the other for interstate travellers. Test results were obtained for acoustic coupling only with an without floor insertion. STI values on one platform complex ranged from 0.55 to 0.65 and on the other complex from 0.43 to 0.49. The addition of noise floors caused expected reduction in RASTI values. m(F) values with and without floors were in agreement with predictions.

Central Railway Station is of a cavern construction and composed of highly reflective surfaces. The PA system has a compressor system which reduces the frequency bandwidth of the system as the compression ratio is increased from 2.5 to 20 with the intention of influencing the quality of the transmitted message. Measurements were taken on platforms No. 3 and No. 4 with acoustical and electrical coupling and a study was made with noise floor insertion and at "Ref." and "Ref. + 10 dB" RASTI transmitter levels. Under normal measurement conditions, RASTI values between 0.45 and 0.48 were obtained with an increase to 0.63 of when the "Ref. + 10 dB" level was used. Insertion of background levels in the floor produced predictable reductions in the RASTI values.

The results of a study using electrical coupling and different PA system compressor ratios are shown in Figure 4.

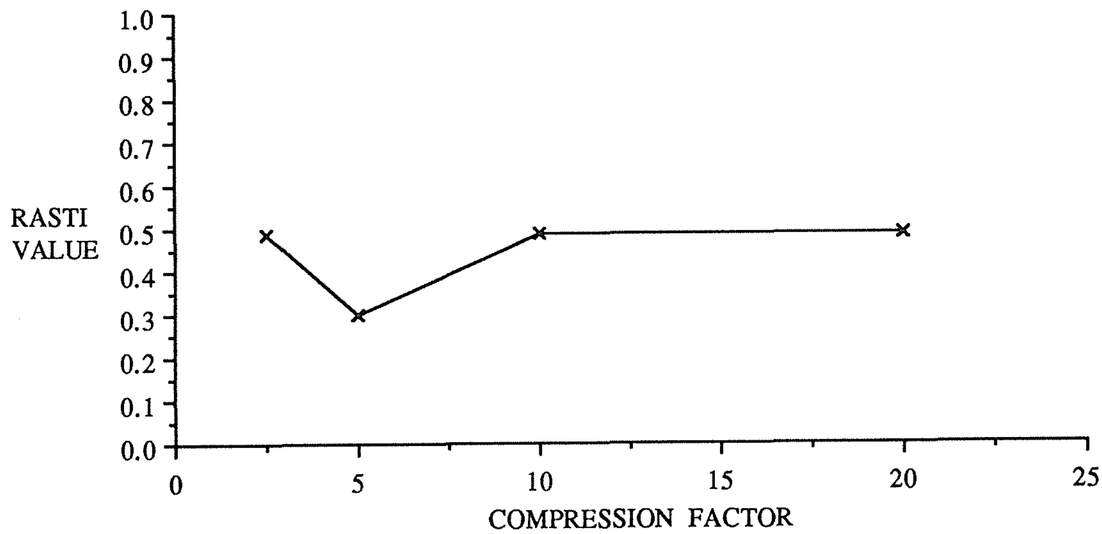


Figure 4. Variation of RASTI value with compression factor.

The compression ratio of 10 is normally used for message transmission, but the ratios 2.5, 10 and 20 give approximately the same RASTI values with a significant reduction at ratio of 5.

4.0 CONCLUSION

In both Airport and Railway terminal locations, RASTI values were found to vary significantly and generally predictably with the various test factors.

The Airport terminal has the advantage that its carriers do not traverse the corridors of its passenger lounges. Consequently, they are not only intrinsically quieter than railway stations, but can also be made less reverberant by the extensive use of carpets and furnishings.

In both cases, RASTI test results confirmed the advantages inherent in having some feedback system to monitor the background noise level in the listening space and controlling the gain in the PA system. Neither the airport nor the railway stations had such a system and the RASTI data could be of value to their management in negotiating budgetary provision for such an addition.

Distance to microphone tests revealed the need for appropriate distancing of announcers from the particular PA microphone in use. This raises the question of training of announcers in use of microphones and speech production. In transmitting spaces where the background level is high consideration should be given to the use of electrical coupling by using pre-recorded messages.

5.0 APPRAISAL OF THE RASTI METHOD FOR ASSESSMENT OF PA SYSTEMS

The RASTI method is suitable for the assessment of the speech intelligibility of PA systems and communication systems which include a PA system. Equipment such as that used in the present investigation is versatile permitting a variety of separate experiments.

The fact that steady-state background conditions are necessary for data acquisition is not a disadvantage. One of the great values of the RASTI system is that it permits a controlled investigation into the effect of raising background levels. When these levels are appropriate to those normally encountered in practice, such as would be obtained from octave-band analysis of real background noise, the RASTI data obtained would be most relevant.

This study has not shown, however, whether or not the abridgement of the STI 98 data points to the RASTI 9 data points is acceptable.

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CLASSROOM NOISE IN ABORIGINAL SCHOOLS

by

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March 1990

ABSTRACT

Results are presented of internal classroom noise measurements carried out in the Northern Territory, particularly at the community school at Yuendumu. This school is typical of schools that serve the local aboriginal communities such as the Warlpri. Buildings are of relatively simple design and construction with the main aim being functionality. It is common practice to mount large air-conditioning units in the windows to provide relief from the summer heat. Unfortunately the resulting noise levels produced within the rooms is excessive and leads to difficulties in classroom communications. This problem is further aggravated by the fact that the pupils are learning in an environment where English is a second language and many have hearing losses ranging from mild to severe.

As a result of measurements and recommendations supplied by NAL all classrooms at Yuendumu are now air-conditioned by remotely mounted units and the air fed by insulated ducts into the classrooms. Marked improvements in internal noise levels have been obtained, in one case the levels have been reduced by 17 dBA.

INTRODUCTION

This project was originally commenced in June 1988 after visits to the Yuendumu community school by a Teacher of the Deaf from National Acoustic Laboratories, Geoff Plant. Mr Plant was in the process of developing an adaptive speech test in Warlpri to be used in assessing the effectiveness of FM hearing-aid fitting. FM hearing-aids are used by children during classroom activities. The test is also to be used as a means of screening childrens' hearing within the school.

Many children in Aboriginal communities throughout Australia suffer mild-to-moderate hearing losses as a result of chronic otitis media. This results in classroom problems which are accentuated by the use of a second language (English) as the major means of instruction. Hence the need for FM hearing aids. Coupled with these two problems is excessive noise in classrooms generated from sources such as air-conditioners. Altogether there exists a very difficult educational environment.

The work at Yuendumu, and subsequent studies at other communities in the Northern Territory, was prompted by the communities wishing to improve the facilities and consequently the standard of educational opportunities available to their children. This work was supported by NAL, the Department of Education, Northern Territory and the Menzies School of Health Research, Darwin.

The three communities mentioned in this paper are Yuendumu, a Warlpri community situated approximately 300 kms north west of Alice Springs; the Yuelamu community at Mount Allan, 275 kms north west of Alice Springs; and Santa Tesesa, an Arrente community, approximately 100 kms south east of Alice Springs.

In any school classroom there are several factors that may contribute to the ability of the pupils to fully understand the teachers speech. The main physical factors are,

i) the intensity or loudness of the teachers voice when it reaches the pupil,
ii) the amount of background noise in the room, and
iii) the reverberation time of the room.

In a study of speech levels in noisy environments, including classrooms, Pearsons, Bennett and Fidell (1977) found that the speech level in a noisy classroom environment was at an average of 73 dBA at one meter. This is regarded as being loud. The level dropped off to 66 dBA at two meters.

This study also showed that the teachers tested tried to maintain their voice level at about 15 dBA above the background noise level. For every one dBA increase in background noise the teachers' voice level rose by one dBA, however, this did not necessarily increase or maintain the speech intelligibility as heard by the students. To sustain a voice level of 73 dBA at one meter for any length of time is very tiring and teachers could not be expected to perform optimally at this level.

All of these factors can be controlled, some more easily than others. For example the reverberation time of the room is primarily a function of room design but it is also adjustable by the use of furniture layout, wall hangings, floor covering etc. Teachers somehow seem to be able to 'intuitively' adjust the reverberation time of a room so that it is acceptable to them, at least to a reasonable degree. However, the control or adjustment of room noise, other than that made by the pupils, is usually beyond their control.

Elliot (1982) in a separate piece of research gave some results that indicated "that environments for young children...need to be designed so that even when the ambient, background noise is very low, the speech signal reaching the young child is more intense than the level that adults require to understand the message." Specifically he found that for three year old children to score 100 percent in a word recognition test, words had to be presented at levels more than 25 dBA higher than that required by adults. This differential decreases with age and by ten years reduces to 10 dBA.

Australian Standard AS 2107-1987 recommends a level of 40 to 45 dBA maximum in classrooms such as those being considered.

METHOD

Noise levels were measured on site using a precision, intergrating sound level meter and from calibrated tape recordings analysed in the laboratory after the event. L_{eq}'s were taken using varying sample times to suit the conditions and type of event being measured. Reverberation times were also calculated back in the laboratory using the decay time of pink noise in the room, generated and recorded on site.

During the period between the initial visit in June 1988 and August 1989 when an opportunity arose to revisit Yuendumu and repeat the measurements, improvements were made to the air-conditionion systems that supplied the Year I and Year II classrooms. A total of $15,466 was spent on installing externally mounted air-conditioning units with the cool air ducted into the rooms via insulated ducts. These replacement units, in fact, service two rooms each so now the situation is such that where a single, window mounted unit serviced one room, now one external unit services two rooms.

RESULTS

Yuendumu

As can be seen from Tables 1 and 2 the noise levels that were measured in the classrooms at Yuendumu in 1988 under the conditions of fully operating air-conditioning units were well over the AS 2107-1987 recommended levels of 40 to 45 dBA maximum. The levels of 77.4 dBA in the Year I classroom and 71.7 dBA in the year II classroom, at distances of two meters and over, lie in the "slightly difficult" to "difficult" range on the Preferred Speech Interference Level (PSIL) for Face-to-Face Communication (1970). These sort of levels necessitate that a 'very loud' to 'shouting' voice be used and are very unsatisfactory for classroom teaching situations.

The tables (1 and 2) show that the remeasured values decreased considerably from those originally obtained. In the case of the Year I room the levels dropped by 17.0 dBA from 77.4 to 60.4 dBA. In the Year II room the level dropped by 5.3 dBA from 71.7 to 66.4 dBA.

While these levels have not been reduced to the levels recommended in AS 2107-1987 of 40 to 45 dBA they never the less represent a marked improvement and are certainly much more realistic conditions under which to try and teach.

Santa Teresa

At Santa Teresa there is quite a large, well maintained school that takes students up to post-primary. The buildings and school rooms have been constructed over many years and consequently range from those that have had air-conditioning added as an after thought to those that have been designed with specific air-conditioning systems in mind. Table 3 gives the results of measurements taken in several classrooms. The measurements taken in the Post-primary room gave the best results with a reading of 58.1 dBA for an empty room with the air-conditioning operating on full power. This is a modern stand alone building with a well designed air-conditioning system. On the same site other classrooms were measured where the air-conditioning was added more as an after thought, either in the windows, as in the Grade I/II room, or on the roof, as in the Transition room. Noise levels in the Grade I/II room were 70.0 dBA and in the Transition room 69.8 dBA. These levels are approximately 12 dBA up on the Post-primary room.

Mount Allan

At Mount Allan the school consists of two large metal caravans that have been immobilised and left on site. One of these caravans is used for formal teaching and two air-conditioning units have been mounted in the windows. Table 4 gives the measurements taken on site. The level in the classroom with both air-conditioners operating on full power was 74.8 dBA. This is a very unsatisfactory level and can only really be changed by extensive modifications.

General

The effects from the reduction of noise levels at Yuendumu school were very pleasing for all concerned, especially the teachers. The Year I room which was previously unusable when the children were particularly noisy is now a satisfactory room in which to work. The Northern Territory Education Department has, since August 1989, spent further funds in improving the air-conditioning systems at the school so that now all rooms, except two, are air-conditioned by indirect rather than direct means. The two rooms that are still directly air-conditioned do not require particularly low noise levels for satisfactory use.

Further improvements to room acoustics are slowly being made by the respective room teachers as funds and materials become available. Now that teachers have some appreciation of acoustic principles such techniques as covering hard surfaces with absorbent materials can be easily implimented. The most dramatic improvements tending to be made with relatively simple techniques in the first instance.

CONCLUSIONS

Noise in the classroom is an on going problem that has been and still is the subject of much study. This noise is very disruptive of classroom activities and in many cases can make teaching very difficult. The simple act of placing an air-conditioning unit in a room may be an excellent solution to a heat problem but almost invariably it will lead to a noise problem. By the application of simple and low cost design principles classroom noise can be minimised to acceptable levels. Potential noise problems must be considered before the construction of new facilities or the refurbishment of old facilities is undertaken. Only by forward planning can interior noise climates be improved.

TABLE 1

YUENDUMU

Year I Classroom
Noise Measurements

Measurement conditions	Weighting	L_{eq} Readings 1988	1989	
vacant room*	linear	69.0 (15)	57.3 (28)	dB
	A	60.3 (24)	45.0 (31)	dBA
class in progress	linear	74.4 (61)	68.8 (61)	dB
	A	71.9 (56)	64.8 (27)	dBA
empty room air-conditioning on full power	linear	88.3 (30)	74.4 (27)	dB
	A	77.4 (22)	60.4 (14)	dBA
Reverberation time		0.7	0.4	seconds

(AS 2107-1987 recommended time 0.4 to 0.5 seconds)

Note: * The general background noise levels in 1989 were lower than the 1988 measurements as there were fewer children in the playgrounds due to "sorry business" in the community. (This is cultural activity associated with tribal mourning activities.)
- Figure in brackets indicates sample time in seconds.

TABLE 2

YUENDUMU

Year II Classroom
Noise Measurements

Measurement conditions	Weighting	L_{eq} Readings 1988	1989
class in progress radio in background	linear	-	71.0 (40) dB
	A	-	66.2 (54) dBA
class talking (talking/shouting)	linear	-	65.2 (48) dB
	A	-	66.6 (51) dBA
class interacting with teacher	linear	69.5 (32)	69.1 (33) dB
	A	63.8 (37)	67.3 (34) dBA
class interacting with teacher with air-conditioner on full power	linear	74.7 (25)	66.9 (22) dB
	A	71.7 (25)	66.4 (30) dBA

Note: - Figure in brackets indicates sample time in seconds.

TABLE 3

SANTA TERESA

Comparison of Noise Levels in Classrooms
with Various Air-conditioning Units

Classroom	Weighting	L_{eq} Readings
Post-primary	linear	79.0 dB (26)
	A	58.1 dBA (24)
Grade I/II	linear	83.4 dB (21)
	A	70.0 dBA (19)
Transition	linear	91.0 dB (25)
	A	69.8 dBA (25)

Note: - Figures in brackets indicates sample time in seconds.
- All of the above measurements were made with the respective air-conditioning units operating on full power.

TABLE 4

MOUNT ALLAN

A one teaching-room school

Measurement conditions	Weighting	L_{eq} Readings
class in progress (no a/c)	linear	78.0 dB (27)
	A	69.7 dBA (36)
class in progress with air-conditioners on full power	linear	82.1 dB (28)
	A	74.8 dBA (15)

reverberation time 0.3 seconds (average)
(AS 2107-1987 recommended time 0.4 to 0.5 seconds)

Note: - Figures in brackets indicate sample time in seconds.

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ABSTRACT

THE DESIGN, TESTING AND USE OF A REVERBERATION CHAMBER

Two adjoining reverberation chambers were constructed on the premises of the Division of Environment several years ago. This paper discusses their design and tests carried out to determine their characteristics. Some discussion follows on the use made of the chambers by the Division and other bodies.

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THE DESIGN, TESTING AND USE OF A REVERBERATION CHAMBER

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INTRODUCTION

The establishment of an acoustical test facility attracts some attention in acoustical circles because it is usually an isolated event due to cost and space restrictions. However, during the last decade facilities have been established by the Australian Defence Academy (Canberra), Brüel & Kjaer Australia (Sydney), National Acoustical Laboratories (Sydney), the University of Queensland (Brisbane) and the Division of Environment (Brisbane).

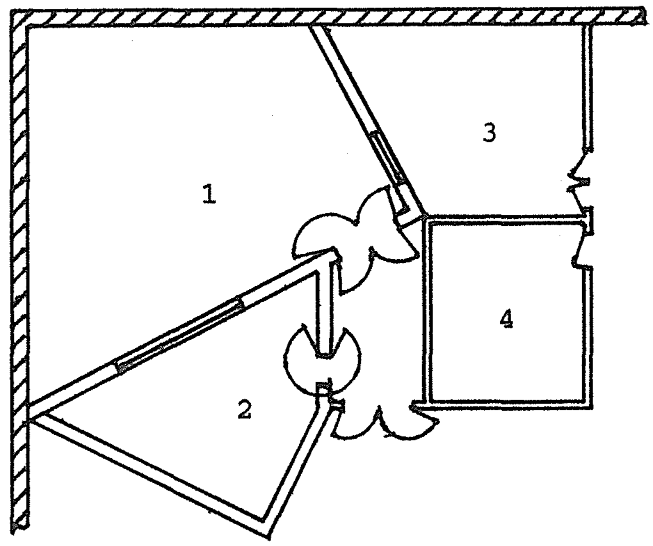
In the early 1980's the Queensland Government decided to construct an acoustical laboratory in the central Brisbane area at Kantara House, home of the then Divisions of Noise Abatement and Air Pollution Control. The work was carried out by the State Works Department which engaged architect John Kershaw and acoustical consultant Ron Rumble for specialised design. Because of space restrictions the laboratory was located on the second floor of the two-storey building, which accordingly presented several design problems.

The laboratory, which was completed in 1982, comprises four separate rooms: an Instrument Room which houses noise measuring equipment and testing instruments, two adjacent Reverberation Chambers and a Control-Calibration Room in which testing in the chambers is controlled. It was intended that the facility be used for acoustical tests such as the determination of Sound Power Level, Sound Transmission Loss and Absorption Coefficients, testing of products and training purposes. Certain of the tests already carried out have been described by Renew (1988).

DESIGN

A location plan of the laboratory is given in Fig. 1. Design criteria for the larger reverberation chamber were:

- a nominal volume of 100 m^3 (to permit a low-frequency cut-off of approximately 125 Hz)
- a capacity for testing panels up to STC50
- an ability to fit into the selected location in the building and not overload it
- adequate isolation from vibrational forces
- an internal noise level not in excess of NR 25.



1. & 2. Reverberation Chamber
3. Control-Calibration Room
4. Instrument Room

Fig. 1. Layout of Acoustical Laboratory

In order to restrict the weight of the reverberation chambers, it was decided to build them from lightweight panels of double wall design.

Design details:

walls - 300 mm thick, double-wall construction consisting of plasterboard panels fixed to a steel channel framework. Two 50 mm layers of fibreglass batts were located inside the panels, as well as a 10 mm sheet of Jointex insulation. Three 13 mm plasterboard panels comprised the exterior wall while the interior wall consisted of two 16 mm panels. Joints were staggered to minimise sound leakage paths.

A glazed opening 1200 mm wide and 900 mm high was provided in the wall between the larger reverberation chamber and the Control-Calibration Room. It consisted of three panels, of 12 mm, 10 mm and 5 mm glass. A test aperture 3000 mm wide and 2400 mm high was provided in the wall between the reverberation chambers.

vibration isolation - it is normal practice to build reverberation chambers on separate, isolated slabs, with masonry walls and ceilings constructed directly on the slabs. However, because of the limitations imposed by locating the chambers in commercial premises (e.g. ceiling height and structural loading limits), the conventional approach could not be used in this instance. It was necessary therefore to fabricate lightweight walls and position them on the existing floor slab. The walls were isolated from the slab with neoprene waffle pads and anti-vibration mounts were installed between adjacent wall members.

ceilings - a similar double-wall construction was used in a 100 mm thick ceiling which was supported from overhead steel beams. Both the walls and the ceiling were given two coats of acrylic paint.

doors - a two-door system was installed, with an outer door of 45 mm solid core timber and an inner door of the same material with a 2.5 mm fibreglas outer lining to limit internal sound reflection. Raven head, jamb and threshold seals were fitted to both doorways. As is shown in Fig. 1, a double door was installed in the larger chamber.

air conditioning - each of the four rooms was equipped with a ducted air-conditioning system with an internal temperature control. The minimum fresh air quantity for each room was 1.5 air charges per hour. Ducts were lined with fibreglass insulation and a silencer was fitted to the air outlet in each reverberation chamber.

room size -

	Volume (m^3)	Surface Area (m^2)
large chamber	131	169
small chamber	55	95

LABORATORY EQUIPMENT

The laboratory is well equipped for carrying out acoustical testing and analysis. Values of sound pressure levels were originally measured in the larger reverberation chamber by means of an array of six microphones mounted on tripods and connected to a Brüel & Kjaer Type 2811 8 Channel Multiplexer, but recent tests made use of a Brüel & Kjaer Type 3923 Rotating Microphone Boom fitted with a Brüel & Kjaer Type 4166 Microphone. A Brüel & Kjaer Type 2307 Level Recorder was formerly used in conjunction with a Brüel & Kjaer Type 4205 Sound Power Source to determine reverberation times. However, a Brüel & Kjaer Type 4418 Building Acoustics

Analyser is now employed for this purpose together with a Brüel & Kjaer Type 2312 Alphanumeric Printer.

When sound power level is being evaluated by the comparison method, a Brüel & Kjaer Type 4204 Reference Sound Source is used. Other equipment includes a Brüel & Kjaer Type 2131 Digital Frequency Analyser, an EDA sound source, an EDA amplifier and two EDA loudspeakers. Calibration is carried out with either a Brüel & Kjaer Type 4220 Pistonphone, a Brüel & Kjaer Type 4230 Sound Level Calibrator or a Quest Model CA-15B Sound Calibrator.

ROOM TESTS

A series of tests was carried out to determine the larger reverberation chamber's characteristics and to assess its suitability for acoustical measurements. The results are given below.

. Noise reduction of walls

The EDA equipment was employed to generate sufficient acoustic power to permit the measurement of the noise reduction provided by various sections of the reverberation chamber walls. Values of noise reduction obtained are:

outer wall of smaller chamber:	55	NIC
wall with viewing window:	58	NIC
wall with double door:	50	NIC

These values may be compared with the following design criteria:

wall between chambers:	STC 65/70
walls between chambers and surrounds:	STC 55

. Reverberation time

Several tests have been carried out to determine the reverberation times of the room at the standard 1/3 octave band centre frequencies. Originally, values of reverberation time were obtained by use of an array of six microphones connected to a multiplexer. The results of subsequent tests in which the rotating microphone boom was used in conjunction with the building acoustics analyser (for a decay of 30 dB in sound pressure level) are shown in Fig. 2. The two sets of values compare quite well except at frequencies below 315 Hz. A maximum standard deviation of 0.4 dB was calculated, for a centre frequency of 160 Hz. Tests were also carried out with decays of 20 and 40 dB and again the measured values of reverberation time corresponded quite well at centre frequencies above 315 Hz. Typical values are shown in Table 1.

Method	Reverberation Time (s)						
	125	250	500	1k	2k	4k	8k
Array of six microphones	2.4	3.6	2.6	2.7	2.9	2.3	1.2
RMB* - 20 dB Decay	2.1	3.2	2.5	2.7	2.8	2.4	1.2
RMB - 30 dB Decay	2.1	3.4	2.4	2.6	2.9	2.3	1.3
RMB - 40 dB Decay	2.3	3.6	2.3	2.6	2.8	2.3	1.2

* RMB = Rotating Microphone Boom

Table 1. Reverberation Time Values Obtained by Various Methods.

Reverberation times at frequencies below 100 Hz were evaluated by use of a Scott Type 811-B Random Noise Generator. Values were found to lie between 1.1 and 1.3 seconds for frequencies from 20 Hz to 80 Hz.

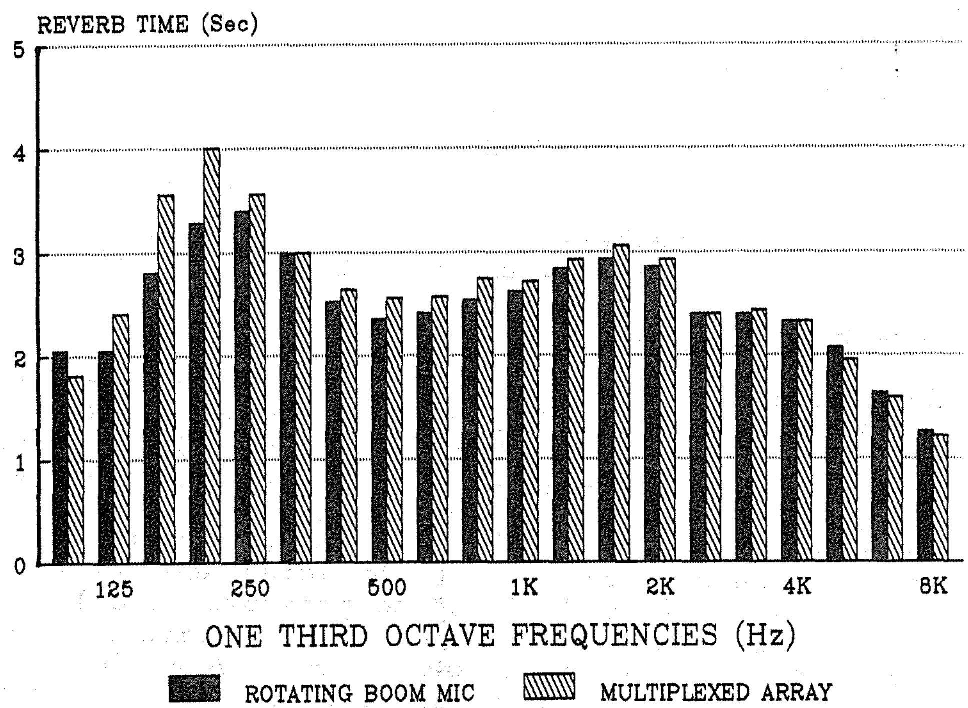


Fig. 2. Reverberation Times for Reverberation Chamber

. Spatial variation in sound pressure level

An investigation was carried out by Hooker (1983) to determine the spatial variation in sound pressure level in the large reverberation chamber. Readings of sound pressure level were taken on the digital frequency analyser for a combination of five positions of the sound source and six microphone locations. The variation was very small, particularly in the upper frequency range, and values of standard deviation did not exceed the maximum allowable values specified in Australian Standard AS1217.2 (1985).

. Room absorption coefficient

The mean absorption coefficient ($\bar{\alpha}$) of the large reverberation chamber was calculated by use of the Sabine relationship,

$$\bar{\alpha} = 0.161\ V/ST$$

where V = room volume (m^3)
S = room surface area (m^2)
T = reverberation time (s)

The value of $\bar{\alpha}$ ranged from 0.06 at a frequency of 100 Hz to a maximum value of 0.10 at 8kHz. It is noted that the provision of AS1217.2 regarding room sound absorption is met in the frequency range 100 Hz to 5 kHz.

. Room background sound pressure level

The background sound pressure level in the large reverberation chamber was evaluated during a series of tests by use of a precision sound level meter. It was determined that sound in the chamber did not exceed the NR 45 criterion when the building air-conditioning system was in operation nor the NR 25 criterion when the air-conditioning unit was switched off. Switching on the chamber's air-conditioning unit raised the sound pressure level to the NR 45 criterion.

. Sound power level of reference sound source

The sound power level of the Brüel & Kjaer Type 4204 Reference Sound Source has been measured on several occasions by Divisional officers. A summary of the results is given in Table 2. The first test involved use of an array of six Brüel & Kjaer Type 4165 Microphones and values of sound power level were determined by the direct method using values of reverberation time obtained from the building acoustics analyser. In subsequent tests the rotating microphone boom was used to evaluate reverberation times corresponding to decays of 20, 30 and 40 dB in sound pressure level. Determinations of sound power level were carried out at 1/3 octave band frequencies from 100 Hz to 8 kHz.

It is apparent from Table 2 and Fig. 3 that agreement is quite close between the maker's calibrated value of sound power level (determined in an anechoic chamber) and the values obtained in a reverberation chamber. The discrepancy at low frequencies that has been commonly observed (ref. Maling (1985)) between reverberant field and free field sound power level values is evident in the tabulated data. For frequencies between 100 and 315 Hz the sound power level of the source measured in the reverberation chamber is less than that measured in the free field.

Method	Sound Power Level (dB)						
	125	**250**	**500**	**1k**	**2k**	**4k**	**8k**
Anechoic Chamber	73.9	75.3	75.7	79.1	79.7	77.3	74.0
Reverberation Chamber - 20 dB Decay	71.1	74.4	75.5	79.6	80.3	78.5	75.1
Reverberation Chamber - 30 dB Decay	70.9	74.1	75.5	79.1	80.2	78.5	75.0
Reverberation Chamber - 40 dB Decay	70.6	74.1	75.2	79.0	79.9	78.2	74.8
Reverberation Chamber - Array	71.6	74.5	75.8	79.5	80.0	77.4	74.1

Table 2. Sound Power Level of a Reference Sound Source

The variation in values of the sound power level obtained during the tests was considerably lower than that indicated in AS1217.2. Values of maximum standard deviation are given in Table 3 for the 30 dB decay tests.

1/3 Octave Band Frequency (Hz)	**Standard Deviation (dB)**
100 - 160	0.8
200 - 315	0.6
400 - 5k	0.3
6.3k - 8k	0.2

Table 3. Standard Deviations for Sound Power Level Tests

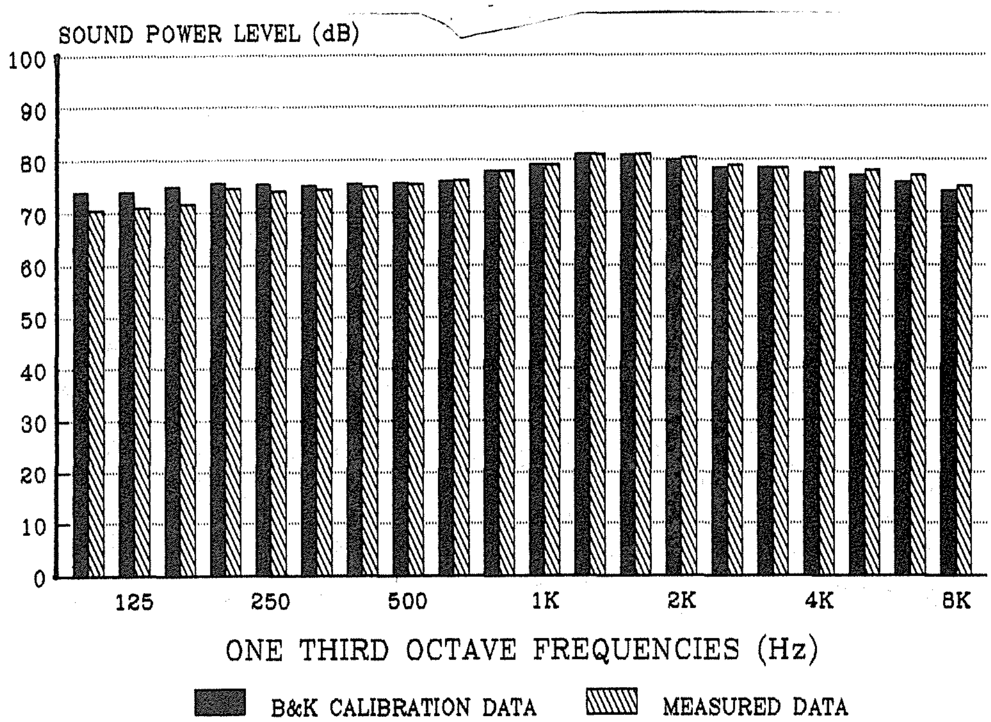


Fig. 3. Sound Power Level of a Reference Sound Source

. Sound power level of air-conditioners

In 1983 the then Division of Noise Abatement participated in a round robin series of tests to determine the sound power levels of two domestic air-conditioners. Both the direct and comparison methods were employed by the Division in these tests which were organised by the Australian Environment Council. Use was made of an array of six microphones and the building acoustics analyser. All results lay well within the 95% confidence limits contained in the report published by the Australian Environment Council (1984).

. Transmission Loss Tests

According to Australian Standard AS1191 (1985), the volume of each reverberation chamber should preferably be at least 100 m^3 in order to provide sufficient diffusion for transmission loss tests. With a volume of 55 m^3 the small reverberation chamber is suitable for tests at and above a recommended 1/3 octave-band frequency of 160 Hz. An investigation was carried out to determine the degree of diffusion in this room. Microphones were set up in eleven locations opposite the test wall and measurements of sound pressure level were taken with the sound source in three positions and for two microphone elevations. The tests revealed a high level of diffusivity in the room.

A series of tests was carried out in the reverberation chambers by Rumble (1987) to determine the low-frequency insertion loss provided by twelve panels of different construction in connection with a noise problem affecting staff in a power station office. The tests were carried out to AS1191 (1985), using both broad-band and pure tone excitation. The results were highly repeatable and enabled an economical panel design to be selected.

DISCUSSION

Examination of AS1217.2 indicates that the larger reverberation chamber qualifies for the determination of sound power level in the one-third octave band frequency range, 160 Hz to 5kHz. Judging from the results of tests to measure the sound power level of a reference sound source, it is likely that the chamber would produce sufficiently accurate results over the range 125 Hz to 8kHz. This likelihood is reinforced by the results of the air conditioner tests where values of sound power level determined by the Division generally lay near the centre of the 95% confidence limits obtained in the round robin series.

An analysis was carried out of the chamber's performance based on the modal analysis described by Bies (1976). The results indicated a need to add absorption to the room for frequencies up to 160 Hz. Installation of a rotating diffuser as suggested by Baade (1987) would be expected to improve the room's performance and this option will be explored in the near future.

The tests carried out by Rumble (1988) indicate that the reverberation chambers are suitable for the measurement of transmission loss, at least over the range 160 Hz to 8kHz. To date the larger chamber has not been used for determining absorption coefficients. Because of the relatively low reverberation times this use of the chamber has not been explored.

CONCLUSION

It has been shown that the Division of Environment's acoustical laboratory is an extremely useful facility for determining sound power level and transmission loss. This usefulness is somewhat limited by the necessarily restricted size of the reverberation chambers. The potential addition of diffusing elements is expected to improve the larger reverbertion chamber's performance, particularly at lower frequencies.

ACKNOWLEDGEMENTS

The authors wish to express their gratitude to staff of the Division of Environment for assistance in carrying out tests in the acoustical laboratory.

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SOUND TRANSMISSION LOSS OF FLUTED AND CORRUGATED PANELS - DAMPED AND UNDAMPED

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ABSTRACT

Standard prediction schemes for isotropic flat panels generally overestimate the sound transmission loss of commercial corrugated and fluted building panels. The work described here investigates the usefulness of three prediction procedures for orthotropic panels by comparing predicted results (based on the panel physical properties) with laboratory measurements for six types of commercially available corrugated or fluted, steel or aluminium building panels. It is demonstrated that simple approximate expressions derived from the analysis of an infinite orthotropic panel generally provide adequate results for a finite panel.

INTRODUCTION

In the construction of industrial buildings, corrugated or fluted (stiffened) panels are often used as they have a high strength to weight ratio, thus being less expensive and lighter than flat panels of the same strength. Unfortunately stiffened panels suffer from a poor sound transmission loss characteristic, being much worse than flat panels of the same weight. Corrugated or fluted panels also have much greater stiffness in one direction than the other which is why they are often called orthotropic panels, and why they are characterized by two critical frequencies rather than a single critical frequency which characterises an isotropic panel.

As corrugated and fluted panels are often used for buildings with high interior noise levels, there is considerable interest in being able to predict their sound transmission loss (TL) from a knowledge of only their geometrical and material properties.

The analysis of sound transmission through finite orthotropic panels has not been covered in the literature; thus the purpose of the work here is twofold. Firstly it is intended to evaluate the usefulness of Statistical Energy Analysis (SEA) for the prediction of the TL of finite orthotropic panels. Secondly, the applicability of both exact and approximate prediction schemes for infinite orthotropic panels will be evaluated by comparison of theoretical predictions with measurements for a number of commercially available corrugated and fluted panels.

REVIEW OF TL PREDICTION SCHEMES FOR SINGLE PANELS

Isotropic Panels

The prediction of the sound transmission loss for both thin and thick isotropic single and double panels has been treated extensively in the literature[1,2,3]. It has been shown that transmission loss data measured using reverberant rooms are usually about 3 dB higher than values calculated by assuming the panel to be infinite in extent[1]. Better agreement has been achieved by taking into account the finite size of the panels[1,2]. Assuming that the bandwidth of the incident sound is one third of an octave and taking into account the finite size of the panel, the transmission loss for frequencies below half the critical frequency and above twice the first resonance frequency for an isotropic panel subjected to randomly incident random noise is given by:[1]

$$TL = TL_N - 5 \quad (dB) \tag{1}$$

where TL_N is the normal incidence transmission loss for an infinite panel and is given by:

$$TL_N = 10\log_{10}\left[1 + \left(\frac{\pi f m}{\rho c}\right)^2\right] \approx 20\log_{10}\left[\frac{\pi f m}{\rho c}\right] \tag{2}$$

where f is the one third octave band centre frequency (Hz)
m is the mass per unit are of the panel (kg/m^2)
ρ is the density of air (1.205 kg/m^3)
c is the speed of sound in air (343 m/sec).

Note that TL in equation (1) is referred to as the field incidence transmission loss, and it is not equal to the theoretical random incidence transmission loss for an infinite panel which is given by:

$$TL_D = TL_N - 10\log_{10}[0.23TL_N] \tag{3}$$

At frequencies equal to and above the critical frequency of an isotropic panel, the transmission loss is given by:[1]

$$TL = 20\log_{10}\left(\frac{\pi f m}{\rho c}\right) + 10\log_{10}\left(\frac{2\eta}{\pi}\frac{f}{f_c}\right) \tag{4}$$

η = panel loss factor
f_c = panel critical frequency which is given by:

$$f_c = \frac{c^2}{2\pi}\sqrt{\frac{m}{B}} \quad \text{(Hz)} \tag{5}$$

B = panel bending stiffness per unit width (kg m^2 s^{-2})

Alternatively, for an isotropic panel the critical frequency f_c may be expressed as:

$$f_c = c^2(1-\nu^2)^{\frac{1}{2}}/1.81\ c_L h \quad \text{(Hz)} \tag{6}$$

ν = Poisson's ratio
c_L = longitudinal wave speed in the panel (m/s)
h = panel thickness (m)

Orthotropic panels

The transmission loss of infinite orthotropic panels has been considered by Heckl[4] who gave the following expression for the transmission coefficient, τ:

$$\tau = \frac{2}{\pi}\int_o^{\pi/2}\int_o^1 \frac{d(\sin^2\theta)d\phi}{\left|1 + \frac{Z}{2\rho c}\cos\theta\right|^2} \tag{7}$$

where the panel input impedance Z is given by:

$$Z = i2\pi fm\left[1 - \left(\frac{f}{f_{c1}}\cos^2\phi + \frac{f}{f_{c2}}\sin^2\phi\right)\sin^4\theta\right] \tag{8}$$

and where f_{c1} and f_{c2} are the two critical frequencies for the panel, calculated using the stiffness (per unit width) B_1 along the panel and the stiffness B_2 across the panel respectively, and equation (5). The transmission loss is related to the transmission coefficient by:

$$TL = -10\log_{10}(\tau)$$

The stiffnesses, B_1 and B_2, per unit width for a corrugated panel may be calculated with reference to Figure 1.

For wave propagation along the ribs, the following equation may be used where the summation is taken over all sections in width ℓ and where the distances z_n from the neutral axis satisfy, $\Sigma\, b_n z_n = 0$:

$$B_1 = \frac{Eh}{(1-\nu^2)\ell}\sum_n b_n\left(z_n^2 + \frac{h^2+b_n^2}{24} + \frac{h^2-b_n^2}{24}\cos 2\theta_n\right) \tag{9}$$

E = Young's Modulus (Pa)
h = Panel thickness (m)

θ_n, z_n, b_n are defined in Figure 1; all distances are in metres.

For wave propagation across the corrugations:

$$B_2 = \frac{Eh^3}{12(1-\nu^2)} \left[\frac{\sum_n b_n}{\ell}\right] \qquad (10)$$

Equation (9) reduces to that for an isotropic flat panel when $\sum_n b_n = \ell$.

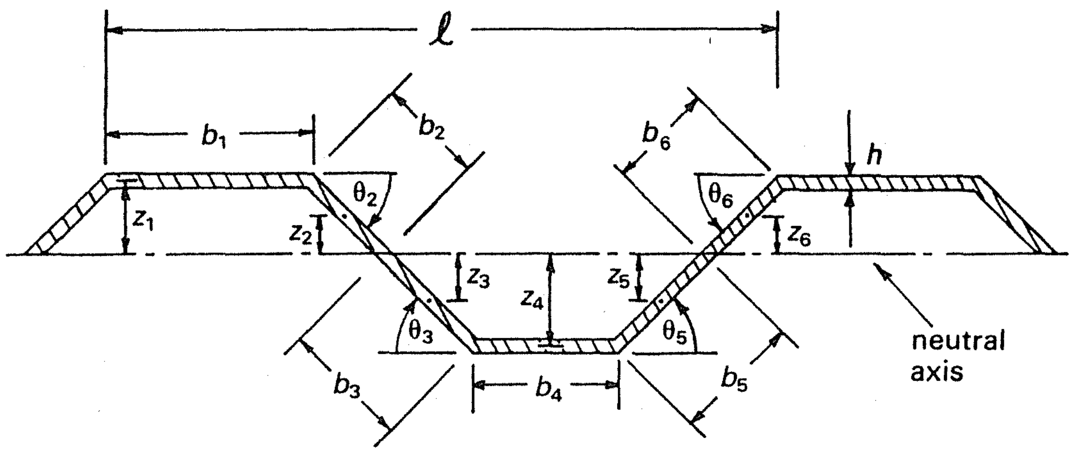

Figure 1. Orthotropic panel profile

Heckl's expression (equation (7)) which applies to an infinite orthotropic panel can be approximated as follows[4]

$$\tau \approx \frac{\rho^2 c^2}{\pi^2 f^2 m^2} \ell n \left(1 + \frac{\pi^2 f^2 m^2}{\rho^2 c^2}\right) \qquad f \le f_{c1}/2$$

$$\tau \approx \frac{\rho c}{2 f m \pi^2} \frac{f_{c1}}{f} \left(\ell n \frac{4f}{f_{c1}}\right)^2 \qquad f_{c1} \le f \le f_{c2} \qquad (11)$$

$$\tau \approx \frac{\rho c}{2fm} \frac{\sqrt{f_{c1} f_{c2}}}{f} \qquad f > f_{c2}$$

For $f_{c1}/2 \le f \le f_{c1}$, a straight line is drawn between the TL calculated for $f = f_{c1}/2$ and that calculated for $f = f_{c1}$ on a plot of

TL vs log frequency.

where f_{c1} is the lower critical frequency of the panel and
f_{c2} is the higher critical frequency.

The prediction of the transmission loss through finite single isotropic stiffened panels using statistical energy analysis has been considered by Crocker and Price[5], using Maidanik's work[6] as a basis. Heckl[7] has also used Maidanik's work as a basis for deriving a simple expression for the transmission loss of a stiffened isotropic panel in the frequency range below the panel critical frequency. Venzke et al[8] provided some measured data for a single panel for the following cases: no stiffeners, longitudinal stiffeners only, and both longitudinal and lateral stiffeners. They compared their data with Heckl's predictions for the cases of no stiffeners and both longitudinal and lateral stiffeners. Each of these latter two cases effectively represent an isotropic panel.

USE OF SEA TO PREDICT THE TL OF ORTHOTROPIC PANELS

The transmission loss of a single isotropic panel derived using SEA is given by:[5]

$$TL = 10\log_{10}\left[\eta_{13} + \eta_{rad}^2 (n_2/n_1)/(\eta_{int} + 2\eta_{rad})\right]$$

$$- 10\log_{10}\left[\eta_3 + (n_1/n_3)\eta_{13} + (n_2/n_1)\ \eta_{rad}\right] + 10\log_{10}\left[\frac{A}{S_3\bar{\alpha}_3}\right] \qquad (12)$$

where it is assumed that the panel is placed between two reverberant rooms of volume V_1 (source room, 105.6 m^3) and volume V_3 (receiver room, 179.7 m^3). The receiver room has a wall surface area S_3 and mean Sabine absorption coefficient of $\bar{\alpha}_3$, and the source room has a wall surface area of S_1.

Equation (12) also holds for an orthotropic panel; however in this case expressions for n_2, the panel modal density, and η_{rad}, the panel radiation loss factor, are different.

The coupling loss factor η_{13} at frequency f(Hz) between the two rooms is given by:

$$10\log_{10}\eta_{13} = -TL_m + 10\log_{10}\left(\frac{Ac}{8\pi V_1 f}\right) \qquad (13)$$

where A is the panel area (1.59 m^2)
c is the speed of sound in air (344 m/s) and
TL_m is the theoretical random incidence mass law (non-resonant) transmission loss of the panel given by:

$$TL_m = 10\log_{10}\left[1+\left(\frac{\pi fm}{\rho c}\right)^2\right] - 10\log_{10}\left[\ell n\left(1+\left(\frac{\pi fm}{\rho c}\right)^2\right)\right] \tag{14}$$

where ρ is the density of air and
m is the mass per unit area of the panel (kg/m^2).

Note that equation (14) is equivalent to equation (3).

The panel radiation loss factor η_{rad} is given by[10]

$$\eta_{rad} = \frac{R_{rad}}{2\pi fmA} \tag{15}$$

where the radiation impedance R_{rad} at frequency f is given by

$$R_{rad} = A\rho c \begin{cases} (c^2/f_{c2}fA)2\ (f/f_{c2})g_1(f/f_{c2})+(Pc/f_{c2}A)g_2(f/f_{c2}), & f<f_{c2} \\ (\ell_1 f_{c2}/c)^{\frac{1}{2}} + (\ell_2 f_{c2}/c)^{\frac{1}{2}}, & f=f_{c2} \\ (1 - f_{c2}/f)^{-\frac{1}{2}} & f>f_{c2} \end{cases} \tag{16}$$

Equations (16) are derived using expressions given by Maidanik[6] corrected according to Crocker and Price[5] and adjusted for an orthotropic panel with critical frequencies f_{c1} and f_{c2}. The quantity P is twice the length of the panel ribs plus the panel perimeter and f_{c2} is the higher critical frequency of the orthotropic panel. ℓ_1 and ℓ_2 are the panel dimensions. As almost all commercially available orthotropic panels used in buildings have their second critical frequency above 10,000 Hz, the second and third of equations (16) are of little practical interest.

The functions $g_1(f/f_{c2})$ and $g_2(f/f_{c2})$ are defined as:

$$g_1(f/f_{c2}) = \begin{cases} (4/\pi^4)\ (1-2\alpha^2)/\alpha(1-\alpha^2)^{\frac{1}{2}} & f < \frac{1}{2}f_{c2} \\ 0 & f > \frac{1}{2}f_{c2} \end{cases} \tag{17}$$

$$g_2(f/f_{c2}) = (2\pi)^{-2}\ \{(1-\alpha^2)\ \ln\ [(1+\alpha)/(1-\alpha)] + 2\alpha\}/(1-\alpha^2)^{3/2} \tag{18}$$

where $\alpha = (f/f_{c2})^{\frac{1}{2}}$ (19)

The modal densities (modes/radian) n_1, n_2 and n_3 of the source room, panel and receiver room respectively are given by:[10]

$$n_1 = \frac{2f^2V_1}{c^3} + \frac{fA_1}{4c^2} \tag{20}$$

$$n_2 = \frac{Am^{\frac{1}{2}}}{8\pi}\left[B_1^{-\frac{1}{2}} + B_2^{-\frac{1}{2}}\right] = \frac{A}{4c^2}\left[f_{c1} + f_{c2}\right] \tag{21}$$

$$n_3 = \frac{2f^2V_3}{c^3} + \frac{fA_3}{4c^2} \tag{22}$$

The panel internal loss factor η_{int} must either be estimated or measured.

EXPERIMENTAL PROCEDURE

Each panel to be tested was placed in an opening between two reverberant rooms as shown on Figure 2. The smaller room had dimensions of 6.085m x 5.175m x 3.355m and the larger room had dimensions of 6.840m x 5.565m x 4.720m. Each room contained a large rotating diffuser which, for all measurements, was set to rotate at 30 rpm to increase the sound field diffusivity. Care was taken to provide an airtight seal around the panel edges and supporting framework to minimize flanking transmission.

Two speakers were placed in two opposite corners in the smaller of the two rooms (the source room) and driven by random "white" noise passed through a one third octave band pass filter. The resulting sound pressure level in each reverberant room was sampled using a microphone traversing continuously diagonally across the room. The microphone signal was averaged to give L_{eq} values using a Bruel & Kjaer 3360 analyser with the averaging time equal to approximately the time taken for two full microphone traverses. In all cases background noise levels (with the speakers off) in both rooms were monitored and in some cases (where the background was within 5 to 20 dB of the signal) the background noise levels were subtracted from the signal levels.

Reverberation times were measured in the receiver room using a single speaker and a stationary microphone for one third octave bands of noise from 100 Hz to 10,000 Hz. Data were averaged over a number of microphone positions as described in reference 3 and measured T_{60} data are given in Table 1. All measurements were in accordance with AS1191[11] and AS1045[12], except that the panel size was 1.5 m x 1.05 m instead of the recommended $10m^2$. This had a small effect on low frequency transmission loss values.

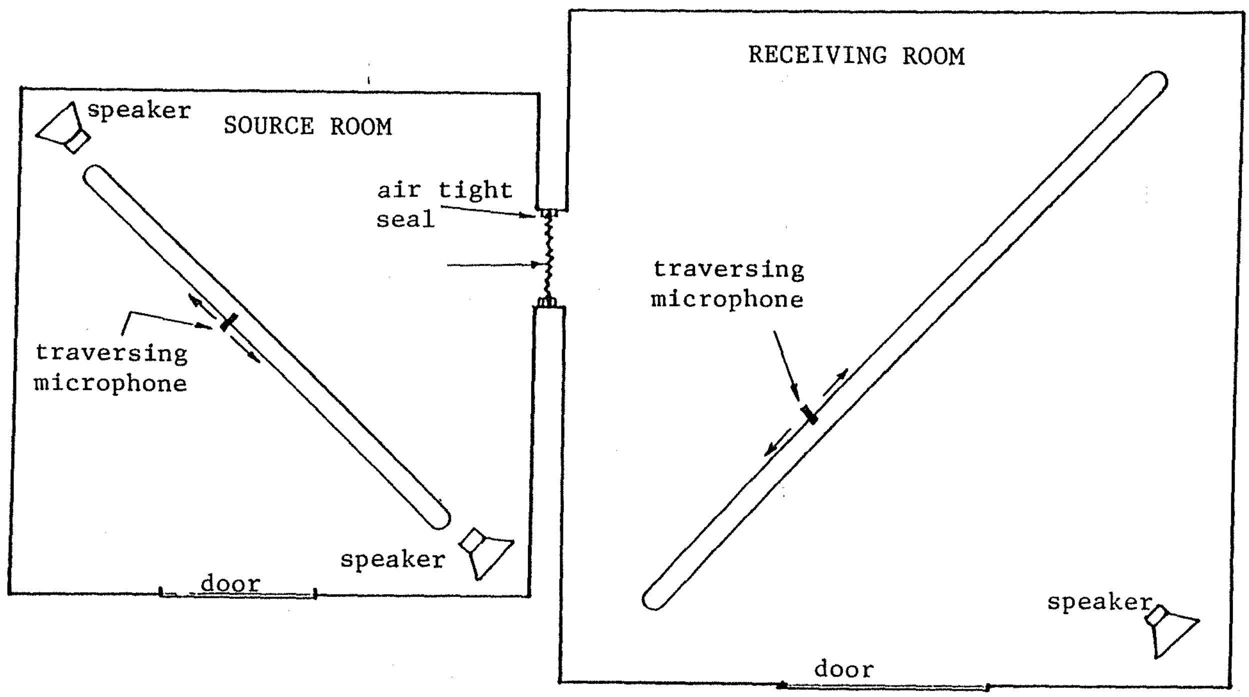

Figure 2. Test arrangement for measuring panel TL.

Table 1. Reverberation times for the receiver room.

one-third octave band centre frequency (Hz)	reverberation time (secs)	one-third octave band centre frequency (Hz)	reverberation time (secs)
50	7.81	800	6.40
63	7.53	1000	6.30
80	6.70	1250	5.45
100	6.58	1600	4.72
125	6.50	2000	4.21
160	6.68	2500	4.05
200	6.02	3150	3.43
250	6.22	4000	2.80
315	5.32	5000	2.17
400	5.41	6300	1.60
500	6.22	8000	1.19
630	6.44	10000	0.79

The sound transmission loss was calculated from the average sound pressure level measured in each room and the receiver room reverberation time (see Table 1) using[11]

$$TL = L_{PS} - L_{PR} + 10\log\left[\frac{AcT_{60}}{55.3V_3}\right] \qquad (23)$$

where L_{PS} = sound pressure level in source room
L_{PR} = sound pressure level in receiver room
A = area of panel ($1.586m^2$)

When placing the test panel between the two reverberant rooms, care was taken to ensure that an airtight seal was maintained to minimise the possibility of flanking transmission. This entailed filling the gaps between the corrugations and the wooden support frame with bitumenized foam strips and butyl mastic filler.

Test Panels

Six orthotropic and one flat panel were tested. The latter panel was used as a check on the validity of the test procedure and results are not presented here. The characteristics of the other six panels which were all tested in both the undamped and damped condition are listed in Tables 2 and 3.

Table 2. Characteristics of orthotropic test panels (undamped)

Panel Trade Name	Surface Density m (kg/m^2)	f_{c1}(Hz)	f_{c2}(Hz)	$f_{1,1}$(Hz)
*Panelrib	3.9	3,520	25,400	5
*Custom blue orb (reg)	5.8	610	19,900	26
*Kliplok	6.2	260	19,900	61
+V-rib (0.7 mm)	2.4	290	17,200	54
*V-crimp	4.2	750	5,400	21
+D-rib	2.1	540	17,200	29

* Steel
\+ Aluminium
$f_{1,1}$ is the lowest resonance frequency of the panel

Panel loss factors were measured by exciting the panel in its frame outside of the reverberant rooms in a semi-anechoic space with one third octave bands of random noise and measuring the decay time. Average values were 0.01 for undamped panels and 0.1 for damped panels. These values were used in the theoretical calculations of TL. The damping compound used was "DECIBAR" which was brushed on, and dried to a hard flexible resin in about 10 hours. Test panel profiles are shown in Figure 3.

Table 3. Characteristics of orthotropic test panels (damped)

Panel Trade Name	Surface Density m (kg/m^2)	f_{c1} (Hz)	f_{c2} (Hz)	$f_{1,1}$ (Hz)
Panelrib	9.1	5,420	39,100	3
Custom blue orb	11.9	920	30,200	17
Kliplok	11.6	350	27,300	44
V-rib	8.8	540	32,500	29
V-crimp	10.7	1,190	40,600	13
D-rib	5.6	880	28,000	18

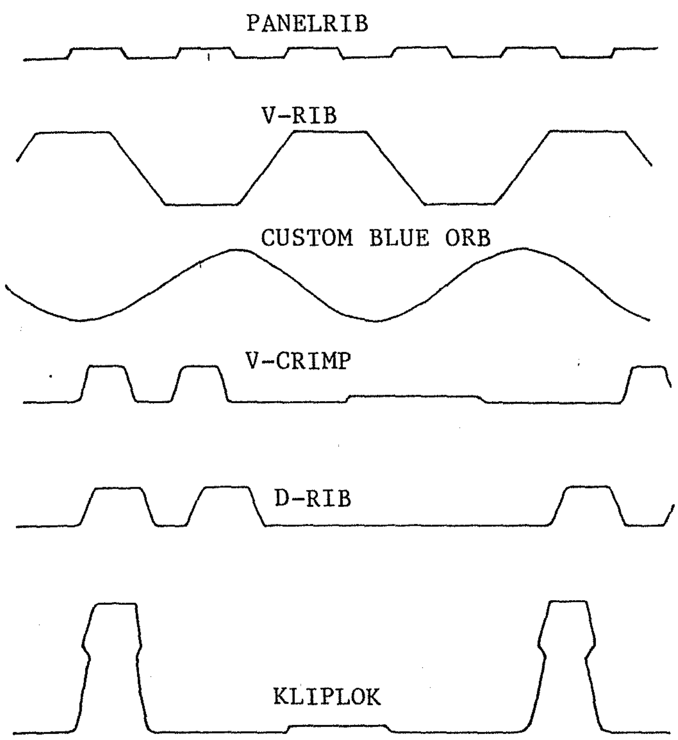

Figure 3. Test panel profiles

DISCUSSION OF RESULTS

Experimental data are plotted for each test panel, together with predictions using three different methods, in Figures 4 to 9. The three theoretical curves correspond respectively to an infinite orthotropic panel (exact integral expression), an infinite orthotropic panel (approximate expression - solid line) and a finite ribbed panel (analysed using Statistical Energy Analysis).

It can be seen from the data that the simple approximate expressions for an infinite orthotropic panel provide remarkably good agreement with the measured data for both damped and undamped finite panels, except for custom blue orb at high frequencies (maximum error 8 dB) and V-rib in the mid-frequency range (maximum error 6 dB).

CONCLUSIONS

It is clear from the data that is presently available that transmission loss prediction procedures for orthotropic panels generally give reasonable results for commercially available corrugated and fluted building panels. However, the smooth calculated curves do not reflect the uneven scatter of the TL data as a function of frequency, although the scatter is roughly about the calculated curves. The results show that the calculations of TL made by assuming the panel to be of infinite size are more useful than calculations made by using Statistical Energy Analysis for a finite size ribbed panel. Furthermore, it appears that the approximate simple expressions for an infinite panel provide better results than those obtained by using the exact expression for an infinite orthotropic panel.

The data also show that attempts to calculate the transmission loss of corrugated or fluted panels by assuming them to be equivalent to isotropic flat panels of the same weight will result in significant overestimations of the panel performance, as stiffened panels generally have a much poorer transmission loss performance than non-stiffened panels.

ACKNOWLEDGEMENTS

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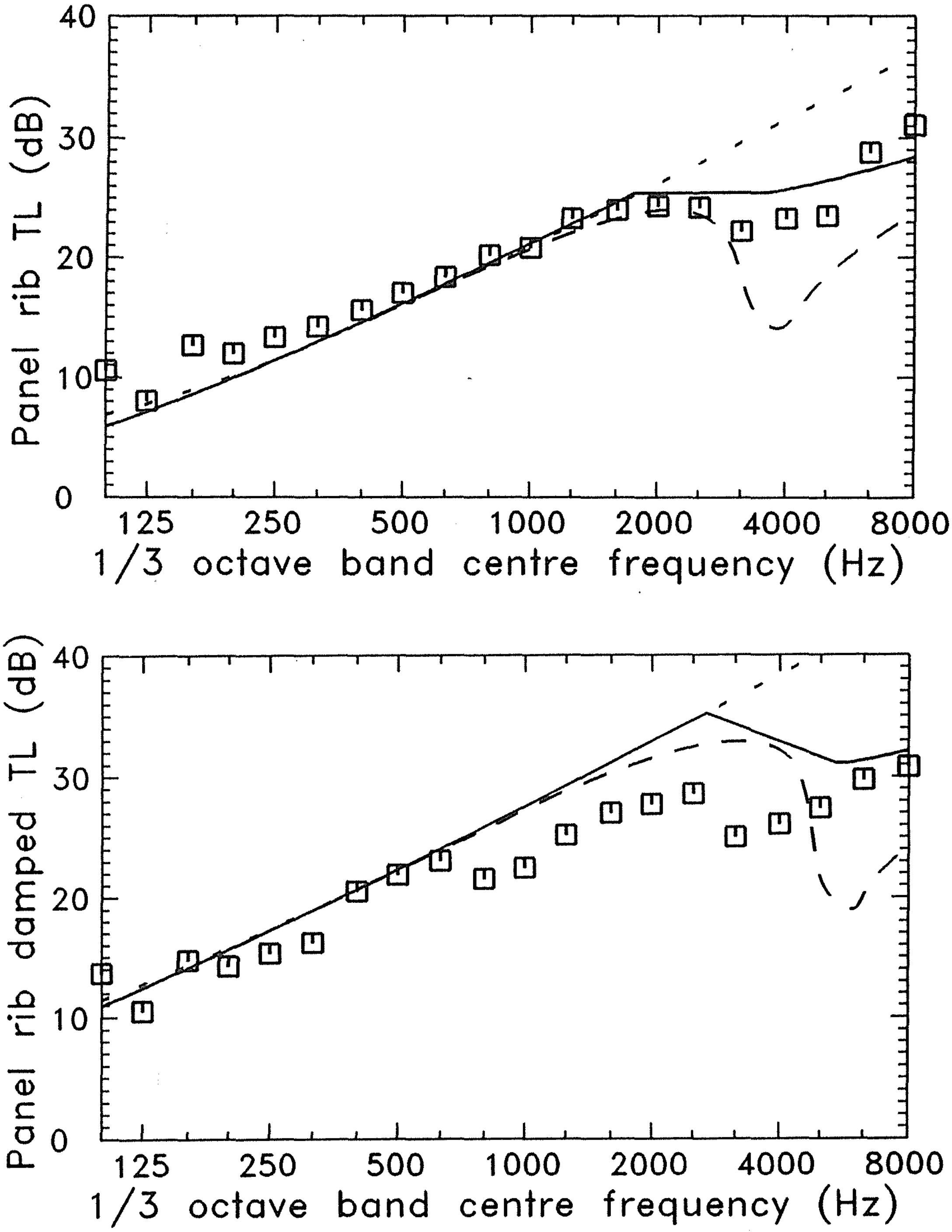
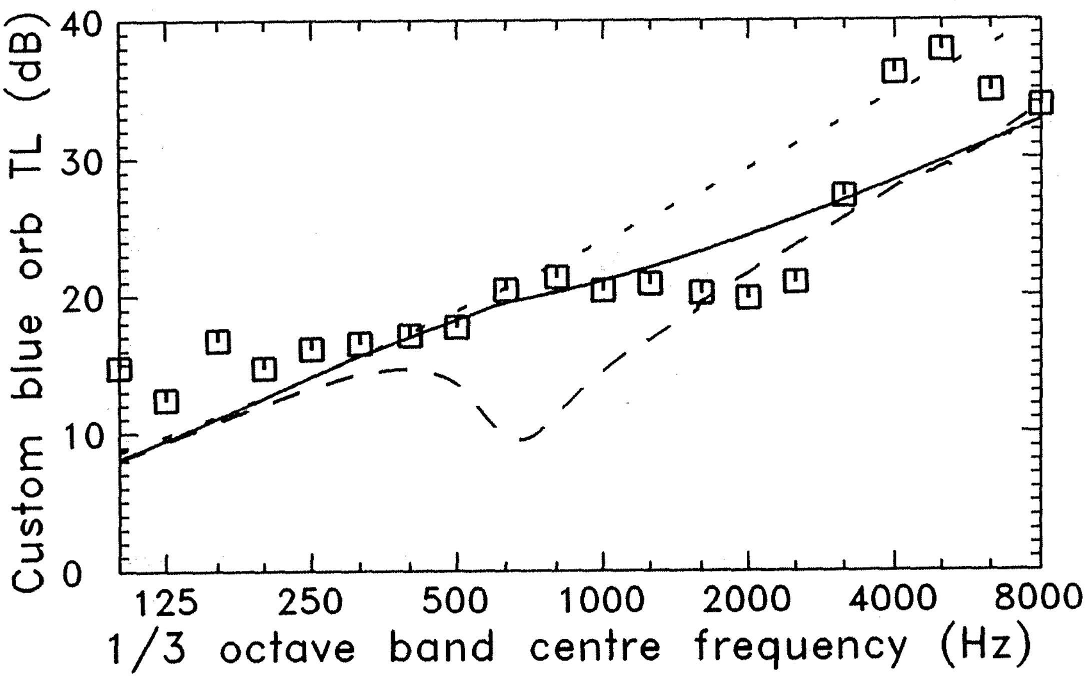


Figure 4. Sound transmission loss for "Panelrib" panel.
(a) Undamped
(b) Damped
——— Approximate theory.
— — Exact theory for an infinite panel.
- - - SEA theory for a ribbed finite panel.



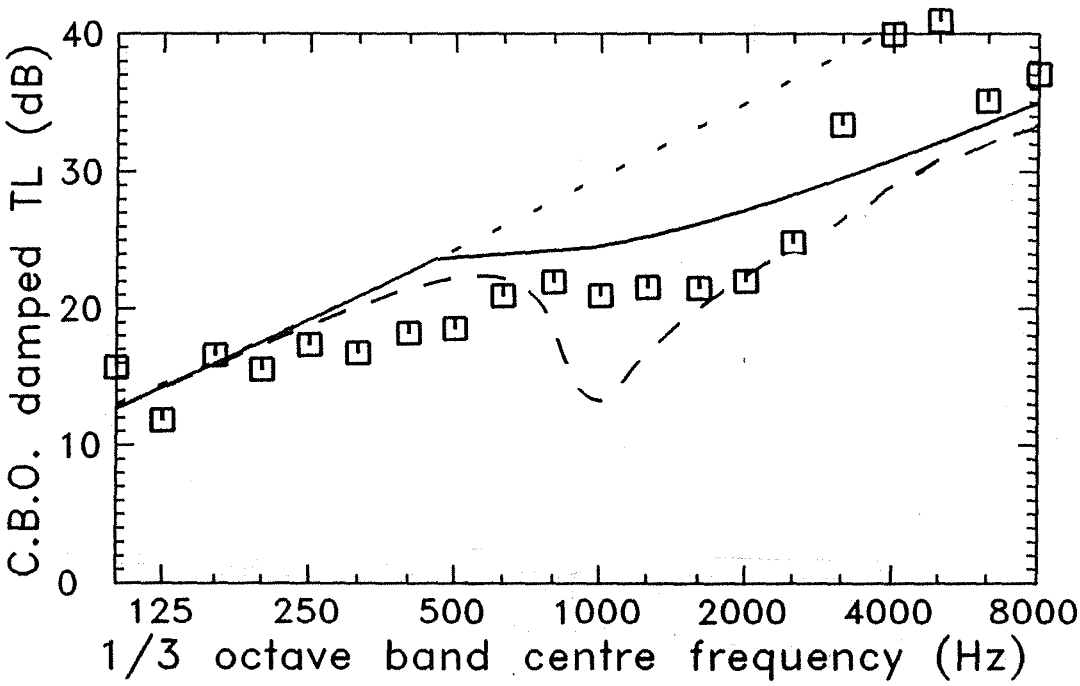
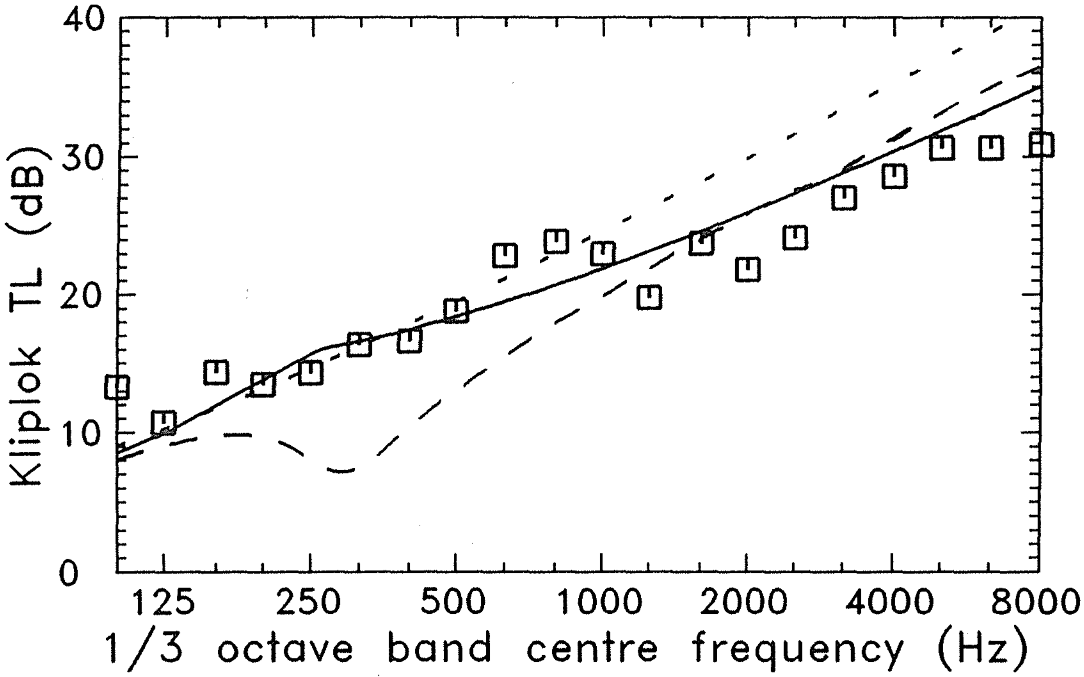


Figure 5. Sound transmission loss for "Custom blue orb" panel.
(a) Undamped
(b) Damped
——— Approximate theory.
— — Exact theory for an infinite panel.
- - - SEA theory for a ribbed finite panel.



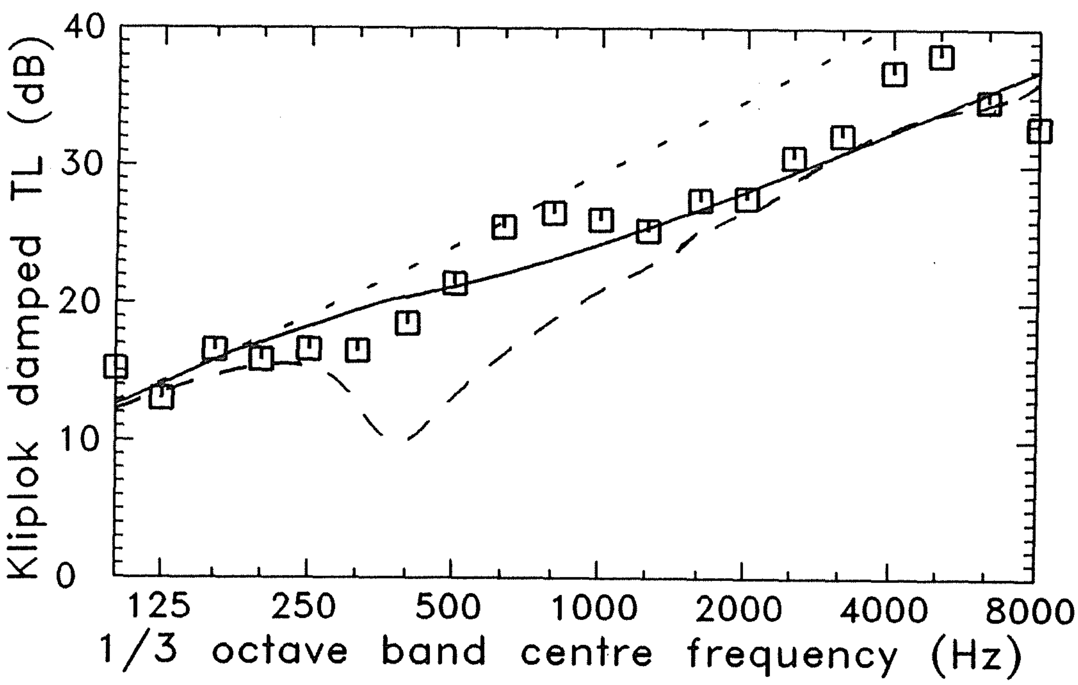


Figure 6. Sound transmission loss for "Kliplok" panel.
(a) Undamped
(b) Damped

————	Approximate theory.
—— ——	Exact theory for an infinite panel.
- - -	SEA theory for a ribbed finite panel.

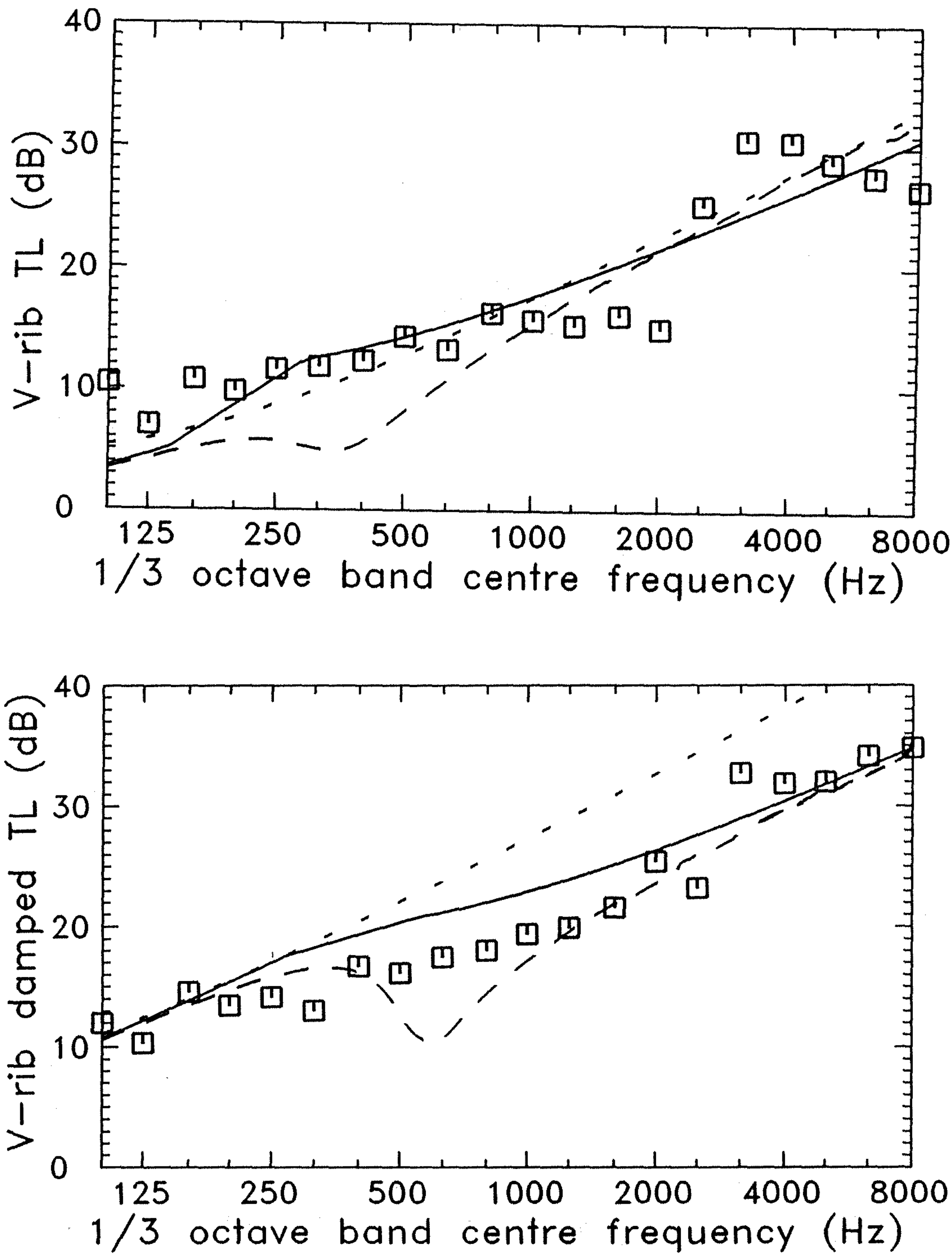


Figure 7. Sound transmission loss for "V - rib" panel.
(a) Undamped
(b) Damped

——— Approximate theory.
— — Exact theory for an infinite panel.
- - - SEA theory for a ribbed finite panel.

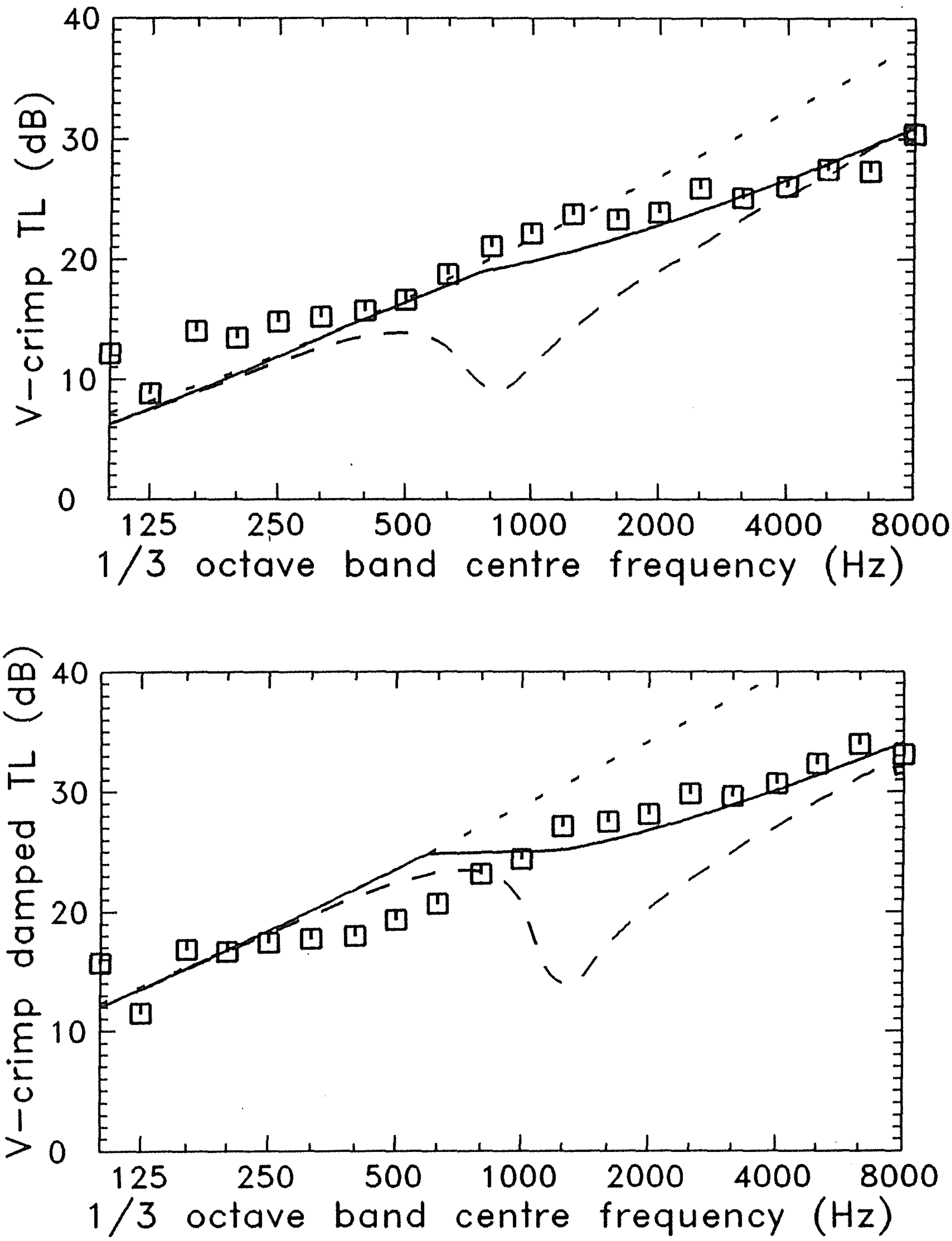
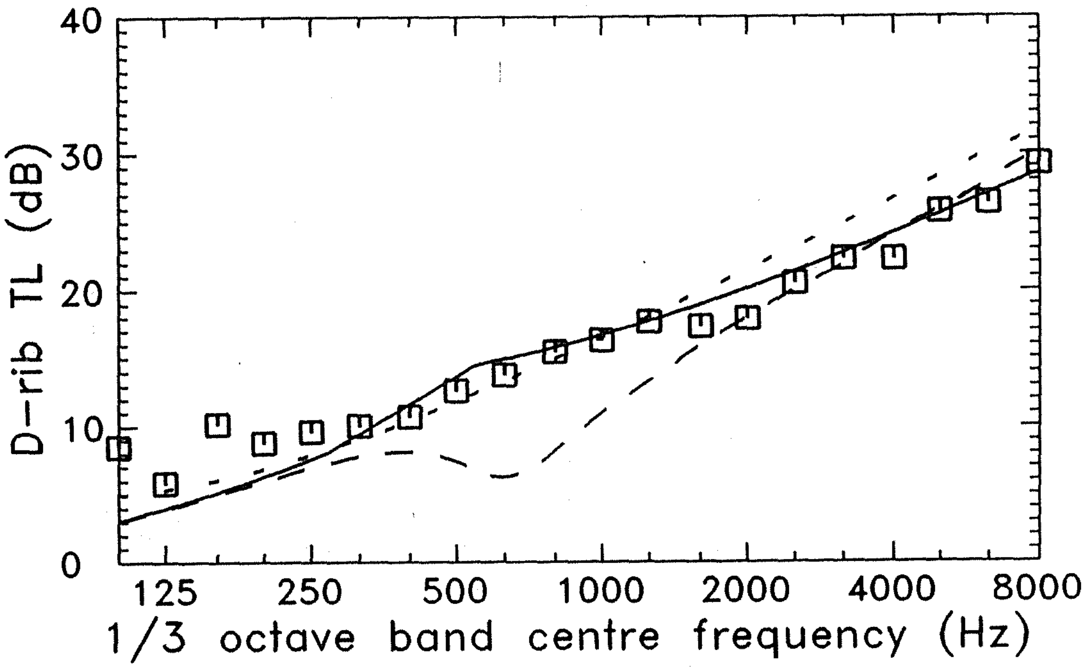


Figure 8. Sound transmission loss for "V - crimp" panel.
(a) Undamped
(b) Damped
——— Approximate theory.
— — Exact theory for an infinite panel.
- - - SEA theory for a ribbed finite panel.



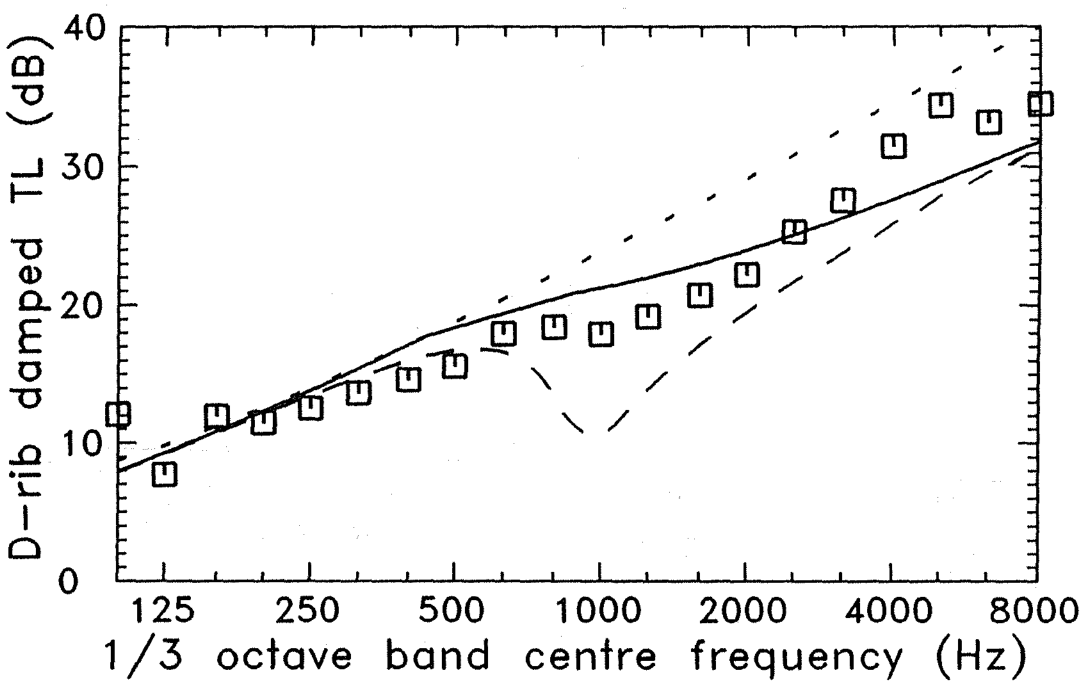


Figure 9. Sound transmission loss for "D - rib" panel.
(a) Undamped
(b) Damped
——— Approximate theory.
— — Exact theory for an infinite panel.
- - - SEA theory for a ribbed finite panel.

AUSTRALIAN ACOUSTICAL SOCIETY

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A CATALOGUE OF ACOUSTIC SHORTCOMINGS IN THE SCHOOL MUSIC DEPARTMENT

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ABSTRACT

Over the past several years, Victorian schools have been engaging acoustical engineers to investigate a range of acoustic problems associated with use of their music tuit--ion and practice rooms, in regard to both adequate sound isolation and provision of an atmosphere conducive to music study.

This paper identifies major acoustic shortcomings in the Music Department, which need to be addressed by architects, builders and clients alike to create an acoustically desirable music practice facility.

The need for vigilance during each phase of the project is essential to ensure an optimum acoustical environment is created, with appropriate selection of materials and methods of construction.

A CATALOGUE OF ACOUSTIC SHORTCOMINGS

IN THE

SCHOOL MUSIC DEPARTMENT

1. INTRODUCTION.

Music tuition has become, over the past decade, an integral part of the Victorian school curriculum, and most schools are providing either a token acknowledgment of the needs of music teachers and their students, or are in fact embar--king on ambitious projects to establish a facility for the study of music.

In establishing a music facility, the architect has the resp--onsibility to design an aesthetic structure with acoust--ically 'friendly' practice areas, and hopefully, adequate sound isolation. However with modifications in design, cost restraints during construction, incorporation of additional features, and for a variety of other reasons, acoustical considerations are overlooked until the facility is opened.

In a building recycling exercise, many of the inherent acoustic weaknesses of the original building are hidden in the modification work, but become apparent later.

Most acoustic pitfalls are well known, and the causes are documented in acoustic reference books, but inspite of this general awareness, the acoustic difficulties continue.

Music teachers very quickly discover the acoustic shortcom--ings in their new department, and endeavour to work around them, as there is generally no funding left to 'fix the flaws'.

The purpose of this paper is to identify the major acoustic shortcomings in the hope that the information will pass on to the parties who were responsible for their creation.

2. DISCUSSION.

In carrying out this study on acoustic shortcomings in the school music department, ten Victorian school music depart--ments were studied.

Billanook College, Mooroolbark,

Blackburn High School,

Boronia Heights Secondary College,

Caulfield Secondary College,

Cleeland High School, Dandenong,

Kingswood College, Box Hill,

Luther College, Croydon,

McKinnon High School,

Tintern C.E.G.G.S., Ringwood East,

Victoria College of Music.

For satisfactory music tuition and practice it is essential that the teacher and student are not disturbed by intrusive sounds from other sources such as nearby music tuition or instrumental practice. Similarly, the tuition or practice sessions should not, themselves, intrude on other classes.

In addition, it is desirable to practice in an area which is of an appropriate size, and is 'friendly' in that it responds positively to the type of music being played.

The first line of defence for sound passing from the pract--ice area to the adjacent rooms is the enclosing envelope. This envelope includes the walls, ceiling, roof and floor.

2.1 APPROPRIATE SELECTION OF SOUND ISOLATING ENVELOPES

2.1.1 ***Sound Transmission Loss*** performance data (***Table 1***) is available for a range of wall, ceiling, roof and floor systems, (1) and if appropriately selected, will ensure that the envelope has the potential for a high degree of sound isolation.

Table 1. Sound Transmission Losses of Building Specimens

Specimen	Airborne STL (dB) with Centre Frequencies (hz) 125	250	500	1k	2k	4k	*STC**
1. Single brick wall 110mm thick.	28	37	36	44	53	60	45
2. Double brick wall 220mm thick.	37	42	49	54	63	68	55
3. Precast concrete slab, 102mm thick.	36	39	41	49	51	63	49
4. Two layers 16mm plasterboard both sides of 64mm stud.	26	32	43	50	48	53	46
5. Above with 50mm mineral wool infill.	31	36	50	55	54	58	52
6. Glazing, 6mm & 10mm glass laminated, with thickness 16mm.	28	32	33	30	39	47	36
7. Double glazed, 10mm & 13mm spaced 76mm, thickness 99mm	27	38	42	43	46	53	46
8. Door, sheeted with 5mm hardboard, 1mm lead, 50mm thick,	22	32	37	37	39	45	39
9. Ceiling 50mm thick compressed strawboards faced with 5mm cement sheet & perforated hardboard.	24	27	35	36	38	37	37
10. 1.6mm thick steel. case of ductwork (2)	14	21	27	32	37	43	-
11. Acoustic air vent.	33	39	45	50	55	60	-
12. Prefabricated Musolab music practice facility.	30	42	50	55	59	61	53

* ***Sound Transmission Class (S.T.C.)*** is a single number evaluation of the property of a building element to attenuate sounds carried out in accordance with AS 1276. (3)

2.1.2 *Instrumental tonal & loudness characteristics*

The type of musical instrument being played, with its part--icular ***tonal character***, and ***loudness***, dictates the mag--nitude of sound isolation required for the enclosing room envelope.

It is a known fact, and confirmed in the data presented in (***Table 2***) that the clarinet is a significantly ***quieter*** instrument than a trombone for example, and similarly the flute has less ***low tonal*** components in its range than the percussion instrument. Thus the clarinet and flute require less stringent sound isolation from the practice room envelope.

The differing needs for sound isolation should be given consideration at the planning stage of the music facility. The ***louder*** instruments may be located remote from the ***quieter*** instrument practice areas, with provision of increased sound isolation measures. This may allow some economy in construction costs as a mix of partitioning may be adopted,

TABLE 2. Typical Sound Pressure Levels for Musical Instrument Practice

Measurements taken at 1 metre from musician.

	Peak S.P.L.s (dB) at Centre Frequencies (hz)							
	63	125	250	500	1k	2k	4k	dB(A)Leq
1. Flute	58	66	93	94	79	77	67	86.8
2. Clarinet	58	67	61	74	88	77	60	82.5
3. Bass Clarinet	70	84	96	88	76	62	51	79.8
4. Alto Sax.	50	78	96	92	77	62	52	94.0
5. Baritone Sax.	98	81	92	92	77	69	52	83.1
6. French Horn	55	73	101	103	92	61	48	96.6
7. Trumpet	53	70	88	108	107	93	78	100.3
8. Trombone	68	78	102	109	104	80	71	102.1
9. Timpani	66	87	101	97	88	77	73	90.7
10. Bass Drum	95	103	100	97	68	52	47	87.1

2.2 ACOUSTIC ENVIRONMENT WITHIN MUSIC PRACTICE ROOMS

The size, shape of room, and use of sound absorbing surfaces all influence the acoustic character of a music practice room. There are optimum room conditions for acceptable instrumental rehearsal or individual practice.

These three room criteria determine the Reverberation Time (T) of the music practice room. The R.T. is expressed in seconds, and may be readily calculated using the Sabine formula. (4) The optimum Reverberation Time (T) which is frequency depen--dent should be 0.7 - 0.9 seconds. (5) for a practice room.

Students receiving private tuition, or practicing their inst--rument require and feel secure in a small area which is suitably sized for the comfortable playing of the instrument, is well illuminated, ventilated and imparts a sense of privacy. For ensemble, stage band or orchestra rehearsal, larger rooms or small auditoria are more appropriate.

Incorporation of an acoustic ceiling, carpet, and use of soft furnishings may effectively modify the reverberation time of a music practice room or rehearsal area. To enable fine tuning of the reverberation time, effective use is made of hinged acoustic wall panels with one sound absorbent face and an acoustically reflective facing on the other side.

For optimum music practice conditions, some acoustic response from the practice room is desirable.

2.3 SOUND FLANKING PATHS.

Sound flanking paths are sound paths other than the direct path between source and receiver through the separating element. (6) These set the the limits of acoustic performance of the music room envelope. The sound transmission loss data in ***Table 1*** is determined from sound isolation measurements of test specimens with strictly controlled ***sound flanking paths*** which do not influence the performance.

Sound flanking paths have the capacity to negate a selection of appropriate sound isolating room elements, rendering, in the worst scenario, a room envelope which is effectively transparent to sound.

In the built music facility, these paths will all, to a lesser or greater extent, adversely affect the acoustic performance of the practice room building elements.

All potential ***sound flanking paths*** should be addressed at the design stage, and must be addressed during the building stage of the school music facility.

In the event that the ***sound flanking paths*** are not eliminated before occupancy of the music facility, their presence will usually become apparent to the music instructor, as the sound from nearby music practice passes along the flanking path.

Undertaking a field sound transmission loss evaluation of the music practice room building elements and comparing the results with the expected acoustic performance will establish the magnitude of the sound flanking path. (7)

The major sound flanking paths confronted are the following.

(a) Sound passing through and around the door.

(b) Sound passing through glazed viewing panels in doors.

(c) Sound passing through the window.

(d) Sound passing through air relief louvres.

(e) Sound passing above the ceiling.

(f) Sound passing under the floor.

(g) Sound passing along ventilation ductwork.

(h) Sound passing through holes in the partition.

(i) Sound passing through wall and floor junctions.

(j) Impact sound passing through structure.

In providing optimum acoustic isolation between the music practice room and adjacent rooms. it is necessary to reduce the contribution of each flanking sound path, to match the sound isolation of the room envelope components.

Acoustic integrity of the structure will, at this point be achieved, and no real benefit will result in further elimin--ation of flanking paths, without additional upgrading of the building structure.

2.4 Analysis of Observations of School Music Departments

On visiting the ten schools, close inspection of each music practice room indicated that consideration had been given to the overall acoustic requirements in the initial design. In many cases however, there were common areas where little or no thought had been given during the construction phase, to the acoustic requirements of the music facility.

A list of acoustic considerations which should be incorp--orated in a school music practice facility is catalogued, and in *Table 3* the ten schools are scored according to compliance.

2.4.1 List of relevant acoustic considerations

(a) Appropriate selection of interconnecting partitions.

(b) Appropriate selection of ceiling system.

(c) Appropriate use of music practice areas.

(d) Use of sound rated doors.

(e) Use of sound locks.

(f) Installation of acoustic door seals.

(g) Small door viewing windows.

(h) Double glazed observation or external windows.

(i) Acoustic air relief vents.

(j) Adequately silenced ventilation ductwork.

Where attention has been paid to each specific acoustic consideration in all, some or in no music practice room, compliance is shown in Table 3 thus:

A = All practice rooms comply.

S = Some practice rooms comply.

N = No practice rooms comply.

- = Not applicable.

Table 3. Incorporation of Acoustic Considerations.

SCHOOL	OBSERVED ACOUSTIC CONSIDERATION (a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)
1.	S	S	S	N	-	N	N	N	-	-
2.	A	A	S	A	-	A	A	N	-	-
3.	S	N	A	N	-	S	N	N	N	N
4.	A	A	A	A	S	A	-	S	A	A
5.	S	A	S	N	-	N	-	N	N	-
6.	A	A	A	A	-	A	A	N	N	N
7.	A	A	A	A	-	A	A	N	-	-
8.	S	S	A	N	-	A	N	N	N	N
9.	A	A	A	A	-	A	A	N	A	A
10.	A	A	A	A	S	A	A	S	A	A

Acoustic shortcomings observed in the music departments of these schools which will affect the optimum use of the music department, are catalogued below.

2.4.2 Catalogue of Acoustic Shortcomings.

(a) Poor sound isolation of music practice room walls.

(b) Inappropriate use of some music practice rooms.

(c) Sound passing through or around door perimeter.

(d) Sound passing through viewing panels in doors.

(e) Sound passing through external windows.

(f) Sound passing through air relief vents or grilles.

(g) Sound passing through ceiling above wall line.

(h) Sound passing along ventilation ductwork.

(i) Sound passing through holes for service pipes.

(j) Sound passing through floor, wall & ceiling junctions.

In ***Table 4*** the presence of these acoustic shortcomings in some or all music practice rooms within the school music departments are scheduled.

x = observed acoustic shortcoming.

Table 4 Schedule of Observed Acoustic Shortcomings.

SCHOOL	OBSERVED ACOUSTICAL SHORTCOMINGS									
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)
1.	x	x	x	x	x	-	-	-	-	-
2.	-	x	-	-	-	-	-	-	-	-
3.	-	-	x	-	-	x	x	x	-	x
4.	-	-	-	-	-	-	-	-	-	-
5.	x	x	x	-	-	x	x	x	-	-
6.	-	-	x	-	-	x	-	x	-	-
7.	-	-	x	-	-	-	-	-	-	-
8.	x	-	x	-	x	-	x	-	x	x
9.	-	-	x	-	-	-	-	-	x	x
10.	-	-	-	-	-	-	-	-	-	-

2.5 Overcoming the Acoustic Deficiencies.

The cost of rectification work in the music department to solve the acoustic problems of music practice rooms is often substantial. The desire of the author of this paper is to encourage the architect and builder to build the facility correctly in the first place.

However, when rectification work is essential to enable the facility to be used to its full potential, a staged approach to the work should be contemplated.

(a) Identify the acoustic problem being experienced.

(b) If more than one acoustic problem is present, rank them in significance.

(c) Budget for the cost of the rectification work over an appropriate period of time.

(d) Commencing with the most serious shortcoming, undertake a rectification schedule.

(e) Assess the effectiveness of each improvement and continue the schedule until a satisfactory result is obtained.

Figure 1 shows a typical section of a music department, and highlights the major acoustic weaknesses detailed in this paper. In contrast to this layout, ***Figure 2*** provides an illustration of a considered approach to the elimination of potential acoustic weaknesses.

2.6 Design Flexibility in the Music Department

Some schools, such as Caulfield Secondary College have made use of surplus large classrooms to create a Music Depart--ment. Within these classrooms the school has installed a series of prefabricated music practice booths.

These music practice booths incorporate high sound isolation excellent lighting, power and ventilation, and are sized to accommodate concurrently, a range of music tuition and practice functions within the confines of the outer classroom.

As the Music Department develops, the music practice booths may be readily relocated as required, providing the school with an economical flexibility.

3. CONCLUSION

It was apparent from the inspection of the ten school music departments that their Architects were aware of the basic requirements for construction of a music teaching facility. However the finer points of design which ensure that the broader considerations in fact work, are often overlooked.

The acoustic weaknesses in the music department, experienced by teachers are not being successfully communicated back to those who have the responsibility to rectify the faults.

In future construction of music practice facilities within schools, it is the Author's hope that appropriate communic--ation between Architect, Acoustical Engineer, Builder, School Board, and Music Staff will take place to ensure that the new Music Department, as well as looking superb, will function equally as well.

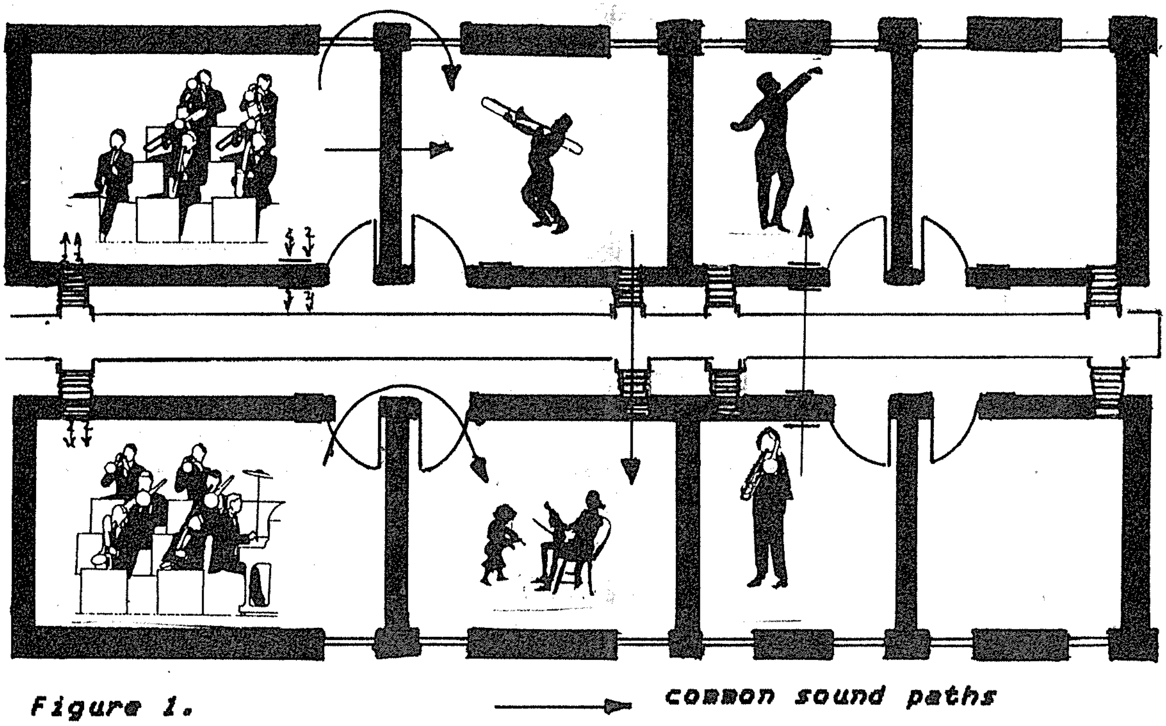

Figure 1.

ACOUSTIC WEAKNESSES IN THE TYPICAL MUSIC DEPARTMENT

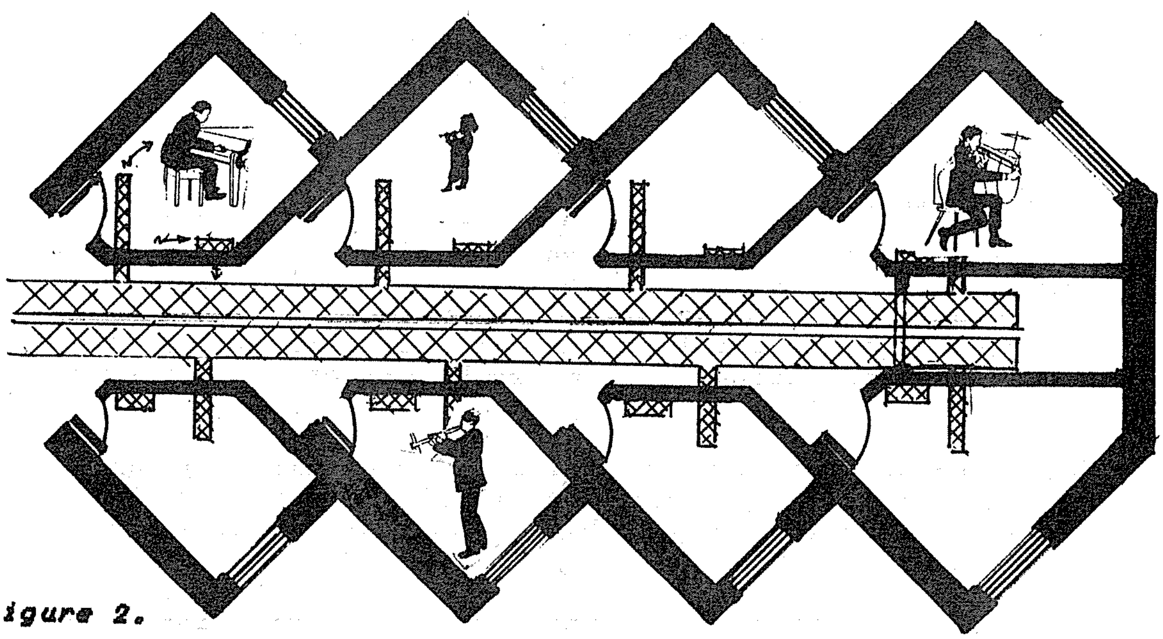

Figure 2.

AN ACOUSTIC APPROACH TO THE MUSIC DEPARTMENT.

4. ACKNOWLEDGEMENTS

The Author would like to thank the Music Co-ordinators from each School for kindly granting permission to inspect and acoustically assess their Music Practice Facilities.

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A STUDY OF NOISE CRITERIA FOR CLEAN WORKSHOPS

CHEN, Qian* LIU, J. H**

ABSTRACT

Clean workshops are widely used in electronic, metallurgical, chemical and precision instrument manufacturing industries, as well as in hospitals. To establish a national standard of China, a nation-wide survey was carried out in 576 clean workshops of 59 factories, which are located in 12 provinces of China. The noise levels in the workshops are found to observe a normal distribution $N(70.4, 8.25^2)$. The relation between the noise level and the rating level of the room cleanliness P (number of dust with $d \geq 0.5 \mu m$ per cube feet) is obtained as $L_A = 43.14 \exp(0.81/\lg P)$. The differences in the noise levels between in-operation and at-rest states, between assembled and constructed workshops, and between different airflow organizations(laminar or non-laminar flow) are obtained, too.

Based on an investigation to 1,154 workers, a study of subjective assessment of noise in clean workshops indicates an interesting regression equation between highly-annoyed percentage and noise level: $HA\% = 1.96 L_A - 98.5$ which is very similar to the OECD formula for environment noise rating $HA\% = 2 (L_{dn} - 50)$. A conclusion is drawn from ergonomics study to the effects of noise on mind-concentration, human performance accuracy and working speed: the effect of noise under 60 dBA is negligible; the one for 65-80 dBA is small and the one above 90 dBA is considerable.

A suggestion for the national standard that gives a target level of 65 dBA and a tolerance level of 70 dBA is produced eventually.

INTRODUCTION

Clean workshop is an artificial, enclosed dust-free space. It is broadly used in electronic, metallurgical, chemical and precision instrument manufacturing industries, as well as in hospitals. The effects of noise on workers in such environment and on products produced in it have gained a lot of attention. Generally speaking, noise in clean workshops causes annoyance of the workers, makes them easier to get tired, reduces the productivity, and increases the ratio of waste products. Due to noise induced vibration, some small particles may escape from various gaps within the materials or structures so that the cleanliness of the room may also be damaged.

With the improvement and development of the clean technology, the effects of noise on production become more and more important. To work out a criteria for clean workshops and develop a guide-line for noise control in clean workshop design is then an urgent task.

Some regulations have been published in various countries. Among others, the Federal Standards 209a and 209b of United States demand that noise in clean rooms must be controlled so that necessary communication is applicable, the requirements of operation or products can be fulfilled and comfort and safe of the working staff can be maintained. The British Standard BS 5295 states that the maximum noise level in the condition while the room, clean (working) equipments or clean air facilities are in operation but with no people in it should not be higher than 65 dBA. The Australian Standard AS 1386 regulates that the noise level in clean rooms should not be higher than 68 dBA.

An regulation saying that noise in clean rooms for electronic industry application should be lower than 70 dBA was put forward by the Ministry of Fourth Machinery Industry of China

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in 1977. In the same year the Ministry of Seventh Machinery Industry published another standard that requires noise level to be lower than or equal to 65 dBA. The regulation of 65 dBA can also be found in "The Technical Measures for Air Clean" which was published by the State Construction Commission of China in 1979.

The above mentioned regulations made in China are based on the general rule for effects of speech interference and annoyance. Although some work has been done on the assessment of clean workshop noise, the all-round survey and systematical investigation which are necessary to establishing a standard have not been seen in China.

Granted by the State Planning Commission of China and the Ministry of Electronic Industry of China, the authors used one year to carry out a noise test and survey to 576 clean rooms in 59 factories which are located in 12 provinces of China. Among other investigation, a subjective assessment survey to 1154 workers was made. A database of noise levels in clean rooms and emitted by the production equipments and air clean facilities is to be built up.

NOISE TEST

LAYOUT.

In order to know the noise levels of the existing workshops and the distribution rule of them, noise test was carried out in all industries which use clean workshops, i.e. electronic, metallurgical, chemical and precision instrument manufacturing industries, as well as hospitals.

Clean workshops with different cleanliness, including 100, 1000, 10000, 100000 rated levels* were tested. The modes of clean workshops tested include assemble-type, construction-type. The locations of these workshops include urban and rural areas, coastal and in-land areas, north and south areas.

Since many factors in the artificial environment, e.g. the air components, lighting, temperature, humidity etc., are combined together with noise, a numerous but carefully selected workshops were visited. To excludes the effects of factors other than noise, clean rooms with the same noise level but with various physical-chemical environments were chosen.

A SUMMARY OF THE TEST

The A weighted noise levels measured in the 59 clean factories indicate that noise levels in the clean workshops in China, in general, are high. The highest level averaged in one factory is 86 dBA. The highest record of the measured point is 102 dBA. The average noise levels in 59 factories are found to observe normal distribution law with mean value of 70.4 dBA and standard deviation of 8.25 dBA. This can be seen from Figure 1 which shows the correlation between the accumulated percentage of the workshop number with noise beyond a certain level and the noise level.

Shown in Figure 2 are some typical noise spectra in which the main noise sources are the ventilation system with medium /lower frequencies as main noise components.

SOME COMPARISONS

Based on an analysis of the measured data, some rules can be seen as follows:

1. Comparison between different industries. As can be seen from Table 1, the medical clean

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* The cleanliness rated levels 100, 1000, 10000, and 100000 mean, in the production environment, the number of the dusts with dimension ^ 0.5lm is \ 100, 1000, 10000, 100000 respectively.

rooms are most quiet and the precision instrument / machinery manufacturing (processing) industry, the second. The noise levels measured in 216 clean rooms of 27 electronic factories were found to observe normal distribution law with mean value of 66.64 dBA and standard deviation of 6.98 dBA. The metallurgical industry is noisier than above mentioned three industries and the worst thing appears in the chemical industry.

Table 1. Comparison of Noise between Various Industries

Industry	Range of Noise Levels	Mean Value	Typical Rooms
Medical	51-63	59.5	Operating Rooms
Precise Processing	54-75	62.5	Micro-bearing
Electronic	57-92	66.6	Vacuum deposition
Metallurgical	64-84	72.5	Crystal silicon furnace
Chemical	64-97	79.1	Magnetic tape

It has been found that the noise from production machines is dominated by medium-high frequencies.

2. Comparison between in-operation and at-rest noise levels. Usually, the clean rooms are called in-operation if they are in the normal production state; the clean rooms are called at-rest if production equipments are turned off with the ventilation, air-condition and air-clean facilities on.

The at-rest noise mainly comes from fans, freezers and water pumps etc. and is transmitted into the rooms through the ducts.

Facilities for air clean in local space such as clean workstations are considered as production machines.

The differences between in-operation and at-rest noise levels are investigated in 14 clean rooms. The average at-rest noise level of the 14 rooms is 55.4 dBA and, in-operation one, 70.3 dBA. The average difference between them is 14.9 dBA and the maximum difference in a room is 32 dBA.

It can be concluded that a well organised standard for clean workshop noise should carefully define the scope or treat the two situations separately.

3. Comparison between assemble-type and construction-type clean rooms. It can be seen from the at-rest noise measurements in 24 assembled clean rooms and 20 constructed ones that the average noise level of constructed clean rooms is lower than the one of the assembled clean rooms by about 10 dBA. The noise levels of the assembled rooms are from 42 to 73 dBA with 60.3 dBA of average value; and the ones from the constructed rooms are from 41 to 62 dBA with 50.1 dBA of average value.

The reason for this distinct difference is that mostly adopted in the constructed clean rooms are central air-clean/air-condition systems which usually use insulated air conditioning machine rooms and, hence, sound insulation can be achieved much easier.

4. Comparison between different airflow organization types. A great difference between the noise levels in clean rooms with different airflow organization types has been found. The noise tests to 16 laminar airflow rooms indicates a noise distribution from 51 to 75 dBA with a average of 65 dBA. In the meantime the measured results for 24 non-laminar airflow rooms show a noise range from 41 to 64 dBA with a average of 53.9 dBA. This is to say that the noise levels in non-laminar airflow rooms are expected 11 dBA lower than laminar airflow ones.

This is believed to be caused by the less needed number of air replacement* for the non-laminar airflow rooms. For example, the numbers of air replacement for non-laminar airflow rooms are commonly 20-50 times/h; meanwhile, the ones for laminar rooms are 200-400 times/h. As can be seen, the smaller the wind volume is, the smaller the ventilation system is and, hence, the lower the noise transmitted into the room is.

What is interesting is, however, that there is no obvious difference between the in-operation noise levels from different airflow rooms. On the contrary, the average in-operation noise levels in non-laminar air flow rooms are a little bit higher than the ones in laminar rooms. This is because that the laminar rooms are mostly used for precise processing such as masking, photoresist, and fine reduction etc. in electronic industry. On the other hand, most noisier production equipments such as vacuum deposition systems are located in non-laminar airflow rooms.

5. Correlation between noise level and cleanliness. An obvious rule can be seen from the measured noise levels in 46 clean rooms which can be catalogued according to their cleanliness. As can be seen from Figure 3, the higher the cleanliness is (the number is smaller), the higher the noise level is. A regression equation can be obtained:

$$L_A=43.14\exp(0.81/\lg P) \tag{1}$$

where

L_A = The space-averaged A weighted noise level in the room, dBA;
P = The cleanliness of the room, dimensionless rated degree.

The residual standard regression coefficient of the equation is S = 0.14 dBA.

6. Correlation between noise level and the effort of noise control. It has been found in the investigation that in most existing clean rooms in China noise control measures have been adopted neither to air-clean/air-condition systems and local air-clean devices nor to the production machines. Some acoustically unreasonable designs were seen incidentally. For instance, a diffusion room in an electronic factory has a good at-rest acoustic condition (60 dBA in winter). Unfortunately, when the cooling wind fan operates, noise level in the room goes up to 80 dBA. A corridor air-return system was set up in a Beijing electronic factory which makes the corridor as noisy as 79-88 dBA.

There are some good examples to follow, too. A precision bearing assemble room with 61 m^2 area in a Northeast China's factory gains an operational noise level as low as 54 dBA due to the effective mufflers in both wind supply and air return ducts. All measured noise levels in masking rooms in an electronic factory of Sichuan Province are lower than 65 dBA which was pre-set as the design goal and achieved through a synthetical treatment. Also, there are burn and blood-cancer sick-rooms in Shanghai's hospitals with at-rest noise levels of 51 dBA and 54.7 dBA.

Reducing the noise emitted by the local air clean devices such as clean workstations can greatly improve the acoustic quality of the clean rooms. Most of these workstations emits noise of about 65 dBA at worker's ear position in China. The lowest measured level is 59 dBA and the highest one, 73 dBA. Three clean workstations in a Beijing's electronic factory increase the room noise from 55 dBA to 74 dBA.

Reducing the noise from production equipments is quite effective. An air exhaust system in a washing room of a Shanxi's factory makes 73 dBA of noise; however, the ultrasonic washing machine increases it to 95 dBA. To be even worse, another ultrasonic washing machine in a Wuhan's factory makes the room 47 dBA more noisy than at-rest level of 56 dBA. A noise control engineering project has been carried out in a crystal silicon production line in Sichuan Province. Before the project the hall with furnaces and pumps arranged in the same room experienced 77 dBA of average noise and more than half of the workers were

* Number of air replacement is for how many times the air in the room is entirely replaced per unit time, times/hour.

highly annoyed; however, after the treatment enclosing all the pumps with a sound insulation room, noise level in the hall was reduced to 70 dBA and 90 percent of workers were happy with that.

Waste/poisonous air exhaust systems are main noise sources in some clean factories. Switching on an exhaust system in a epitaxial room of an electronic factory increases the room noise from 64 dBA to 83 dBA. 97 dBA of such noise was recorded in a washing room of a Beijing's electronic factory. It has been found that this noise source has not attracted sufficient attention and, in fact, it is very easy to control such noise source.

SUBJECTIVE ASSESSMENT

STUDY METHOD.

General speaking, the noise in clean rooms is not as high as in common workshops. According to the research both in China (Fang and Chen, 1987) and abroad (say, Schults, 1982), the effects of noise under 80 dBA on human health can be negligible. Nevertheless no further information will be reported in this paper.

The effects of noise at about 70 dBA levels in working places usually include annoyance, speech/communication interference, and effects on human performance.

Seven rating level method was used to increase the accuracy of the survey. The seven levels for the annoyance are: 1. Extremely quiet; 2. Very quiet; 3. Comparatively quiet; 4. A bit noisy; 5. Comparatively noisy; 6. Very noisy, 7. Extremely noisy.

For speech/communication interference five rating level method is used in the survey. They are: 1. No influence at all; 2. Only a bit influenced; 3. Loud communication is applicable; 4. Very difficult; 5. Impossible at all.

The survey on noise ergonomics includes three parts: the effects on mind-concentration, on accuracy of performance and on working speed. Five rating level method is used here, too.

The response levels in the survey are treated as functions of the noise levels under which the worker works. The noise levels are divided into zones with 5 dBA as a zone, i.e. 37.5 -- 42.4 dBA is considered as 40 dBA; 42.5 -- 47.4 dBA, 45 dBA; etc..

The exposure noise levels of the workers are determined by L_{eq} measurements in ear positions. In most measured rooms the noise levels are very stable on both space and time axes. This is because, generally speaking, the space of clean rooms are small and the walls, ceilings and floors of the rooms are good sound reflective surfaces. Nevertheless, the exposure noise level of the workers are, very often, just the average noise level in the room where they work.

To avoid the other effects, including age, healthy situation, and widely existing psychological trends that overestimating the hazard of noise may gain more economic compensation, the 1500 survey forms were distributed one by one and face to face with careful inquiry and explanation. More than 300 forms were excluded because of the above mentioned factors. The data from workshops where there are distinct effects such as poison etc. was excluded.

ANNOYANCE

The effect of human's annoyance is one of bases for establishing the noise standard of clean workshops.

There are 926 effective forms for annoyance investigation in the survey. All responses stating "Extremely noisy" and "Very noisy" are considered as HIGHLY ANNOYED. The percentage of highly annoyed people to the total number exposed in the same noise level (zone), HA%, is plotted as a function of noise level in Figure 1, too. The investigation result shows a linear regression equation as follows:

$$HA\% = -98.6 + 1.96\ L_A \tag{2}$$

This is to say that, in clean workshops, increasing noise level by 5 dBA will cause nearly 10% more "highly annoyed".

It is worthwhile pointing out that this result is very similar to the results of environment noise survey made by OECD:

$$HA\% = 2\ (L_{dn} - 50) \tag{3}$$

This similarity indicates that noise in working environment with medium/low noise levels has got very similar effects as environmental noise.

According to this result, if more than 2/3 workers should be protected out of "highly annoyed", a 65 dBA of standard level would be recommended.

ERGONOMIC STUDY

A literature review on ergonomic study of noise was put forward as a part of the project which shows that the effects of meaningless noise under 90 dBA on human performance are unimportant (see, for example, Harris, 1979). It was also mentioned that effects of high frequency noise (f>2kHz) are more obvious than the ones of low frequency.

The investigation results on the effects of noise on mind-concentration(MC), accuracy of human performance(AHP), and working speed(WS) are shown in Table 2.

Table 2. Results of Ergonomic Study on Noise

Effect of noise	Number of effective forms	Correlation between Highly Interfered and noise level		Average response level		
				50-60	65-80	≥90
MC	911	$HI(\%)=0.284\exp(0.054L_A)$..(4)		2.3	2.7-2.8	3.1
AHP	918	$HI \leq 6\%$ for $L_A \leq 80$	$HI \geq 30.8\%$ for $L_A \geq 90$	1.8-1.9	2.1	2.8
WS	859	$HI(\%)=0.01\exp(0.084L_A)$....(5)		2.0	2.3	2.8

In this investigation responses at the first two levels ("The influence is Extremely big" and "The influence is very big") are considered as Highly Interfered. The average response level is the average of all response levels in the same noise zone with "The influence is Extremely big" as 5; "The influence is very big" as 4;...and "No influence at all" as 1. It can be concluded that:

--- A obvious THREE STEPS FEATURE is obtained. That means, for noise from 50 dBA to 60 dBA, the effects of noise on performance are very small and does not change with noise levels; for noise from 65 dBA to 80 dBA, the effects are small and does not change with noise levels, either; for noise above 90 dBA, the effects are comparatively high.
--- The highly interfered percentages turn up and become obvious when noise increases to 90 dBA.
--- The effect of noise on mind-concentration is the biggest one among the three effects under consideration.

4. Speech/communication interference In clean workshops no much talk is expected.

However, in some areas the only way for the people in rooms to contact outside ones is by telephone. 904 forms of investigation gives a regression equation as follows:

$$HI(\%)= -402.3+ 245.7 \log L_A \qquad (6)$$

The formula indicates that HI% reaches 50% when noise goes up to 70 dBA.

NOISE STANDARD FOR CLEAN WORKSHOPS

Drawn in Figure 1 is the summary of the results of the noise survey. As can be seen that a 65 dBA of noise standard for clean workshops will make highly annoyed percentage less than 30% and the effect on mind-concentration will be limited under 10% of highly interfered. All other effects are negligible.

However, unfortunately, 75% of existing clean workshops in China are beyond such a standard and 67.4 of existing clean rooms in electronic factories are beyond, too.

On the other hand, a 70 dBA of noise standard for clean workshops will make highly annoyed percentage as high as 39% and the effect on mind-concentration will be 12.4% of highly interfered. Although other ergonomic effects are still very small the speech/communication interference will become serious.

However, more than half of existing clean workshops can meet the requirement of such a standard and only 35% of electronic industry clean rooms would beyond the value.

That means, from point of view of reality 70 dBA of standard is acceptable; from view point of noise influence 65 dBA of standard is better.

As above discussed, if noise countermeasures are taken reasonably, noise levels in most of clean workshops which are at present beyond 70 dBA can be reduced to lower than 70 dBA.

Based on the study a standard draft is accomplished with main points as follows:

--- The operational noise level in clean rooms should not be higher than 70 dBA.
--- The at-rest noise level in non-laminar airflow clean rooms should not be higher than 60 dBA; and in laminar airflow clean rooms, 65 dBA.

A guide-line for noise control in clean workshop design is accomplished, too. It is left out of this paper for the limited space.

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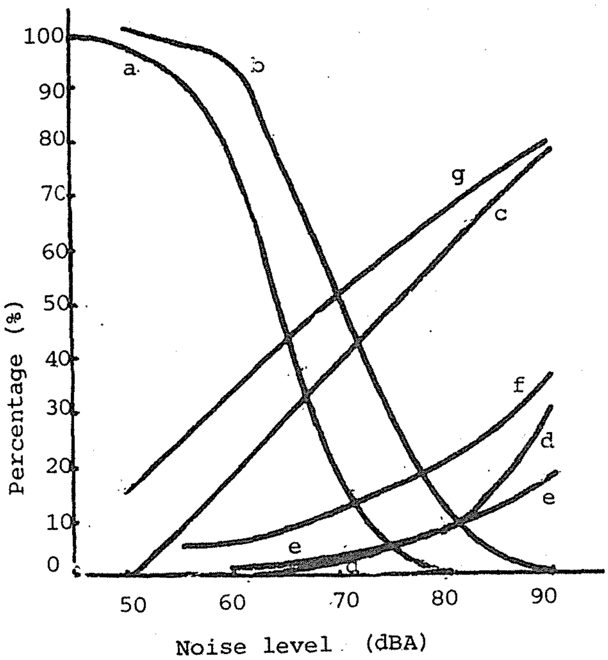
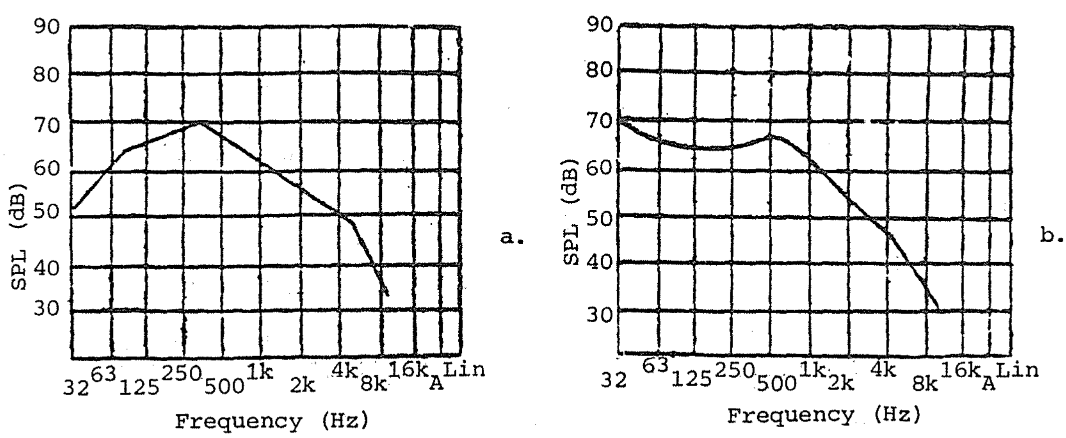


Fig. 1. Cleanroom noise distribution and assessment.
a. Percentage of accumulated workshop number with noise beyond a certain level to the total number vs that noise level. (measured in 59 workshops)
b. ditto, measured in 216 clean rooms of electronic industry.
c. Highly annoyed percentage.
d. Highly interfered on accuracy of human performance.
e. Highly interfered on working speed.
f. Highly interfered on mind-concentration.
g. Highly interfered on speech/ communication.



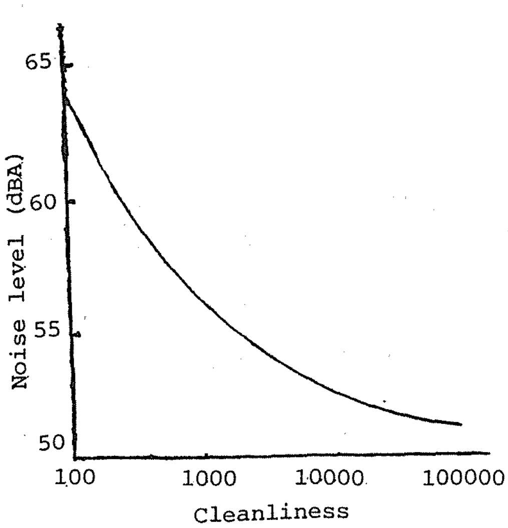


Fig. 2. Typical at-rest noise spectra.
a. A non-laminar airflow cleanroom in Shanghai.
b. A fine reduction room with laminar airflow, assemble-type.

Fig. 3. Correlation between noise and cleanliness.

Conference: Australia Acoustical Society, 1990

Title of Paper: Speech Privacy Design Guide

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Abstract:

NSW Public Works Department has recently published the second edition of its *Speech Privacy Design Guide*. This Guide is a reference document, stating the Departments Acoustic Criteria, and methods of Noise Control.

The Guide includes:

* Acoustic Ambient and Privacy criteria for designed spaces.
* Acoustic wall performance criteria between spaces
* Selection of materials and equipment to achieve the Acoustic Criteria
* Drawing details of noise control in specific applications.

Ambient noise levels specified are in accordance with Australian Standard AS 2107-1987. The selection of room conditions is in accordance with Public Works experience (over 17 years). Sound Transmission Class criteria for walls have been calculated in accordance with Australian Standard AS 2822-1985.

1. Speech Privacy

Speech Privacy is the acoustic quality of a room that relates to the degree to which a person's voice in one room can be heard in an adjacent room, and the degree to which that person can hear voices from that other room. For example in a situation where people are being interviewed by Social Workers, eg. family dispute, their voices should not be heard in the adjacent waiting room.

2. Speech Privacy: The PWD Experience

NSW Public Works is the largest construction authority in Australia with a total expenditure of $711 million in 1988/89.

The Public Works' Acoustics Group provide a comprehensive and competitive consultancy service to the Department's Architecture, Engineering and Construction Divisions.

The Acoustics Group has completed over 3,500 jobs since 1974. Almost every acoustic job had a component relating to Speech Privacy, and often Speech Privacy was the major component.

2.1 Speech Privacy - A Project Management Issue

PWD Acoustics Group experience is that for good Speech Privacy, acoustic input is required at all stages of a project, and include;

- Speech Privacy requirements to be introduced at the beginning of the project.
- Tender documents and plans to specify adequate acoustic criteria (ambient noise and STC rating of walls and ceiling), and provide Acoustic details of construction.
- "Variations to Contract" are agreed to only if the Speech Privacy requirements are maintained
- Site inspections and compliance checks are conducted to ensure good construction methods are employed throughout the project.

2.2 Speech Privacy: Where it fails

The major observations made by the Acoustics Group, where inadequate Speech Privacy in offices occured, include;

- some Project Architects and Engineers did not appreciate the importance of Acoustic quality of spaces and consequently gave little or no design consideration to this aspect.

- Some Project Architects and Engineers hold the view that Acoustic design can be introduced at any stage of a project, including occupation of the building, at minimal additional cost to the project. This is a convenient view and helps the Designer in;
  - reducing initial project cost estimates by deleting acoustic requirements
  - reducing "blow out" costs on "variations to contract", by not including acoustic requirements during construction.
- some Project Architects and Engineers considered their acoustic knowledge sufficient to design spaces for acoustic quality themselves. **Consequently some spaces were under designed due to lack of understanding of the loss in noise performance between laboratory test data and onsite performance.** Furthermore Designers did not supply acoustic construction details in their drawings. This implies they did not appreciate that good on site acoustic performance relies heavily on quality workmanship.
- Even when adequately designed and documented plans were prepared, some constructions failed because there was no provision made for site inspections during construction and compliance checks at end of construction, to monitor acoustic quality.
- Some builders did not;
  - construct according to the acoustic details provided
  - erect walls and ceilings with materials specified
  - install air conditioning systems with the components specified
- on site construction Architects and Engineers agreed to "variations to contract" documents which derated the Speech Privacy performance criteria.

2.3 Priority of Practical Problems

The major practical problems with speech privacy encountered in PWD constructions in approximate order of priority are;

- Poor wall to ceiling joint
- Infill panel between wall and window mullions inadequate
- Perimeter zone console unit not blocked in line with partition
- Door and wall return air grille not acoustically treated
- Skirting board sealing inadequate on jacked up type partitions
- Inadequate door sealing
- Inadequate room to room via ceiling transmission loss
- Poor above-ceiling barriers
- Poor partition details or inadequate design

The partition itself has lowest priority. It is unusual to have problems with it unless its original specifications are inadequate.

2.4 Speech Privacy: Need for a Design Guide

Public Works clients include Education, Health, Police and Courts, Arts and Sports Departments. All require offices for their employees, each demanding different levels of speech privacy in their offices. In order to design offices for good speech privacy, several Australian Standards and manufacturers catalogues need to be referenced.

To encourage Project Achitects and Engineers to include Speech Privacy in their designs, without their having to consult an Acoustic expert on every job, it is important to provide them with all the acoustic information they require, in a convenient, easy to use format.

The PWD Speech Privacy Design Guide contains the information likely to be required by a Designer, and presents it in a "Recipe Style" step by step procedure.

To account for differences between laboratory noise test data and on site construction performance, it is important to provide indicative noise performance losses for various materials. It is also important to provide acoustic details that can be subsequently reproduced in plans, specifying good construction methods.

The PWD Speech Privacy Design Guide provides both on site performance losses and acoustic detail drawings, within the "Recipe Style" presentation.

3. Speech Privacy - Design Factors

Speech privacy in offices depends on a variety of factors, including;

- sound transmission loss of the partition between rooms
- ambient noise in the listeners room and the noise level of speech in the speakers room
- distance from the listener to the partition or door
- acoustic absorbency in the listener's room

4. The Speech Privacy Design Guide

4.1 Highlights of the Design Guide - 2nd Edition

Relationship Diagram

The Speech Privacy Design Guide provides a "Recipe style" solution to acoustic design for offices, with "default" limits to guide the user. Only when the Designers consider they are designing outside these limits do they need to consult an expert in Acoustics. This "recipe" appears as an Acoustic Criteria Relationship Diagram in figure 1.

The Ambient Noise Levels specified are in accordance with Australian Standard AS 2107-1987 for "Satisfactory" recommended design sound level.

The Sound Transmission Class (STC) criteria have been calculated in accordance with Australian Standard AS 2822-1985, with the excess signal level equal to 0 dB(A).

The room conditions, eg. Privacy factor, voice level, floor area and furnishing are in accordance with PWD experience for typical situations.

Once a Designer has identified the room activity, and the adjacent room activity, it is a simple matter to read off the Relationship diagram, the STC ratings of the room walls, ceiling and floor. Having this information, the Designer proceeds to the subsequent sections in the Guide, to select suitable constructions and construction details.

Alternative above ceiling barriers.

With so much air conditioning ductwork installed in limited ceiling spaces, it is extremely difficult to install effective above-ceiling barriers, to isolate speech transfer via the ceiling space. An alternative is the installation of additional layers of plasterboard and fibreglass, over the existing ceiling, and extending for 2m either side of the common perimeter walls. This information is provided together with instructions on treatment to ceiling penetrations, eg. light fittings and air terminals.

Plasterboard partitions with 50mm steel studs

With a move to lightweight, thinner constructions, there is a preference for 50mm steel studs partitioning. Test data for these constructions are provided.

Compliance checking

Even with every good intention at the design stage, there is a need to conduct an Acoustic Compliance check for work completed, prior to occupation.

The Guide establishes the need for several site visits during construction (to arrest and correct a potential problem before it becomes very expensive to rectify at the final check), and recommends that compliance levels should be within +3dB(A), - 5dB(A) of the Design levels. The building contractor is required to make good the construction, should it exceed these limits.

4.2 A walk through the Design Guide

Ambient noise levels and Partition Noise ratings

This is the starting point for the Designer to establish the ambient noise levels appropriate for each space and then the noise ratings of walls separating the spaces. This section is covered in the "Highlights of the Design Guides - Relationship Diagram".

Suitable Construction

After selection of the required STC rating of walls, the Designer proceeds to the Materials section which describes various wall and ceiling arrangement, together with their STC ratings.

Categories include;
- Masonry Partitions
- Plasterboard Partitions, fixed and demountable
- Glass Partitions
- Above ceiling barriers
- Doors
- Operable Walls and Accordion Doors
- Composite walls

This section recommends "loading factors" for various types of construction to account for loss in performance due to standard construction. For example a 3 STC loading is given to masonry partitions and up to 20 STC for operable walls and accordion doors. Also, there are details for recommended door seals (perimeter, bottom of door and meeting styles).

It is typical to have composite walls of 2 materials, eg. door in partition or fan light glazing. A look up table for typical wall areas is provided with associated STC ratings.

Common Problem Areas

After the materials have been selected, the next section identifies common Acoustic Construction problem areas, including;
- Wall to window junction
- Wall to ceiling junction
- Above ceiling barriers
- Skirting boards
- Power points

Drawings are provided showing good acoustic solutions to these problems.

Mechanical Services

Most spaces these days are air conditioned. Acoustic problems for speech privacy arise at;

- Return air transfer ducts and grilles
- Supply air cross talk
- Above ceiling penetrations
- Perimeter zone inductions air conditioners

Again, drawings are provided showing good acoustic solutions to these problems.

Compliance Check

An Acoustic compliance check should always be conducted on a building prior to occupation. Criteria for this check are in the Guide's final chapter, and are discussed under "Highlights of the Design Guide".

5. Future Improvements to Design Guide

Many recommended criteria and construction details do not come with quantifiable noise performances, rather they have evolved from the Acoustics Group's collective experience, spanning 17 years.

The Building industry trend is to; lighter constructions, less overdesign, standardised construction techniques and cost saving opportunities. Consequently, Acousticians are increasingly required to justify, in quantifiable terms, their requirements for heavier materials and "better than standard" workmanship. Both aspects increase building costs. Some areas for which there are no acoustic test data currently available include;

- STC rating of walls and ceilings having typical penetrations, eg. power points, door grilles, lightfittings, and typical perimeter gaps
- Noise breakout from air conditioning flexible ducts
- Room to room and ceiling to room STC data for various modified ceiling arrangements, eg. fibreglass and plasterboard over mineral fibre tile panel grid ceiling, and for the wall erected off the grid.

Also there is a need to;

- improve AS 2107, AS 2822 by
  - better describing activity spaces
  - better explaining the terms Satisfactory and Maximum Design, and relating these levels to Compliance Levels.
- conduct "Acoustic Awareness" training seminars for Project and Site Architects/Engineers.

6. Acknowledgement

The author thanks NSW Public Works Department for the resources to prepare the second edition of this Speech Privacy Design Guide, and colleagues in the Acoustics Group for their technical contribution.

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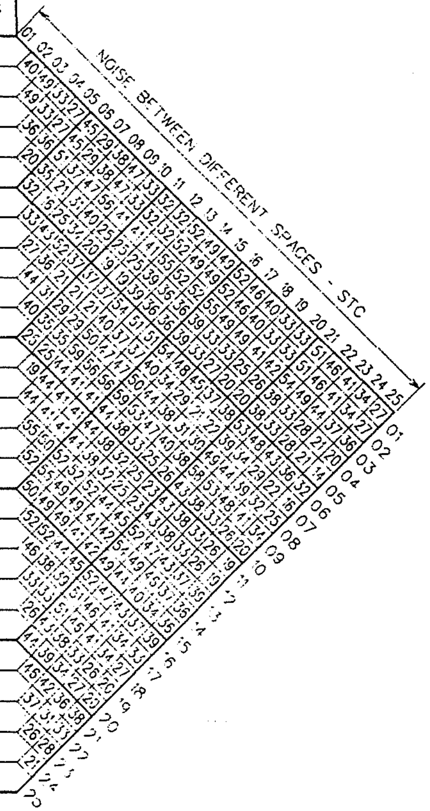
   BTR Ludeng
   Modernfold
   Rintoul
   Silverwood & Beck

	SOURCE ROOM Privacy Factor	SOURCE ROOM Voice Level	FLOOR AREA m2	RECEIVE ROOM Ambient Level dB(A)	TYPE OF OCCUPANCY/ ACTIVITY	NOISE BETWEEN IDENTCL SPACES STC
01	NORMAL	CONVSE	12	35	ADMIN. OFFICES	40
02	NORMAL	CONVSE	12	35	AUDITORS OFFICE	40
03	CONFID	RAISED	25	30	BOARD ROOM	52
04	NORMAL	CONVSE	50	45	CALC./TABULATION	20
05	LOW	CONVSE	50	45	COMPUTER ROOM	14
06	CONFID	RAISED	50	30	CONFERENCE ROOMS	47
07	LOW	CONVSE	35	45	CORRIDORS/LOBBYs	17
08	NORMAL	CONVSE	25	35	CRT REP/T'SCRIPT	35
09	CONFID	RAISED	70	25	COURT ROOMS	50
10	NORMAL	CONVSE	50	40	DESIGN OFFICES	25
11	LOW	CONVSE	50	40	DRAUGHTING OFC.	19
12	LOW	CONVSE	50	40	GEN. OFFICE AREAS	19
13	CONFID	RAISED	12	40	INTERVIEW ROOMS	47
14	CONFID	RAISED	25	30	JUDGES CHAMBERS	52
15	CONFID	RAISED	25	35	MINISTERS OFFICE	47
16	CONFID	RAISED	12	35	PERSONNEL ADMIN.	52
17	NORMAL	RAISED	12	35	PRIVATE OFFICE	46
18	NORMAL	CONVSE	12	35	PROF. OFFICES	40
19	LOW	RAISED	50	40	PUBLIC SPACES	25
20	LOW	CONVSE	25	40	RECEPTION AREAS	24
21	CONFID	RAISED	15	35	SENIOR EXE. OFFC.	51
22	NORMAL	RAISED	12	40	SUPERVISORS OFF.	41
23	LOW	RAISED	6	50	TEA ROOMS	32
24	LOW	CONVSE	12	50	TOILETS	19
25	LOW	CONVSE	50	45	TYPING POOL AREAS	14



<u>NOTES</u>

1. <u>Privacy Factor</u>
   - low; speech which can be intelligible to a person in an adjacent room, and for which the speaker is not overly concerned.
   - normal; speech which is partly intelligible to a person in an adjacent room, but not intrusive, to that person
   - confidential; speech which is not intelligible to a person in an adjacent room, except when that person concentrates on hearing.

2. <u>Voice Level</u>

   The level of a speaker, judged at 1m from the speaker, conversational and raised voice (if shouting anticipated, then consult Acoustics Group).

3. <u>Ambient Level</u>

   The noise level existing in the room, when unoccupied, but with normal building services operating in the room, e.g. air conditioning, and adjacent rooms to be occupied with normal activities.

4. <u>Sound Transmission Class (STC)</u>

   The noise rating of walls and ceilings required to ensure that speech in a room is not intelligible to persons in an adjacent room.

5. <u>STC Design Guide Limits</u>

   Advice from the Acoustics Group must be sought when the space to be designed differs greatly in requirements from that recommended as typical in the Table. The safe limits are listed.

Speaker's privacy:	As recommended - OK
Speaker's voice:	As recommended - OK
Speaker and Listener Positions:	Greater than 1m from any wall - OK
Floor area:	Within 50% of recommended value - OK
Ceiling height:	Less than 3m - OK
Room absorbancy:	Furnished room - OK
Composite walls:	STC ratings are within 10 units of each other - OK
Ambient noise in rooms:	As recommended - OK

NOTE: Poor perimeter sealing, e.g. wall to ceiling, skirting joints and gaps not set, e.g. brick mortar, plasterboard sheets, or penetrations not treated, eg. air relief door grilles, power points, will all derate the noise performance of the partition. Special attention is required here.

NEW CHALLENGES AND SOLUTIONS FOR INTERIOR NOISE CLIMATES

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Environmental issues have assumed a level of significance in our society which few of us would have believed possible when we embraced acoustics as our profession. Twenty five years ago, before I turned my back on underwater acoustics and electronics, and before devoting my professional efforts to the wider field of architectural and industrial acoustics, I decided to assess what was happening in Europe and America. On the basis of the trends I perceived in America, I believed that Australia would most probably follow the American pattern with acoustics assuming a far greater level of importance in architectural and industrial design.

Whilst my crystal ball gazing was basically correct, the 'ebb and flow' of Governmental and developers indecision (which some would even describe as technical ineptitude) over the last 25 years and particularly when they were faced with forseeable problems, has provided me with a multitude of anecdotes which are sufficiently noteworthy and voluminous to warrant writing a book.

In the initial years of that period, I spent most of my time correcting acoustical blunders, the majority of which had been perpetrated by engineers and architects whose general appreciation of either the need for acoustical design or the procedures through which such design should be implemented, were disturbing to say the least.

Learning from other people's mistakes is a less disturbing and far less painful means of learning than the alternative which is to learn from one's own mistakes. With those observations and experiences as a basis, I have developed a series of documented procedures through which I can assist my clients.

What is clear to me is that when the planners, architects or engineers failed to adopt a comprehensive plan for the assessment of acoustically related factors, there were generally acoustical problems downstream and the project generally suffered.

The achievement of appropriate acoustical performance for an architectural or industrial project requires that both the external and internal acoustical environment should conform to minimum standards. These include the achievement of appropriate background noise levels, and other internal and external attributes which are primarily determined by the nature of the usage of the subject premises, as well as those that are adjacent. To achieve these fundamental requirements, one must adopt an appropriate plan through which these as yet undefined goals may be achieved.

The process should theoretically start with the selection of the building site and continue through each subsequent stage of the design. Obviously in most practical situations, one or more of the potential steps is pre-ordained or may be dictated by site conditions over which you will have no control. Notwithstanding, it behoves the architect, engineer or planner (as well as the acoustical consultant) to avoid potential design errors by adopting check lists in which the appropriate acoustical precautions are identified so that they may be addressed.

A typical identification process will involve the following steps:

(a) The selection of a site which should ideally be in the quietest possible surroundings consistent with the other planning requirements and financial constraints, (excluding of course the requirements of noisy industrial sites).

(b) The evaluation of the diurnal statistical variability of extraneous noise sources, through which the potential building façade design criteria may be appropriately assessed.

(c) The development of an appropriate design configuration which places those rooms requiring the quietest environment well away from the potential sources of intrusive noise.

(d) The selection of appropriate construction systems for the building envelope, with due consideration being given to the design of glazing, doors, entrances, air intake and discharge penetrations for both ventilation and exhaust systems.

(e) A comprehensive assessment of the potential noise and vibration sources within the building (and where relevant, external to the building), to ensure that the most appropriate barrier to airborne sound and structure-borne vibrations may be integrated into the design.

(f) An assessment of the design configuration for each type of room or space for which specific acoustical requirements have been identified. This assessment should take into account background noise levels, the need for enhanced sound propagation requirements in terms of speech and/or music, or conversely, the attenuation requirements of the space, if sound intrusion needs to be minimised.

(g) The selection and distribution of absorptive and reflective surfaces and selection of materials to either optimise the build-up, maintenance or where necessary, the decay of sound within that room or space.

(h) The acoustical design requirements for the air conditioning system and for the associated mechanical services, in order to preclude an adverse impact on specific areas within the building (as well as an adverse impact on adjacent buildings).

(i) The design of electronic and electro-acoustic systems associated with communications, computing networks, speech amplication systems, emergency evacuation systems, electronic masking systems or internal paging systems, in order to ensure compliance with the fundamental or competing acoustical requirements of each room or space within the building.

(j) The acoustical performance requirements, field or laboratory test requirements and associated quality assurance programs to be adopted for the selection and specification of critical acoustical materials, products, services or systems in the building. (This approach is essential if one is to ensure that the basic design criteria will be achieved by each of the systems on the completion of the building.)

(k) The development of an acoustical supervisory program for the production of the relevant acoustical elements during the various phases of construction, to ensure that contractors fulfill the specific acoustical requirements identified in the drawings and/or specifications. Such a program must incorporate intermediate, as well as final acoustical performance test evaluations, through which the standards of nominated construction and assembly may be ensured.

(l) Preparation of appropriate maintenance instructions and specifications for special construction, periodic inspections, special testing and even sources of replacement material relating to :-

(i) how the acoustical materials or special facings can be cleaned and redecorated; or alternatively with what materials they may be replaced;

(ii) the acoustical characteristics and extent of specific surfaces, furnishings, carpets and underlay, absorptive screens or wall linings, acoustical murals or tapestries so as to maintain a specified level of sound absorption, or variable absorption when and where it is required;

(iii) the environmental design requirements which should be specified in terms of temperature and/or humidity, supply and return air velocities, main and branch duct velocities, hydraulic pipe velocities and other associated physical parameters which have been adopted as specific design criteria for low noise areas, such as sound recording and broadcast or television studios, court rooms, parliamentary and debating chambers, auditoria or spaces in which special electro-acoustic requirements have been identified;

(iv) the performance parameters of electro-acoustic systems, as well as their design requirements for use in rooms in which it is proposed to provide assisted resonance, digital sound processing, and ambiophany, in order to achieve designated standards of 'variable acoustical' performance.

Obviously the adoption and implementation of an appropriate acoustical check list, and one which leads to the identification of potential problems at the earliest possible stages of the design, invariably assists both the acoustical consultant and his client or architect, to avoid the majority of critical acoustical pitfalls. Although such procedures are invoked in major architectural projects, they are equally applicable to the planning process of industrial and transportation facilities, educational institutions and in a wide range of commercial buildings.

There are relatively few opportunities for an acoustical consultant to comprehensively implement all or even most of the technological requirements of the above list. There are of course a few projects in which virtually all of these check list items become important, or even critical, if the project is to proceed smoothly.

New Parliament House Canberra was just such a project, and when its acoustical consultancy was offered to me, I just couldn't refuse what I forsaw as being potentially the most challenging brief of my career.

Much has been written about many of the more newsworthy issues such as cost over-runs, industrial disputation, construction delays and even the 'avante garde' architecture. By contrast, very little has been said and even less written[1,2] about the complex acoustical problems which we faced during our eight years of hectic involvement on Capital Hill.

We were involved in many interesting, quite a few unusual and many more exciting problems, only a few of which are described below, and more of which will be subsequently described during the lecture which will expand on these issues.

Our Ref 1960A/505A

One of the most significant implications of the 'fast track' design and construction process was that at the initiation of construction in the formative period 1981 to 1983, the most important aspects of the structural design (and also the associated construction program) were well in advance of the detailed architectural design.

At that time it was not possible for either the architects or the structural engineers to positively identify where the majority of future sound or fire rated walls would be constructed, as those details had not yet been resolved.

Of the many 'fast track' problems which affected the acoustical design, one of the earliest in which I became embroiled was that associated with the prior adoption by the structural engineers of a waffle slab reinforced concrete floor configuration. That form of construction offered maximum strength and load bearing capacity, irrespective of where, or what form, those future walls might take.

Nobody perceived that this system might impose some potential acoustical or even zoning limitation. It was only after the formwork for the first area of suspended floor slabs was stripped, and scaffolding and formwork was already being placed for the first floor in the northern precinct of the Parliament that the more insidious implications resulting from the selection of this form of floor construction was fully perceived.

On viewing the first of the suspended waffle slab floors, the architects, Mitchell Giurgola and Thorp, realised that they (and their client, The Parliament House Construction Authority) faced a significant and a potentially expensive problem, as the architectural design just could not be artificially restrained by the waffle slab rib configuration, and yet it was obvious that the wall configurations would require extensive and potentially complex fire and acoustical sealing at the irregular line of intersection underneath the lower face of those waffle slabs.

After examining a large number of potential conventional solutions, (none of which appeared to offer appropriate functional or economic advantages), and after unsuccessfully seeking help from the structural engineers, I was asked to offer a solution or develop a design which would satisify the acoustical requirements of the various types of walls and ideally satisfy the fire rating requirements at the same time.

Our Ref 1960A/505A

The problem was potentially more serious than even I had anticipated, because the relatively thin upper central sections of the waffle slab would in many places constitute a 'weak link' in terms of its airborne sound attenuation performance.

The depth of these sections was typically 125mm thick, and as such this thin element would constitute the primary factor limiting the airborne sound attenuation performance of the floors in those areas where low noise levels were required in the area immediately above basement mechanical plant rooms.

In a number of critical locations, such as in the library and cabinet rooms, which were to be constructed directly over the central energy plant room, the limited sound attenuation capabilities of the slab, would prejudice our ability to achieve the background sound levels designated for those areas.

A further problem which only added 'fuel to the fire', was the extent of walls for which such a solution would be required. The initial estimates indicated that more than 10,000 metres of wall length would require such treatment, and as time progressed this figure was found to be conservatively low. With a problem of such major proportions, it was also apparent that the solution would need to be economical to avoid a major cost blowout.

Having considered the problem, I prepared a sketch for the only solution which I believed could effectively resolve the acoustical and fire related problems and which would provide an appropriate transmission loss for those sections of the waffle slab floor overlying mechanical plant and/or noisy equipment.

The solution was based on the fabrication of a series mass produced moulded plasterglass or GRC panels with a profile which precisely matched the open aperture at the base of each waffle cavity. I proposed that each panel would be structurally reinforced by a set of galvanised steel rods with a cruciform configuration, supplemented by glass fibres to optimise the strength and provide deformation resistance.

Our Ref 1960A/505A

Based on the sketches, a series of test panels were produced, and the results of those tests were regarded as being sufficiently impressive to proceed with fire tests at the Experimental Building Station, and subsequently acoustical tests to confirm their acoustical adequacy. Production of some 30,000 of these panels followed over the next four years, and the nett saving in construction costs for the the client, as a result of the use of these panels, was estimated to be in excess of $1,000,000.

One of the other 'fast track' problems about which I have already written[1,2], related to our acoustical scale modelling. What has not been previously documented was our use of a 12mm thick glass miniature reverberation chamber to evaluate the 1/10th scale materials which we utilised in our subsequent scale modelling program. Miniature glass reverberation chambers are quite dandy when you have to see what is happening inside, however, as we discovered, Lubasky's Law of Cybernetic Entomolgy (there's always one more bug), always seems to take control of such circumstances and the glass reverberation chamber developed its own resonance characteristics when excited by our high energy spark source. Such problems of course don't occur when you utilise a model as well made as that which the late Peter Birmingham had constructed for the House of Representatives modelling program. As we soon discovered such problems are capable of being rectified by the judicious use of damping materials including multiple layers of lead vinyl covered foams, which neatly solved that problem.

During the course of the project, we were forced to adopt some unusual stratagems to solve what many of you may well consider to be relatively simple problems. Foremost amongst these was our need to find a source for the procurement of doors conforming to the design brief specification, which required members and senators office doors with STC ratings ranging between STC 42 and STC 50, and a simultaneous requirement that the doors be no more than 50mm thick.

Doors with lower ratings, or with greater thicknesses, were readily available from local manufacturers. When they were approached, not one of the local manufacturers was able or prepared to offer a 50mm thick door which would conform to the specification.

Our Ref 1960A/505A

Faced with the knowledge that nobody wanted to offer what we required I realised that if we couldn't show that it was feasible - and respecting the government edict of "NO IMPORTED DOORS", I decided that the only option would be to design the door myself.

With the architect's assistance, I successfully requested a small capital grant to design a prototype door, as well as its seals, so that I could arrange for a contractor to construct the door and test it.

Faced with the problem that the architect would not accept any of the current range of high performance Australian door seals, (which he vocally described in unprintable terms), I was then forced to design a seal which would be visually innocuous and yet would still be acoustically effective. This I did, and although the prototype door was half a point below the magical STC 42, the client and the architect were satisfied that this would suffice. More than 300 single panel doors were constructed in accordance with that prototype design, and all of those doors were tested and confirmed as providing an installed rating of better than 40 STC.

Of the many complex design problems which we successfully faced during the design and supervisory phases of New Parliament House, one minor one stands out, and is worthy of recounting. That problem occured during the fit-out stage of the House of Representatives when I was convinced that the construction manager had not ensured that the interface between the double glazed central skylight structure and the up-turn of the concrete roof of the building had been effectively sealed.

My problem was compounded by the construction manager's insistence that the scaffolding, which would provide access to confirm my suspicions, was to be removed and without delay.

What do you do when you have a skylight 20m above the floor, with an array of fragile glass elements extending down below the core of the skylight and you want to prove that there is an acoustical deficiency without risking damage to the glass or the skylight, nor to the carpeted floors or furniture which was already being installed. Although we knew that conventional techniques could resolve this impasse, the construction manager steadfastly refused each proposal and of course denied that there was any problem.

The approach that I adopted was I acknowledge a trifle unusual, although I must admit that it tickled my fancy (and that of my able assistants in the Quality Assurance Group).

I had previously purchased a meteorological balloon, to which we fitted a sub-miniature Sennheiser microphone and Sennheiser sub-miniature microphone cable to keep the weight down. The helium filled balloon and its microphone was then floated gentle up to the outer edges of the skylight area so that the microphone was level with the interface around the edges of the skylight. In that position we could use it with an interconnected sound level meter to measure the level of sound transmission at the interface around each of the four sides of the skylight.

As we had half expected, the construction manager was unwilling to accept our measured results on the basis that the microphone could not be positioned close enough to the interface to possitively identify the 'exact points' at which the sound leakage problem was purported to be occuring.

He requested that we should reverse the process and put the sound source inside with the microphones outside (without using the scaffolding and without using rigid supporting elements which might risk damaging the suspended glass elements). Now that was a real problem, yet it was also solved using similar techniques. We attached an array of flat disc electro-acoustic transducers, (of the type used for the buzzers in calculators and computers) around the what I would describe 'the balloon's stomach' had it had one. These provided us with a 100 dB of sound pressure level at 1m distance at selectable frequencies of between 2-4 kHz, which conforms to their resonant frequency peaks. With the amplifier on the floor, and the signal being fed up the same miniature cable, we really confounded the construction manager and positively identified the points from which there was excessive sound leakage around the base of the skylight, and at a number of other points in the skylight for good measure. The construction manager seemed to be impressed, but much to our surprise was still reluctant to put up the scaffolding required to seal the inner edges of the skylight/roof interface.

Two days later a 'higher body' took control, and during an extremely heavy deluge, rain was observed to be leaking through the skylight and thus immediately prejudiced the on-going fit-out of the House of Representatives. Faced with the added pressure of the client requesting the double fix (acoustics plus water leakage), the scaffolding was installed, and we were able to inspect and re-identify each of the points through which sound leakage had been identified and have them fixed.

New Parliament House Canberra was undoubtedly one of the most exciting and professional rewarding projects on which I have had the pleasure to work. I was lucky to be supported by a team of able engineers, a responsive architect, a very able quality assurance group, quite apart from a number of competent contractors.

No project runs smoothly, and the problems which we experienced on that project required an inordinate amount of time to develop solutions so as to ensure appropriate rectification. In retrospect, it was the problem solving and the wide range of 'fixes' during that project which made the whole exercise worthwhile.

At the end of the road when you have assessed what you have done and what aspects of your work have provided you with your greatest 'job satisfaction', you will undoubtedly agree that 'problem solving' is what acoustics is all about. As long as we are prepared to take off our 'rose coloured glasses' and avoid the problems of 'tunnel vision', which may affect any of us, but more likely our clients, then acoustics will continue to be one of the most exciting technological fields in which we can enjoy our work and improve the 'quality of living' of those around us.

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THE EVALUATION OF SOUND INTENSITY FROM TAPE RECORDED SIGNALS VIA THE QUARTER-SQUARE MULTIPLIER PRINCIPLE

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ABSTRACT

In many practical situations it is desirable to be able to tape record data in the field and analyse the sound intensity in a laboratory. This is significantly more convenient than having to use a large laboratory based analyser into the field, particularly for diagnostic purposes where one might want to repeat the measurement/spectral analysis with a different bandwidth.

This paper demonstrates how sound intensity can be measured with a simple tape recorder (without any consideration being given to phase), and how it can be subsequently analysed with a single channel spectrum analyser (narrow band or octave etc.).

The paper outlines how the well established quarter-square multiplier principle (which operates directly on analog pressure signals) can be used (i) to obtain a broadband sound intensity measurement; (ii) as a pre-processor to accurately tape record sound intensity, and (iii) to accurately determine octave, one-third octave and narrowband sound intensity measurements from the tape recorded signals with two filter sets which do not have to be phase matched. Both the tape recording and the filter sets do not introduce any phase errors and the system does not require the sound field to be time invarient - i.e. the two signals are not measured sequentially but are acquired and processed in parallel.

INTRODUCTION

The use of acoustic energy flux or sound intensity in the area of noise characterization and noise control has significantly increased since the advent of fast signal processing techniques, and the many other advances in analog and digital electronics and instrumentation. One important property of sound intensity is that it represents only the active part (i.e. the propagating part) and not the reactive part (i.e. the non-propagating part) of a sound field, whereas sound pressure is the sum of both [1].

Sound intensity is defined as the average rate at which sound energy is transmitted through a unit area in a direction normal to that area. It is computed as the expected value of the product of the instantaneous sound pressure and the corresponding particle velocity. Since velocity is a vector quantity and pressure is a scalar, the sound intensity is a vector quantity pointing in the direction of the velocity vector. The component of sound intensity in the i-direction can therefore be written as

$$I_i = \langle p(t)\, u_i(t) \rangle, \qquad (1)$$

where I_i is the sound intensity in the i-direction, p(t) is the acoustic pressure and $u_i(t)$ is the particle velocity in the i-direction. The <> denotes the expected value for stationary sound fields, the expected value being equivalent to the time-averaged value.

The required pressure for the measurement of the sound intensity is easily measured using a pressure microphone. The difficulty in measuring the sound intensity arises in trying to determine the particle velocity. The technique which is now universally adopted for the estimation of the particle velocity can be traced back to a seminal paper by Schultz [2]. The particle velocity is deduced by integrating the pressure difference measured between two microphones, which are a small distance apart. This method rests on the linearized Euler equation assuming no mean fluid flow, where the pressure gradient is estimated by a finite difference approximation. An important requirement of this method is that the characteristics of the pressure microphones must be identical, i.e. they must at least be phase matched, magnitude variations being easily accounted for in the measurement.

The sound intensity measurement based on the two microphone technique has been formulated by several authors, amongst others Pavic [3] and Gade [4], and is not repeated here. It suffices to state the final result, i.e.

$$I_i = -\frac{1}{2\rho\Delta x_i}\left\langle (p_1(t) + p_2(t))\int(p_2(t) - p_1(t))\, dt \right\rangle, \tag{2}$$

where $p_1(t)$, $p_2(t)$ are the pressures from the two pressure microphones, which are a distance Δx_i apart, and ρ is the density of air.

Even with the formulation given by equation (2), it is still not an easy matter to implement a simple system for accurate sound intensity measurement. A direct method would require either analog or digital techniques to process the pressure microphone signals. In the case of digital processing, and since the direct signal processing includes a time-averaging operation, a large amount of computer memory is taken up with the digital storage of the microphone pressure signals. This means that a large computing machine is required for this task. This is not a very useful situation for practical field investigations. A more practical signal processing technique, which was independently developed by Fahy [5] and Chung [6], shows that the sound intensity can be calculated from the imaginary part of the cross-spectrum of the two microphone pressure signals. This technique is nowadays easily implemented using a dual-channel FFT analyzer, allowing for the calculation of sound intensity in the standardized octave and one-third octave bands, as well as in narrow frequency bands. Mathematically, the sound intensity is computed from the cross-spectrum G_{12} between p_1 and p_2 as

$$I_i = \frac{-1}{2\pi\rho\Delta x_i}\int_0^{\infty}\frac{\mathrm{Im}(G_{12})}{f}\, df\,. \tag{3}$$

The alternative to the digital signal processing is to operate directly on the analog pressure signals produced by the microphones. This entails using an analog computing circuit which implements equation (2). A variation on the direct implementation of equation (2) was proposed by Bolt & Petrouskas [7], who applied the pressure-sum pressure-difference principle to measure the acoustic impedance of a surface. This quarter-square multiplier principle was also used by Fahy [8] in a system which used the two-microphone method, and included a portable sound level meter fitted with octave band filters to measure sound intensity. The particular system used by Fahy required the sound field to be time invariant, since the two signals had to be measured sequentially. The two-microphone technique with a similar analog processing system in conjunction with a two-channel spectrum analyzer was used by Norton & Soria [9] to study the effects of bounding surfaces on the radiated sound power of sound sources. This system also included a processing unit which evaluates and digitally displays the sound intensity directly, and therefore does not suffer from the limitations of a sound level meter based system, since the two signals are acquired and processed in parallel.

The analog signal processing methods have certain advantages over the digital cross-spectrum method for practical field measurements - (1) the analog instrumentation is usually much smaller and easier to handle, (2) a direct reading is often available and (3) matched filters are often incorporated giving a direct octave or one-third octave sound intensity spectrum. The advantage of the digital system is that once the pressure signals have been digitized, no further phase errors can be introduced between both channels. Hence, the only phase error in the cross-spectral signal processing method is due to the microphone phase mismatch. Phase matching between both analog pressure channels is essential for accurate sound intensity measurements, because the particle velocity measurement involves the measurement of a phase gradient. This is the reason why the tape recorded microphone pressure signals cannot be used to accurately determine the sound intensity. The tape recording of the two-microphone signals and the subsequent play-back of the two tape recorded signals introduces large phase errors, which makes the accurate measurement of the sound intensity impossible.

In many practical situations it would nevertheless be desirable to be able to tape the analog sound intensity signal for archiving, since this type of data storage is still cheaper than digital storage. Also, it is not always possible or desirable to use dual channel FFT analyzers in industrial environments, whereas a small tape recorder is easily used and maneuvered in any industrial environment.

This paper outlines how the quarter-square multiplier principle can be used (1) to obtain a broadband sound intensity measurement, (2) as a pre-processor to accurately tape record sound intensity and (3) to accurately determine octave, one-third octave and narrow band sound intensity measurements from the tape recorded signals. Instead of the tape recorder, two filter sets which do not have to be matched can be substituted and the spectral analysis of the sound intensity performed in situ. It is also shown in this paper that the filter sets do not introduce any additional phase error.

THE SIGNAL PROCESSING UNIT

The block diagram of the signal processing unit developed in this study is shown in Figure 1. The primary difference between this unit and a convential quarter-square multiplier (e.g. Fahy [8]) is that the processing unit evaluates and digitally displays the sound intensity directly since the two signals are acquired and processed in parallel. The two microphone voltages V_1 and V_2 are applied to the respective inputs on the unit. The microphone voltages are first multiplied by their respective calibration constants α_1 and α_2. At this stage, both signals are ac-coupled and low-pass filtered. This operation uses matched broadband amplifiers with individual gains in conjunction with matched low-pass filters. The phase variation between both channels for this section of the circuit was electronically tested, and found to be less than 1.5^0 in the frequency range 100 Hz - 12.8 kHz.

The analog circuit is then divided into two legs. In one leg, the difference between the two pressure signals is fed into an integrator with a gain accuracy better than 0.1 dB, and a phase accuracy of less than 2^0 in the frequency range 100 Hz - 12.8 kHz. In the other leg, the two pressure signals are summed. The signals from both legs are then combined to form two different analog signals, one representing the sum, and the other representing the difference between the signals from both legs.

The sum and difference amplifiers in the signal processing unit have a gain error smaller than 0.06 dB, and a relative phase difference of less than 1^0 in the frequency range 100 Hz - 12.8 kHz. It must be pointed out, that the error values given in the preceding paragraphs are the worst case values, and they occur at the high frequency end of the operating range with the exception of the integrator. The low frequency phase error for the sum and difference amplifiers and filters is approximately $\pm 0.1^0$.

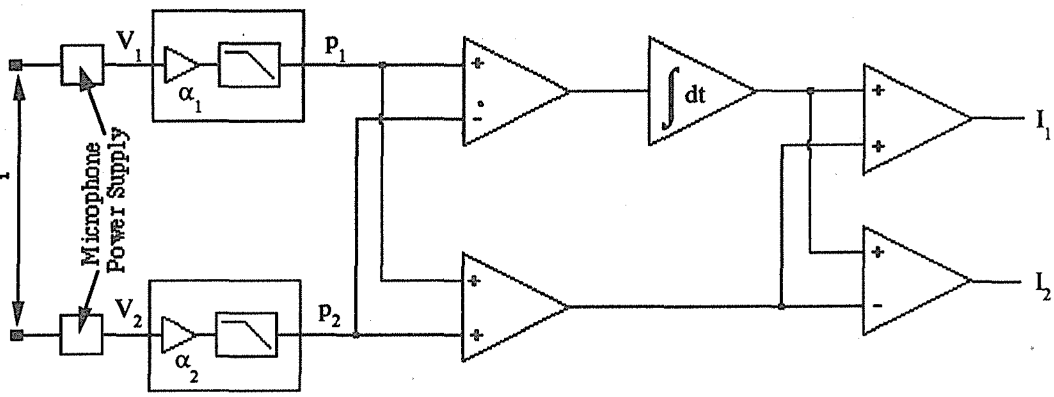

Figure 1. Block diagram of the signal processing unit.

The performance of the signal processing unit has been tested in a standing wave tube. A Bruel & Kjaer type 3519 sound intensity probe system was used for this purpose. This system can be used with 6.35 mm diameter type 4135 microphones or 12.7 mm diameter type 4165 microphones. The microphones are arranged in a face-to-face configuration, separated by plastic spacers of either 6 mm or 12 mm for the 6.35 mm microphones and 12 mm or 50 mm for the 12.7 mm microphones. For these tests, the probe microphones were calibrated using a B & K pistonphone type 4220. The complete results of these test are given in Norton & Soria [9]. It suffices to say, that the measured sound intensity for the frequency range of interest had a maximum deviation of 1.3 dB compared to the calculated sound intensity based on the standing wave ratio, with most variations being typically less than 0.3 dB.

MATHEMATICAL CONSIDERATIONS

In this section, the mathematical justification for tape recording the pre-processed signals is presented. The signal analysis required to evaluate the sound intensity component I_i in terms of the signals $I_1(t)$ and $I_2(t)$ (Figure 1) produced by the signal processing unit is quite straightforward. The output from the summing amplifier is given by

$$I_1(t) = (p_1(t) + p_2(t)) + \int (p_1(t) - p_2(t))\, dt, \tag{4}$$

and the output from the difference amplifier is given by

$$I_2(t) = -(p_1(t) + p_2(t)) + \int (p_1(t) - p_2(t))\,dt. \tag{5}$$

Subtracting the expected square value of equation (5) from the expected square value of equation (4) yields

$$\langle I_1(t)^2\rangle - \langle I_2(t)^2\rangle = 4 \left\langle (p_1(t) + p_2(t)) \int (p_2(t) - p_1(t))\,dt \right\rangle. \tag{6}$$

From equations (1) and (6) it is now an easy step to deduce an expression for the sound intensity I_i in the i-direction, i.e.

$$I_i = -\frac{\left(\langle I_1(t)^2\rangle - \langle I_2(t)^2\rangle\right)}{8\rho\Delta x_i}. \tag{7}$$

In order to evaluate the sound intensity in a given bandwidth, one notes that the expected square value of the signals $I_1(t)$ and $I_2(t)$ can be evaluated from their respective auto spectra (Newland [10]), i.e.

$$\langle I_1(t)^2\rangle = \int_0^\infty G_{11}(f)\,df \tag{8}$$

and

$$\langle I_2(t)^2\rangle = \int_0^\infty G_{22}(f)\,df. \tag{9}$$

$G_{11}(f)$ and $G_{22}(f)$ are the one-sided auto spectra of $I_1(t)$ and $I_2(t)$ respectively, and f is the frequency in Hz. The sound intensity in a given bandwidth between the frequencies f_1 and f_2 is easily computed by changing the limits in the integrals of equations (8) and (9), hence

$$I_i(f_1, f_2) = -\int_{f_1}^{f_2} \frac{\left(G_{11}(f) - G_{22}(f)\right)}{8\rho\Delta x_i}\,df. \tag{10}$$

The preceding analysis has shown, how the broadband sound intensity or the sound intensity in a given bandwidth is deduced from the two output signals of the signal processing unit. The operations performed on the output signals $I_1(t)$ and $I_2(t)$ are (i) the expected square operations, and (ii) the difference between the resultants of these operations. Phase variations between both output signals from the signal processing unit do not affect this computation, since the differencing operation only involves constant numeric values and not time-dependent values. In principle, it is therefore possible to tape record the pre-processed sound intensity signals $I_1(t)$ and $I_2(t)$, and then, to accurately determine the sound intensity from the recorded signals without the tape recording process introducing any additional phase error.

For exactly the same reason, it is possible to use filter sets, which do not need to be phase matched, to band-pass the output signals $I_1(t)$ and $I_2(t)$. It is then possible to deduce the sound intensity from these filtered signals. This operation is given by equation (10). For stationary sound fields (most practical sound fields are stationary) the expected value is replaced by a time-averaged operation.

EXPERIMENTAL EVALUATION OF THE PRE-PROCESSOR

The B & K type 3519 sound intensity probe system fitted with the 12.7 mm diameter type 4165 microphones is used for the experimental evaluation. The microphones are arranged in a face-to-face configuration separated by a 12 mm plastic spacer. The microphones and the amplification/low-pass filter stage of the signal processing unit (SPU) have been calibrated with a B & K sound level calibrator type 4230. The sound intensity is evaluated from the microphone signals by two methods, i.e. (i) from the imaginary part of the cross-spectrum (equation (3)) between both microphone signals, and (ii) by feeding the microphone signals into the SPU and recording the two outputs with a Nagra IV-SJ tape recorder, and then subsequently analysing the recorded signals by using equation (10).

The signal analysis for both techniques is performed by a four channel, programmable Data Precision Data 6000 digital spectrum analyzer controlled by a Apple Macintosh SE microcomputer via a RS 232 C interface. The analog signals were digitized into time records 1024 points long with an acquisition period of 78 μs. Spectral ensemble averaging over 50 records is performed as part of the signal analysis. The input sound for the experimental evaluation is produced by a sound power source (B & K type 4205), and consists of broadband noise between 100 Hz - 10 kHz, with a constant power level setting. A schematic of the experimental set-up is shown in Figure 2.

The basis of both the sound intensity measuring methods is still the two-microphone principle, which measures the phase difference between the signals at points 1 and 2. A phase error between both measuring channels causes an error in the sound intensity measurement as discussed in detail by Gade [11], [12]. This type of error is confined to the low frequency range of the measurements. The upper frequency limit of the measurement range is principally due to the approximation of the pressure gradient by the finite pressure difference. For the microphones/spacer combination used in these tests, the upper frequency limit is ~ 5 kHz. The error in the sound intensity due to phase mismatch is a result of the measuring system and the sound field. As Gade [12] showed, this error is given by

$$L_\varepsilon = -10 \log\left(1 \pm 10^{(L_{K,0} - L_K)/10}\right). \tag{11}$$

where L_K is the measured pressure-intensity index, which is defined as the difference between the measured sound intensity level and the measured pressure level, and $L_{K,0}$ is the residual pressure-intensity index.

The residual pressure-intensity index is an indication of the bias error that is present between the two channels of the complete measuring system, and therefore, gives an indication of the quality of the system. The residual pressure-intensity index of both measuring systems has been estimated by feeding the same broadband signal to both microphones in an acoustic coupler. The results of these residual pressure-intensity measurements for both sound intensity measuring systems are presented in Table 1.

Based on equation (11) Gade [12] stipulates, that the measured pressure-intensity index must be numerically 7 dB smaller than the residual pressure-intensity index of the measuring system to ensure that any error due to phase mismatch is within ± 1 dB. It should be noted that the SPU/Tape measuring system runs into trouble in the 150 Hz octave band. This problem is easily alleviated by increasing the microphone spacing, i.e. doubling the microphone spacing results in $L_{K,0}$ = -10 dB in the 150 Hz octave band.

Another source of error is the random error associated with the sound intensity measurement. Gade [11] has shown that the required BT product (B - bandwidth, T - averaging time) depends on the desired statistical accuracy and the pressure-intensity index. Based on Gade's data, the expected normalized random error for the one-third octave sound intensity measurements in this investigation for a 68 % confidence interval is estimated to be less than 10 %.

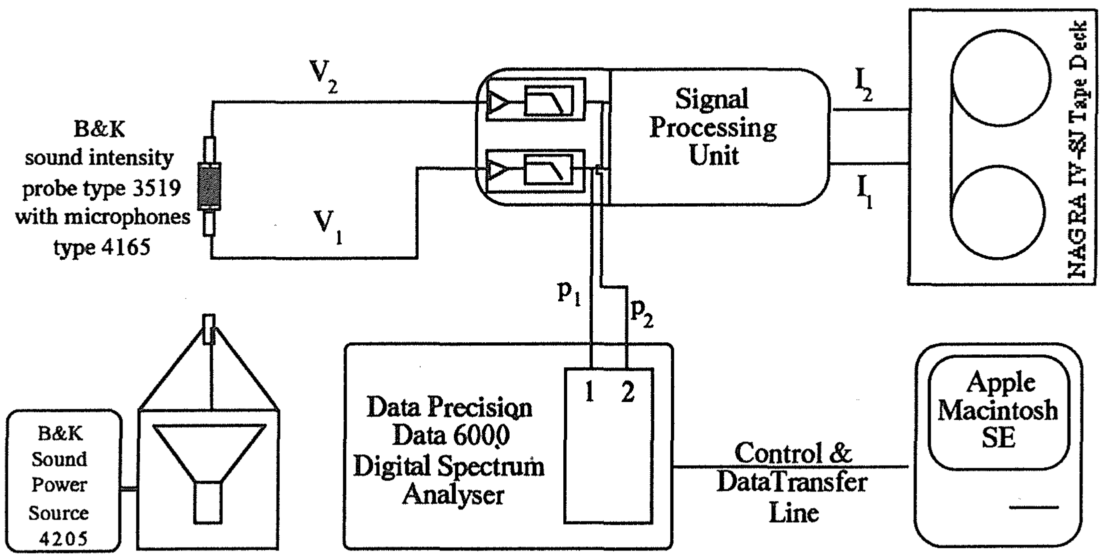

Figure 2. Schematic of equipment and experimental set-up for this investigation.

The broadband sound intensity from the sound power source has been estimated from the imaginary part of the cross-spectrum by direct measurement of the amplified and low-pass filtered pressure signals as depicted in Figure 2. Concurrently with this sound intensity measurement, the processed signals from the SPU are tape recorded using the Nagra IV-SJ tape recorder. The tape recorded signals are subsequently analysed with the Data Precision Data 6000 spectrum analyzer by performing the mathematical operation given by equation (10). That is, the ensemble averaged auto-spectra of the two tape recorded signals are evaluated, and this is followed by applying equation (10). The results of these measurements are presented as (i) 200 Hz bandwidth narrow band sound intensity measurements, (ii) one-third octave band sound intensity measurements, and (iii) octave band sound intensity measurements in Figures 3, 4 and 5.

The results presented in Figures 3, 4 and 5 show that the sound intensity determined from the pre-processed tape recorded data compare favourable with the more accurate sound intensity measurement calculated from the imaginary part of the cross-spectrum between the two microphone signals. The maximum deviation in the sound intensity measurement occurs towards the high frequency limit of the measurement. This discrepancy is due to limitation of the particular microphone/spacer combination used, and because the tape recorded signals in this frequency range are very small, digitization errors of the tape recorded analog signals are present at the high frequency limit.

Table 1. Residual pressure-intensity index for both sound intensity measuring systems and typical measured pressure-intensity indices at the measuring position as shown in Figure 2.

1/3 Octave f_c (Hz)	$L_{k,0}$ (dB) cross-spectrum system	$L_{k,0}$ (dB) SPU/ tape deck system	L_k (dB)
159	-12	-7	-3
200	-14	-9	-1
252	-16	-10	-2
317	-17	-11	-2
400	-19	-12	-2
504	-20	-13	-1
635	-21	-13	-2
800	-21	-13	-1
1008	-21	-13	-1
1270	-21	-13	-2
1600	-21	-14	-2
2016	-21	-14	-2
2540	-21	-14	-2
3200	-25	-15	-2
4032	-27	-15	-6
5080	-23	-15	-7

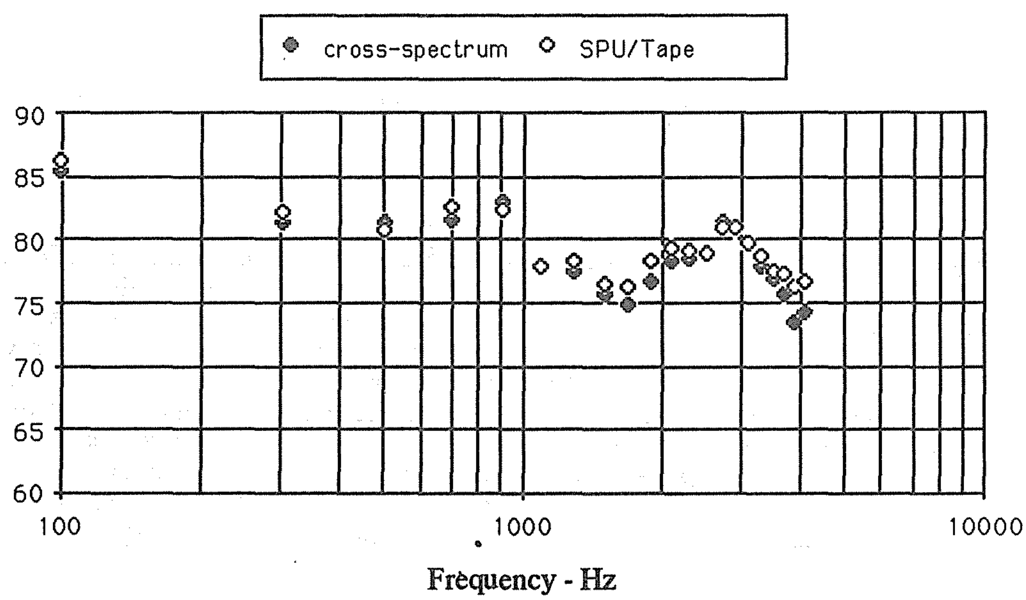

Figure 3. Comparison of 200 Hz bandwidth narrow band sound intensity measurements (dB) using the imaginary part of the cross-spectrum method and the SPU/tape recording method.

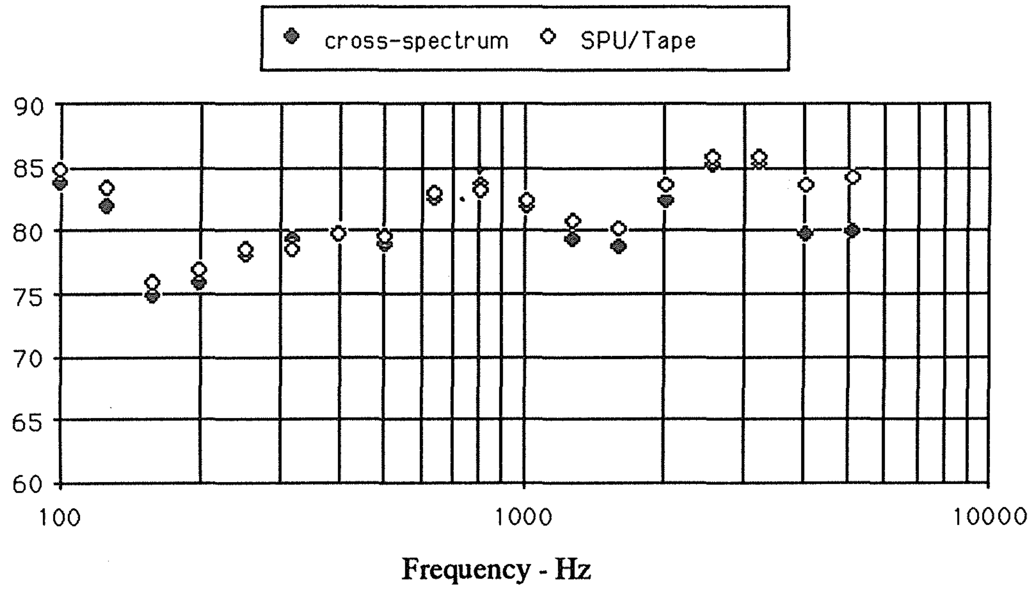

Figure 4. Comparison of one-third octave band sound intensity measurements (dB) using the imaginary part of the cross-spectrum method and the SPU/tape recording method.

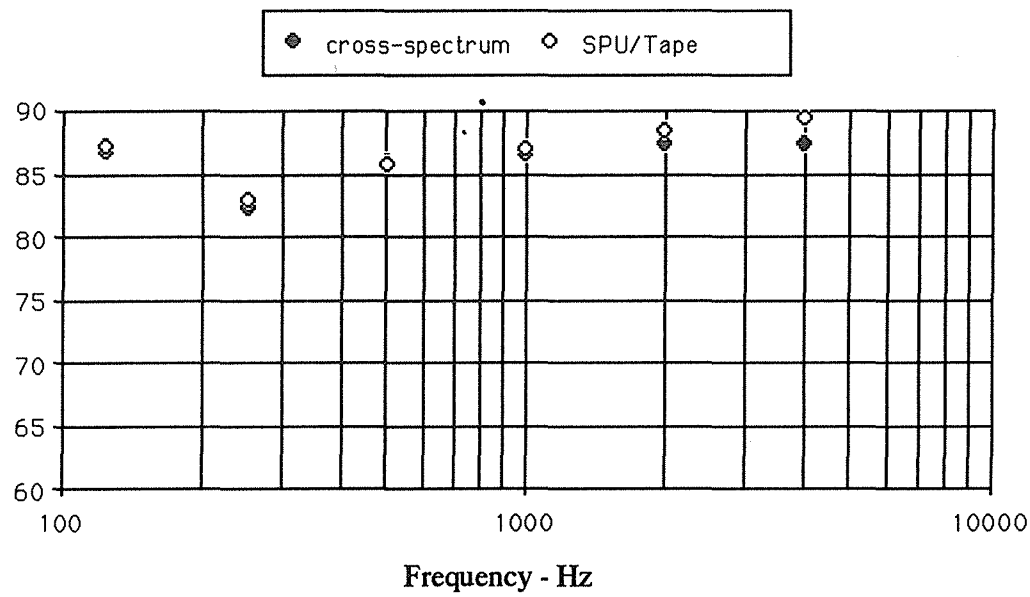

Figure 5. Comparison of octave band sound intensity measurements (dB) using the imaginary part of the cross-spectrum method and the SPU/tape recording method.

Better matching between the tape recording gain and the analog-to-digital conversion range can reduce this second problem, whilst the first source of discrepancy can be alleviated by using a smaller spacing between the microphones. Some deviations are also observed at the low frequency end of the measurements particularly for the one-third octave bands which provide finer resolution at the low end of the frequency spectrum. These deviations are due to phase mismatch in the two channels of the SPU as shown in Table 1. ($L_{K,0}$ - L_K) is -9 dB for the cross-spectrum method compared to -4 dB for the SPU/tape deck method, indicating the much better quality of the cross-spectrum method in this low frequency range. The better quality of the cross-spectrum method is not unexpected, since the only phase mismatch possible in this system is due to the microphones and the gain/filter section of the SPU. Once the two pressure signals are digitized, no additional phase error can be introduced in the cross-spectrum analysis method. The quality of the SPU/tape deck method can be improved by increasing the microphone spacing, but this in turn has the effect of decreasing the high frequency range of the system. Therefore, it is necessary to chose the appropriate microphone and the microphone spacing carefully for a particular measurement depending on the frequency range of interest.

CONCLUSIONS

It has been shown in this paper that the two-microphone sound intensity method can be used in conjunction with a signal pre-processor based on the quarter-square multiplier principle, to tape record the two resultant signals, and then accurately deduce the sound intensity from these tape recorded signals. The technique has been compared against the cross-spectrum signal analysis method and the results are good. The SPU/tape deck system appears to be therefore a useful alternative for field measurements of sound intensity, which allows easy subsequent narrow band analysis in the laboratory and inexpensive archiving of the sound intensity data.

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NOISE SOURCE IDENTIFICATION IN BUILDING VENTILATION SYSTEM: An Application of the Sound Intensity Technique

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ABSTRACT

One of the major sources of noise in buildings is that due to the ventilation system. The noise from a plant room may be transmitted into the offices via the air-borne, structure-borne or duct-borne paths. This paper presents field investigations of low frequency noise occurring in an open-plan office. The sound intensity technique was used to identify the noise source and its propagation path. Flow measurements using hot-wire anemometry were used to deduce the operating efficiency of the fans and to determine whether there is any correlation between the noise generation and the dynamics of the fluid flow. Results of field transmission loss measurements of the wall dividing the plant room and the open-plan office made by using the sound intensity technique indicate that the wall provides sufficient attenuation to the noise in the plant room. The source of low frequency noise was located to be in the fan by conducting sound intensity scan in the plant room. Flow calculations suggest that this low frequency noise could be due to vortex shedding and coalescing. The air flow has been found to be highly turbulent (over 30%) within 350 mm of the fan outlet. Such high turbulence further enhances the low frequency noise generated at the fan blades.

1.0 INTRODUCTION

Ventilation system noise in an office building is often the major source of concern for ensuring that the ambient sound levels comply with the Standards (such as AS2107) for the particular type of occupancy. Conventional methods of evaluating the acoustical performance of ventilation systems involves the determination of the sound power generated by fans and the attenuation of ducts in special laboratory environment as documented in the Standards (such as BS848 Pt. 2). Such methods are time-consuming and expensive and often may not correlate well with the field performance. With the advent of the sound intensity technique, it is possible to make in-situ measurements on site, although the application of the technique to flow ducts is still under research (see, for example, Fahy (1985) and Seybert (1988)).

In this paper, a case study is presented for an open-plan office in which the ambient sound level, particularly in the low frequency range between 63 and 125 Hz, is too high. The objective of this study is to demonstrate the use of the sound intensity technique to identify the noise source and its propagation path and to determine if there is any correlation between the noise generation and the fluid dynamics of the flow.

2.0 THE CASE STUDY

As shown schematically in Figure 1, the open-plan office space being studied is located next to the plant room which supplies ventilated air to the office. The air-conditioning system in the plant room consists of two 51-blade forward curved centrifugal fans driven by a 5.5 kW-rated electric motor through a pulley belt drive system. The motor speed was 1450 rpm while the fans were rotating at 1070 rpm so that the blade passage frequency (BPF) was 910 Hz. As seen in Figure 2, the ambient sound levels in the office space at low frequencies (particularly below 160 Hz) are too high and are the major source of concern here. The reverberation times in the open-plan office were measured with an Bruel & Kjaer (hereafter referred to as B&K) type 2231 modular precision sound level meter with BZ7104 reverberation processor module and B&K 4224 sound source. The reverberation time of the office space of volume 1400 m^3 at 500 Hz was measured to be about 0.8 s.

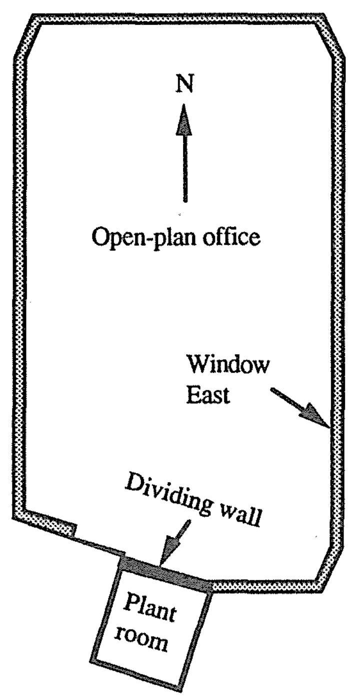

Figure 1 Schematic showing open-plan office and plant room.

3.0 EXPERIMENTAL TECHNIQUE AND INSTRUMENTATION

3.1 Sound Intensity Technique

The sound intensity technique is an advanced technique developed in the 1970s and became commercially available in the 1980s for noise source identification, sound power measurements and noise ranking. Unlike sound pressure which is a scalar quantity, sound intensity is a vector quantity (that is, it possesses both direction and magnitude). Hence, by knowing the sound intensity in a sound field, the source of noise or noise propagation path can be identified. A detailed review of the principle of the sound intensity technique, its limitations and accuracy has been described by Fahy (1989). Basically, the technique involves the use of two microphones which are used to detect the sound pressure as well as the phase change of the sound field over the two microphones. By processing this information with a dual channel fast Fourier transform analyzer (FFT), the sound intensity can be deduced.

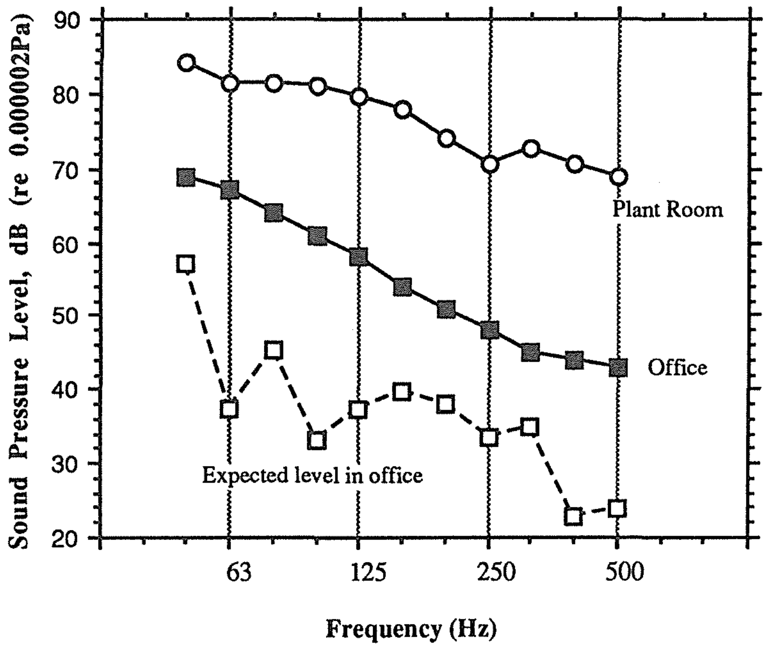

Figure 2 Ambient sound pressure level and expected level in open-plan office and ambient sound pressure level in the plant room.

3.1.1 Instrumentation

For all the measurements presented here, the sound intensity system used comprises a B&K 2032 dual channel FFT analyzer and a sound intensity probe made up of a pair of B&K 4181 phase-matched 1/2 inch microphones mounted in a face-to-face configuration. The microphones were separated at a distance of 50 mm for low frequency investigation (that is, below 500 Hz.). The narrow band sound intensity data were processed and synthesized into 1/3 octave bands from 50 to 500 Hz with a Hewlett-Packard series 300 microcomputer. The microphones were calibrated with B&K 3541 sound intensity calibrator and the calibration in sound intensity was within 0.1dB of the value specified by the manufacturer.

The sources of error of sound intensity measurements have been discussed by Gade (1985) and are primarily due to phase mismatching between the two microphone channels at low frequencies. It has been shown by Gade (1985) that the error L_ε due to phase mismatching is given by

$$L_\varepsilon = 10 \log [1 \pm 10^{(L_K - L_{K,0})/10}] \qquad (1)$$

where L_K, the pressure-intensity index, is the difference between the measured sound pressure L_P and sound intensity L_I level;

L_{K0}, the residual pressure-intensity index, is the difference between the pressure and intensity level when the microphones are subjected to a sound field with 0^o phase difference between the two microphone positions.

For a measurement accuracy to within 1dB, L_K has to be at least 7dB lower than L_{K0}. The dynamic capability, which is defined as (L_{K0} -7) dB, for the measurement system used is shown in Figure 3.

3.2 Instrumentation for Flow Measurements

Mean streamwise velocity and turbulence intensity traverses were made in the supply ducts in the plant room at 350 mm downstream of the two supply fans with a TSI (Thermal Systems Inc.) constant temperature hot-wire anemometer. The sensor used was an TSI1210-60 platinum hot-film with a diameter of 152μm, operated at a constant resistance ratio of 1.5. As the relationship between voltage output and velocity is nonlinear, the hot-film signal was first linearized, then passed through a digital voltmeter and an rms (root-mean-square) meter to give the mean velocity and the turbulence intensity. The frequency spectrum of the streamwise turbulence intensity was obtained by an B&K2032 analyzer and stored in an B&K7400 digital cassette recorder.

4.0 IDENTIFICATION OF NOISE PROPAGATION PATH

With reference to Figure 1, the noise generated in the plant room may propagate into the office space through the dividing wall, the supply ducts or the windows. In order to identify the propagation path, the field transmission loss (FTL) of the dividing wall was measured using the sound intensity technique as reported by Lai & Burgess (1990). Sound intensity scans were made in various parts of the office space with the ventilation system operating.

4.1 Field transmission loss measurements of the dividing wall between the open-plan office and the plant room

The field transmission loss of the dividing wall between the open-plan office and the plant room was measured by using an B&K 4224 sound source to produce a pink noise in the plant room. The sound pressure level L_{ps} in 1/3 octave frequency bands in the plant room was measured and the transmitted intensity L_{It} through the dividing wall in the open-plan office was measured by scanning the wall with the sound intensity probe at a distance of approximately 200 mm from the wall. The results have been used to calculate the field transmission loss of the dividing wall which is shown in Figure 3, together with the pressure-intensity index of the measurements and the dynamic capability of the measuring system. Since the pressure-intensity index in the 63 Hz frequency band is higher than the dynamic capability of the sound intensity system used, the field transmission loss value in this frequency band cannot be determined with confidence. If the dominant transmission path from the plant room to the open-plan office is through this wall, then an estimate of the expected ambient sound pressure level in the office can be obtained by using the FTL values in Figure 3 and the sound pressure levels in the plant room with the ventilation system operating. The results are shown in Figure 2 which indicates that the low frequency noise levels in the open-plan office would be at least 10 dB lower than that had been measured, thus suggesting that the dominant transmission path is not through the dividing wall between the plant room and the open-plan office space.

4.2 Sound intensity scans in the open-plan office

Sound intensity measurements were made in the office on surfaces of interest by scanning a sound intensity probe at about 200 mm from the surfaces.

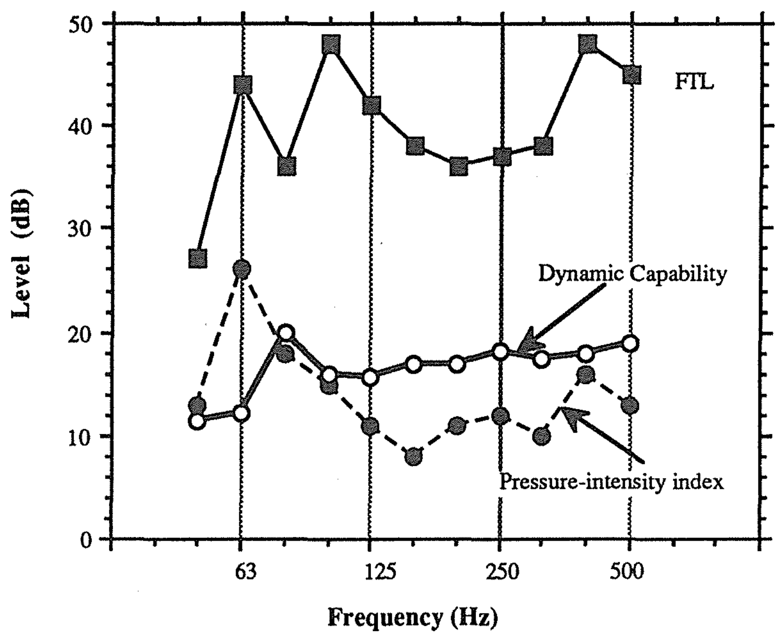

Figure 3 Field sound transmission loss of the dividing wall, dynamic capability of the measuring system and pressure-intensity index of the measurements.

4.2.1 Dividing wall and windows

The results of the sound intensity scan on the dividing wall are shown in Table 1.

Table 1 Sound intensity level and pressure-intensity index within 200 mm of the dividing wall

1/3 octave band centre frequency (Hz)	L_I (dB)	Pressure-intensity index $L_P - \|L_I\|$ (dB)
50	57	12
63	47	20
80	50	14
100	39	22
125	-43	15
160	46	8
200	35	16
250	33	15
315	33	12
400	<20	>24
500	26	17

In general, a high value of the pressure-intensity index indicates that there is a significant component of sound travelling in the opposite direction to the transmitted intensity component and if this component is larger than the transmitted component, the measured sound intensity will be negative. For example, in the 125 Hz 1/3 octave band, the sound intensity value of -43 dB indicates that the sound propagates from the open-plan office towards the dividing wall rather than from the wall into the office space (as would be expected). As the pressure-intensity value is fairly high for most of the 1/3 octave frequency bands, it is suspected that a significant component of sound is propagating from the office space to the wall. It is, therefore, worthwhile to examine the sound intensity measured in narrow frequency bands. As shown in Figure 4, the sound intensity is negative over a fairly extensive range of the frequencies of interest, indicating that the low frequency noise propagates from the open-plan office to the wall. The results, therefore, suggest that the low frequency noise originated in the plant room does not get transmitted through the dividing wall; this is consistent with the field transmission loss measurements described in section 4.1.

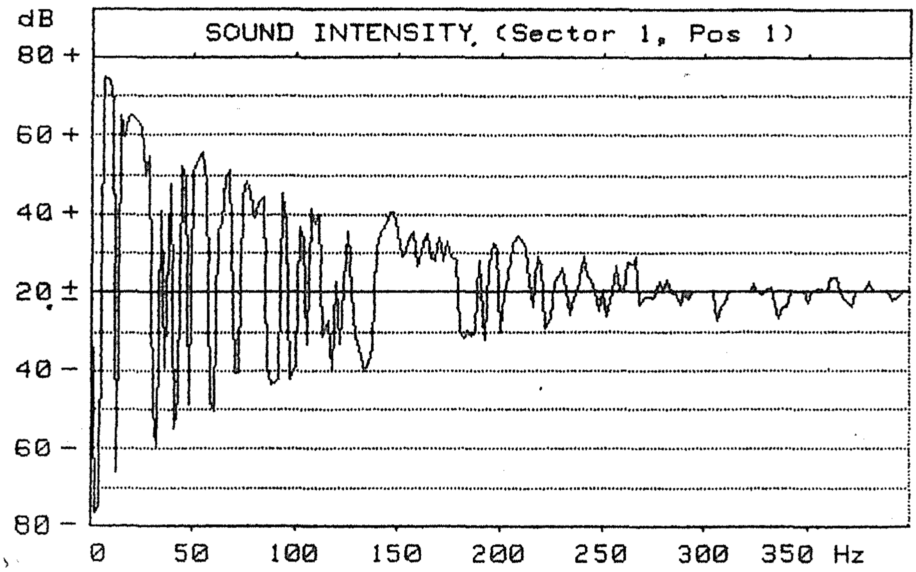

Figure 4 Measured sound intensity spectrum over the dividing wall.

The results of the sound intensity scan on windows (south) and windows (east), though not shown here, are similar to those obtained for the dividing wall, thus indicating that the high noise levels in the low frequency bands are not transmitted into the open-plan office from the plant room through these building components.

4.2.2 Below main supply duct within 2 m of the dividing wall

The results of the sound intensity scan below the main supply duct within 2 m of the dividing wall are given in Table 2. The low values of the pressure-intensity index in Table 2 indicates that the sound in the 1/3 octave frequency bands below 250 Hz is transmitted into the office space through the supply duct and the ceiling here. This is indeed confirmed by the measured sound intensity spectrum in narrow frequency bands (not shown here) which shows that apart from a few frequency lines, the intensity is positive. In fact, there is very little difference between the narrow band sound intensity spectrum and sound pressure spectrum.

Sound intensity scans were also made below the main supply duct at turn, approximately 6 m from the dividing wall. The results, not shown here, indicate that the low frequency noise transmitting into the office space does not get attenuated much and is still radiating through the ceiling.

Table 2 Sound intensity level and pressure-intensity index below main supply duct.

1/3 octave band centre frequency (Hz)	L_I (dB)	Pressure-intensity index $L_P - \lvert L_I \rvert$ (dB)
50	67	1
63	60	0
80	58	3
100	52	5
125	55	1
160	53	2
200	47	4
250	41	6
315	<30	>15
400	<30	>14
500	42	0

5.0 NOISE GENERATION IN THE PLANT ROOM

5.1 Sound intensity scan over the ventilation unit

Sound intensity scans were made in the plant room over different parts of the ventilation unit, namely, the two fans, the electric motor, the air intake section and around the ducting system. The sound intensity scan over the electric motor shows that the levels are significantly lower than that from the fan. As shown in Figure 5 which displays the narrow band sound intensity spectrum for fan number 1, high levels of low frequency noise are being generated at the fan. The sound intensity spectrum for the air intake section in Figure 6 indicates that the low frequency noise generated in both fans radiates through the air intake section into the plant room. The average air speed at the air intake section was about 2.5 m/s.

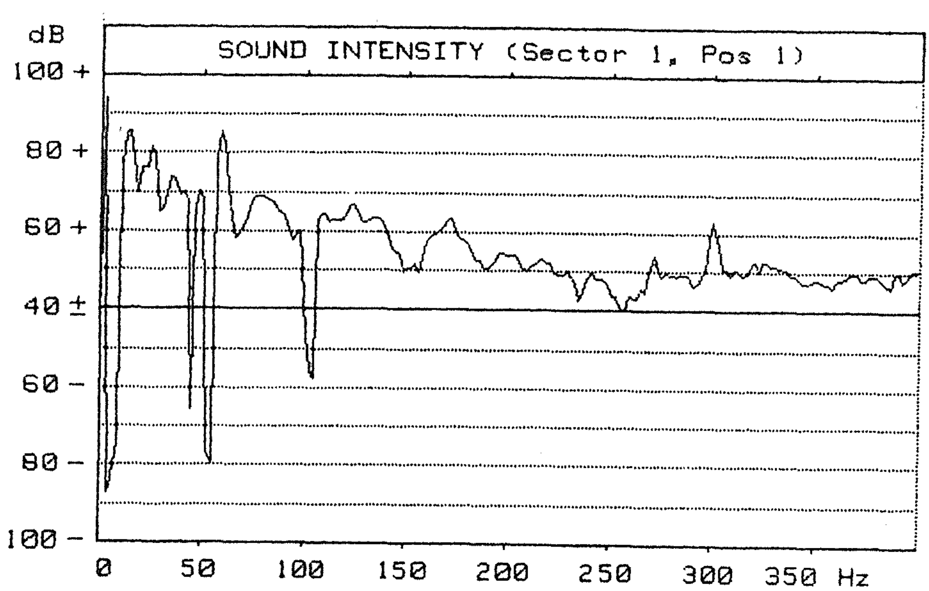

Figure 5 Narrow band sound intensity spectrum over fan number 1.

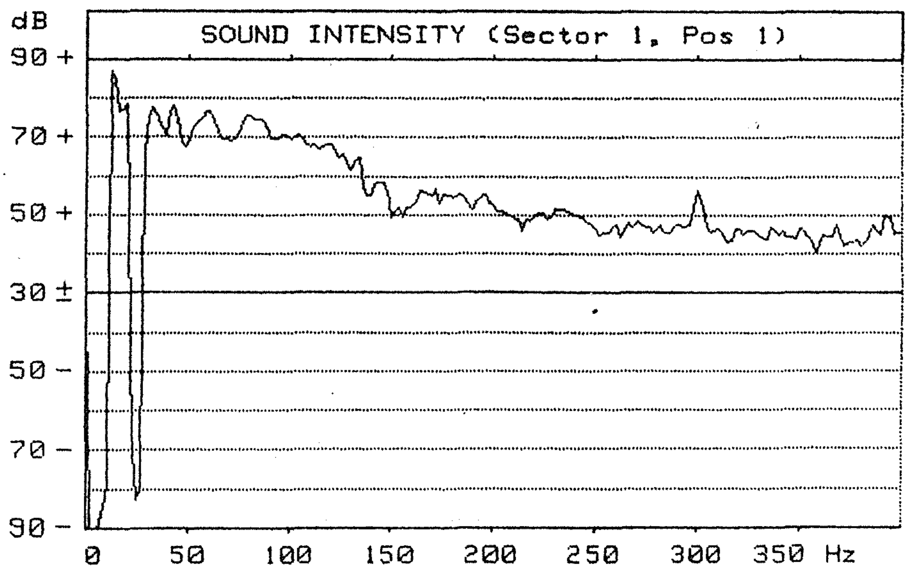

Figure 6 Narrow band sound intensity spectrum over the air intake section.

5.2 Streamwise velocity and turbulence intensity profiles

Streamwise mean velocity and turbulence intensity profiles were made in the supply ducts at 350 mm downstream of fan no.1 and no.2 by traversing a hot-film probe in a midplane parallel to the longer dimension of the supply duct. The results of these traverses are displayed in Figure 7. Here H is the half-width of the supply duct and y/H = 0 corresponds to the centreline of the duct. The results indicate that the two fans may not be properly balanced as the flow rate in the two ducts is quite different. As shown in Figure 8, the streamwise turbulence intensity is at least 30% over 60% of the cross-section of the supply ducts. The two fans appear not to be well balanced. The streamwise turbulence intensity spectrum is typical of that published in the literature. Discrete frequency peaks near 50 - 63 Hz, however, cannot be discerned.

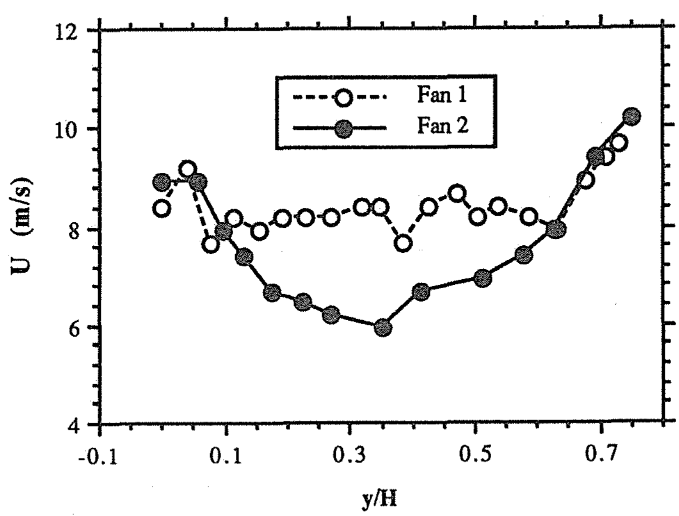

Figure 7 Mean streamwise velocity profiles in the outlets of fan no. 1 and no.2.

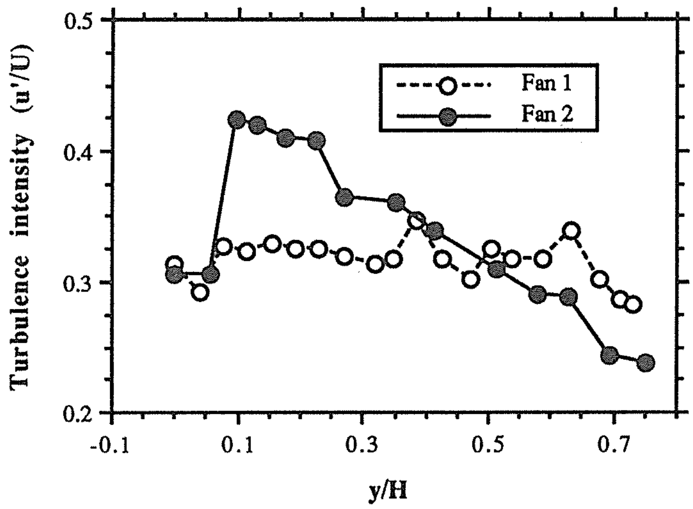

Figure 8 Streamwise turbulence intensity profiles in the outlets of fan no. 1 and no. 2.

5.3 Efficiency of Fans

The fan shaft rotational speed was measured to be 1070 rpm. The static pressure drops across fan no. 1 and no. 2 were 520 and 532 Pa respectively. By integrating the streamwise velocity profiles in Figure 7 and by assuming uniform flow, the volume flow rate was deduced to be 3.63 m^3/s. The electric motor was rated as 5.5 kW at 1450 rpm. By assuming an efficiency of 90% for the pulley belt drive between the motor and the fans, the fans have been estimated to be operating at an efficiency of 40%.

5.4 Vortex shedding of fan blades

Studies of vortex shedding for circular cylinders reported by Roshko (1953) indicate that for a Reynolds number range of 400 - 200,000, the Strouhal number is approximately 0.21. Although the fan blades are more aerofoil in shape, they are at high angles of attack. It is, therefore, not unreasonable to assume that the vortex shedding for the fan blades will occur at around a Strouhal number of 0.21.

Figure 9 shows a schematic of the fan blades and the inlet velocity triangle. For a rotational speed of 1070 rpm, the blade velocity u is 20.6 m/s. For a volume flow rate of 3.63 m^3/s, the radial velocity v is about 9.5 m./s and the angle of incidence α is about 106^o, thus yielding a velocity w relative to a fan blade of about 22.7 m/s.

Since the angle of incidence of the flow to the fan blade is large, the fan blade can be treated as a bluff body with its chord length (33mm) as the relevant length scale. In order to account for the effects of neighbouring fan blades on the vortex shedding frequency, it is assumed that only a blade on either side of the blade of interest is important so that the effective length scale has been deduced to be 55mm. Hence, with a Strouhal number of 0.21 and a relative velocity of 22.7 m/s, the vortex shedding frequency of a fan blade corresponds to about 87 Hz.

It is, therefore, postulated here that the main source of low frequency noise generation is due to noise arising from vortex shedding and coalescing. This discrete frequency, however, cannot be detected at the measurement locations as it is superimposed on a highly turbulent flow. Nevertheless, the sound generated at this frequency propagates along the main supply duct into the open-plan office.

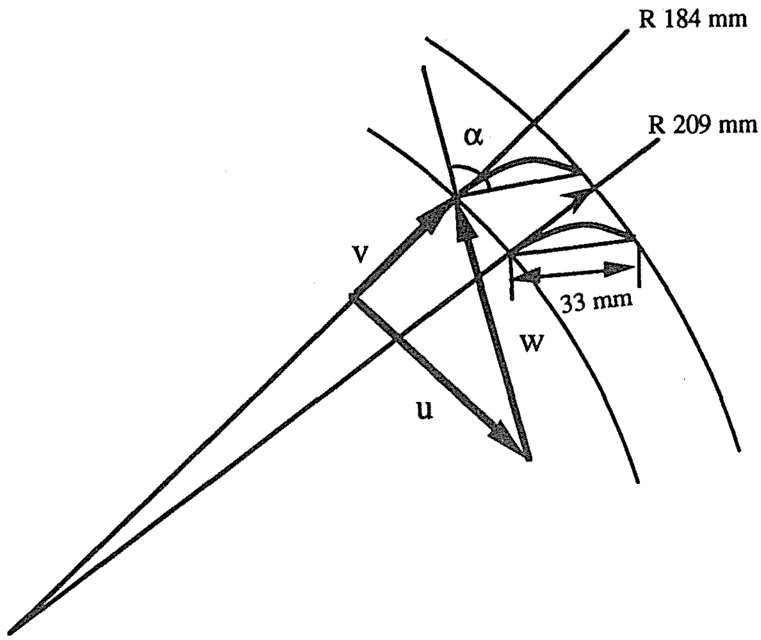

Figure 9 Schematic showing fan blades and velocity triangle.

6.0 CONCLUSIONS

Sound intensity measurements and flow measurements using hot-wire anemometry for identifying the low frequency noise in a ventilation system have been described.

The sound intensity measurements in the plant room indicate that the low frequency noise is generated by the fan. It has been postulated through calculations that this low frequency noise is primarily due to vortex shedding and coalescing. The flow within 350 mm of the outlet of the fan has been found to be highly turbulent, which tends to enhance the low frequency sound generated at the blades. .

Field transmission loss measurements of the plant room wall and sound intensity scan in the open-plan office indicate that the low frequency noise generated in the plant room is transmitted into the office space primarily through the main supply duct.

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ROAD TRAFFIC AND INTERIOR NOISE

A SURVEY OF NOISE LEVELS IN HOUSES EXPOSED TO TRAFFIC NOISE

Sue Macalpine and Stuart McLachlan

NSW State Pollution Control Commission

1. Introduction

Road traffic noise is a major environmental problem wherever high volume roads exist. A recent estimate of human exposure to traffic noise based on OECD standards indicated that 350,000 people are seriously affected and a further 1.25 million are moderately affected in Sydney alone.

Over the years considerable effort has been made throughout the world to reduce the problem with more stringent vehicular noise controls. The gains through reducing noise at the source have not been enough to reduce noise to acceptable levels in every instance. So non vehicular techniques are considered to be an important and urgently needed supplement to the vehicle noise controls. These non vehicular methods include better road construction techniques, better planning and development control and the use of traffic management techniques to reduce noise.

However, regardless of the overall effectiveness of these controls, there will still be a large proportion of people living along busy roads for which such strategies may have limited effect. For these people the design of their house plays a vital role and last line of defence in determining the noise environment that they are exposed to. The aim of this paper is to briefly examine and quantify the traffic noise levels inside some houses situated near busy roads.

One of the houses evaluated in this paper was the focus of the Commission's Quiet House education project. The house was designed to keep out traffic noise through the use of commonsense acoustical concepts and the limited use of acoustical materials. The aim of the project was to demonstrate that quiet houses can be built at little additional cost. The house is the first prize winner of an architectural design award competition. It was subsequently built with the help of private industry and was on display to the public in 1989. Sponsors contributed most of the building materials and also furniture, fittings, carpets and appliances so that during the exhibition period, the home was fully furnished and the grounds totally landscaped.

A comparison of internal noise levels is made with the use of Australian Standard 2107-1987 "*Ambient Sound Levels For Areas of Occupancy Within Buildings*". The paper briefly discusses typical design deficiencies and reviews possible remedies.

2. The Study Method

Internal noise levels at six Sydney dwellings located on roads with daily volumes from 5500 to 38400 were examined. The houses were selected to represent a number of architectural periods from a dwelling constructed in the 1920's to a house constructed and specifically designed to reduce traffic noise, the Quiet House. Measurements were taken over a reduced period of 3 hours as provided in the CORTN method.

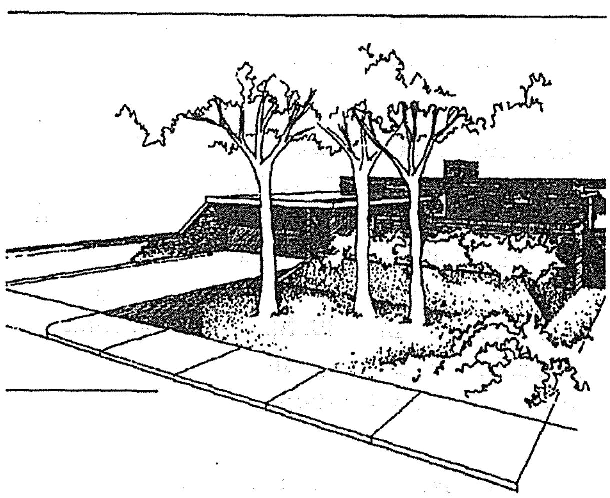
It was realised that use of this approximation could produce some small error, however, it was felt that 24 hour measurements would be an imposition on the residents and cause the measuring of unwanted domestic noise.

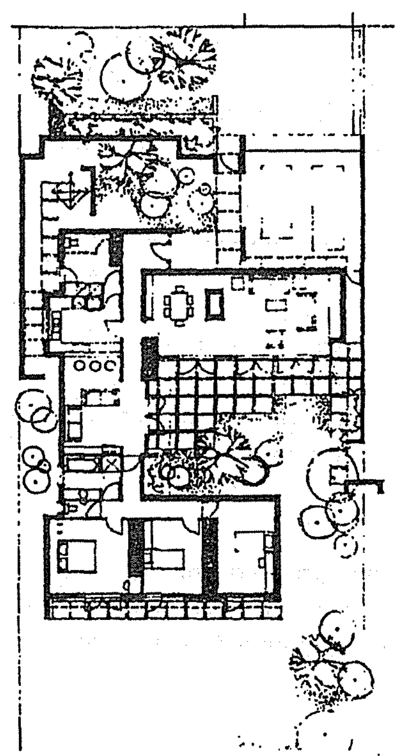
3. The Measurement Sites

The location of sites chosen for measurement and some site and traffic flow information are mentioned below:

S1 - The Quiet House - 156 Pennant Hills Rd, Dundas.

The plan and elevation of this house are shown. The facade of the house is 13m from the road. Annual average daily total (AADT) traffic flow for both directions is 34,000.





S2 - 154 Pennant Hills Rd, Dundas.

The residence is immediately beside the Quiet House. It is a brick building built in the 1920's when Pennant Hills road was a small country road. The facade of the house is 12m from the road. Annual average daily total (AADT) traffic flow for both directions is 34,000.

S3 - 153 Pennant Hills Rd, Carlingford.

A contemporary brick home. The facade of the house is 26m from the road. Annual average daily total (AADT) traffic flow for both directions is 38,436.

S4 - 20/1-11 Pennant Hills Rd, Wahroonga.

A town house situated between Pennant Hills Rd, the Wahroonga expressway and the old Pacific Highway. An earthmound is situated between this townhouse and Pennant Hills Rd. The windows of the bedroom were slightly open during measurement and overlooked the expressway in the distance. The facade of the house is 24m from the road. Annual average daily total (AADT) traffic flow for both directions is 32,200.

S5 - 28 Hill St, Wentworthville.

A fibro house with its rear adjacent to Hart Drive. A fence and garage partially obscure line of sight to the road. The facade of the house is 11m from the road. Annual average daily total (AADT) traffic flow for both directions is 5486.

S6 - 239 Old Prospect Rd, Greystanes.

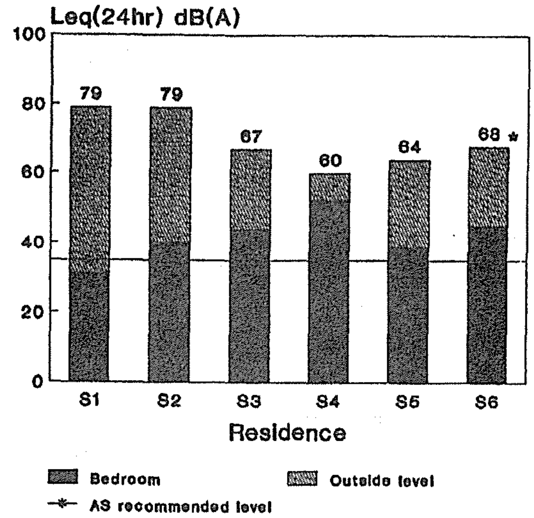
A contemporary brick home. The facade of the house is 12m from the road. Annual average daily total (AADT) traffic flow for both directions is 11,310.

4. Discussion of the Measurements

The comparison between the measured bedroom levels is shown in Figure 1. Comparison of these results with the Australian Standard 2107 (anon, 1987) shows that, with the exception of the Quiet House, all the bedroom levels were well in excess of the recommended maximum level of 35dB(A). These high levels are of concern as they can result in sleep disturbance and a deterioration of sleep quality. Langdon reports that effects of traffic noise on sleep have been noted from 40dB(A), (Langdon, 1975). He found that traffic noise can delay the onset of sleep, cause involuntary awakening and disruption of the sleep pattern.

Bedroom noise levels can be reduced by shielding and the placement of those rooms as far as possible from the road. The importance of bedroom location is clearly indicated by Figure 1. In the cases of sites S1, S2 and S5, the bedrooms are shielded by other rooms nearer the road causing substantially greater noise attenuation resulting in lower levels in the more sensitive sleeping areas.

Figure 1. Noise Levels
Bedrooms

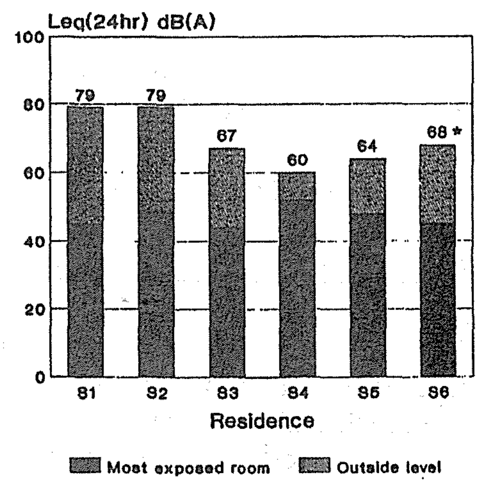


* Outside level measured at 1m from facade instead of 10m

The interior ambient noise levels in the most noise exposed rooms of the houses is graphed in Figure 2. These rooms have a variety of functions so it is not appropriate to compare them with a single recommended noise level. However the majority of these noise levels exceed the highest recommended level for any room in a residential building.

All of the residences studied have brick veneer or double brick construction with the exception of S5, which was predominantly fibro. The brick residences with windows closed yielded on average a noise reduction of 29 dB. This result compared favourably with the noise reduction of 16 dB produced by the fibro home. These reductions include a component due to distance attenuation.

Figure 2. Noise Levels
Most Exposed Room



* Outside level measured at 1m from facde instead of 10m

S4 shows an unexpectedly small noise reduction for a brick home, but, this was due to measurements being taken with open windows. In addition an earth mound had been constructed outside the residence and shielded the lower story of the town house while rooms upstairs had direct "line of sight" to the road.

Three hour measurements as described in the CORTN method (CORTN, 1988) were undertaken in both the main bedroom and the room most exposed to the road. Measurements of the outside traffic noise measurements at 10m from the facade facing the road were taken at the same time. The traffic count and composition were also monitored.

The LA10 (18hr) was calculated from the LA10(3hr) measurements in accordance with the shortened measurement procedure described in the CORTN method. The formula is given below:

$$L10(18hr) = L10(3hr) - 1dB(A)$$

$$\text{where } L10(3hr) = 1/3 \sum_{10<t<14}^{t+2} L10\ (\text{hourly})$$

The L10(18hr) data was then converted to Leq(24hr) using the accepted convention:

$$Leq(24hr) = L10(18hr) - 3dB(A)$$

5. Conclusions

A large number of people are exposed to excessively high levels of traffic noise within their homes and conventional housing design and construction practices are not sufficient to reduce traffic noise to acceptable levels.

The substantial differences between the Australian Standard requirement and the measured levels, differences up to 17 dB(A), suggest that traffic noise levels within homes of conventional design can be excessively high.

The Quiet House illustrates that it is possible to build dwellings next to major roads and still retain an acceptable living environment. As a part of the NSW State Pollution Control Commission's traffic noise control program, the Quiet House project was designed to inform all sections of the community about the viability of good housing design for traffic noise control.

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A DISCUSSION OF MEASURES BY THE ROAD AND TRAFFIC AUTHORITY IN NSW TO REDUCE TRAFFIC NOISE, PARTICULARLY ALONG THE WAHROONGA-BEROWRA SECTION OF THE SYDNEY-NEWCASTLE F3 FREEWAY.

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DISCLAIMER.

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A. INTRODUCTION

It is widely accepted that traffic noise is the No. 1 noise problem in the community, that it is the principal form of pollution from road traffic, and that it is one of the most important environmental concerns of the community. A probable reason that traffic noise has become increasingly predominant in the total spectrum of community noise is that it is an intransigent problem, for example, unlike other troublesome noise sources, traffic cannot be enclosed in a factory shell or similar, except at inordinate cost.

However, there are also some important differences between noise and other areas of environmental concern:-

(a) Unlike the Green House Effect and the Ozone Layer, it does not affect the long term capacity of this planet to sustain life.

(b) Unlike air and water pollution, it does not have major health effects, even of hearing loss, except under exceptional circumstances. The principal concern is that of annoyance, and maintenance of a peaceful environment.

(c) Measures to reduce traffic noise can have other adverse environmental effects such as increased use of fossil fuel, or loss of scenery.

(d) Although a new road is likely to reduce the total noise to which a community is subjected, nevertheless it is now a political fact that the greater political pressure will come from those adversely affected along the new road rather than from those who are already subject to high traffic noise levels, and who will benefit. This may be already threatening the future capacity of cities to function economically and efficiently.

(e) To some it appears that sections of the Australian Public are able to pressure their governments to take any measures which they perceive will improve or "save" the environment. However there is an increasing realization that economic realities also need to be considered, particularly in our present situation. Debt has been a contributing factor to the loss of rain forests in the Amazon Basin, and to farmland degradation in Australia.

    We need to focus on those measures which will achieve the greatest benefit in relation to the cost incurred.

(f) At 25%, transport costs in Australia used to be as high a proportion of total costs as in any other country, and this may still be so. It might not be most appropriate for Australia to be absolutely abreast of the leading nations in the area of traffic noise reduction.

B. DEVELOPMENT OF A POLICY BY THE RTA IN NSW ON MEASURES TO REDUCE TRAFFIC NOISE.

1. INITIAL POLICY

The NSW Department of Main Roads (now the Roads and Traffic Authority) first formulated a policy in this area in December 1982, the Wahroonga - Berowra section of the Sydney-Newcastle Freeway being seen as the first major project where this policy was likely to be applied. This policy, which nominally applied to all new roads, was based on a similar policy by the Country Roads Board in Victoria, which applied to expressways and freeways only. At the time, NSW was among the last states to formulate a traffic noise policy.

The principal points were as follows:-

(a) Provision of noise mitigation measures to be considered where L10 (18 hrs) is estimated to exceed 68 dBA at a building facade, and to have risen by 2 dBA. The word "considered" was a safeguard in the light of the DMR's inexperience. All treatments were to be considered by "Head Office", before approval.

(b) Estimation (and measurement) of noise levels was to be based on the UKDoE CORTN procedure as amended for Australian conditions. This method had been the subject of much research, culminating in the ARRB validation by Steven Samuels (Ref 1), which found an average error of underestimation of 1.7 +/- 2.5 dBA at a house. Generally this departure was disregarded, and became a safety factor, so that over-estimation theoretically occurred on most occasions.

(c) Estimated noise levels are to be based on traffic 10 years after the road is opened to traffic, taking account of the expected road network at that time.

(d) A departure from the Victorian policy was that corrective measures were not to be considered to be justified for the benefit of a property which had been purchased or built after the road proposal had been made public. In practice, this clause has only been applied at complying individual properties which were difficult to treat. The counter view, that people will be exposed to noise in any event has generally carried greater weight.

Significantly the most troublesome location on the F3 Freeway, Curtin Avenue/ Uralba Place at North Wahroonga, was a relatively recent residential development which does not comply with this criterion. An alternative land use here would have solved many problems as these streets are high above the freeway (see Figure 1), and will be difficult to shield.

(e) The policy also applied to recently completed roadworks. However complaints regarding noise from long standing roads were to be considered on their merits. This clause was not widely publicised until 1987.

(f) Cost, effectiveness and aesthetics etc. were to be considered.

(g) Corrective measures were generally to be limited to the road reserve. There was a legal principle which limited expenditure to properties where some acquisition was required.

This policy was applied initially to major projects. It was issued to Divisional Offices in January 1984, with the inclusion was that minimization of traffic noise should always be taken into account in a road design, when cost or other design aspects would not be significantly compromised.

2. POLICY REFINEMENTS

This policy was further refined in June 1985 when the following principles were enumerated. My comments are included:-

(a) Traffic noise is to be considered in choosing a road location and grading. In most urban situations, the scope for road design factors to reduce noise levels may be limited:-

   (i) Where a road is moved further from one group of properties it may be moved closer to another.
   (ii) Generally we cannot afford to sterilize land solely for the purpose of noise reduction. Vacant land which is subject to high levels of traffic noise will be less than ideal as a public reserve.
   (iii) Placing a road in cut to reduce noise levels will usually be unduly expensive.
   (iv) New urban roads have usually been planned for many years, and scope to vary their design is usually limited.

(b) Noise is to be considered in choosing a pavement surfacing.

   (i) A smooth impervious surface is likely to cause skidding accidents in wet weather.
   (ii) A textured surface will be safer but noisy.
   (iii) The purpose of open graded asphalt is to allow both water pressure and air pressure to dissipate under a tyre, reducing both noise and splash, whilst maintaining skidding resistance.

(iv) However this material is of little structural benefit. It does not seal water from the pavement. It is weak and may lose some of its void content with time, which is its "raison d'etre".

(v) There may be a need for the effect of pavement surfaces to be investigated in Australia also using loudness scales other than the "A" weighting. These generally give greater weight than the "A" weighting to noise frequencies above 2000HZ and below 500HZ .

(vi) The Authority recently commissioned the ARRB to investigate the effects of road surface texture on traffic and vehicle noise, on this project and other roads. A report is available comparing the effects of different surfacings on roadside noise levels.

(v) Pavement surfacings have least effect on peak noise levels and greatest effect on tyre drone.

(c) Dense planting should be used where this will reduce noise. Again, other loudness scales, and experience indicate that this may be more effective than is indicated by "A" weighted measurements.

(d) Use of 2m high opaque fencing in lieu of chain wire fencing, although the latter is economical, durable and unobtrusive. The fencing used on the freeway was of treated pine, 40mm thick as shown in Fig. 2.

(e) Use of surplus fill to provide earth mounds. In practice if earth mounds would be of benefit, the road design would usually be modified to provide this material.

(f) The location of the boundary fence is to take noise levels into account, eg. at the top of cuts. Where appropriate, guard rail should be "infilled" to act as a noise barrier, or "New Jersey Kerb" preferred.

(g) In general, I consider that this policy appears to have attempted to achieve what was possible, whilst avoiding the use of blatant noise barriers.

3. THE GUIDELINES ARE PUBLISHED.

In June 1987, the Department's guidelines "Road Traffic Noise" was published. This described in detail how traffic noise levels should be estimated using the CORTN procedures. The most important addition was that noise attenuation measures are "generally considered to be warranted" where the noise level will rise as follows:-

(i) by 2dBA min. to 68dBA min. (eg. along an existing route)
(ii) or by 15dBA min. to 63dBA (ie. along a new route)

It may be that the wording of this policy is less conciliatory than was necessary or than is the usual practice.

As recommended by a NSW Inter-departmental Committee 63dBA, has recently become the most generally used criterion subject to cost-effectiveness. This is only 3dBA higher than the 60dBA (L10, 18hrs) criterion, adopted by the State Pollution Control Commission for new residential subdivisions in NSW.

The above policies were developed while George Glazier was the Engineer for Environmental Matters within the DMR.

C. EVENTS FOLLOWING THE FREEWAY OPENING.

The freeway was opened on 22nd March 1989. The design incorporated the following measures:-

(a) Opaque (timber) fencing 2m high within residential areas.

(b) Earth mounds where these were feasible, with tree planting (83,000 trees), and usually with (a) above.

(c) For engineering durability, the pavement surfacing was concrete, either lightly grooved or ungrooved (hessian-dragged) in the designated residential areas which stopped short of Curtin Avenue/Uralba Place (see Fig 1). With the benefit of 20-20 hindsight, it is obvious that a decision did not give proper weight between engineering and environmental factors, and was inappropriate. It has adversely affected public concern regarding noise from new road projects planned in the Sydney area.
In addition to cost, the following factors influenced this decision:-

 (i) That open-graded asphalt, and other measures, could be installed after the opening.

 (ii) Decentralization of the RTA had only recently occurred. The ramifications of regional independence vss working as one organisation was still being resolved.

(iii) Uncertainty regarding noise outcomes, which would be resolved soon after the freeway was opened to traffic. The determination of the Environmental Impact Statement required monitoring of the noise levels after the road was opened to traffic.

Within two weeks it was clear that there was widespread public concern regarding the noise levels. The gist of these complaints were that:-

(a) The noise levels were higher than expected and were unreasonable.
(b) The best technology had not been used.
(c) Undertakings made in the EIS had not been complied with.

A program of noise measurements was set in motion, hampered significantly by weather and other problems. A series of 16 preliminary (short CORTN) measurements in mid April were between 59 and 64 dBA with one at 70 dBA (Curtin Av) and one at 68 dBA (1st floor, Chelwood).

Measurements in May were as follows:-

North Wahroonga. 17 Sites, 2 at 69, 13 at 59-65dBA, 2- 56 dBA.

Mt. Colah. 16 Sites, 1-67 dBA, 2-66 dBA, 1-64, 11 at 59-63 dBA, 1-53.

Wahroonga. 17 Sites, 1-67 dBA, 4-66, 2 at 65, 11 at 59-62.

By the 15th of May an undertaking had been given as follows:

(i) For a 1.2km trial length of open graded asphalt to be laid at North Wahroonga, for assessment of the noise level reductions along this length. The ARRB study, previously mentioned was commissioned to examine noise levels on pavement surfacings at this and 11 other sites, and an examination of overseas literature regarding these surfacings was made.

(ii) Modelling of 11 sites to assess the effectiveness of higher noise barriers, using the CORTN and ENM models.

(iii) A 2.8m high noise fence was erected along the road shoulder at Uralba Place, at the request of residences, despite concern that it would not be effective (see Fig 2). In fact, this barrier was reported to have slightly increased the noise levels opposite, by reflection.

In mid June, the speed limit was reduced from 110kph to 80 kph for 6 km, and a commitment had been made to lay open graded asphalt from Mt. Ku-ring-gai south, 6km in total, through the residential areas, a cost of about $2 million.

After further evaluation, a commitment was made to increase the height of the 2m noise fences to 4m at a cost of several million dollars. Again, treated pine panels are proposed in the main, but with steel RSJ posts and improved gap sealing. This was made on the basis of night-time noise levels, as L10(18hrs) had or would have been generally reduced to below 63dBA by the measures proposed.

At Curtin Avenue and Uralba Place, the final design of noise reduction measures is still not yet resolved. The use of concrete and/or absorptive barriers is being considered.

D. CRITERIA

In designing noise treatments, one is confronted by a range of noise levels at various properties for a range of noise barrier scenarios, a range of shades of grey. How high or how far should one go? The decision how much to reduce traffic noise occurs at every barrier situation.

(1) Do nothing. This policy gave trouble free operation for many decades at minimal cost. Except where a route was formulated and constructed soon afterwards, roadside residents did not perceive undue hardship, as the road project was known well in advance.

(2) On the other hand, there is a legal principle that where an activity benefits a whole community, then an unreasonable burden should not fall on a small group.

(3) Compliance with criteria such as 68 dBA or 63 dBA are general rules, applied after consideration of overall economic factors. However this approach has some disadvantages.:-

  (i) It may be easy and very worthwhile to achieve better than 63 dBA at many locations.

  (ii) For some properties, it will be most difficult to achieve 63 dBA.

  (iii) Government bodies are under constraint for expenditures to return a greater benefit. Consider a hypothetical situation where there is one house on one side of a road and 200 on the other. Should both be given the same treatment? Common sense and economic reality dictate that the 200 houses should benefit from a higher noise barrier.

(4) To overcome the shortcomings of (3), then perhaps some form of mathematical model can provide an answer. This has been attempted over the years, along the following lines:-

(i) The results of many studies worldwide of property values has concluded that each dBA rise in traffic noise above 50 dBA will reduce property values by 0.5% on average, but that this rate should rise with the noise levels. Ref 2 summarizes the present position, however local data could be used.

(ii) In a paper (Ref 4) at the 1988 Conference (of this Society) I amplified (i) above, and proposed that roadside scenery may be of generally greater value than the cost of the noise barriers and should be considered. For example 40,000 vehicles per day at $1 per hour (night time vehicles will offset the number of occupants per vehicle) and 80 kph would indicate a scenery value of $183,000 per annum per km. or $2.6 million/km at a 7% discount rate ie. $1,300 per metre of roadside. If 100 residences per km were affected, this becomes $26,000 per residence or $35 per week at 7% discount rate. At many sites, a noise barrier would not obliterate the roadside scenery completely.

If only say 10 houses are affected then roadside noise barriers may be inappropriate. Some would argue, on the basis that the user should pay, that barriers should be erected in this situation, as apparently occurs in some European countries. However, this would be expensive in relation to the net benefit. It is likely, in fact, to cause a net deterioration to the environment, and we are all road users at some time.

E. STUDIES ON THE F3 FREEWAY

In parallel with the previously outlined announcements the Authority has initiated three other investigations:-

(a) A "before" and "after" study of noise levels along the freeway and along roads which previously carried the freeway traffic.
(b) A before and after noise opinion survey, in conjunction with (a).
(c) Various studies in conjunction with the retrofitted noise treatments.

All these studies have been or are being carried out by independent consultants.

F. RESULTS OF INVESTIGATIONS

The "before and after" noise and social surveys are not yet completed. Anyone who is familiar with the many alternative routes, along normal urban roads through Hornsby, will be aware of the considerable potential which is available for a net reduction in community exposure to traffic noise, due to this project.

Preliminary indications are that along the Pacific Highway routes, L10 (18hrs) fell from 74 dBA to about 64 dBA, and was also reduced along other routes. Along the freeway before levels of 40 dBA (L90) and 50 dBA (L1o,18hrs) were raised to below 63 dBA (L10,18hrs), except for the outliers previously mentioned, however night-time levels remain the primary community concern.

Two investigations have been carried to some degree of finality:-

(a) The effect of open graded asphalt was measured to be 4.5 dBA average when compared with shallow-grooved concrete.

It should be noted that the 1988 CORTN procedure gives greater attention to the effect of pavement surfacings, and is now complicated by the need to measure the pavement texture using the sand patch method, for which data is not yet available. At this site, there is a very high number of heavy trucks. Accordingly it is unlikely that open graded asphalt would be more than 2.5 dBA more quiet than dense graded asphalt.

It is possible that the strong public reaction to the lightly grooved concrete is a strong indication that sole use of the "A" weighting to assess the effect of pavement surfacings may be unsatisfactory.

(b) The effect of reducing the speed limit from 110 kph to 80 kph reduced noise levels by 2.5 dBA, average speeds having been 115kph and 90kph respectively. The CORTN model indicates almost 2 dBA reduction between these speeds. At Eden Drive, Asquith, the measured reductions were 3-5 dBA during the day and 2-4 dBA at night on grooved concrete. These reductions are likely to have been less pronounced over open graded asphalt.

This speed reduction theoretically delays traffic by 1 minute. At 50,000 vehicles per day, and $10 per vehicle per hour, the cost of this delay was $3 million per annum, or $40 million at 7.5% discount rate.

G. PERCEPTIONS OF BARRIER EFFECTIVENESS:

A persistent public reaction was that the 40mm thick timber barriers were not stopping the noise. Ref 5 indicates that 7.5 - 1.5 kg/ sq m (0.9 - 1.7mm thick steel) is satisfactory for barriers up to 6m high. CORTN design data is similar.

Consider a wall material which attenuates all noise by 20 dBA. This is only 1% of the original noise energy. In theory, someone immediately behind the barrier would hear the noise at least a quarter as loud, due to the noise coming "through" with other noise coming "over", and is likely to perceive the barrier as being ineffective.

However at the normal position of the nearest houses, 15-50m from the barrier, the noise energy passing through the barrier is clearly inconsequential in relation to the noise energy passing over.

This was a significant aspect of public reaction to the barriers.

H. HIGH TRUCK VOLUMES, PARTICULARLY AT NIGHT.

The LEN (Less Expressway Noise Inc) Survey indicated 53% of respondents to their Community survey indicated sleep disturbance and 60% night time noise as being the principal noise problem.

A most important factor was that the heavy truck volumes (3 or more axles) were much greater than anticipated. Ref 3, P14 indicates that these rose from 7% of 43,000 v.p.d. along the Pacific Highway to 12% of 43,000 v.p.d. along the freeways with 1100 medium and heavy vehicles between midnight and 6am which was 50-60% of all vehicles. The source of these extra trucks is unclear.

Under the TNL criterion, being proposed by the SPCC in NSW for night time noise levels, (TNL = Leq (24 hrs) + 0.1N where N is the hourly volume of night time heavy trucks), the heavy vehicle correction would be 17 dBA, which appears to be quite unrealistic.

At North Wahroonga/Curtin Avenue, the most troublesome location, the freeway gradient results in slow trucks at full power uphill and the use of air brakes downhill. Noise levels higher than CORTN estimates are believed to be principally associated with high numbers of heavy trucks, steep gradients being a further contributing factor.

Truck exhaust noise, and its emission height made a troublesome contribution to peak noise levels, whilst being less predominant in the total traffic noise.

I. CONCLUSIONS.

The opening of this freeway section resulted in considerable, if not unprecedented public reaction to the traffic noise.

Within a few months, additional abatenant measures were provided which reduced noise levels to below generally accepted criterion levels, albeit at a cost of several million dollars.

This paper describes events leading up to and following the opening of this freeway section.

The paper underlines the difficulty of satisfying all public perceptions in an appropriate, practical and economic manner.

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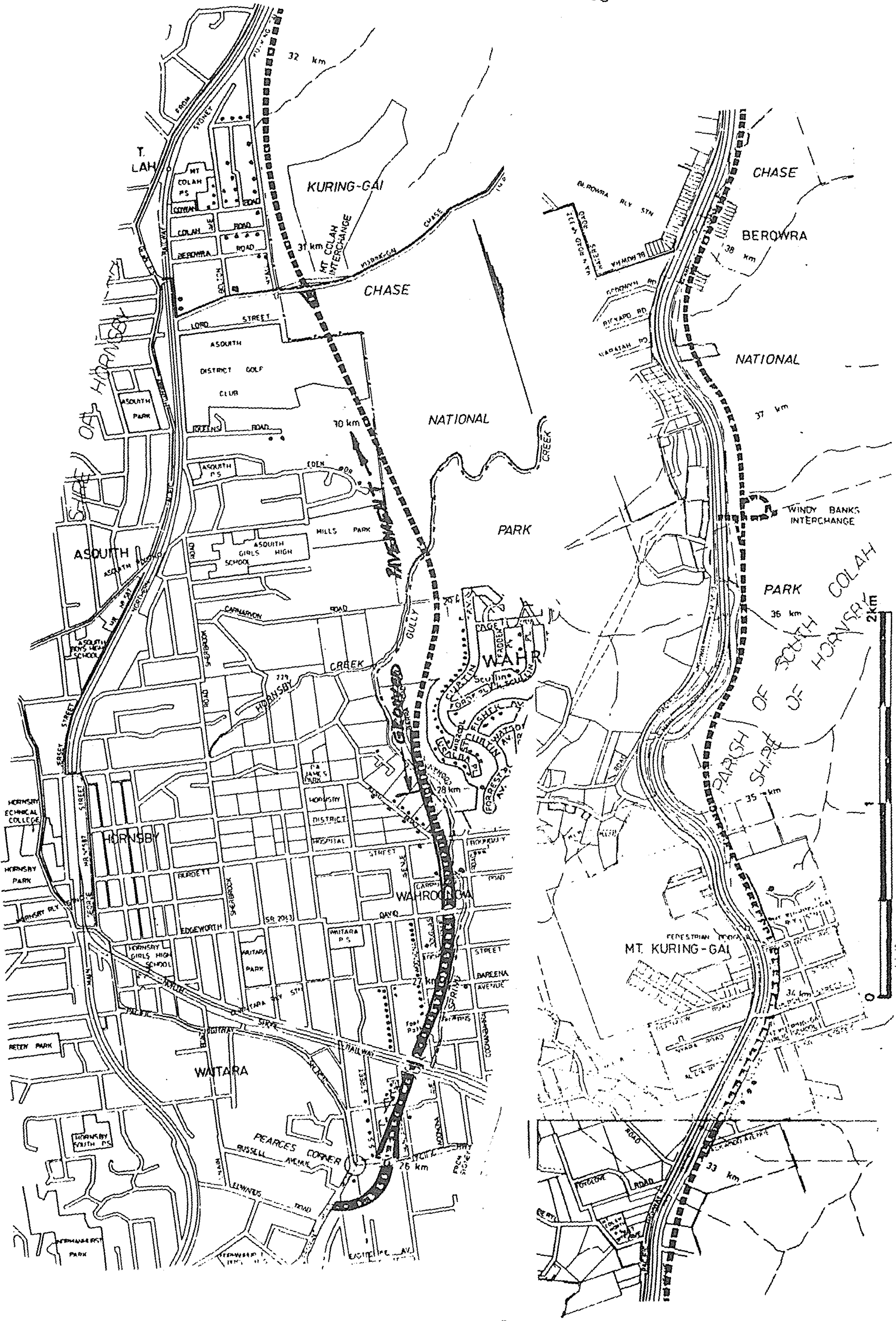

SYDNEY NEWCASTLE FREEWAY
Wahroonga–Berowra Section
FIG 1. LOCALITY PLAN

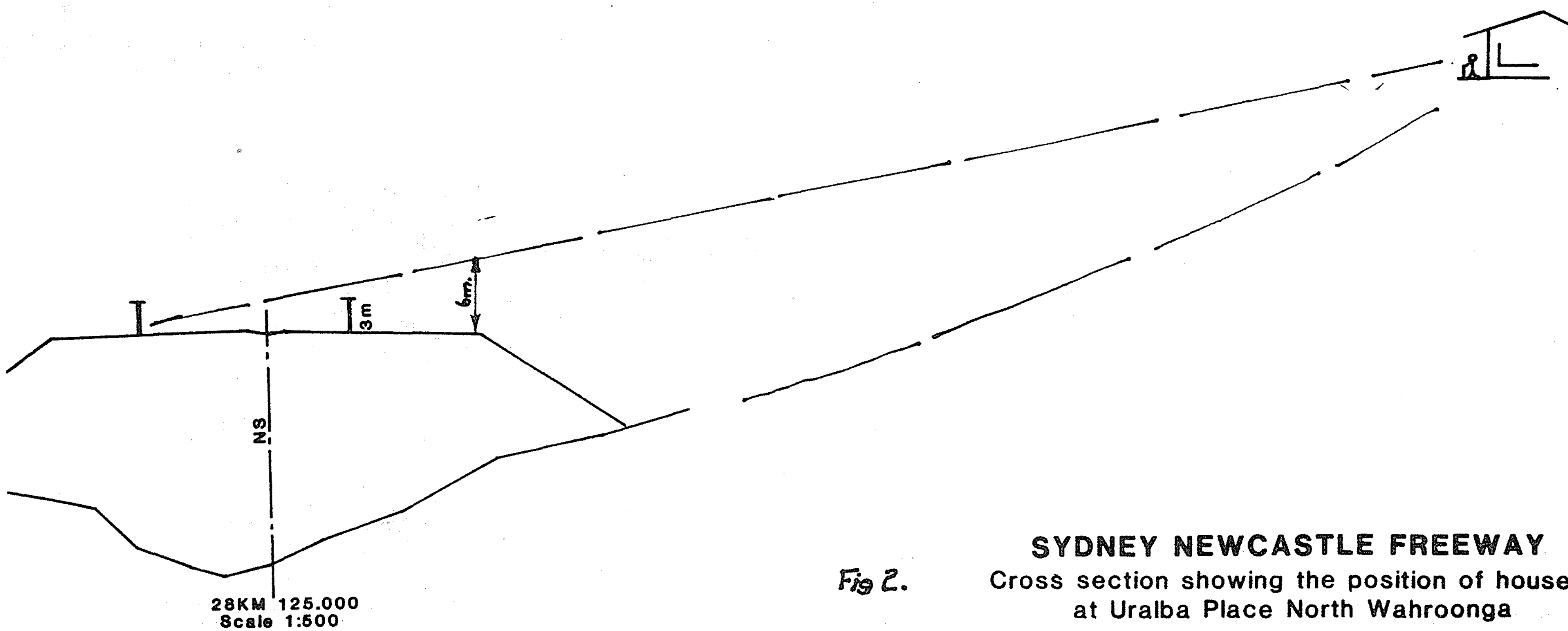

Fig 2.

SYDNEY NEWCASTLE FREEWAY

Cross section showing the position of houses at Uralba Place North Wahroonga

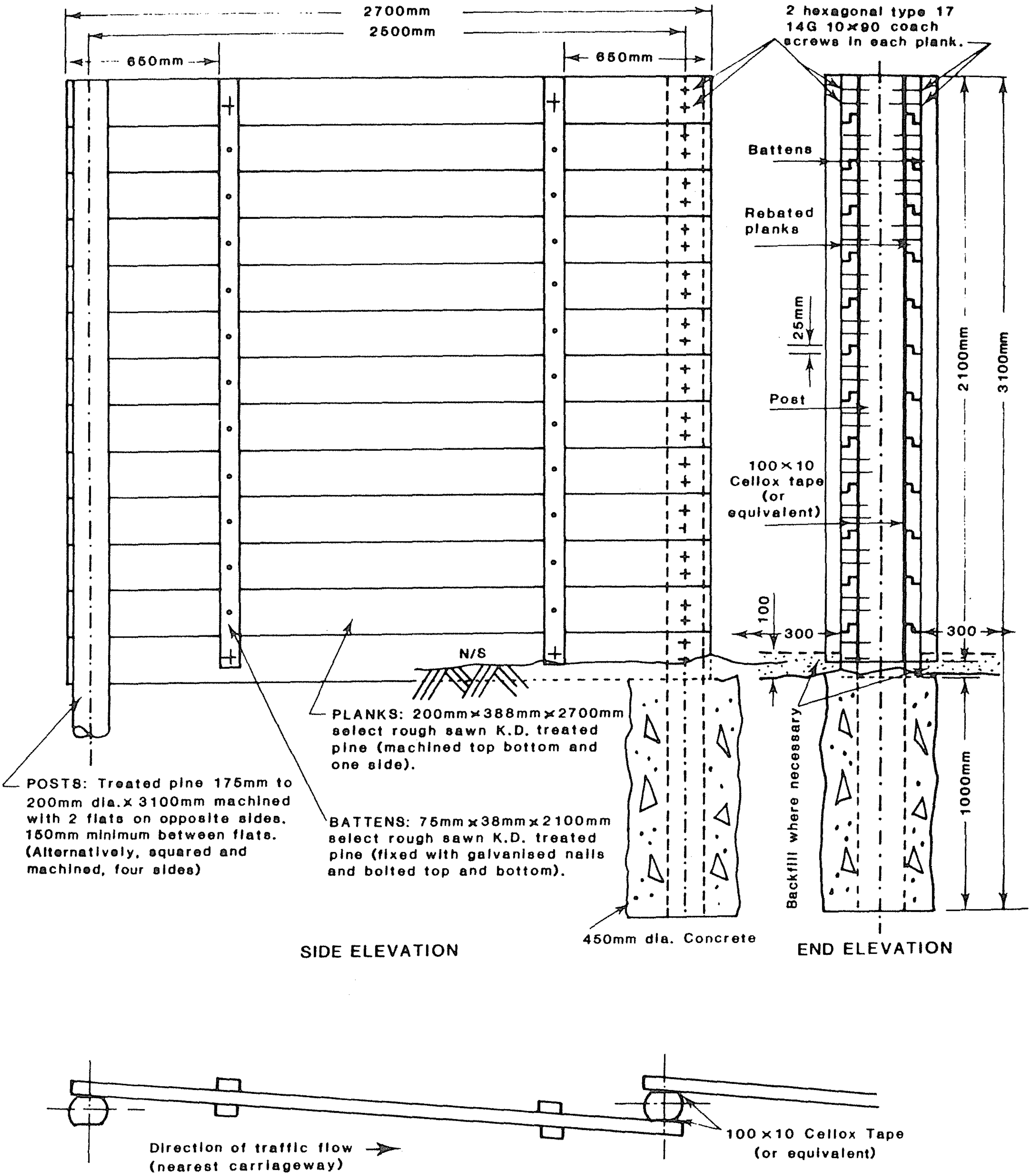

FIG 9. TYPICAL NOISE BARRIERS

1989/1990 AAS Conference, Perth

FIELD MEASUREMENT OF TRANSMISSION LOSS USING THE SOUND INTENSITY TECHNIQUE

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ABSTRACT

In newly completed buildings, or following modifications and alterations, it is often necessary to determine if the dividing wall meets the required acoustic specifications. The conventional methods for the determination of the field transmission loss are complex and time consuming and normally only the measurement of noise reduction is attempted. The sound intensity method can be used for determination of the field transmission loss and offers the advantage of providing the transmission loss data for elements of walls, such as doors and windows. The acoustic characteristics of the receiving room are not critical - if necessary absorption can be added and this is much easier than providing a reverberant space. The sound intensity method has shown good correlation with the conventional methods for laboratory test procedures.

The results from some field sound transmission loss measurements of composite partitions using the sound intensity technique are presented. The importance of checking that the pressure-intensity index of the measurement is within the dynamic capability of the measuring system is discussed.

1.0 INTRODUCTION

The sound transmission loss and the sound absorption characteristics of building materials are important characteristics which are used by the architects, designers and acoustic consultants involved with new building constructions and with alterations to existing buildings. The sound transmission loss of a dividing partition is important when the transmission of sound from one area to another must be controlled. The sound absorption characteristics of the materials in the space are important when it is necessary to control the distribution of sound within that space.

The conventional methods for determining these properties of building materials are defined in Standards (for example, AS1191-1985). For the determination of transmission loss (TL), the methods fall into two categories:

- laboratory methods which require the test partition to be installed in a specially designed specimen holder between two reverberant rooms
- field methods which specify the procedures to be adopted when the partition to be tested is installed in a building

The advent of sound intensity analysing systems offers opportunities for the development of new measurement procedures for the determination of these characteristics. The main advantage of using sound intensity measurements is that a direct measurement of the sound power which is incident upon, reflected from or transmitted through the material, can be obtained. Thus the characteristics of the source and receiving spaces are not as critical as for the conventional methods where the sound energy is determined from sound pressure level measurements. In particular, field measurements of sound transmission loss of individual components of a partition become feasible.

There have been numerous studies on the measurement of sound transmission loss of building materials under laboratory conditions using the sound intensity method (see, for example, Halliwell & Warwick (1985), Cops & Wijnants (1988)). A detailed literature review on the use of the sound intensity technique for the determination of sound transmission loss under laboratory conditions has been made by Burgess (1989). In general, the results show reasonable agreement between the conventional method and the sound intensity method.

The objectives of this on-going study are to apply the sound intensity technique to field measurements of sound transmission loss and to discuss the advantages and limitations of the method.

2.0 THEORY OF SOUND TRANSMISSION LOSS MEASUREMENTS

2.1 Conventional Method

The Sound Transmission Loss (STL) is defined as (AS 1191 - 1985):

$$STL = 10 \log 10 (W_s/W_r) \quad (1)$$

where W_s is the sound power incident on the partition, in Watts
W_r is the sound power transmitted through and radiated by the partition, in Watts

It can be shown (see, for example, Halliwell & Warnock (1985)) that if both the source and receiving rooms are diffuse, then the STL of a partition with surface area S can be expressed in terms of the sound pressure level in the source room (L_{ps}) and in the receiving room (L_{pr}) as

$$STL = L_{ps} - L_{pr} + 10 \log (S/A) \quad (2)$$

where A is the absorption of the receiving room.

2.2 Sound Intensity Method

To date no standard procedures have been specified for the determination of STL using measurements of sound intensity. In terms of sound intensity, equation (1) becomes:

$$STL = 10 \log_{10}(I_i / I_t) \quad (3)$$

where I_i is the incident sound intensity, W/m^2
I_t is the transmitted sound intensity, W/m^2

This can be expressed as:

$$STL = L_{Ii} - L_{It} \quad (4)$$

where L_{Ii} is the incident sound intensity level, dB
L_{It} is the transmitted sound intensity level, dB

As the test partition usually forms part of a room enclosure, it is often more convenient to determine the incident sound intensity from measurements of the space averaged sound pressure levels L_{pi} in the source room. Using the relationship for a diffuse field and the normal values for density of air and the speed of sound, equation (4) becomes:

$$STL = (L_{pi} - 6) - L_{It} \qquad (5)$$

From the comparative measurements on the same wall, Halliwell & Warnock (1985) found that the STL values determined using the sound intensity technique were lower at the low frequencies and higher at the high frequencies than those determined using the conventional technique. They suggested that the discrepancies at low frequencies related to the increase in energy density at the surfaces and junctions in the room because of the interference between the incident and reflected waves. This was pointed out by Waterhouse (1955) who determined a correction:

$$10 \log (1 + S_s\lambda / 8V_s) \qquad (6)$$

where λ is the wavelength of sound at the centre frequency of the filter band
S_s is the total surface area of the source room
V_s is the volume of the source room.

This "Waterhouse correction" was applied to their receiving room values for the conventional test methods to obtain an estimate of the radiated power comparable with the measured intensity. The correction was greater at the low frequencies and, when applied, the agreements between the two methods improved, although the differences at the higher frequencies remained. The authors suggested that the high frequency differences could be related to the the finite difference error and better agreement could be gained by repeating the tests with smaller microphones at a closer spacing.

3.0 EXPERIMENTAL SET-UP AND INSTRUMENTATION

3.1 Conventional Method

Field transmission loss (FTL) measurements of the partition (which includes the wall (W) and the door (D)) between the source room and the receiving room were made according to the Australian Standard AS1191-1985 in a set-up as shown in Figure 1. The masonry wall (W) has a layer of plasterboard attached to it with an air gap in between. The source room has dimensions 9.6 m x 5.16 m x 5.55m with a reverberation time of 0.7s at 500 Hz. The receiving room has a total volume of 2410 m^3 with a reverberation time of 1.2s at 500 Hz. A Bruel & Kjaer (hereinafter referred to as B&K) 4224 sound source was used in the source room (as shown in Figure 1) to generate a wide band noise with an overall linear sound pressure level of approximately 100 dB. Sound pressure levels in 1/3 octave bands were made with a B&K 2231 precision sound level meter in the source room and in the receiving room. The field transmission loss of the partition was then calculated from equation (2).

3.2 Sound Intensity Method

Field transmission loss (FTL) measurements for two doors (labelled D and OD), the door frame (labelled DF) of door D, one masonry wall (labelled W) and a louvre (labelled L) within door OD were made by using the sound intensity technique in an arrangement as shown in Figure 1. One side of the louvre is covered with a 1 mm thick aluminium sheet. Sound intensity measurements for the outer door and louvre were made in the open air. The field transmission loss of each building element was then calculated from equation (5).

The theory of sound intensity measurements using the two microphone and Fast Fourier Transform (FFT) technique has been well known (see, for example, Gade (1985)) and will not be described here. The sound intensity system used comprises a B&K 2032 dual channel FFT analyzer and a sound intensity probe made up of a pair of B&K 4181 phase-matched 1/2 inch microphones mounted in a face-to-face configuration. Most of the measurements were made with the microphones separated at a distance of 12 mm. The narrow band sound intensity data were processed and synthesized into 1/3 octave bands from 100 to 5000 Hz with a Hewlett-Packard series 300 microcomputer. The microphones were calibrated with a B&K 3541 sound intensity calibrator and the calibration in sound intensity was within 0.1dB of the value specified by the manufacturer.

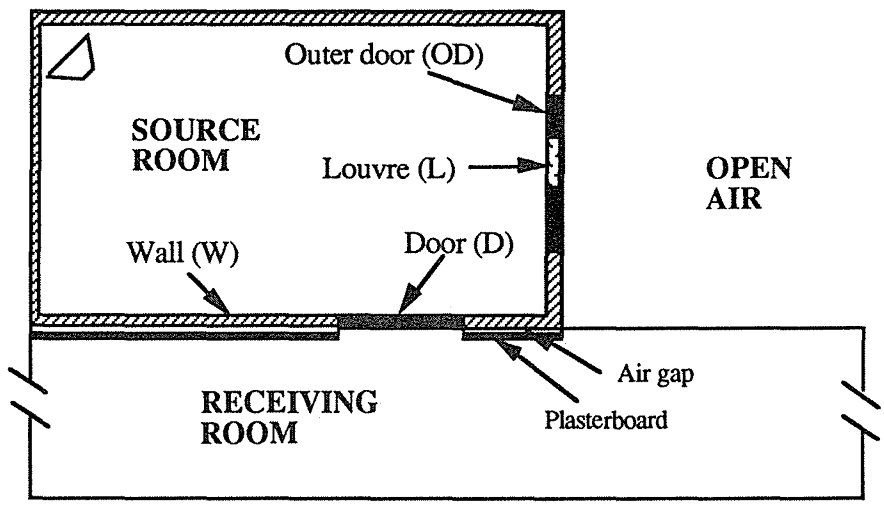

Figure 1 Set-up of source and receiving rooms showing measured elements.

The sources of error of sound intensity measurements have been discussed by Gade(1985) and are primarily due to phase mismatching between the two microphone channels at low frequencies and to the finite difference approximation at high frequencies. While the error at high frequencies can be reduced by reducing the separation distance between the two microphones, the error at low frequencies is dictated by the phase matching characteristics of the two microphone channels, which indicates the quality of the sound intensity system. It has been shown by Gade (1985) that the error L_ε due to phase mismatching is given by

$$L_\varepsilon = 10 \log [1 \pm 10^{(L_K - L_{K,0})/10}] \tag{7}$$

where L_K, the pressure-intensity index, is the difference between the measured sound pressure and sound intensity level

L_{K0}, the residual pressure-intensity index, is the difference between the pressure and intensity level when the microphones are subjected to a sound field with 0^o phase difference between the two microphone positions.

For a measurement accuracy to within 1dB, L_K has to be at least 7dB lower than L_{K0}. The dynamic capability, which is defined as (L_{K0} -7) dB, for the measurement system used is included in Figure 3.

4.0 RESULTS AND DISCUSSIONS

The sound pressure level in the source room was measured with a B&K 2231 sound level meter at various positions and the results at different positions were within 1 dB of each other over the whole frequency range.

All the sound intensity measurements reported here were made by scanning the sound intensity probe over the partition of interest and the sound intensity spectrum was obtained by averaging 200 spectra using a Hanning window with 50% overlap.

The B&K 2032 analyzer is a 800 line constant bandwidth analyzer and for a 6.4 kHz range, the resolution is only 8 Hz. which may not be sufficient for low frequency measurements. If a frequency range of 1.6 kHz is used, the resolution of the narrow band data improves to 2 Hz per line. The spacer distance and the scanning distance from the partition can both influence the accuracy of the measurements. Figure 2 shows the transmitted intensity through the door (D) for various spacer distances, scanning distances and frequency ranges.

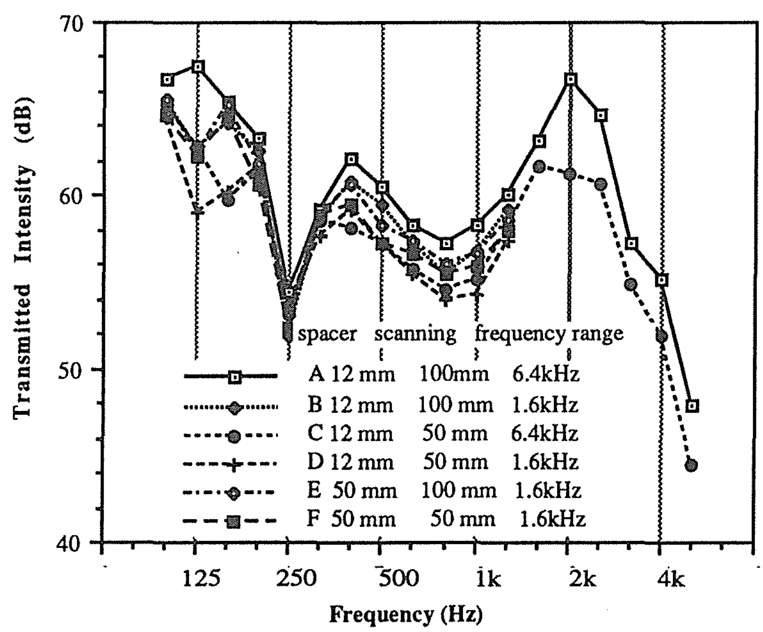

Figure 2 Transmitted Intensity Through Door (D)

For the frequency range from 200 Hz to 1.6 kHz, there is reasonable agreement among the data because the scatter is quite typical of statistical variation. The data scatter at frequencies below 200 Hz and above 1.6 kHz (particularly at 2 kHz) is quite severe. It is, therefore, important to check the accuracy of the measurements by comparing the pressure-intensity index for each set of measurements with the dynamic capability of the measurement system as shown in Figure 3. It can be seen that for tests C and D at frequencies below 250 Hz and above 1.6 kHz, the measured pressure-intensity index is either close to or greater than the dynamic capability of the sound intensity system, indicating that these measurements may not be reliable. Thus, based on this comparison, all subsequent sound intensity measurements were made over a frequency range of 6.4 kHz with a 12 mm spacer and a scanning distance of 100 mm. The pressure-intensity index for all measurements subsequently reported here was monitored and was at least 4 dB below the dynamic capability of the sound intensity system.

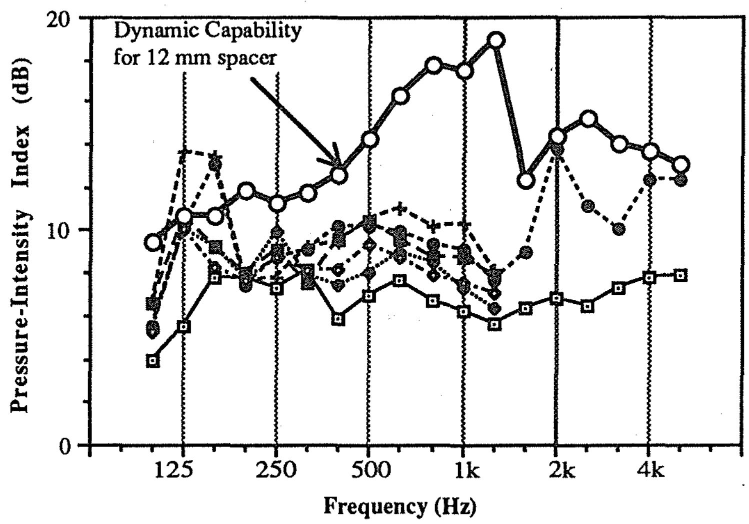

Figure 3 Pressure-Intensity Index for Measurements of Door (D)
(symbols same as in Figure 2)

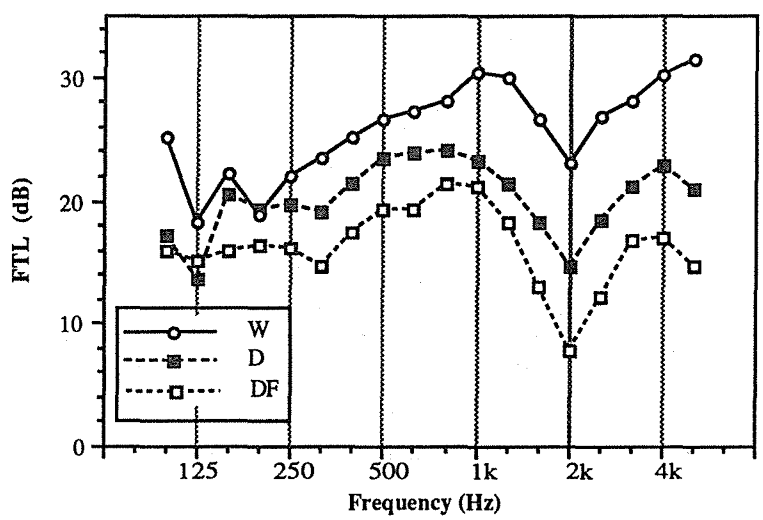

Figure 4 Field Transmission Loss for Wall (W), Door (D) and Door Frame (DF)

The field transmission loss (FTL) values determined by the sound intensity method for the wall (W), door (D) and door frame (DF) are plotted in Figure 4 and the differences between the performance of these three components can be discerned. The overall FTL of the partition between the source room and the receiving room has been calculated from the data in Figure 4 and plotted in Figure 5 for comparison with the values obtained by the conventional method. The values obtained by the sound intensity method are lower than those obtained by the conventional method at frequencies below 1 kHz, a result that is consistent with those reported in the literature for sound intensity measurements under laboratory conditions. Nevertheless, the FTL obtained by the sound intensity method follows a similar trend to that obtained by the conventional method and the extent of agreement is quite encouraging. It can be seen that for the set-up in this study, the influence of the Waterhouse correction is limited to no more than 1.5 dB at 125 Hz and is negligible for frequencies over 500 Hz.

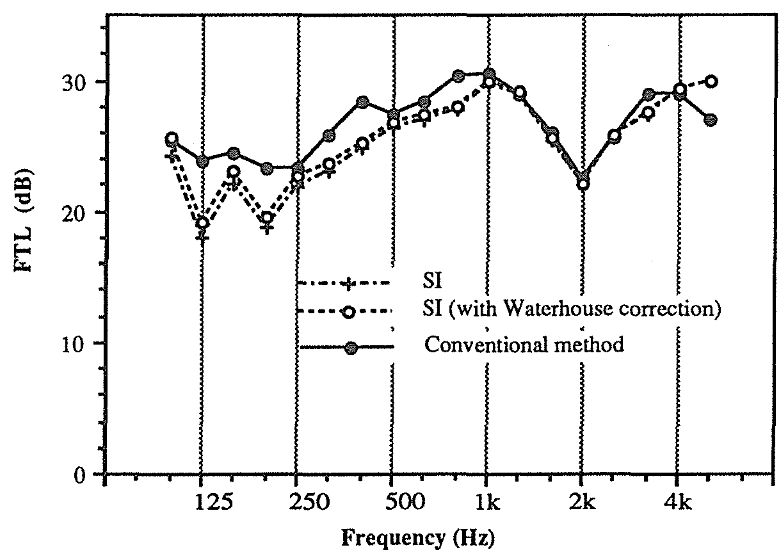

Figure 5 Comparison between Sound Intensity Method and Conventional Method

The field transmission loss values for the outer door (OD) and louvre (L) are shown in Figure 6, highlighting the potential of using the sound intensity technique in locating faulty elements in a partition.

5.0 CONCLUSIONS

Field transmission loss measurements have been made using the sound intensity technique and the results compare reasonably well with those determined by the conventional method. Furthermore, the sound intensity technique offers the advantage of determining the sound transmission loss for various elements in a partition. It is, however, important to monitor the pressure-intensity index to ensure that the measurements are made within the dynamic capability of the sound intensity system.

ACKNOWLEDGEMENTS

This study has been supported by the Rector Special Research Grant from the University College of the University of New South Wales.

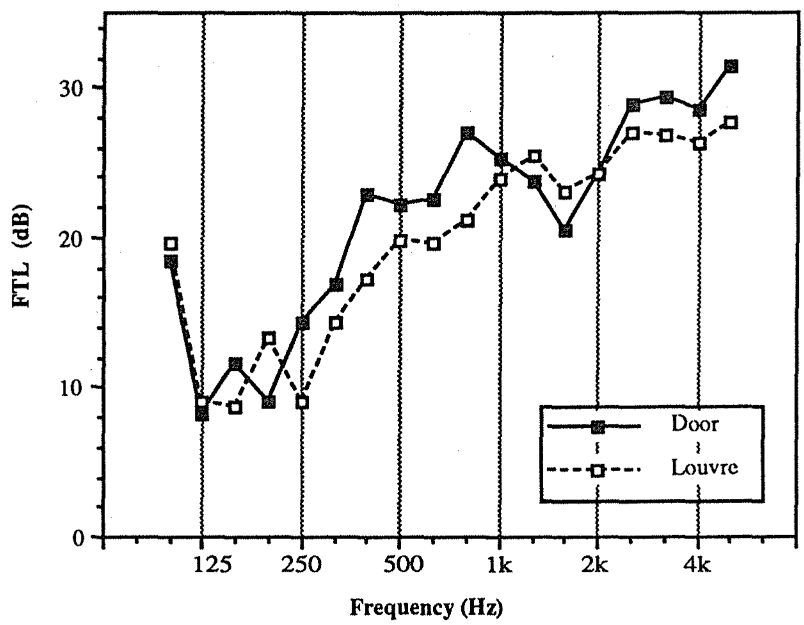

Figure 6 Field Transmission Loss for Outer Door & Louvre

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A new look at the description of acoustic enclosures

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Abstract

The traditional description of the sound field in an enclosure begins with the assumption that the walls are locally reactive and that they may be characterised by a normal acoustic impedance. This assumption allows the introduction of sound absorption coefficients and a formalism which is intended to predict adequately the acoustic response of the enclosure. The assumption of locally reactive walls and the concepts which follow from it have been applied to architectural acoustics, apparently without question, for several decades. Describing the boundaries of an enclosure as locally reactive is a satisfactory approximation only in some cases. However, the interiors of aircraft and motor vehicles are examples where this description is useless.

Experimental work carried out in a reverberation room indicates that the walls are not locally reactive and that the coupling between wall structural modes and room acoustical modes controls the reverberation time. This paper reviews some aspects of a systematic study of the modes of the sound field in an enclosure, the modes of boundary vibration of the enclosure and how each is modified by interaction with the other. The effect of this interaction upon the characteristics of the acoustic modes (in the low frequency range) and upon the average decay behaviour of the sound field and boundaries (in the higher frequency range) will be discussed. These discussions present a new understanding of the fundamental acoustical quantities and explain some of the discrepancies between different test facilities and between tests and field experience which have been known, but ignored, for 50 years.

Introduction

Fifty years ago, Morse (1939) developed a theory of sound absorption for a rectangular room in terms of the locally reactive normal acoustic impedance. The locally reactive surface is defined as one for which, on each portion of the surface, the response is only dependent on the local sound pressure and is independent of the response at any other part of the boundary. When the parts of the surface are coupled together, this surface is called extensively reactive (Morse and Ingard 1968). The assumption of locally reactive boundaries decreased the complexity of Morse's analysis, and enabled the understanding of some basic physical properties of the acoustical modes in the room. Although most of the sound absorption material tested at the time behaved in a locally reactive manner, Morse hypothesized that future experiments might expose inadequacies in the locally reactive boundary assumption.

Since then, this assumption and the concepts which follow from it have been applied to architectural acoustics, apparently without question. Even in the recent work by Ando (1985), the evaluation of subjective preference for characteristics of a concert hall assumed that walls are locally reactive. The theoretical principles arising from this assumption have provided a conceptual framework for studying the transient and steady-state behaviour of sound waves in an enclosure (Morse and Bolt 1944).

Describing the boundaries of an enclosure as locally reactive is a satisfactory approximation only in some cases. The interiors of aircraft and motor vehicles are examples where this description is useless. Even in a standard reverberation room, the locally reactive assumption for the wall surfaces can be violated (Pan and Bies 1988). It has been shown that in the latter case, the interaction between the sound field in the room and the wall structure depends upon modal coupling.

Where modal coupling exists, the sound wave behaviour cannot be described by using the locally reactive boundary assumption and it is necessary to take account of the air-structural coupling when formulating the boundary conditions of the sound field. Because the sound wave behaviour is directly related to the nature of the interaction between the sound field and its boundaries, it is necessary to explore whether this behaviour can be altered significantly by this coupling, and if so, what useful information can be obtained from a study of it.

In this paper, evidence is provided to show that the walls of a standard reverberation room are not locally reactive and that the coupling between wall structural modes and room acoustical modes controls the reverberation time. By including the air-structural coupling effect, a new analysis of the sound field in an acoustical enclosure has been developed. Some novel effects of this coupling upon the acoustical characteristics of a room is demonstrated.

I. The locally reactive assumption

The use of locally reactive assumption was first questioned because of the failure of Morse's solution to predict the modal decay rate in a rectangular reverberation room. In 1982, Munro and Bies (1982) examined a theoretical model for predicting the decay rate of individual modes. This model was Morse's solution for a very lightly damped room and relied upon the assumption that the walls can be accurately modelled as locally reactive. They also assumed that the wall impedance is uniform over the surface and is a slowly varying function of frequency. The 60 dB modal decay times of the sound field in a reverberation room were measured for comparison with this model. Figure 1 shows that this model cannot accurately predict the relative decay times of the modes.

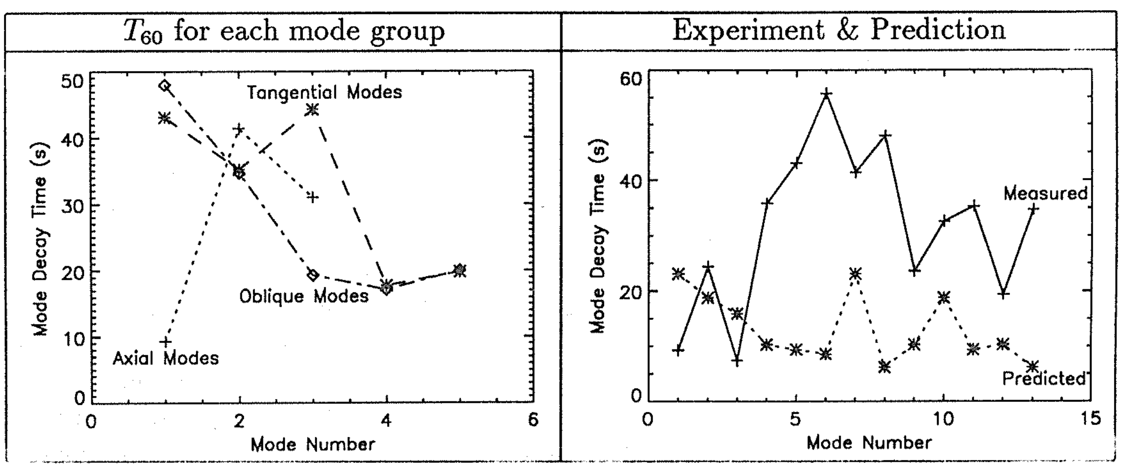


Figure 1: Measured mode decay times in a reverberation room and Morse's decay model predictions (Munro 1982).

For example, one of the interesting results from Morse's solution is that the acoustic modes in a rectangular room can be classified into seven groups. Three groups are axial modes, three are tangential modes and the remaining group is of oblique modes. If the impedance of the boundaries varies little over the frequency range of interest, then every mode in a given group should decay at the same rate. However, the measured modal decay times within each group varied substantially. The calculated values of the decay time varied by as much as a factor of eight from the measured values. The most probable reason for the failure of this model to predict the modal decay rate is that the locally reactive assumption is not valid in the reverberation room used for their experiments although the room was constructed according to accepted standards.

The modal coupling between the sound field in the reverberation room and wall vibrations can be identified on any wall surface at any frequency. If room mode of sound field is generated by a loudspeaker, the modal vibrational patterns can be found on all the walls. If one wall mode of the room is generated by a mechanical shaker, the modal structure of the sound pressure in the room can be identified by ear. Figures 2 shows a collection of modal patterns identified in the reverberation room. The modal patterns can be dominated by single mode or may contain combination of modes, depending on the driving frequency. Nevertheless, they all response modally.

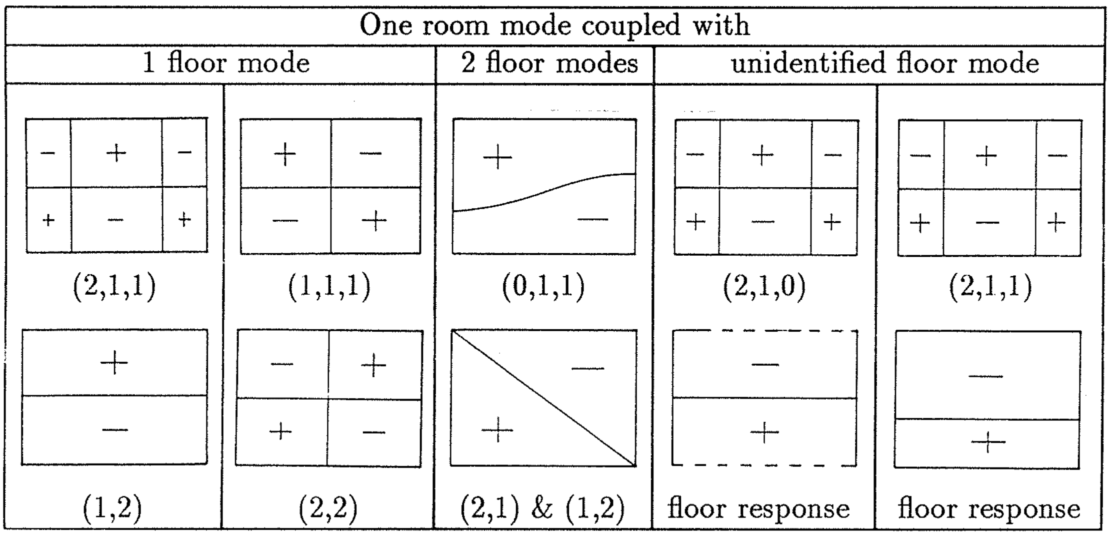


Figure 2: Identified modal patterns in the room and in the floor.

The effect of the modal coupling on the acoustical behaviour in the room can be demonstrated by varying the quality factor of wall structural modes and measuring the change in the modal decay times of the room. For example, at a driving frequency of 84.6 Hz, the coupling between the sound field and the back wall is stronger than with any other wall. Under these conditions, the room acoustic mode is (2,2,1) and the back wall mode is (1,2). The quality factor of the back wall mode was varied by wedging wooden blocks between wall of the building and the back wall of the reverberation room at the mode antinodes (Figure 3). The decay time of the room mode and the quality factor of the back wall were measured, as the number of the blocks was increased.

Although these experiments were done in a reverberation room, the results have provided a close look at sound fields in enclosures in a very general sense. Most boundaries of enclosures, such as those of theaters and studios and those of aircraft or motor vehicles, are acoustically similar to the walls of the reverberation room in that they are extensively reactive, although they may have different shape, dimension and material construction.

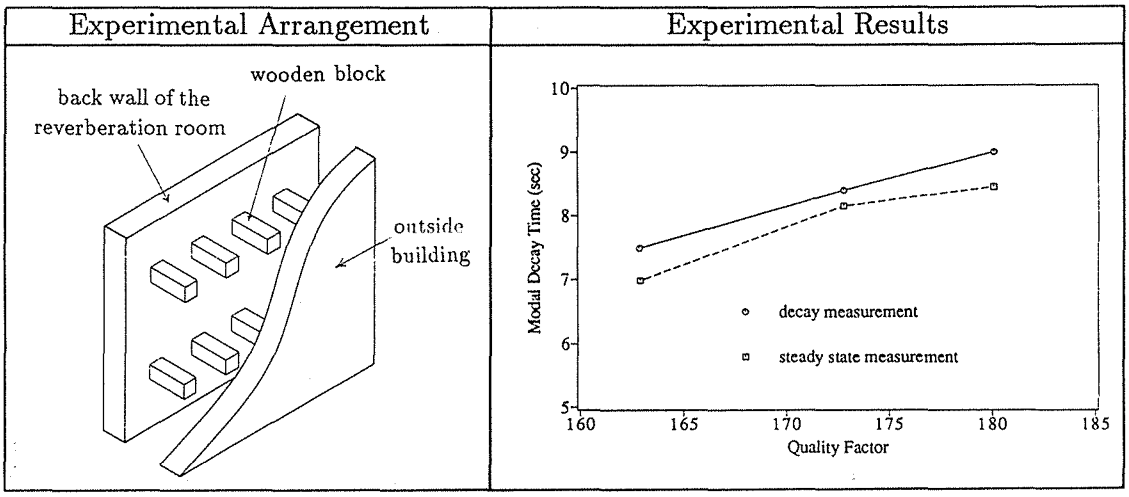


Figure 3: Experimental arrangement for varying the quality factor of the (1,2) back wall mode and the measured 60 dB decay times of the (2,2,1) room mode.

II.Effect of modal coupling

The failure of the sound decay theory in predicting modal decay times in the reverberation room suggests that a new model is required. The new model must include the effect of modally reactive boundaries. The model presented in this paper describes the acoustical behaviour of an enclosure in terms of the modal coupling of the sound field system and the boundary structural system. At low frequencies, the steady and transient response of the coupled system are represented as a collection of normal modes. In the high frequency range, statistical energy analysis is used to describe the average acoustical behaviour of the room and the boundaries.

a. low frequency range

Using modal coupling analysis (Pan and Bies 1989), the sound pressure field $p(\vec{r},\omega)$ in an enclosure with both locally reactive (A_l) and modally reactive (A_m) boundaries (Figure 4), is expressed as linear combination of the rigid wall mode shape functions Φ_I,

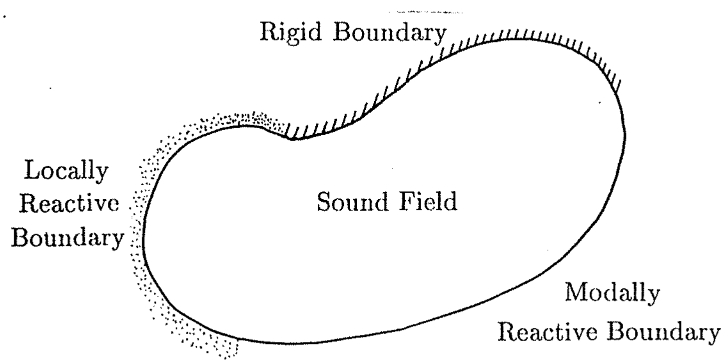
$$p(\vec{r},\omega) = \sum_{I=1,N1} P_I \Phi_I(\vec{r}), \tag{1}$$

where P_I is a frequency dependent coefficient of Ith cavity mode. The velocity distribution $v(\vec{\sigma},\omega)$ on the modally reactive boundaries can be expressed as a combination of mode shape functions S_I, which is determined by the dimensions, material and edge conditions of the boundary structures.

$$v(\vec{\sigma},\omega) = \sum_{I=1,N2} V_I S_I(\vec{\sigma}) \tag{2}$$

where V_I is the coefficient of Ith structural mode. The sound pressure in the enclosure is described by the wave equation $\nabla^2 p - \frac{1}{c_o^2}\frac{\partial^2 p}{\partial t^2} = 0$

Structural-Acoustic Coupling Model



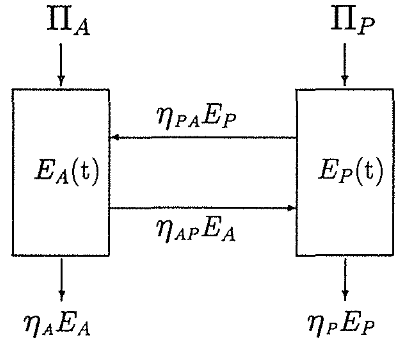
Low Frequency Range	High Frequency Range
Sound Field: $\Phi_i(\vec{r},\omega), k_{ai}, \zeta_{ai}$ Structural Vibration: $S_j(\vec{\sigma},\omega), k_{pj}, \zeta_{pj}$ $(i,j=1,2,\ldots)$	
$(\lambda^2 - 2\zeta_{aN}\lambda + k_{aN}^2)M_N^a P_N$ $-\lambda C_o \rho_o^2 \sum_I B_{I,N} V_I = 0$ $(\lambda^2 - 2\zeta_{pM}\lambda + k_{pM}^2)M_M^p V_M$ $+\lambda \frac{1}{C_o} \sum_J B_{J,M} P_J = 0$ $(N,M=1,2,\ldots)$	$E_A(t) = A_{A1}e^{s_1 t} + A_{A2}e^{s_2 t}$ $E_P(t) = A_{P1}e^{s_1 t} + A_{P2}e^{s_2 t}$

Figure 4: Illustration of the structural-acoustic coupling model.

and the vibration velocity of the flexible boundaries is described by the biharmonic wave equations (Junger and Feit 1986). The continuity of the air particle velocity and the velocity of the boundaries, is then used to couple these two wave equations. The eventual general result for the free vibration of the coupled room-boundary system is the eigenvalue problem,

$$(\lambda^2 - 2\zeta_{aN}\lambda + k_{aN}^2)M_N^a P_N - \lambda C_o \rho_o^2 \sum_I B_{I,N} V_I \;=\; 0 \tag{3}$$

$$(\lambda^2 - 2\zeta_{pM}\lambda + k_{pM}^2)M_M^p V_M + \lambda \frac{1}{C_o} \sum_J B_{J,M} P_J \;=\; 0 \quad \text{for} N, M = 1, 2, \ldots \tag{4}$$

where $M_N^a = \rho_o \int_V \Phi_N^2(\vec{r})dv$ and $M_M^p = \rho h \int_{A_f} S_M^2(\vec{\sigma})ds$ are modal masses for the enclosure and structure respectively. $k_{aN} = \dfrac{\omega_{aN}}{C_o}$ is the wavenumber of the Nth cavity mode, $k_{pM} = \dfrac{\omega_{pM}}{C_o}$

is the "wavenumber" of the Nth panel mode and $\lambda = -ik$. The damping constant ζ_{aN} of the Nth cavity mode is related to the specific acoustical impedance Z_A on the locally reactive surfaces A_l (Morse and Bolt 1944). The damping constant of the Mth structural mode, which is represented by ζ_{pM}, is related to mechanical damping of structure (Snowdon 1970). $B_{I,J}$ is the modal coupling coefficient between the Ith structural mode and the Jth cavity mode,

$$B_{I,J} = \int_{A_m} S_I(\vec{\sigma})\Phi_J(\vec{r})ds \ . \tag{5}$$

For $N1$ cavity modes and $N2$ structural modes, Equation (3) becomes a $(N1+N2)\times(N1+N2)$ matrix equation with $(N1+N2)$ solution eigenvalues. The imaginary part of the eigenvalue λ_i is related to the resonance frequency of the ith acoustic mode by $(f_i = \frac{Im(\lambda_i)C_o}{2\pi})$ and the real part is related to the 60dB decay time of the same mode by $(T_{i60} = \frac{6.91}{Re(\lambda_i)C_o})$.

The eigenvector $(P_{i1},\ldots P_{iN1}, V_{i1},\ldots V_{iN2})$ associated with each eigenvalue λ_i, describes the shape of the acoustical mode. This means that each acoustic mode Ψ_i has both a sound pressure part Ψ_i^P and a boundary vibration part Ψ_i^S. The sound field part of the ith acoustical mode can be written as

$$\Psi_i^P(\vec{r},\lambda_i) = \sum_{I=1,N1} P_{iI}\Phi_I(\vec{r}), \tag{6}$$

and structural vibration part can be written as

$$\Psi_i^S(\vec{\sigma},\lambda_i) = \sum_{J=1,N2} V_{iJ}S_J(\vec{\sigma}). \tag{7}$$

The parameters determining the characteristics of the above expressions are all modal parameters (mode shape, resonance frequencies and modal damping constants) of the enclosure and the boundary. This mathematical description of the coupled system does not make any assumptions about the geometrical shapes of enclosures and boundaries. For simple geometries (e.g. rectangular rooms), analytical expressions for the modal parameters are available. If the geometry is complicated, these parameters can be determined numerically.

b. high frequency range

In the high frequency range, the behaviour of the coupled system shown in Figure 4 can be described in terms of the average vibration energies stored in the boundaries and in the room, the energies dissipated within them and the energy flowing between them. All of these energy terms are related to the average modal parameters of the system and the input power. For example, the loss factor of the room (η_A) is the ratio of the stored energy to the energy dissipated in the room. The coupling loss factor from the room to the boundary (η_{AP}) is the ratio of the stored energy to the energy flow to the boundary. The loss factor of the boundary η_P and its coupling loss factor η_{PA} are defined in the same way.

When the coupled system is initially in a steady state and then the input power is suddenly cut off, the time dependent energies in the room and in the boundary are

$$E_A(t) = E_A(0)[\frac{s_1+d_1}{s_1-s_2}\exp(s_1t) - \frac{s_2+d_1}{s_1-s_2}\exp(s_2t)] \ , \tag{8}$$

$$E_P(t) = E_P(0)[\frac{s_1+d_2}{s_1-s_2}\exp(s_1t) - \frac{s_2+d_2}{s_1-s_2}\exp(s_2t)] \ , \tag{9}$$

where $E_A(0)$ and $E_P(0)$ are the initial energies in the room and in the panel respectively and

$$s_1 = -\frac{1}{2}[(\eta_A + \eta_P + \eta_{AP} + \eta_{PA}) - ((\eta_A + \eta_{AP} - \eta_{PA} - \eta_P)^2 + 4\eta_{AP}\eta_{PA})^{\frac{1}{2}}] , \tag{10}$$

$$s_2 = -\frac{1}{2}[(\eta_A + \eta_P + \eta_{AP} + \eta_{PA}) + ((\eta_A + \eta_{AP} - \eta_{PA} - \eta_P)^2 + 4\eta_{AP}\eta_{PA})^{\frac{1}{2}}] , \tag{11}$$

$$d_1 = (\eta_P + \eta_{PA} + \eta_{PA}\frac{E_P(0)}{E_A(0)}) , \tag{12}$$

$$d_2 = (\eta_A + \eta_{AP} + \eta_{AP}\frac{E_A(0)}{E_P(0)}) . \tag{13}$$

Both experimental and theoretical methods have been developed to evaluate the average modal parameters of the coupled system (Lyon 1975). From these parameters, both the steady-state and the transient state behaviour of the average energies in the enclosure and in the boundary can be estimated from Equations (8) and (9).

III. Results and discussions

Two different acoustical-structural systems have been used to verify the new models and to demonstrate some novel characteristics of enclosures with modally reactive boundaries. The rectangular panel-cavity used for low frequencies consists of a 6 mm thick, simply supported aluminium panel on top of a 0.868m × 1.150m × 1.000m cavity. The remaining walls of the cavity are 200 mm-thick concrete slabs. For the study in the middle and high frequency ranges, flexible panels of 1.2m × 1.5m × 10mm particle board were installed in a 6.84m × 5.57m × 4.72m (rectangular) reverberation room

a. measured and predicted results

To verify the low frequency model, the resonance frequency and the 60 dB decay time of some acoustical modes of the first coupled system were measured. These experimental results are compared with the theoretical predictions in Table 1. The parameters of the first 20 uncoupled cavity and panel modes are required as inputs for the prediction. These were measured independently and substituted into Equations (3) and (4).

The results of the SEA calculations and the experimental measurements in the high frequency range are shown in Figure 5. The average modal parameters of the empty reverberation room and test panel (i.e. the uncoupled components of the system) were obtained by conventional measurement techniques (Lyon 1975). These parameters were used in Equations (8) to calculate the reverberation times of the coupled system consisting of the reverberation room and the test panels.

The agreement between the experiments and predictions shown in Table 1 and Figure 5 indicates that the coupling theory can be used to evaluate the reverberation time in enclosures where the boundaries are modally reactive.

Table 1: Measured and predicted resonance frequencies and modal decay times of the panel-cavity system.

mode(l,m,n)	Measurement		Prediction	
	$f_{l,m,n}$ (Hz)	T_{m60} (s)	$f_{l,m,n}$ (Hz)	T_{m60} (s)
0,1,0	149.82	10.1	149.91	10.02
0,0,1	170.23	7.9	172.23	7.90
1,0,0	198.34	9.2	198.66	9.2
0,1,1	227.9	5.5	229.32	6.07
1,1,0	248.9	1.5	249.79	1.83
1,0,1	260.9	3.2	262.81	5.43
0,2,0	297.7	4.9	299.68	4.73
1,1,1	299.3	3.8	302.45	4.49
0,0,2	339.2	7.4	344.61	6.84
0,2,1	342.6	5.0	345.72	5.11
1,2,0	356.6	4.3	359.25	4.60
0,1,2	369.7	4.0	375.51	5.92
2,0,0	393.9	5.0	396.53	5.52

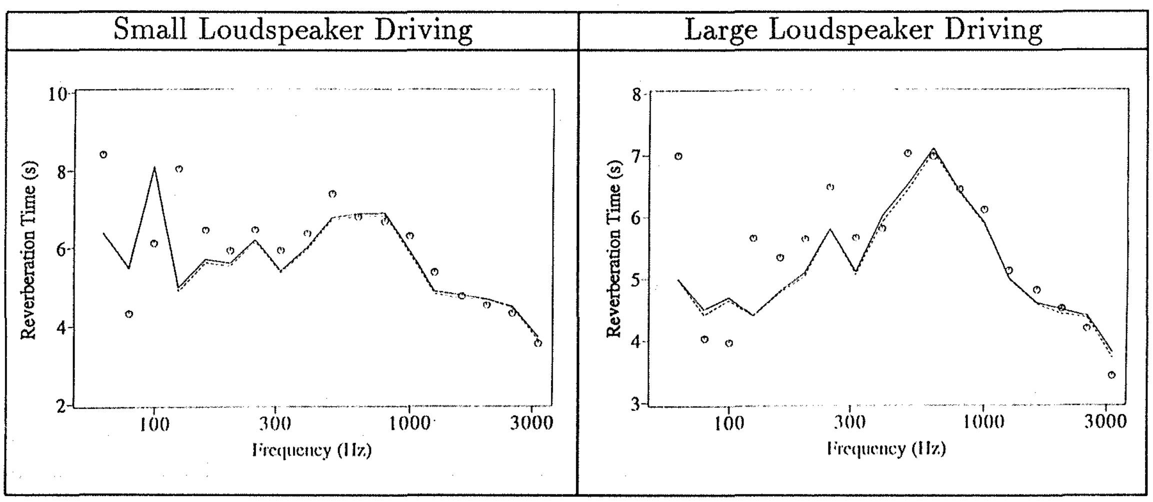


Figure 5: Measured (o) and predicted (— by Eq. (8); $\cdots$, by Eq. (15)) average reverberation times in the reverberation room with 6 panels installed.

b. eigenvalues of the coupled system

The eigenvalues of the modes are important characteristics of the coupled system. They determine the width and the location of the resonance peaks of the steady-state response. They also determine the decay rate and shape of the decay curves, of the transient response.

Because of sound field coupling with the boundary structure, the phase of the boundary velocity with respect to the phase of the incident sound pressure in the room will affect the superposition of the traveling waves in the room. Therefore the resonance, which occurs when the reflected waves coincide with the incident waves, will be perturbed and so will the resonance frequency. The energy flow between the boundary structure and sound field in the presence of the coupling will directly affect the decay of the sound field. The magnitude of this energy flow is determined by the coupling condition between individual structure and room modes. To demonstrate this effect, the eigenvalues were altered by varying the modal density of the boundary structure. Although the interior boundary surface is not changed, significant changes in the resonance frequency and the 60 dB decay time of each mode were observed. In fact, by selecting certain modal density values, it was possible to change the decay time of any particular mode by a factor of ten or more. These large changes in decay time are achieved both theoretically and experimentally with very small changes in boundary modal density. Figure 6 shows the modal decay times of three acoustical modes as a function of panel modal density. The measured resonance frequencies of the same modes are plotted in Figure 6. As the panel modal density increases the resonance frequency of each mode has one or more jumps (from low to high frequency) which coincide with its decay time minima.

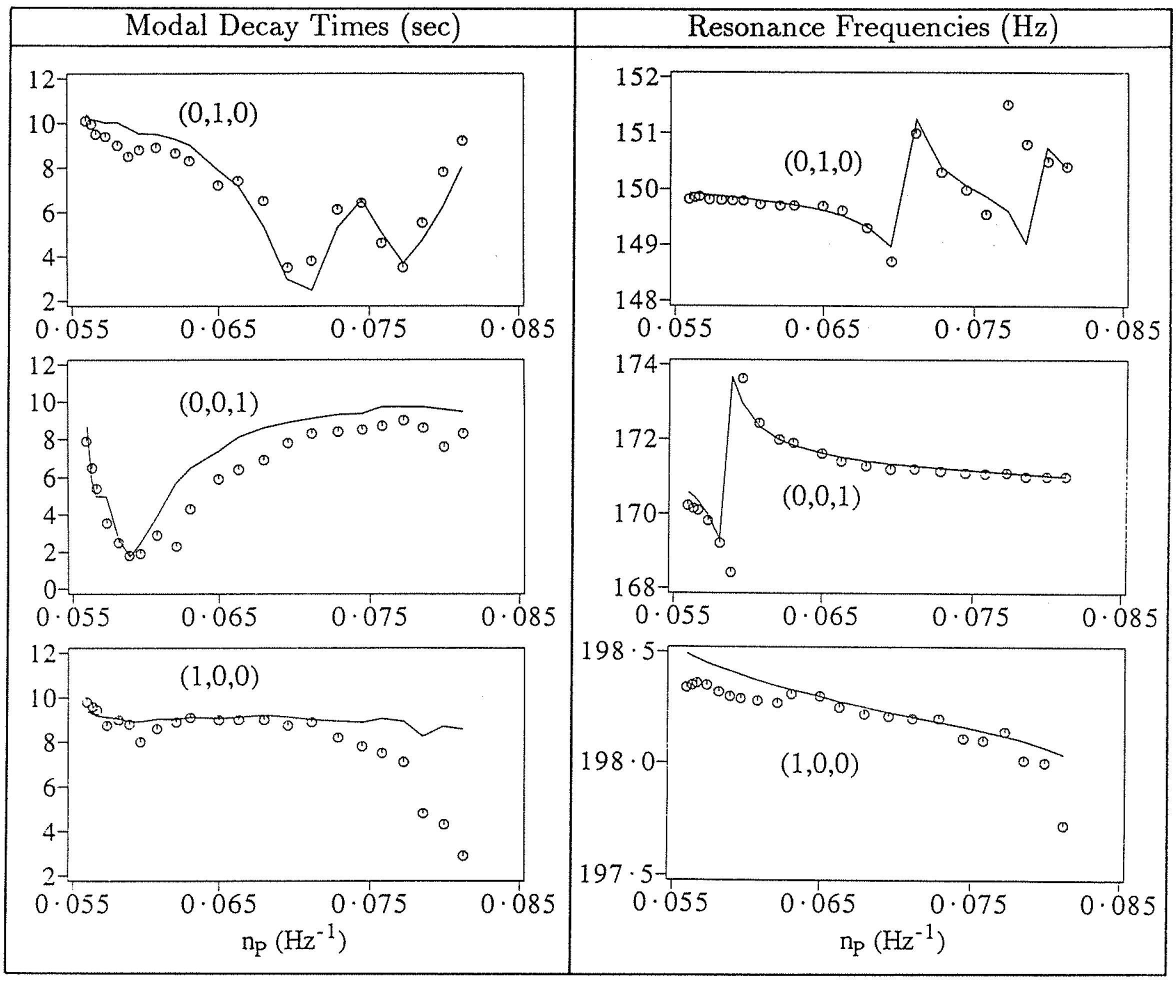


Figure 6: 60 modal decay times and resonance frequencies of three acoustical modes as a function of panel modal density (o measured; — predicted).

c. cavity controlled and panel controlled modes

The solution of Equations (3) and (4) shows that the air-structural coupled system is described by what may be called acoustical modes. The total number of the acoustical mode is the sum of the numbers of cavity modes and the panel modes in the frequency range of interest. An acoustical mode is a mode of the entire panel-cavity system. If we measure the relative amounts of energy contained in the two parts of the acoustical mode (i.e. in the cavity sound field and in the panel vibration), we can identify two types of acoustical modes : "cavity controlled" and "panel controlled". A cavity controlled mode has most of its energy stored in the cavity sound field, while a panel controlled mode has most of its energy stored as panel vibrational energy.

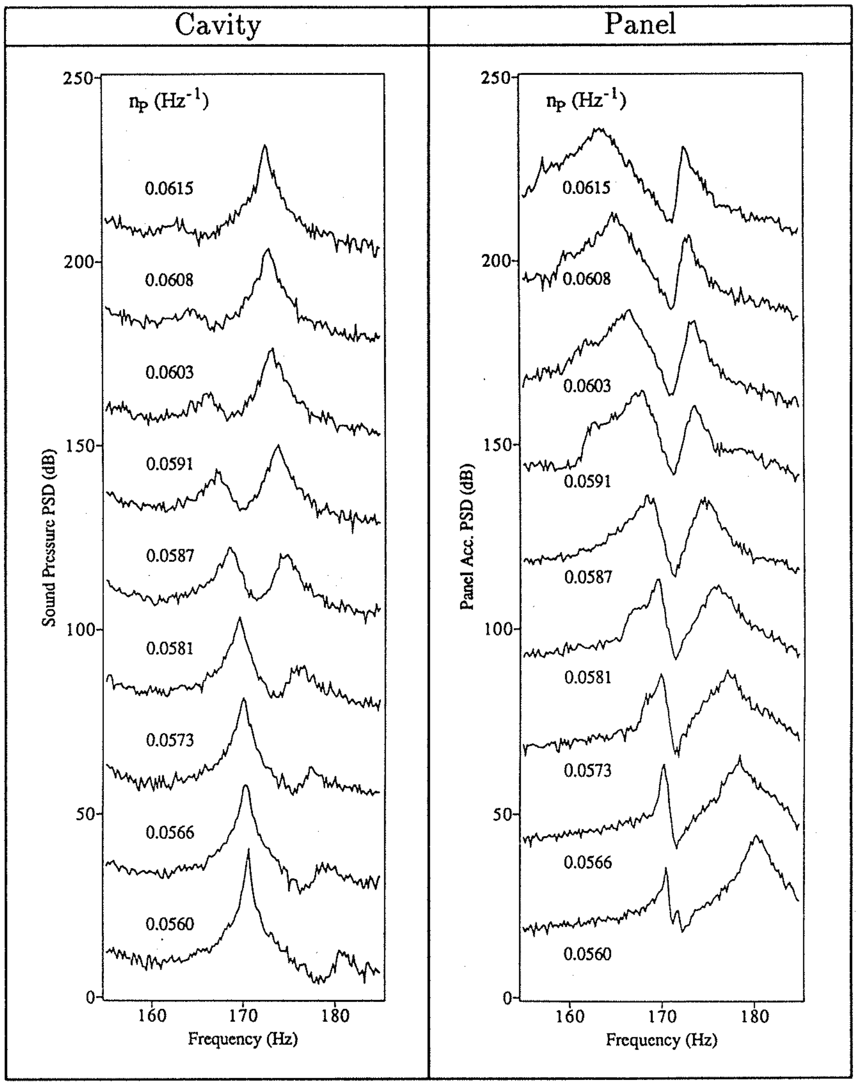


Figure 7: Power spectral density of the cavity sound pressure and the panel acceleration for each increment of panel modal density. Each curve is offset from the one underneath it by 25 dB.

These two kinds of acoustical modes have been identified experimentally. A typical example may be used to explain some of their characteristics. The power spectral densities of the sound field and the panel vibration in the region of the resonance frequencies of the cavity controlled mode (0,0,1) and the panel controlled mode (3,1) are shown in Figure 7 for a range of panel modal densities. At a modal density of $0.0560Hz^{-1}$, one low frequency peak in the PSD of the cavity sound field can be identified clearly. A high frequency peak of much lower amplitude is also visible. However, in the corresponding panel vibration spectrum, the high frequency peak has much greater amplitude than the low frequency peak. In this case, the cavity controlled mode (0,0,1) and the panel controlled mode (3,1) can be distinguished easily from relative amplitudes of the peaks.

As the modal density is increased (moving up the page), the higher frequency peak in the sound field PSD becomes larger while the lower frequency peak moves slightly toward lower frequencies and becomes smaller. Eventually only the high frequency peak remains and the low frequency peak disappears. The PSD of the panel vibration shows almost the opposite behaviour. The lower frequency peak becomes bigger and its bandwidth becomes wider, while the higher frequency peak becomes smaller and narrower.

The low frequency peak in the cavity sound field spectrum and the low frequency peak in the panel vibration spectrum are resonance peaks for the same coupled acoustical mode. The peak of larger amplitude in the cavity sound field spectrum belongs to a cavity controlled mode. Similarly the peak of larger amplitude in the panel spectrum belongs to a panel controlled mode. As the panel modal density is increased in the region of maximum sound absorption, the cavity controlled mode becomes panel controlled and the panel controlled mode becomes cavity controlled.

d. double decay slopes

Figure 8 shows the decay curves of a test panel in the reverberation room and the decay curves for the sound field. The decay curves of the sound field are linear. However, the panel decay curves each have two distinct decay rates. The initial decay rate corresponds closely to the internal loss factor of the panel and the second decay rate corresponds closely to the internal loss factor of the room. If the panel and the room are physically separated, each decays at a rate determined by its individual loss factor. The air-structural coupling is relatively weak, and so when a panel and a room are coupled, their decay rates will not be greatly changed.

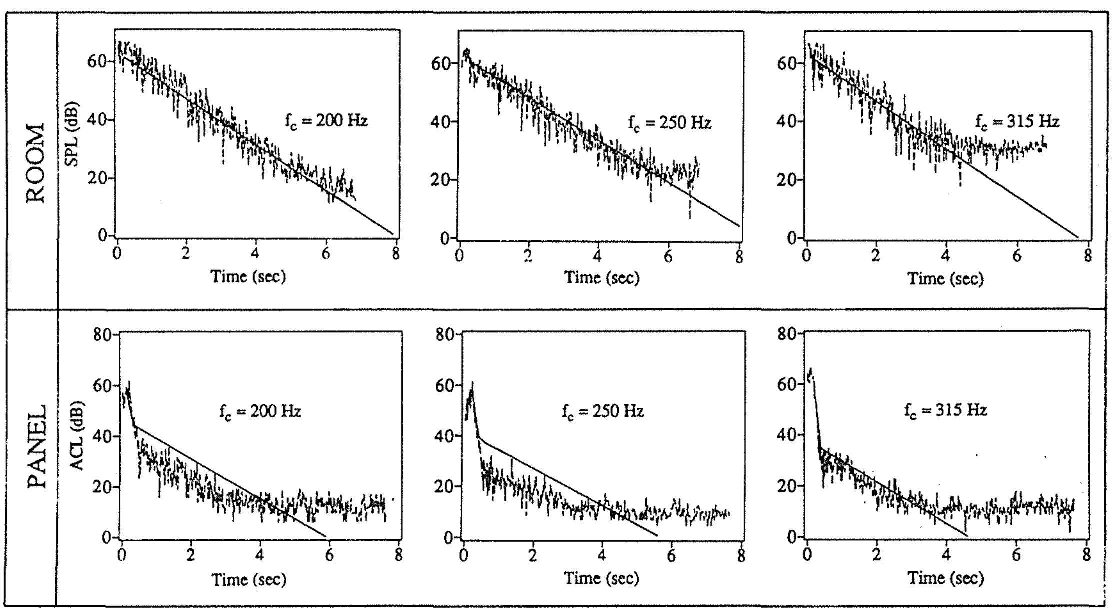


Figure 8: Decay curves of the sound pressure level and one panel acceleration level in the reverberation room (- - - measured; — calculated).

If we drive the sound field and measure the sound decay in the room and if the reverberation time of the uncoupled room is longer than that of the panel, the energy in the room will always flow into the panel during all of the decay period. Linear decay is expected in this case and this is what is observed in the measurement. On the other hand, if the uncoupled room reverberation time is shorter than that of the panel, then after the energy in the room becomes smaller than the energy in the panel, the energy in the panel will flow back into the room. During this time, the energy in the room decays at the decay rate of the panel. In this case two slopes in the decay curve are expected. The same phenomenon can be observed when the panel is driven by a magnet and the panel decay is measured by an accelerometer. In the experiment, the panel damping is much higher than that of the room damping, which is why double decay slopes were obtained in the panel decay curves (Figure 8).

e. Sabine absorption coefficient

The conventional measurement of the Sabine absorption coefficient of the panels in a reverberation room is based on the formula,

$$\alpha_{Sab} = \frac{55.3V}{SC_o}\left(\frac{1}{T'_{60}} - \frac{1}{T_{60}}\right) , \tag{14}$$

where T_{60} and T'_{60} are the reverberation times before and after the test panels are put in the reverberation room; S is the total surface area of the panels.

Because the modal density of the room is much larger than that of the panel in the high frequency range, we can use the first term in Equation (8) to approximate the decay behaviour of the sound field in the room. Therefore the approximate reverberation time of the room can be written as

$$T_{60} \cong \frac{27.6}{|(\eta_A + \eta_P + \eta_{AP} + \eta_{PA}) - ((\eta_A + \eta_{AP} - \eta_{PA} - \eta_P)^2 + 4\eta_{AP}\eta_{PA})^{\frac{1}{2}}|} . \tag{15}$$

The reverberation times calculated from this formula are plotted in Figure 5. They are in agreement with the results calculated using Equation (8).

By substituting Equation (15) into Equation (14), we obtain

$$\begin{aligned} \alpha_{Sab} &= \frac{55.3V}{27.6SC_o}(|(\eta_A + \eta_P + \eta_{AP} + \eta_{PA}) \\ &\quad -((\eta_A + \eta_{AP} - \eta_{PA} - \eta_P)^2 + 4\eta_{AP}\eta_{PA})^{\frac{1}{2}}| - 2\eta_A) . \end{aligned} \tag{16}$$

This result shows that the Sabine absorption coefficient of a modally reactive boundary represents the average effect of air-structural coupling on the sound decay in the room. Equation (17) demonstrates that the Sabine absorption coefficient α_{Sab} of the test panels depends upon the characteristics of the room in which it is measured. Different reverberation times and dimensions of the empty room (which are used to determine to calculate η_A and η_{AP},) will give rise to different values of Sabine absorption coefficient α_{Sab} for the same set of panels. For example, the room loss factors are affected by the sound absorption of driving loudspeaker used for the measurement. Figure 9 shows the Sabine absorption areas $\frac{55.3V}{C_0}(\frac{1}{T'_{60}} - \frac{1}{T_{60}})$ of the panels for two different loudspeakers. In a few frequency bands, there is about thirty percent difference between the two sets of experimental results. A similar difference was produced by the SEA calculation.

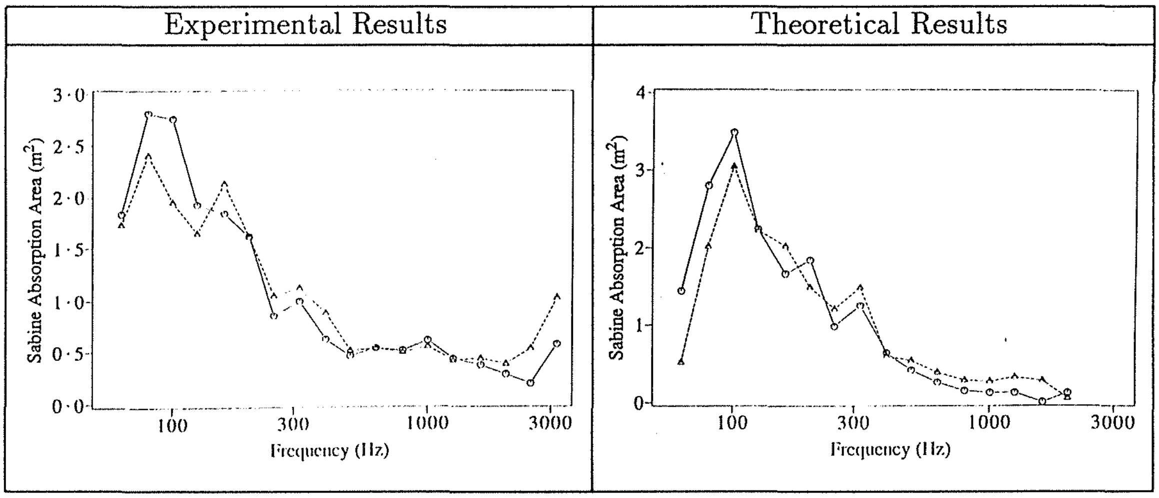


Figure 9: Sabine absorption area of the test panels for two different loudspeakers (— large loudspeaker; - - - small loudspeaker).

IV. Conclusions

This paper shows that the locally reactive assumption can produce an incorrect description of the sound field in a room, and can produce experimental discrepancies. It has been demonstrated that the classical acoustical decay solution can only be used when the boundaries of a room are locally reactive. These boundaries are often modally reactive rather than locally reactive. If the boundaries (or part of the boundaries) are modally reactive, the sound field behaviour in a room should be described as part of the behaviour of a coupled system. The vibration behaviour of the boundaries is the other part. Results from the modal coupling analysis and SEA provide a proper description for the acoustical behaviour of the coupled system. This new model, which is not restricted to simple geometries and includes both locally and modally reactive boundaries, has revealed and explained some characteristics of the sound field which the classical theory cannot predict. Some of the results reported here may explain some of the discrepancies between different test facilities and between tests and field experience. They may provide insights useful for improving interior acoustical design.

Acknowledgements

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TOWARDS A CODE OF PRACTICE FOR OCCUPATIONAL NOISE IN THE ENTERTAINMENT (MUSIC) INDUSTRY IN WESTERN AUSTRALIA

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ABSTRACT

The use of high sound levels in entertainment venues where live or recorded music is presented has been of concern to regulatory authorities as well as to many musicians, venue managers, staff, audience members, and of course "the neighbours". While environmental regulatory authorities have had some success in dealing with the neighbourhood noise problem through architectural treatment of buildings, the occupational noise problem has tended to lie in the "too hard" basket. In Western Australia the proclamation of the Occupational Health, Safety and Welfare Act 1984-87 and Regulations in 1988, covering all workplaces, including entertainment venues, has caused a reappraisal of the problem. A code of practice is being developed in consultation with relevant industry groups to provide practical strategies for compliance with the regulations. This paper outlines the unique aspects of this code, in particular the proposed method for noise surveying and provision of information on noise, and examines the underlying assumptions.

TOWARDS A CODE OF PRACTICE FOR OCCUPATIONAL NOISE IN THE ENTERTAINMENT (MUSIC) INDUSTRY IN WESTERN AUSTRALIA

INTRODUCTION AND BACKGROUND

The entertainment (music) industry presents a number of unique problems to those legislators who wish to achieve a degree of regulation over environmental or occupational noise. It is well known that the same sounds which are delightful to an audience may also be damaging to their ears, (or to those of employees at the venue), and may be distressing to the neighbours! This paper will focus primarily on occupational noise exposure aspects.

The characteristics of the industry which make it unique from a regulatory point of view are:-

1. Musical sound (noise) is the "product", not an unwanted "by-product" as in other industries[1]. Thus the normal control methods such as noise reduction at source or containment in the noise path are not so readily applicable.

2. There is a high degree of variability in sound levels generated and exposures received in this industry, as a result of performers moving from venue to venue, variations in the sound levels of performances and the great range of musical styles being presented. Thus it is difficult to establish viable measurement and assessment methods.

3. Employer/employee relationships are often difficult to define, making the apportioning of legal responsibility a major task. Thus the application of standard occupational noise regulations in this industry, with their requirements for engineering noise control, noise surveys and provision of personal hearing protection, have not usually been practical.

Previous noise control codes, specifically for this industry, have focussed on the particular problem of licensed premises where live or recorded rock music is presented[2,3,4]. The approach has generally been to specify a "sound level limit" or an L_{eq} limit in respect of either patrons, employees or neighbours. This is often administered by means of electronic compressors or sound level detectors with automatic power shut-off. Anecdotal evidence suggests that such devices, particularly the latter, are likely to be by-passed or disabled wherever possible, to allow the performance to continue unhindered. The problem with this control model is that, by being based purely on enforcement of a sound level, it places all responsibility on the performers and none on the venue operator; it has no real regard for factors such as acoustic properties of the venue, performance style, or audience demands; and importantly has no educative component. Thus in recent experience, performers, audiences and venue management have tended to show little patience with such methods.

Legislators on the other hand are becoming increasingly aware that high noise exposures are likely to be encountered not only in rock bands and discos, but also in brass bands, symphony orchestras, aerobics classes and some forms of ethnic music, and that practical control measures are needed. While there is some debate as to whether the dose-response relationship for music is the same as that commonly used for industrial noise[5,6], legislators have insufficient evidence to hand to depart from accepted industrial noise damage risk criteria. It is not proposed to join that debate in this paper.

In view of the need for a fresh approach to the particular problem of occupational noise exposure in this industry, this paper outlines a measurement and assessment method intended to address some of the above problems. This method forms part of a draft Code of Practice for Noise Control in the Music/entertainment Industry which is currently under consideration by a tripartite working party established under the auspices of the Commission for Occupational Health, Safety and Welfare of Western Australia. The Code if eventually adopted, will provide strategies complementary to the existing general Code[7] for compliance with the requirements of the Occupational Health, safety and Welfare Act 1984-87 and Regulations 1988.

BASIC CONCEPTS

In order to define the level of risk of hearing damage related to music and to apportion responsibilities in dealing with the hazard, the following parameters are proposed as the basis for the Code:-

"Music level" — the average sound level of a representative portion of a typical performance, measured at a nominated "reference position" in a venue;

"Reference position" — a nominated measurement position within the venue sufficiently close to the stage area that the sound level is dominated by the music;
(for venues which have live bands performing regularly the reference position should be 5 metres from the front of the stage at least 1.8 metres above floor level and centrally located in front of the performers, while for other venues the reference position may be any nominated monitoring point where the music dominates the sound levels).

"Room loss" — the average drop in sound level from the reference position back to the locations occupied by employees, measured during a typical performance with a typical crowd in attendance;

"Received noise" — the average noise level measured at the employee's ear during a representative portion of a performance;

"8-hour exposure" — the noise level averaged out over an 8 hour period and measured at the employee's ear.

"Action level" — (as defined in regulation 304 of the Occupational Health, Safety and Welfare Regulations 1988):-

(a) a peak noise level (L peak) of 140 dB(lin); or

(b) an 8 hour noise exposure level ($L_{Aeq,8h}$) of 90 dB(A).

The "music level" defines the overall sound level of the performance. A rock band will tend to produce a fairly constant "music level" from performance to performance, while an orchestra or theatre company will produce different "music levels" from performance to performance, depending on the musical work.

The "room loss" is a characteristic of the room acoustics and room size, and is independent of "music level". It defines the extent to which employees are separated from the sound source, and can be increased by architectural means in many cases.

"Received noise" can then be estimated for any performance where the "music level" is known, by subtracting the "room loss" from the "music level". This "received noise" value can then be converted to an "8-hour exposure" by knowing the duration of the performance (see page 6).

The "8-hour exposure" can be compared with the "action level".

The examples presented in Appendix 1 outline typical scenarios in the entertainment industry and some suggested means of meeting the duty of care.

The advantages offered by this procedure are:-

1. The responsibilities of performers and venue operators are clearly defined. The performers are responsible for providing "music level" information and adhering to it, while the venue operator is responsible for maximising "room loss" and minimising "8-hour exposure";

2. The procedure provides a practical strategy for fulfilling statutory requirements for noise survey reports and provision of information on noise, where it was believed that noise surveys may be needed nightly!;

3. The use of a "reference position" for monitoring "music level" ties in well with environmental noise control conditions which may require adherence to an internal noise limit.

MEASUREMENTS IN TYPICAL VENUES

In order to identify any practical problems with the measurement technique, and to assess some typical "music levels", "room loss" and "received noise" values, a series of noise measurements was conducted in five (5) typical hotel and cabaret venues in Perth.

The measurements were conducted using a fixed measurement system, with simultaneous measurements carried out at various locations using two portable sound level meters. The fixed measurement system comprised a Bruel & Kjaer 2230 Integrating Sound Level Meter and Bruel & Kjaer 2318 Printer. This system was programmed to print out $L_{Aeq,T}$, L_{Amax} and L_{Amin} levels at five-minute intervals, continuously. The portable meters were Rion type NL-11, set to obtain single five-minute $L_{Aeq,T}$ samples which were taken concurrently with the fixed system samples.

For the purposes of this series of measurements, the fixed system was set up at the bar location nearest the stage, where the sound level would be dominated by the music.

The portable meters were used to obtain samples at the following locations:-

- other bars;
- door staff;
- glass collectors;
- reference position (5 metres from stage);
- stage (or console for discotheque);
- mixing desk (for live bands).

Where the employee was mobile, e.g., glass collector, the observer followed the employee around the venue for the duration of the sample.

This procedure enabled typically two to four five-minute samples to be taken at each position of interest during the performance.

The simultaneous samples were then compared to enable estimation of "room loss", thus:-

1. The logarithmic average of the samples at the reference position was calculated;

2. From this, was subtracted the logarithmic average of the concurrent samples recorded at the fixed measurement position;

3. This difference was added to the logarithmic average of all samples recorded at the fixed measurement position, to give the "music level", i.e., $L_{Aeq,T}(M)$ for a representative section T of the performance duration;

4. Similarly, the samples recorded at each location were averaged and normalised by comparing with the fixed measurement position, to obtain a "received noise" level $L_{Aeq,T}(R)$, normalised over the same time T;

5. "Room loss" was calculated for any location (R) by subtracting "received noise" from "music levels", as follows:-

$$\text{"Room loss"}(R) = L_{Aeq,T}(M) - L_{Aeq,T}(R).$$

The "music level", "received noise" and "room loss" values were determined for the five venues investigated, together with the "8-hour exposure" level ($L_{Aeq,8h}$). This was calculated by adjusting the "received noise" for performance duration. By assuming that noise exposure outside the performance duration was negligible compared with that from the music, the 8-hour exposure for Position R was simply calculated thus:-

$$L_{Aeq,8h}(R) = L_{Aeq,T}(R) + 10 \log (Tp/8)$$

where Tp = duration of the performance (hours).

The results for the five venues are presented in Table 1.

TABLE 1:

SUMMARY OF RESULTS
OF MEASUREMENTS IN PERTH ENTERTAINMENT VENUES

Location		Room Loss dB	Received Noise $L_{Aeq,T(R)}$ dB(A)	8-hr exposure $L_{Aeq,8h}$ dB(A)
Venue 1 - Nightclub, disco music, music level = 102 dB(A), duration = 5 hours				
Bar	B1	9	93	91
	B2	8	94	92
	B3	11	91	89
	B4	10	92	90
Disk Jockey Box		9	93	91
Venue 2 - Hotel, live band, music level = 105 dB(A), duration = 2 hours				
Bar	B1	6	99	93
	B2	7	98	92
Hot Dog Bar		4	101	95
Glass Collector, bouncer		4	101	95
Mixer (part of band)		3	102	96
Crew at side of stage		0	105	99
Venue 3 - Tavern, live band, music level = 103 dB(A), duration = 2 hours				
Bar	B1	5	98	92
	B2	4	99	93
Glass Collector		5	98	92
Door attendant		12	91	85

Location		Room Loss dB	Received Noise $L_{Aeq,T}(R)$ dB(A)	8-hr exposure $L_{Aeq,8h}$ dB(A)
<u>Venue 4</u> - Hotel, live band, music level = 106 dB(A), duration = 2 hours				
Bar		5	101	95
Mixer (part of band)		6	100	94
<u>Venue 5</u> - Nightclub, disco music and floor-show, music level = 101 dB(A), duration = 6 hours				
Bar	B1	5	96	95
	B2	7	94	93
	B3	8	93	92
DJ Console		1	100	99

DISCUSSION

The measurement procedures and results outlined above show that the proposed measurement methodology is feasible. The following observations can be made on the practicalities of this measurement procedure:-

1. The choice of a 5-minute sample time appears to be a good compromise. A shorter sample time would allow for more samples to be obtained from portable meters, but would also permit greater variation in level from sample to sample. The fluctuations during performances were typically within $\pm$ 2 dB of the arithmetic mean of the $L_{Aeq,5min}$ values. Moving to a larger sample period, say 10 minutes, would smooth these fluctuations out even more, but would severely restrict the number of samples able to be taken with portable meters.

2. The fixed position, rather than being at a bar, should ideally be at the reference position. This would involve some inconvenience in setting up, as the microphone would probably need to be suspended from the ceiling and connected to a remote meter via an extension cable. However, the reference position needs to be a location at which the sound levels are dominated by music, and not significantly affected by crowd noise. By using a nearby bar for the fixed microphone, one is relying on this position also being dominated by music, in order to be able to interpolate back to a "music level" at the reference position. If the data at a fixed position at a bar were found to be corrupted significantly by crowd noise or bar noise, it may not be possible to estimate "music level" or "room loss" with sufficient precision. In the case of live band performances, it is likely that the sound levels at a nearby bar would be dominated by music, (Venue 4 in Table 1 "Received noise" = 101 dB(A)). At a discotheque, however, crowd noise may be a significant component when measuring at a bar location, and the fixed microphone would be better located at the reference position.

3. The choice of the "reference position" at 5 metres from the stage is a practical compromise between proximity to the loudspeakers and access to the position. Dance floors in licenced venues are commonly about 5 metres wide, so that position is usually at the edge of the dance floor. At greater distances, the room acoustics are likely to have a more significant effect on the music level, while at closer distances access for measurements may be difficult. If one approaches too close to the loudspeakers, near field effects may be encountered.

4. It is a simple matter during this type of measurement to measure $L_{Ceq,T}$ levels where it is suspected that a personal hearing protection programme will be needed. A few checks during our measurements indicated that C-weighted levels are generally likely to be 5 to 10 dB above the corresponding A-weighted levels. As the $L_{Aeq,8h}$ exposures were typically 90-95 dB(A) at the venues, the attenuation of hearing protectors needed to reduce $L_{Aeq,8h}$ to an indicative value of 85 dB(A) at the protected ear, would thus be an SLC_{80} in the range 10-20 dB.

5. It is also a straight forward matter to check whether the peak noise level, L_{peak}, is likely to exceed the Western Australian action level of 140 dB(lin), at any point adjacent to the loudspeakers. At Venue 2, with a "music level" of 105 dB(A), peak levels of 140 dB(lin) were encountered at a distance of approximately 300 mm from the mid-range drivers in the front-of-house system, indicating an area where access should be restricted. Under the proposed Code, it would be the responsibility of entertainment providers to define their "music level" and restrict access to any areas where the peak noise level was likely to exceed 140 dB(lin).

CONSIDERATION OF BASIC ASSUMPTIONS

The successful operation of this proposed Code will depend on venues where the performers change regularly needing to carry out one or two noise surveys only, this being the basis for adequately estimating "8-hour exposure" on any subsequent occasion. There are a number of assumptions inherent in transposing data from one typical performance to other performances with different performers in the venue.

In examining the following assumptions, it should be borne in mind that the precision required in this type of assessment may be likened to that of an $L_{Aeq,8h}$ determination in any normal industrial environment.

These assumptions are discussed below:-

1. That the performers generate the same "music level" for each performance at each venue - The question is to what extent the "music level" of any performance may vary from the baseline level established during a typical performance at a typical venue. (This would apply mostly to rock bands, who would need to advise the venue of their "music level". In large concert halls and theatres, the "music level" would be likely to be monitored by an in-house system. Likewise many disc jockeys would monitor the level of their system or "calibrate" the volume scale, such that while their "music level" may vary, the DJ would be able to provide the "music level" information).

   The two main factors likely to cause a variation in "music level" are the room acoustics and the performance level itself. Since the measurement is carried out at 5 metres ("reference position"), the sound level at that point is largely determined by the direct sound from the loudspeakers of the front-of-house PA system. In most venues, these loudspeakers would be set up in two stacks, positioned each side of the stage. Any significant reflections would normally come from the dance floor in front of the stage and the ceiling - Consequently there is a consistency in the physical set-up from venue to venue which would mitigate for consistency in "music level". The second factor is the level of performance itself. Many performers prefer to play at a set level appropriate for their PA system. At venues where sound levels are controlled (for environmental noise reasons), the "music level" would normally be a known quantity anyway. Where performers hire additional PA equipment to perform at certain venues, they would need to have the sound mixer monitor the music level (and adhere to the agreed level).

   Obviously where major changes in personnel, equipment or music style occurred, the "music level" may need to be re-measured. It can be argued, therefore, that a group of performers could define their "music level" from one set of typical measurements, with sufficient accuracy for the purposes of this code.

2. That the "room loss" in a venue is consistent from performance to performance.

   Barring major architectural changes, the main factors likely to alter "room loss" are the frequency content of the music and crowd size. In most amplified rock music, the sound sources which dominate the A-weighted sound level are the lead vocal and snare drum. These sources normally have their main energy in the frequency region around 500 Hz to 2k Hz. Since it is likely to be this same frequency range which dominates, the "room loss" is unlikely to be sensitive to changes in bass content in the music.

   The second factor, crowd size, could have a significant effect on "room loss" as a result of the acoustic absorption and possible screening effects. The measurement of "room loss" should be made with a moderate or average audience in attendance, covering a period in which the crowd builds up during the performance. This average condition would provide a conservative estimate of "room loss" under "packed house" conditions. The worse case would occur with a small audience, when "room loss" would be at a minimum. In this case, the performers would be likely to reduce their music level below the volume for a full audience, thus compensating to some extent. It is therefore considered that "room loss" measured during one performance with an average audience will suffice for the purposes of the Code.

3. That "received noise" is dominated by music and not crowd noise - The "received noise" for an employee is an estimated level based on "music level" and "room loss" and therefore excludes crowd noise and bar noise. The original measurement of "room loss" does however include the crowd noise component present during the measurements. The level of this crowd noise component will normally rise and fall with the "music level", thus is taken into account fairly consistently by use of "room loss" and "music level" for estimation of "received noise". If the sound level of the music is around 90 dB(A) or less at the employee location, then ambient noise may become a significant factor independent of the music.

   In this case the risk to hearing is much lower than in situations where the music dominates, thus the effect of ambient noise is unlikely to affect operation of the Code.

CONCLUSIONS

A technique for measurement and assessment of occupational noise exposure in the music entertainment industry has been proposed, involving determination of "music level" of the performance and "room loss" of the venue. This scheme has the advantages of:-

- placing responsibility for the "music level" on the performers and "room loss" on the venue operator;
- reducing noise survey requirements to a practical level;
- providing valuable information and control strategies to the workplace;
- combining well with environmental noise control measures.

The measurement technique has been tested in five typical venues, and found to be practically feasible. A number of assumptions underlying the use of the measurement results for subsequent performances have been examined qualitatively.

It is to be hoped that the opportunity will arise to further assess some of these relevant factors in typical venues in the future.

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APPENDIX 1

EXAMPLES OF USE OF PROPOSED METHOD

EXAMPLE 1:

HOTEL WITH LIVE BAND

A hotel owner manager engages live bands on four nights a week. In order to be in a position to fulfil his "duty of care", he arranges a series of sound level tests on a typical night. The Noise Officer who conducts the tests measures a "music level" of 103 dB(A) at the "reference position" (at the edge of the dance floor 5 metres from the stage) and at the same time measures "received noise" levels of 98 dB(A) at the two bars and 100 dB(A) at the glass collector's ear (by means of a portable meter). The Noise Officer subtracts the "received noise" levels measured at the two bars and for the glass collector, from the "music level" at the "reference position", to obtain "room loss" values of 103-98 = 5 dB for the bars, and 103 - 100 = 3 dB for the glass collector. The normal performance lasts two hours (consisting of three forty-minute sets with quieter breaks in between). The adjustment to calculate "8-hour exposure" is - 6 dB for a two-hour performance. The "8-hour exposures" on the night of tests were therefore 98 - 6 = 92 dB(A) for bar staff, and 100 - 6 = 94 dB(A) for the glass collector. The manager now knows that for a "music level" of 103 dB(A), his staff are exposed above the "action level" ("8-hour exposure" of 90 dB(A))) and he should seek appropriate controls and, where this is not practicable, reduce the amount of time the employee receives noise. If the employee still receives noise above the action level, the employer needs to provide personal hearing protection combined with an appropriate education programme. Each night the new band informs the manager what their "music level" will be prior to starting the performance. The manager then knows what the "8-hour exposures" of this staff will be and whether personal hearing protection will be required. For example, a band with a "music level" of 105 dB(A) will cause an "8-hour exposure" of 94 dB(A) for bar staff and 96 dB(A) for the glass collector, based on the above figures for "room loss".

EXAMPLE 2:

DISCOTHEQUE/CABARET VENUE

A discotheque owner presents recorded music via a different disc jockey (DJ) every night. The sound system is a fixed installation with the main loudspeakers above the dance floor and others throughout the room. The disco owner defines a "reference point" at the DJ's desk and has a Noise Officer measure the "room loss" from that point back to the bar staff, glass collector and doorman. With a reasonably practicable "music level" of 98 dB(A) at the "reference position", the "received noise" at the employee locations ranges between 90 and 95 dB(A). For an exposure of 5 hours, the adjustment to calculate "8-hour exposure" is - 2 dB. The "8-hour exposures" therefore range between 88 and 93 dB(A). The owner now knows which staff are exposed above the "action level" on a typical night. He provides personal hearing protection combined with an education programme for these staff. In order to maintain the "music level" at or below 98 dB(A), he installs a sound level meter at the DJ's desk and instructs the DJ not to exceed this level, giving reasons for this instruction.

EXAMPLE 3:

THEATRE FOR STAGE PRODUCTIONS

A large theatre is used as a venue for staging performances ranging from opera to rock musicals and modern dance programmes. For rock musicals the band may perform in the orchestra pit or on a special platform above the stage, using a special public address system brought in for the performance. Modern dance programmes usually use pre-recorded music played back through the in-house PA system. For musicals, operas. etc/,. the orchestra may perform from the pit, and on some occasions vocalists may be amplified slightly through the in-house PA system. The theatre management employ door staff, ushers and technical crew, some of whom work backstage.

The theatre management takes a policy decision, in consultation with the relevant unions, to prepare a noise report for each new show. A "reference position" is selected at the pit conductor's position about 3 metres from the front of the stage. The "music level" for the show is measured at this position, while sound levels are simultaneously monitored at employee locations. The results of the tests for the first few shows indicate that it is only during rock musicals and modern dance productions that there is any likelihood of the "action level" being exceeded. It is found that the "room loss" is about the same during both these types of productions. For future productions, the "music level" is monitored during a dress rehearsal from the "reference position". The "received noise" of employees is calculated from the measured "music level" by subtracting the "room loss" values and the "8-hour exposure" is determined by adjusting for the duration of the performance.

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PREDICTING THE SOUND TRANSMISSION LOSS OF CAVITY WALLS

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ABSTRACT

Work has begun on producing a spreadsheet model for predicting the sound transmission loss (STL) of walls. The initial intention was to survey the literature and use the best available formulae. The formulae were judged by comparing their predictions with laboratory measured results which have been published in a number of STL compilations. This approach has proved fairly successful for thin single layer walls. However, none of the existing formulae has been found to be satisfactory for cavity walls. This has led to the development of new theoretical formulae for the prediction of the STL of cavity walls. Initially the aim was to predict the STL of gypsum plasterboard cavity walls without studs, with wooden studs, with steel studs, with staggered studs and with double studs, all with or without absorption in the cavity. This work is still in its early stages but the advances are sufficient to make a progress report worthwhile. The formulae adopted for thin single layer walls are presented. Formulae for cavity walls have been developed for frequencies between the mass-air-mass resonance and the lower critical frequency of the cavity linings. The formulae cover both airborne and stud-borne sound transmission across the cavity. These formulae are presented and compared with experimental results.

THIN SINGLE WALLS

The sound transmission coefficient (τ) of a wall is the ratio of the sound energy transmitted by the wall to the sound energy incident upon the wall. For an infinite, isotropic, uniform thickness plane wall the sound transmission coefficient of a plane wave depends on the angle (θ) between the direction of propagation of the incident plane wave and the normal to the plane of the wall. To evaluate the diffuse field sound transmission coefficient (τ_d) it is necessary to average the plane wave sound transmission coefficient $\tau(\theta)$ with appropriate weighting across all angles of incidence,

$$\tau_d = 2\int_0^{\pi/2} \tau(\theta)\cos\theta \sin\theta \, d\theta. \tag{1}$$

The $\cos\theta$ term is the cross-sectional area of the plane sound wave that is incident on a unit area of the wall at an angle of incidence of θ to the normal to the wall. The $\sin\theta$ term is due to the fact that the annulus of solid angle between θ and $\theta + \delta\theta$ is $2\pi \sin\theta \, \delta\theta$. The 2 term is a normalisation factor which arises from the fact that τ_d must be 1 when $\tau(\theta)$ is 1 for all values of θ. Equation (1) can be rewritten in a number of forms. Use will be made of the following form

$$\tau_d = \int_0^1 \tau(\theta) \, d(\cos^2\theta). \tag{2}$$

For a thin plane wall with the properties described above (see Cremer 1942)

$$\tau(\theta) = 1 / | 1 + Z\cos\theta / 2\rho_0 c |^2, \tag{3}$$

where Z is the bending wave impedance of the wall and $\rho_0 c$ is the impedance of air, being the product of the ambient density ρ_0 and the speed of sound in air c. If the damping and stiffness of the wall are ignored, then Z is equal to $j\omega m$ where ω is the angular frequency and m is the mass per unit area of the wall. This means that

$$\tau(\theta) = 1 / (1 + a^2\cos^2\theta), \tag{4}$$

where

$$a = \omega m / 2\rho_0 c. \tag{5}$$

The normal incidence sound transmission coefficient is

$$\tau(0) = 1 / (1 + a^2) \approx 1 / a^2, \tag{6}$$

where the approximation applies for the usual case of $a >> 1$.

Evaluating equation (2) with $\tau(\theta)$ given by equation (4) gives the diffuse field sound transmission coefficient

$$\tau_d = (1 / a^2)\, ln\, (1 + a^2) \approx -\tau(0)\, ln\, [\,\tau(0)\,], \tag{7}$$

where the approximation is valid for $a >> 1$. Unfortunately τ_d does not agree very well with experimental results. Better agreement with experiment is obtained by limiting the range of angles of incidence over which the sound transmission coefficient is averaged from 0° to a value θ_l between 78° and 85° (see Sharp (1978)). The value obtained is called the field incidence sound transmission coefficient and is given by

$$\tau_f = \int_{\cos^2\theta_l}^{1} \frac{d\,(\cos^2\theta)}{1 + a^2\cos^2\theta}$$

$$= (1 / a^2)\, ln[\,(1 + a^2) / (1 + a^2\cos^2\theta_l)\,], \tag{8}$$

which if $a >> 1$ becomes

$$\tau_f \approx -\tau(0)\, ln\, [\,\tau(0) + \cos^2\theta_l\,]. \tag{9}$$

If $\tau(0) << \cos^2\theta_l$ then

$$\tau_f \approx -\tau(0)\, ln\,(\cos^2\theta_l), \qquad (10)$$

and τ_f differs from $\tau(0)$ by a constant factor.

The physical reason for the need to introduce a limiting angle is that the experiments are performed on a finite size wall while the theory assumes a wall of infinite extent. The finite size of the wall means that a bending wave of single wavenumber becomes a band of wavenumbers. For θ near 90° some of the bending wave energy will have wavenumbers which are greater than the wavenumber in air and thus will be unable to radiate. Some of the bending wave energy will have smaller wavenumbers for which the predicted sound transmission coefficient is less. Both of these effects mean that to obtain a reasonable answer the upper angle of incidence in the integral has to be limited.

This approach works fairly well for thin single walls but fails badly for cavity walls. The reason is that cavity wall theories are much more sensitive to the value of the limiting angle because they involve the square of the single wall sound transmission coefficient and thus vary with angle of incidence (θ) as $1/\cos^4\theta$ instead of $1/\cos^2\theta$.

The limiting angle is needed because of the finite size of the wall. Thus it would be expected that the limiting angle should depend on the ratio of a typical wall size dimension to the wavelength of sound, rather than be a constant. It turns out that this is indeed the case. Sewell (1970) has shown for a single wall of the type considered in this paper that

$$\tau_f = \tau(0)\, ln\,(k\sqrt{A}), \qquad (11)$$

where k is the wavenumber of the sound in air and A is the area of the wall. All but the first term in the brackets in Sewell's equation (54) have been ignored because they are usually insignificant. Comparing equation (11) with equation (10) shows that the limiting angle θ_l is given by

$$\cos^2\theta_1 = 1 / k\sqrt{A}. \tag{12}$$

This limiting angle is used for both single walls and cavity walls.

The bending wave impedance of a thin wall is (see Cremer 1942)

$$Z = jm\omega\,[\,1 - (\omega/\omega_c)^2\,(\,1 + j\eta\,)\,\sin^4\theta\,], \tag{13}$$

where ω_c is the angular critical frequency of the wall (that is, the angular frequency at which the wavelength of free bending waves in the wall equals the wavelength of sound in air), and η is the damping loss factor of the wall. If the damping loss factor and the angular dependence are ignored, it is not surprizing to find that Sewell (1970) has shown that below the critical frequency the a of equation (5) becomes

$$a = (\,\omega m/2\rho_0 c\,)\,[\,1 - (\,\omega/\omega_c\,)^2\,]. \tag{14}$$

To calculate the field incidence sound transmission coefficient below the critical frequency equations (14), (12) and (8) are used. Above the critical frequency use is made of the equation developed by Cremer (1942),

$$\tau_f = (\,1/a^2\,)\,(\,\pi/2\eta\,)\,/\,(\omega/\omega_c - 1\,), \tag{15}$$

where a is given by equation (5) not equation (14). Both these methods give indeterminate values at the critical frequency. In the region of the critical frequency use is made of the equation developed by Josse and Lamure (1964),

$$\tau_f = (\,1/a^2\,)\,(\,\pi/2\eta\,)\,/\,b, \tag{16}$$

where a is given by equation (5) not equation (14), and b is the ratio of the bandwidth of the filter used in the measurements to the centre frequency of the filter. For a third-octave filter b is equal to 0.2316. Equation (16) is used if it gives a smaller value than equation (8) below the critical frequency or equation (15) above the critical frequency.

AIRBORNE TRANSMISSION ACROSS THE CAVITY

For cavity walls the airborne sound transmission across the cavity and the stud-borne sound transmission across the cavity are treated separately. The starting point for the prediction of the field incidence sound transmission coefficient for the airborne sound transmission across the cavity is equation (C-10) of Rudder (1985) which is derived using the approach of Mulholland *et al.* (1967). This equation can be written as follows,

$$\tau(\theta) = 1 / [\, R^2(\theta) + I^2(\theta)\,], \tag{17}$$

$$R(\theta) = 1 - A_1 A_2 (\, 1 - r^2 \cos 2\beta\,), \tag{18}$$

$$I(\theta) = A_1 + A_2 - r^2 A_1 A_2 \sin 2\beta, \tag{19}$$

$$a_i = (\, \omega m_i / 2\rho_0 c\,)\, [\, 1 - (\, \omega / \omega_{ci}\,)^2\,], \tag{20}$$

$$A_i = a_i \cos\theta, \tag{21}$$

$$\beta = kd \;\cos\theta, \tag{22}$$

where m_i and ω_{ci} are the mass per unit area and angular critical frequency of the ith leaf of the cavity wall (i = 1 or 2), d is the cavity width and r is the reflection factor of the cavity. Some algebra shows that

$$1/\tau(\theta) = \left(\sqrt{(A_1^2+1)(A_2^2+1)} \; - r^2 A_1 A_2 \right)^2 +$$

$$4r^2 A_1 A_2 \sqrt{(A_1^2+1)(A_2^2+1)} \; \sin^2\!\left(\beta - \tfrac{1}{2} \arctan\!\left(\frac{A_1+A_2}{A_1 A_2 - 1} \right) \right). \tag{23}$$

$\tau(\theta)$ has its maximum values at resonance when the sin argument in the second term is equal to an integer multiple of π. The mass-air-mass resonance occurs when the sin argument is equal to zero. Above the normal incidence mass-air-mass resonance frequency, there is always an angle of incidence at which the mass-air-mass resonance occurs. Except for values of the cavity reflection factor r which are very close to one, the 'bandwidth' of the resonance in terms of angle of incidence is broad. This means that $1/\tau(\theta)$ in the integral over angle of incidence can be approximated by ignoring the second term in equation (23). Using the fact that A_i is usually large compared with one, the first term of equation (23) can be approximated to obtain

$$\tau(\theta) = \frac{1}{(q + p\,x)^2}, \tag{24}$$

where

$$q = \frac{1}{2}\left(\frac{m_2}{m_1} + \frac{m_1}{m_2}\right), \tag{25}$$

$$p = a_1 a_2 \alpha, \tag{26}$$

$$\alpha = 1 - r^2, \tag{27}$$

and

$$x = \cos^2\theta. \tag{28}$$

α is the absorption coefficient of the cavity.

The field incidence sound transmission coefficient is

$$\tau_f = \int_{\cos^2\theta_1}^{1} \frac{dx}{(q + p\,x)^2}$$

$$= \frac{1 - \cos^2\theta_1}{(q + p\cos^2\theta_1)(q + p)}, \tag{29}$$

where $\cos^2\theta_1$ is equal to $1 / k\sqrt{A}$ by equation (12). Equation (29) is only valid between the normal incidence mass-air-mass resonance frequency and the lower of the critical frequencies of the two cavity leaves. Further work is planned to extend the theory in both directions.

When the reflection factor r is very close to one a method similar to that used by Cremer (1942) in deriving equation (15) can be used. The idea is that the main contribution to the integral over $\cos^2\theta$ comes from angles of incidence close to the oblique mass-air-mass resonance which occurs when the argument of the sin term in equation (23) is zero. Under these circumstances the sin function can be approximated by its argument and the other terms replaced by their value at the oblique mass-air-mass resonance angle. The range of integration is extended from $\cos^2\theta_1$ to 1, to $-\infty$ to $+\infty$, and use made of the fact that A_1 and A_2 are usually very much larger than one. This approach yields the following formula,

$$\tau_f = \frac{\pi\sqrt{s}}{a_1 a_2 r\,(\sqrt{t} + 1/\sqrt{t}\,)\,(\sqrt{(st + 1)\,(s/t + 1)} - r^2 s)} \tag{30}$$

where

$$s = \frac{M_1 + M_2}{4\rho_0 d}, \tag{31}$$

$$t = M_1 / M_2, \tag{32}$$

and

$$M_i = m_i\ [\,1 - (\omega/\omega_{ci})^2\,]. \tag{33}$$

This formula is not of much practical use because r will always be somewhat less than 1 because of sound transmission through the leaves, air absorption and boundary absorption of the leaves. However it does agree fairly well with the numerical calculations of London (1950) for a limp, lossless cavity wall above the normal incidence mass-air-mass resonance frequency.

The ranges in which reflection factor equations (29) or (30) should be used have not yet been investigated. For the moment the spreadsheet model uses equation (30) only if

$$r > 1 - \frac{1}{\sqrt{a_1 a_2}}. \quad (34)$$

Equation (29) is used for all other values of r. Like equation (29), equation (30) is only valid between the normal incidence mass-air-mass resonance frequency and the lower of the critical frequencies of the two cavity leaves.

STUD-BORNE TRANSMISSION ACROSS THE CAVITY

The stud-borne sound transmission theory was originally developed by Sharp (1973,1978), Sharp *et al.* (1980) and Cremer *et al.* (1973). Fahy's (1985) approach to the theory was followed, but the cavity leaves were allowed to have different masses per unit area and different critical frequencies. It was also assumed that the studs had a mechanical compliance of C_M where $C_M = 0$ gave the rigid stud case of previous authors. Gu and Wang (1983) have also treated the case of resilient studs but their formulae are different to those in this paper and are not obviously an extension of Sharp's formulae as is the case with those in this paper.

This approach produces a value of $\tau(\theta)$ that varies with angle of incidence θ as $1/\cos\theta$. Thus from equation (1) we need to evaluate an integral of the form

$$2 \int_0^{\pi/2} \sin\theta \, d\theta = 2. \quad (35)$$

Strictly speaking the upper limit of the integral should be θ_l, but the value of equation (35) will not be changed significantly by setting the upper limit to $\pi/2$. The field incidence sound transmission coefficient is

$$\tau_f = \frac{64\rho_0^2 c^3}{[\, g^2 + (4\omega^{3/2} m_1 m_2 c C_M - g\,)^2\,] h b \omega^2}, \quad (36)$$

where

$$g = m_1\omega_{c2}^{1/2} + m_2\omega_{c1}^{1/2}, \tag{37}$$

$$h = [1-(\omega/\omega_{c1})^2]^2[1-(\omega/\omega_{c2})^2]^2, \tag{38}$$

and b is the spacing between the studs.

Equation (36) is only a lower limit because it does not include resonant radiation since it assumes an infinite wall and that the studs do not interact. It also assumes that the bending waves induced in the wall by the incident sound are incident normally to the studs. To improve the agreement between theory and experiment and between equation (36) and Sharp's formula, equation (36) was multiplied by an empirical constant of 2. This difference arises because Sharp assumed that the correction factor for averaging over angle of incidence was the square of his factor for a single leaf wall, namely 1.9^2, while our theory shows that according to equation (35) the factor should be only 2. Sharp's formula also includes an empirical 5 dB correction factor which is compensated for in our theory by the square of the ratio of the sum of the wall impedances to the value of one of them, which in the case of identical leaves gives a factor of 4. Surprizingly one of Sharp's (1978) formulae includes both the 5 dB correction factor and the impedance ratio, but it appears that he has not compared this formula with experiment. Formula (36) is, like formulae (29) and (30), only valid between the normal incidence mass-air-mass resonance and the lower of the critical frequencies of the two leaves. Work is planned to extend the theory in both directions.

The total field incidence sound transmission coefficient of a cavity wall is determined by adding equation (29) (or (30)) to twice equation (36).

COMPARISON WITH EXPERIMENT

Figure 1 shows a comparison of theory with experiment for a single leaf wall of 13 mm gypsum plasterboard. Three experimental results are shown. The first is for no studs, while the second and third are for 50 × 100 mm wooden studs spaced at 400 or 600 mm centres respectively. The experimental results show that the studs do not make any significant difference, while the theory slightly but significantly overestimates the experimental results in the lower frequencies. The experimental results were measured by the National Research Council of Canada, and obtained from DuPree's (1981) catalogue. The theoretical results at and above the critical frequency are dependent upon the choice of 0.03 for the damping loss factor of gypsum wallboard.

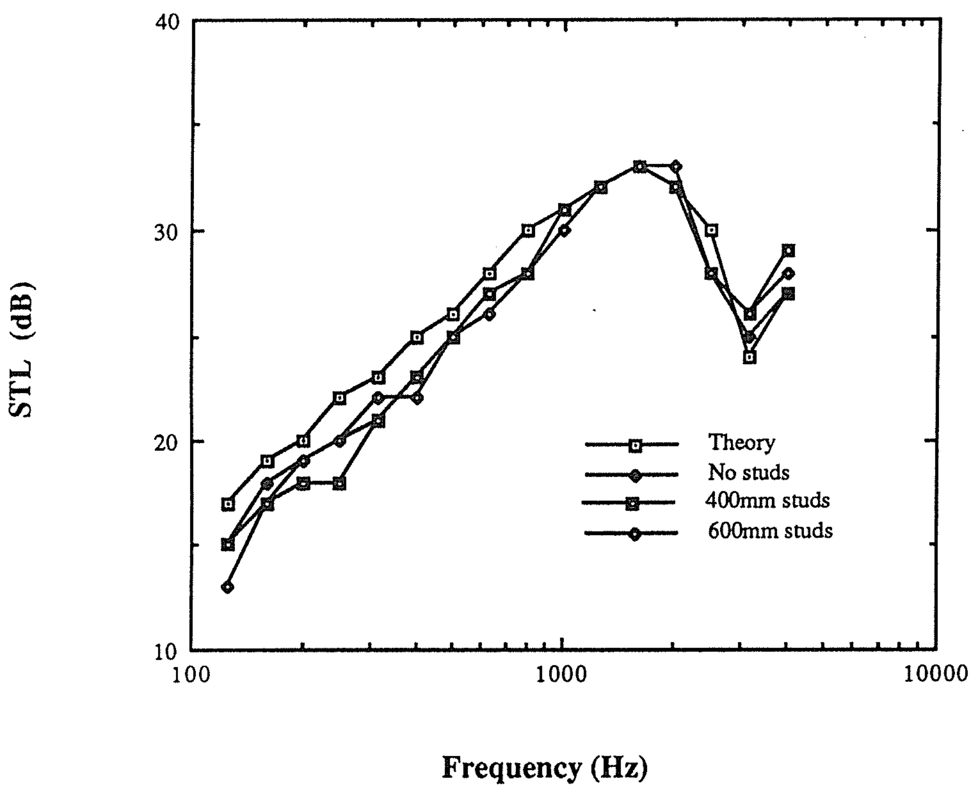

Figure 1. 13 mm gypsum wallboard.

The results for a double stud 16 mm gypsum wallboard cavity wall with cavity absorption are shown in Figure 2. This is a case where there is no vibration connection between the two leaves of the wall and hence only the airborne transmission formula (29) is involved. The theoretical results for cavity walls in this paper are only given for frequencies below the critical frequency since the theory is not valid above the critical frequency. The value one was used for the cavity absorption for cavity walls in this paper since the experimental results show little dependence of STL on type or thickness of cavity absorption. All the experimental cavity wall results in this paper were obtained from the NAHB (1971) manual.

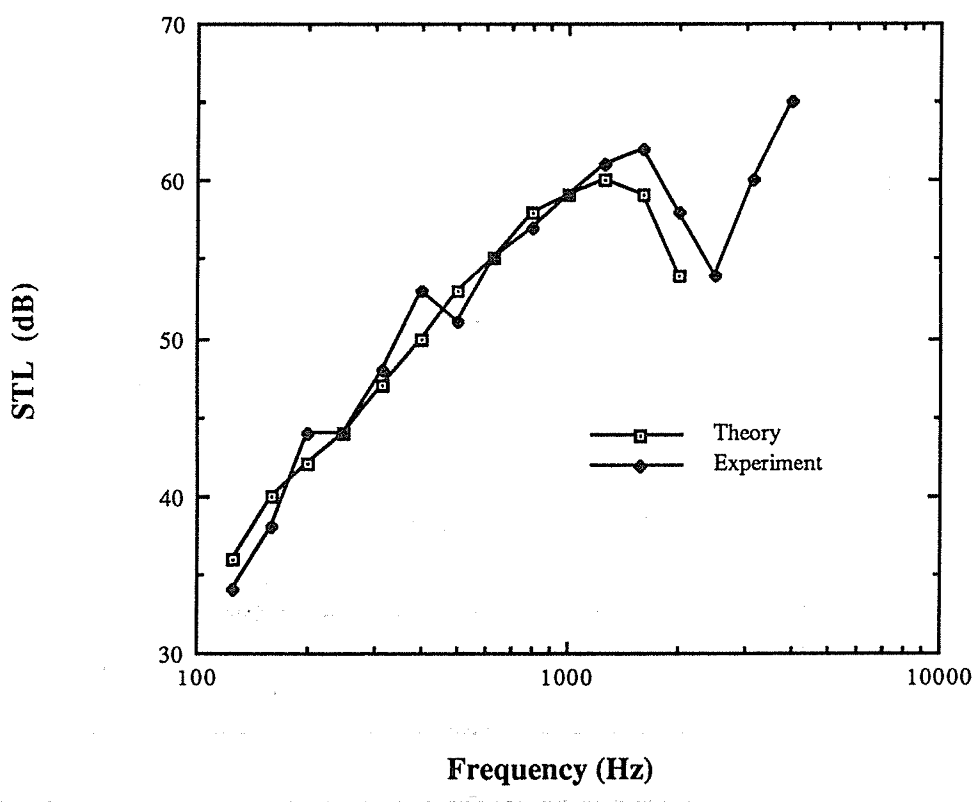

Figure 2. Double stud 16 mm gypsum wallboard cavity wall with cavity absorption.

Figure 3 shows the case for resilient studs. It is for a steel stud 13 mm gypsum wallboard cavity wall with cavity absorption. The stud spacing is 600 mm and the stud mechanical compliance used in the theory was 1×10^{-6} m/N. Experimental results are shown for two different stud widths, namely 63 and 92 mm. It can be seen that cavity width makes little difference to the results. The 63 mm results are the average of four test reports while the 92 mm results are the average of three test reports. It is only at the low frequencies that the transmission through the studs has an effect on the theoretical results.

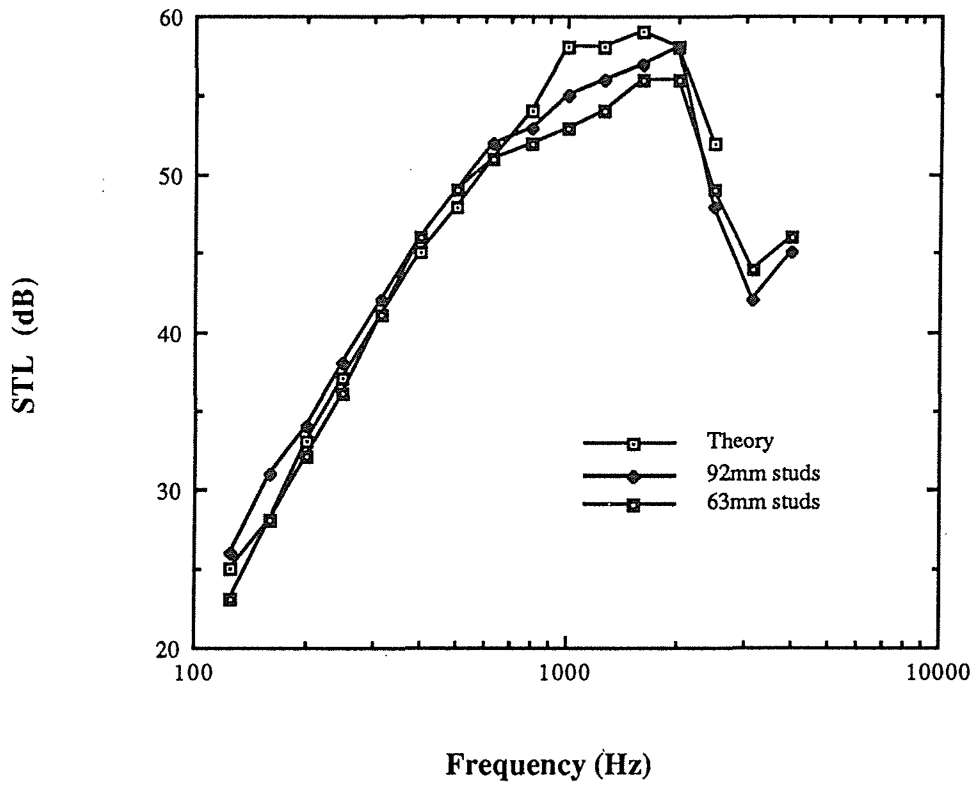

Figure 3. Steel stud 13 mm gypsum wallboard cavity wall with cavity absorption.

The results for a wooden stud 16 mm gypsum wallboard cavity wall with cavity absorption are shown in Figure 4. The studs are 50 × 100 mm spaced at 600 mm centres. The experimental results are the average of two test reports. The theoretical sound transmission is determined mainly by the transmission across the non-resilient studs.

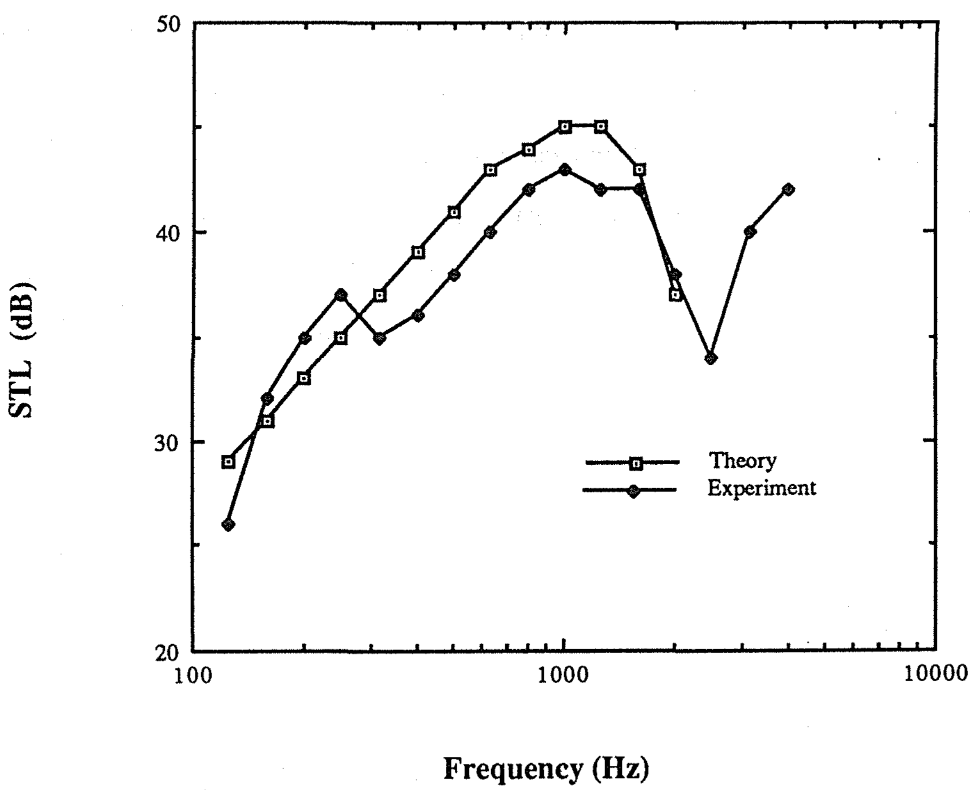

Figure 4. Wooden stud 16 mm gypsum wallboard cavity wall with cavity absorption.

No results for cavity walls without absorption are given in this paper. This is because it is necessary to decide what value of cavity absorption to assign to the cavity. The theory has been evaluated in this case by calculating the values of absorption necessary to give reasonable agreement with experimental results, and seeing if the values of absorption are reasonable. The values of cavity absorption obtained in this way appear to be reasonable for the current theory. However some early theories were rejected because they needed unrealistic values of cavity absorption to agree with experimental results.

CONCLUSION

The theory in its current form gives fairly reasonable agreement with the experimental results for the sound transmission loss of gypsum plaster walls. However, much more development of the theory is needed. In particular, the theory has to be extended to cover the case of cavity walls at and above the critical frequencies of the leaves. The sound absorption coefficients of cavities without sound absorbing fillings need to be investigated and the theory needs to be compared to walls constructed of other materials.

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LABELLING

A USEFUL STRATEGY IN ENVIRONMENTAL PROTECTION

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Abstract

An outline is given of the general labelling environment in Australia. The theory of noise labelling is analysed against this backdrop and some brief reference is made to overseas practice and experience. Strengths and weaknesses are enumerated and practical problems are identified and discussed. Suggestions are invited on the question of whether or not noise labelling might be able to benefit from the green spot initiatives presently being developed.

1.0 INTRODUCTION

There has been a recent rediscovery in Australia, of the potential benefits inherent in *labelling* as an environmental protection strategy. As a result, state governments are being asked to put their weight behind a ***green spot*** scheme which, if successful, would see labelling become a big business in itself. This paper will briefly outline the general features of green spot thinking about labelling. It will also outline the theory of ***noise*** labelling and comment on successful applications of this theory in Australia and overseas. Practical considerations concerning the conduct of a noise labelling strategy and its strengths and weaknesses will be featured in the discussion.

Unfortunately, we are not here able to address the perplexing question as to whether or not noise labelling might benefit by becoming part of the green spot scheme. Green spot thinking to date, neither acknowledges nor excludes noise as an environmental pollutant and there is some interesting research and analyses awaiting someone seeking a challenge.

2.0 RECENT INTEREST IN LABELLING IN AUSTRALIA : GREEN SPOT THINKING

The green spot approach to labelling is based upon the belief that there is, in Australia, a genuine interest in protection of the environment and a strong community support for the idea of an independent labelling scheme to signal to consumers the certainty that particular products are environmentally friendly. The scheme is viewed as but one element of a wide ranging environmental education program properly and professionally marketed on an on-going basis. Under the scheme, a government owned, not for profit firm, governed by a board of directors selected from industry, would establish criteria upon which to subsequently grant products the right (for a small fee to cover running costs) to carry the environmentally friendly green spot logo. There is to be public input into criteria, the firm is to be charged with assuring confidentiality with respect to technical matters, and there is to be close involvement with industry.

The scheme is to begin by establishing criteria which will serve to endorse specific product categories: unbleached paper products; chlorine-free paper products; recycled paper and products made from recycled plastics, and an initial two year licence of the green spot is envisaged. The green spot logo is to incorporate words and there is to be a substantial marketing effort to ensure its acceptance. For the first three years, participating governments are to provide funds to establish the scheme which, (through annual licence fees starting at a minimum of $1,000 and escalating as a function of turnover) is designed to be self-funding by 1992. The objectives of the scheme are stated:

. to complement and add to the general effort to raise consumer awareness of environmental considerations

. to assist consumers to make informed choices based on environmental considerations

. to improve the market share of products which are comparatively beneficial for the environment

. to build consumer confidence in the face of competing claims about the environmental benefit of different products

. to stimulate development of new environmentally sound products

and, if all of the State Governments pull together (sic), there is every possibility that the scheme could be of great benefit. The primary environmental criteria, together with additional environmental criteria with respect to consumer products are stated:

. they cause substantially less pollution than other comparable products in production, usage and disposal

. they are recycled and/or recyclable where comparable products are not

- they make a significant contribution to saving non-renewable resources or minimising use of renewable resources compared with other comparable products

- they contribute to a reduction of adverse environmental health consequences

- the source of the raw materials and the likely impact on the environment in accessing those materials

- the energy used for production of a product

- the environmental effects of wastes arising from the production process and method used to dispose of gaseous, liquid and solid wastes. In these respects, product manufacturers must comply with State and Federal environmental regulations.

There has been some opposition to green spot labelling from the industry lobby which has called it a 'simplistic public relations exercise' which would have 'little credibility because of the lack of objective criteria to determine the environmental acceptability of many products'. But in spite of this criticism the Australian and New Zealand Environment Council has decided to proceed with it.

Is there a case for noise control to benefit from the considerable public awareness and public participation activities that a successfully marketed green spot program would generate? As was stated at the outset, an answer to that question is not attempted. Some hard thinking on this issue by noise professionals could yield worthwhile results. A common sense response to this question suggests that noise labelling could benefit from the green spot movement and it is to a discussion of noise labelling that this paper now turns.

3.0 THE THEORY OF NOISE LABELLING, THE PRESENT STATE-OF-THE-ART IN AUSTRALIA AND OVERSEAS, AND THE BASIC CONDITIONS NEEDED FOR ITS SUCCESS

3.1 <u>The Theory</u>

In its simplest form noise labelling requires that manufacturers and retailers are required to attach to particular categories of equipment a label which informs potential consumers of the noise emitted by the equipment during a prescribed test. Other things being equal, consumers might favour quiet machines in their purchasing, so that in time market forces become an important element in maintaining a quiet environment. In its more detailed form labelling seeks to serve architects, town planners, health authorities, industrial designers, trade unionists, employers and industrial hygienists through a requirement that manufacturers must supply particular noise information in handbooks and specification sheets at or prior to the point of sale. Informed purchase on the basis of noise information will help bias the market towards selecting the quiet environment.

3.2 The Present State-of-the-Art

Work on labelling, started in 1974 by Technical Committee 43 of the ISO, came to fruition in 1984 when ISO Standard 4871 was published. This standard, a revision of an earlier edition, deals with classification and noise labelling of machinery and equipment and specifies the A weighted sound power level as the appropriate labelling descriptor. In 1985 the ISO released Standard 7574 Parts I to IV, dealing with statistical methods for measuring and verifying stated (labelled) noise emission values of machinery and equipment. Part I concerns definition. Part II refers to the case for single machines and details labelled values and compliance with those values. Part III outlines a transition method for managing batches of machines for which the parameters required in Part IV have not been specified. Part IV explains statistical sampling methods for checking the noise emission levels of batches.

ISO 7574 is now in the process of being incorporated into the European Standards, which must in turn be incorporated into the national standards of the member states of the European Standards bodies CEN and CENELEC. EEC directive 86/594, on the labelling of household equipment also applies Parts III and IV of ISO 7574, and Technical Committee 59 of the International Electrotechnical Commission is basing its measurement and labelling codes for household equipment on this standard. EEC directive 86/18, which requires that members specify regulations for the declaration of noise emission of machines likely to cause hearing damage, may well incorporate ISO 7574.

In the USA, American National Standard S 1.23 - 1976 addresses labelling and specifies as a descriptor the Noise Power Emission Level (NPEL), defined as the A weighted sound power level (re. one picoWatt) expressed in bels and rounded to the nearest 0.1 bel. This standard (i.e. ANSI 1.23 - 1976) was adopted in a modified form, by the European Computer Manufacturers Association, as ECMA - 109 and was submitted to the ISO; it now exists as ISO DIS 9296, which is in accordance with ISO 4871 mentioned earlier.

The considerable work on international standardisation means that nations willing to call up the standards in their legislation can collectively contribute to the removal of a potential barrier to trade (differing and confusing standards) and at the same time harness the power of consumer sovereignty in selecting the quiet environment.

Work on labelling has also been progressing independently of the standardisation drive. Australia introduced labelling of chainsaws and grass cutting machines in the 1970s as part of a strategy to enforce maximum permissible limits. A new interest in consumer awareness in 1985\86 led to legislation requiring the labelling of domestic air conditioners, mobile air compressors and pavement breakers. There is also current debate in Australia as to whether Australian Standards on labelling should be in line with the statistical sampling methods used in ISO 7574, which is certainly suitable for large numbers machines, or whether they should follow requirements in current legislative practice. If Australian Standards are not brought into harmony with ISO Standards, some imported machines will require two labels if sold on the Australian market.

In France the Association Francaise de Normalisation (AFNOR) has for some time monitored the acoustic quality of consumer products. AFNOR holds the NF trademark, which if awarded guarantees the product under specified parameters. The NF trademark is guaranteed in law and developments are presently in hand to bring the benefits of this trademark to noise labelling. The proposal is to link the trademark to ISO standards within an overall quality control framework aimed at NF certification of participating companies.

The Federal Republic of Germany has also hosted independent action on noise labelling. Since 1979 there has existed an Ecologically Beneficial Product Labelling Committee. This committee has voted for the labelling of very low noise lawn mowers. The manufacturer, provided his product meets committee specifications, can for a yearly fee avail himself of its label.

In Finland collaboration between the National Board of Labour Protection and VTT/Occupational Safety Engineering has led to a proposal involving labelling. Emission limits are specified for seven categories of machine within three generally defined industrial activity groups. It is the position within this classification system which prescribes for manufacturer the kind of acoustic information they are obliged by law to supply in handbooks and technical specifications. Some general provisions also apply and acoustic information supplied must be determined according to ISO 7574. In Austria, labelling has been of interest since the 1960's. It is presently specified under Austrian standards, which will be reviewed to accommodate ISO 7574 when this standard itself is introduced by CEN. An important feature of the Austrian approach is its model Invitation to Tender form. This form helps purchasers to specify noise parameters in their contracts. At present labelling in Austria is not subject to legislation.

3.3 Basic Conditions Needed for the Success of Noise Labelling

A number of conditions need to be fulfilled to ensure that labelling can function effectively:

(a) There must be a clearly defined and universally acceptable noise descriptor which, when used in labelling, easily communicates to the buyer the noise emission level of the machine, thereby aiding him through simple comparison to rank it within the range of machines on the market.

(b) The characteristic noise descriptor outlined in (a) must be linked to a clearly defined measuring method.

(c) There must be a State guaranteed methodology for verification of the labelled values. This methodology must be a low cost strategy which can bring considerable pressure to areas suspected of fraud.

(d) There must be a continuing and complementary education/marketing strategy which explains labelling to successive generations and which from time to time might be used to inform the public about new technologies and developments.

(e) Consumer rights against false labelling should be well publicised and accessible at law as low cost.

(f) Where possible noise abatement legislation should advantage the manufacturers of low noise products by allocating special favours to consumers of those products. For example in the Federal Republic of Germany owners of low noise lawn mowers are permitted extended hours of use over and above those generally available.

Concerning (a) (clear definition and educated acceptance of a noise descriptor) there is general agreement that the sound power level is the appropriate measure, but disagreement exists about nomenclature, the two contenders being decibels (A) and bels. More detailed labelling information, including an explanation of the noise descriptor and advice about how to implement the most suitable protection measures, should be provided in instruction materials and specifications available at the point of sale. Where warranted the label might contain a simple warning to use hearing protection devices, but it should not be allowed to become cluttered with information; its strength is in its simplicity. It is very important that manufacturer should be obliged to state the actual emission levels of their products, rather than to state that the machines are below upper benchmark levels. This requirement is essential if consumer sovereignty is to dictate the quiet environment.

Concerning (b) and (c) (measurement and testing), the hard work has been done and the reader is referred to the publications of the standardisation bodies and others described previously.

Marketing and education initiative with respect to labelling (d), although relatively simple and inexpensive in itself, has hardly begun in most countries. Policy makers need to be aware of the importance of this strategy and to include it in noise programmes right from the beginning. Effective outcomes from this strategy are typically in the long term indicating that marketing and education must be continuing.

To bring forward an efficient and less costly procedure for the consumer to access his rights at law (e) may prove to be very difficult. However if fought for and won it may be a very powerful incentive for firms to adhere to honest labelling.

Finally, the incentives to manufacture condition (f) is presently part of the labelling strategy in the Federal Republic of Germany; it appears to be working well in that country.

3.4 Strengths and Weaknesses

The labelling strategy cannot begin to operate unless the consumer responds rationally to labelling information, but impediments to communication do exist. Firstly, countries which import most of their consumer and process machinery may opt to rely on the labelling strategies of other countries. This is a mistake because labelling is costly and exporting countries may choose to exempt their exports from labelling requirements. Secondly, other groups have interests in labelling (design, power efficiency, safety/ergonomics, consumer protection, green spot) and too much information on one label, on competing labels, or in a single specification schedule may well result in switch off for some consumers. Synergistic relationships should be exploited wherever possible in labelling if this helps to keep the communicated message simple for the buyer. Thirdly, governments around the world are good at exempting themselves from their own legislation, either directly or indirectly by failing to use their considerable influence in exploiting opportunities. Governments which fail to write labelling into their purchasing criteria and make manufacturers aware of this are depriving the strategy of a very powerful influence. Fourthly, there is the ever present phenomenon of cognitive dissonance where human health and safety are concerned and there is always a substantial proportion of the population who will ignore the label.

4.0 CONCLUSION

Noise labelling in Australia and overseas has begun to emerge as a viable strategy in occupational noise and environmental amenity. This emergence is now being shadowed by a growing awareness of labelling in general as a strategy in environmental protection and it appears that there is considerable support and enough determination to ensure its introduction on a large scale. There is a danger that the green spot could, to some extent, eclipse the substantial and important noise labelling work already done and this raises the question of whether or not noise labelling initiatives should be modified in some way or another so as to be able to benefit from the wide recognition the green spot is likely to bring.

AN AUTOMATED IMPEDANCE TUBE

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ABSTRACT

The acoustic absorptive properties of materials have a significant influence on interior noise climates. The impedance tube (or standing wave apparatus) has been used for measuring acoustic absorption for decades.

The usual measuring procedure using a travelling microphone with probe tube (Australian Standard 1935) is slow and tedious, although various improvements in technique have been developed (for example, tone-burst, impulsive or two-microphone methods). However, the conventional method has merit in the teaching laboratory: it has the splendid advantage of demonstrating convincingly the existence and theory of standing waves.

An account is given of an automated impedance tube of travelling microphone type. The microphone is traversed by a stepper motor with pinion and rack drive. The motor is controlled by a personal computer, and sound pressure is measured at each step, with all parts stationary and motor electrics quiescent.

A data acquisition system to the PC enables data readout in a variety of forms. For student experiments, raw data of S.P.L. v. distance is presented for students to undertake the data analysis and interpretation entirely. Alternatively, full data reduction (absorption coefficient and impedance) can be selected.

The apparatus provides an effective demonstration of acoustic principles with much less tedium and much greater productivity than hitherto.

1. INTRODUCTION

The acoustic impedance tube (a standing wave apparatus) has been used as a means of determining acoustic absorption properties for decades. The principle was noted at least as early as 1902 [1, page 2] and a commercial version has been available since 1955 [1]. The apparatus requirements and experimental procedures are well established [2,3].

The usual measurement technique of using a travelling microphone (often with a long probe tube) is slow and tedious. Many other procedures have been devised and developed to overcome the tedium. A mechanical drive was reported by Dunlop [4]. Methods which avoid the need for microphone traverse have been developed, using tone-burst, impulsive, or multiple microphone techniques. A review of impedance measurement methods (with 82 references quoted) was given by Singh [5]. Nevertheless, the traditional travelling microphone technique has the advantage of demonstrating effectively the existence of standing waves and so has great merit in the educational sense.

An automated apparatus of the travelling microphone type has been designed and commissioned. It removes the tedium but serves the teaching function of showing the standing wave pattern and verifying the theory of impedance tube behaviour. It can also be used for routine testing.

2. GENERAL ARRANGEMENT

The arrangement is basically a conventional impedance tube with probe type microphone - see, for example, [2] or [3]. The microphone is mounted on a carriage which is driven by a stepper motor. Motor drive is controlled by a Personal Computer (PC) and from a set starting point the microphone carriage is traversed over the required distance in operator-selected steps of up to 4 mm. At each step, the drive stops and the acoustic measurement is taken with all parts stationary and motor electics quiescent.

Acoustic instrumentation is conventional (measuring filtered level of pure tone signal) but signal level is supplied through A/D converter to the data acquisition section of the PC. Data presentation to screen or printer can be in either or both of two forms. The PC can be set either to compute and display the acoustic properties of the sample or to give only a schedule of sound levels v. distance. The latter option is used for the teaching laboratory mode of operation - students perform the exercise of determining absorption coefficient and acoustic impedance of the sample. (The supervisor has a useful option of access to the "right answer") .

3. PARTICULAR FEATURES

The tube has a working length of 0.9 m, which allows a certain lower frequency limit of 200 Hz and a sometimes possible lower limit of 100 Hz, depending on the nature of the sample. Cross-section of the tube is 88 mm square. This would normally set the upper frequency limit at 1900 Hz due to the possibility of transverse standing waves (at 1950 Hz, $\lambda/2$ = 0.088 mm = tube width). In this apparatus, a four-pronged end is fitted to the probe tube with the measurement openings at the quarter points of the diagonals of tube cross-section. With this arrangement the first three transverse standing wave resonances cancel, so that the tube can operate up to the condition $d = 2\lambda$, i.e. to 7800 Hz. Such operation implies that the assumption of plane wave motion in the tube is valid. Comparison tests using a smaller tube have verified results up to 8000 Hz using the 0.088 mm tube and four-pronged probe.

In use, the operator must set frequency and level of input signal to the tube loudspeaker. Data is keyed into the computer : Descriptive text of sample details and configuration, frequency, microphone amplifier attenuator setting, distance of specimen face from reference plane, traverse step size.

Operation with the PC is interactive. According to the frequency, a recommended traverse step is displayed, and the program will select traverse range to give two minima (or, at low frequency, maximum traverse available). If two minima are measured, linear extrapolation to compute "true" level difference is provided (Figure 1), plus computation of frequency. Distance from specimen face to reference plane is converted to distance from specimen face to probe and at start position.

Stepping through the traverse can be operator controlled or automatic. With automatic operation, full traverse takes about two minutes, and full data reduction and printout (for one frequency) is then available. Time saving over manual operation is of the order of 10:1, and full data reduction is completed - absorption coefficient, reflected wave phase change, and impedance (real, imaginary, magnitude, phase).

The principal difficulty has been in the precise identification of maxima and minima (levels and positions). The problems are:-

(a) For highly absorbent material and at low frequencies, the maxima are broad. Random variations in amplifier output create a difficulty in identifying the precise location of a maximum. This problem is overcome by a procedure in which a turning point is only recognised if sign change of the slope of the pattern (level v. distance) has remained constant for a chosen number of consecutive level measurements. Typically the number chosen is three (Figure 2).

(b) For reflective material and high frequencies, the minimum level is measured high due to the finite steps in carriage traverse (Figure 3). An analysis of the errors due to this effect has been carried out and the results are presented in Table 1. From the data given it is possible to choose a suitable step size for any given frequency and anticipated material absorption. The screen display which guides the operator in selecting step size for chosen frequency is based material absorption. The screen display which guides the operator in selecting step size for chosen frequency is based on these results.

Operation of the automated tube has been verified by manual testing in this and other tubes.

TABLE 1. ERROR SCHEDULE AT MINIMA

α_a L_o dB	0.2 25.08	0.4 17.92	0.6 12.95	0.8 8.36	0.9 5.69	0.95 3.95
Δ = 0. 01 L_o* α_a* α_a^* - α_a	 23.88 0.226 0.026	 17.67 0.409 0.009	 12.87 0.6035 0.0035	 8.33 0.8012 0.0012	 5.68 0.9003 0.0003	 3.94 0.9503 0.0003
Δ = 0.02 L_o* α_a* α_a* - α_a	 21.53 0.286 0.086	 16.98 0.435 0.035	 12.64 0.614 0.014	 8.26 0.804 0.004	 5.64 0.9015 0.0015	 3.92 0.9508 0.0008
Δ = 0. 03 L_o* α_a* α_a* - α_a	 19.23 0.355 0.155	 16.05 0.471 0.071	 12.29 0.629 0.029	 8.14 0.809 0.009	 5.59 0.9034 0.0034	 3.89 0.9514 0.0014

α_a = absorption coefficient

L_o = standing wave ratio in decibels

* indicates value for worst error conditon

Δ = sampling spacing as proportion of wavelength

4. FURTHER DEVELOPMENT

Many further options, refinements and developments can be suggested. Some of the possibilities are in course of implementation. Some will be denied to the students for class work.

The apparatus could be made more automatic - by provision of means for machine setting of frequencies (of input and filters) and execution of repeated traverses. The ultimate would be a "one key stroke" operation for all tests on one specimen configuration. This could include tabulation and plotting of results.

In the present arrangements, microphone traverse is in one direction only by equal steps over the entire chosen travel distance. Hence much unwanted data between maxima and minima is recorded and time is wasted. There is scope for rapid traverse and omission of data collection in the regions of unwanted data. A further provision could be a logic-controlled search operation for finding maxima and minima. At present a manual (key-press) override is provided to suppress unwanted data and speed up data collection. For a few results there is merit in requiring students to take and plot the full wave pattern (in both level and pressure forms). The override option also enables amplifier scale change during traverse.

Another possible approach is to take fewer measurements, at widely spaced points, not necessarily at maxima and minima, and then perform a special "curve-fit", i.e., to fit a "pattern curve" to the data. Investigation of this prospect is a proposed student exercise.

5. THE EDUCATIONAL VIEWPOINT

The travelling microphone technique demonstrates clearly the existence of standing waves and therefore has an educational advantage over many other techniques. This automated version does not disguise the behavioural pattern, and has a substantial advantage in that data from a useful number of test samples and frequencies can be collected in one practical session of reasonable length. Manual operation of the apparatus does not permit this - by the time a group of students has developed some expertise, the session is over with results obtained for only a few frequencies for one or two test samples. With the new arrangement, several samples can be tested over a wide range of frequencies. Thus the session extends to a small acoustic investigation, for example, on the effect of thickness of material, or the effect of size of air gap behind absorbent material. Further, each student can be given an individual exercise in data analysis and interpretation of results.

Most of the work of design, development and commissioning has been carried out by undergraduate students as individual projects, and this approach has been of educational value. Further development, for example, on economies of search for standing wave extrema, graphics output, automatic parameter setting, will also be carried out as student projects.

6. CONCLUSION

The automated impedance tube has significant benefits in terms of improving the speed of measurement of absorption properties. For measurement alone, there are strong arguments for using a technique other than the travelling microphone. However, the educational value of the arrangement described is substantial.

ACKNOWLEDGEMENT

Grateful acknowledgement is given to M. O'Sullivan, T. Johnson, S. Dias-Jayasinha and J. de Jongh, and also to Mr. B.C. Daniel of the Department of Mechanical Engineering for data acquisition software and hardware development.

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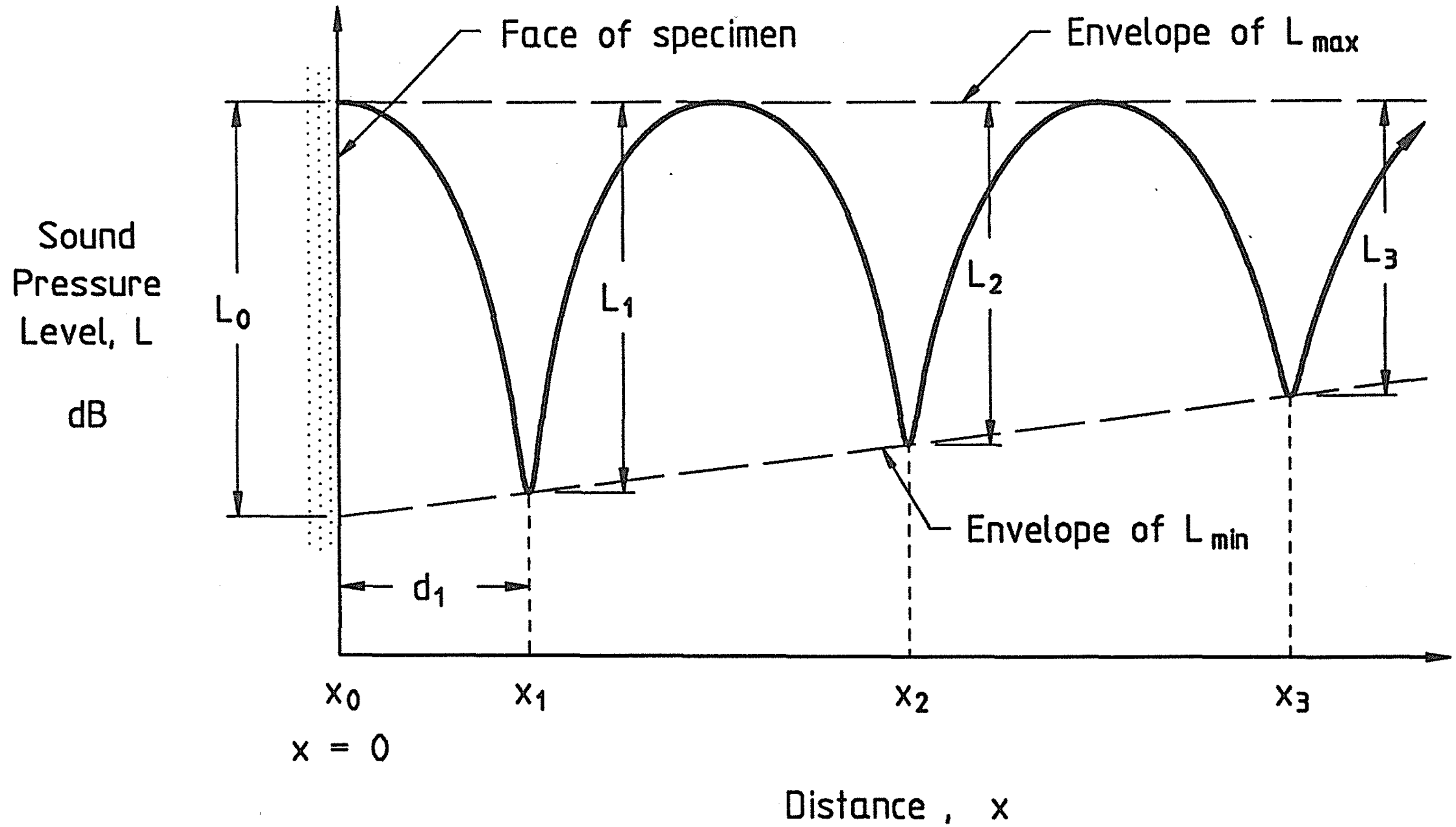

Figure 1. Standing Wave Pattern

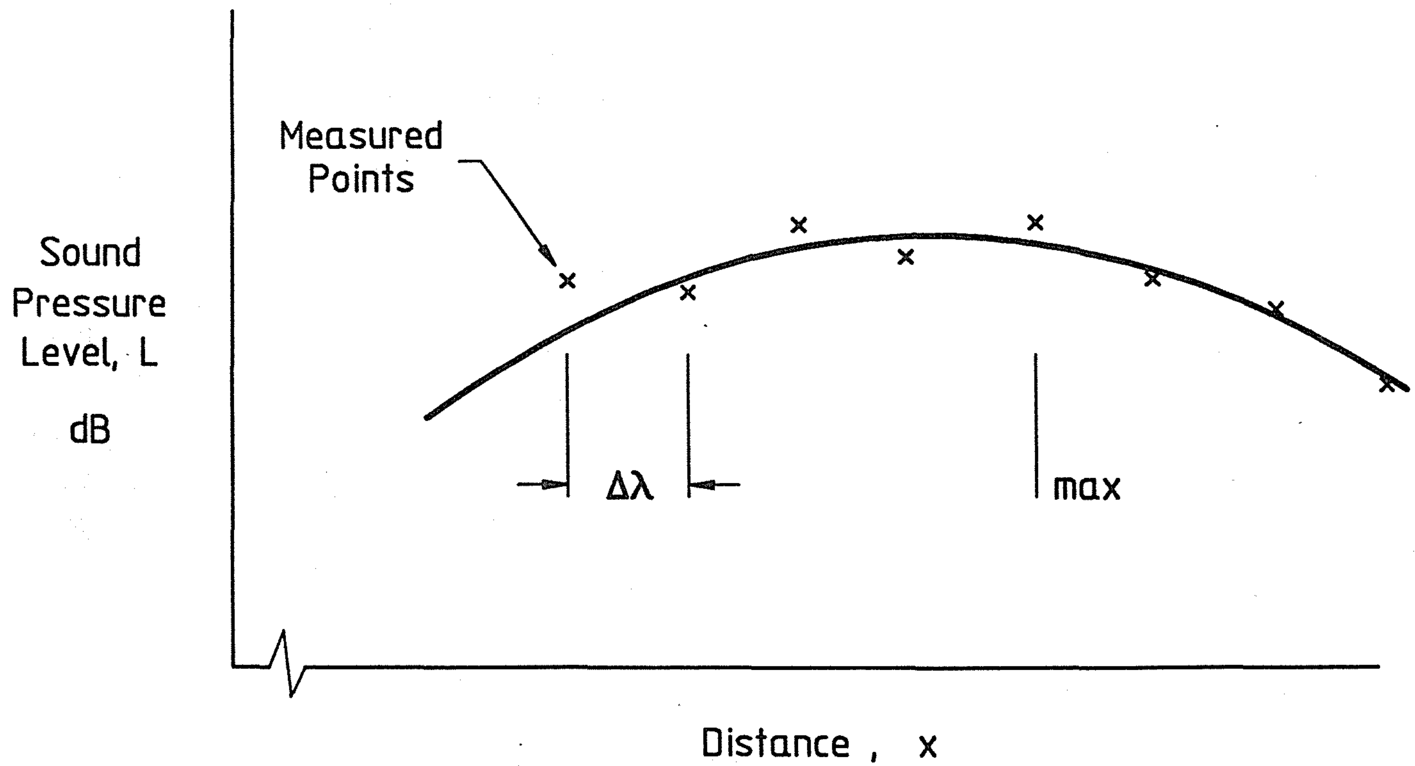

Figure 2. Identification of Maximum

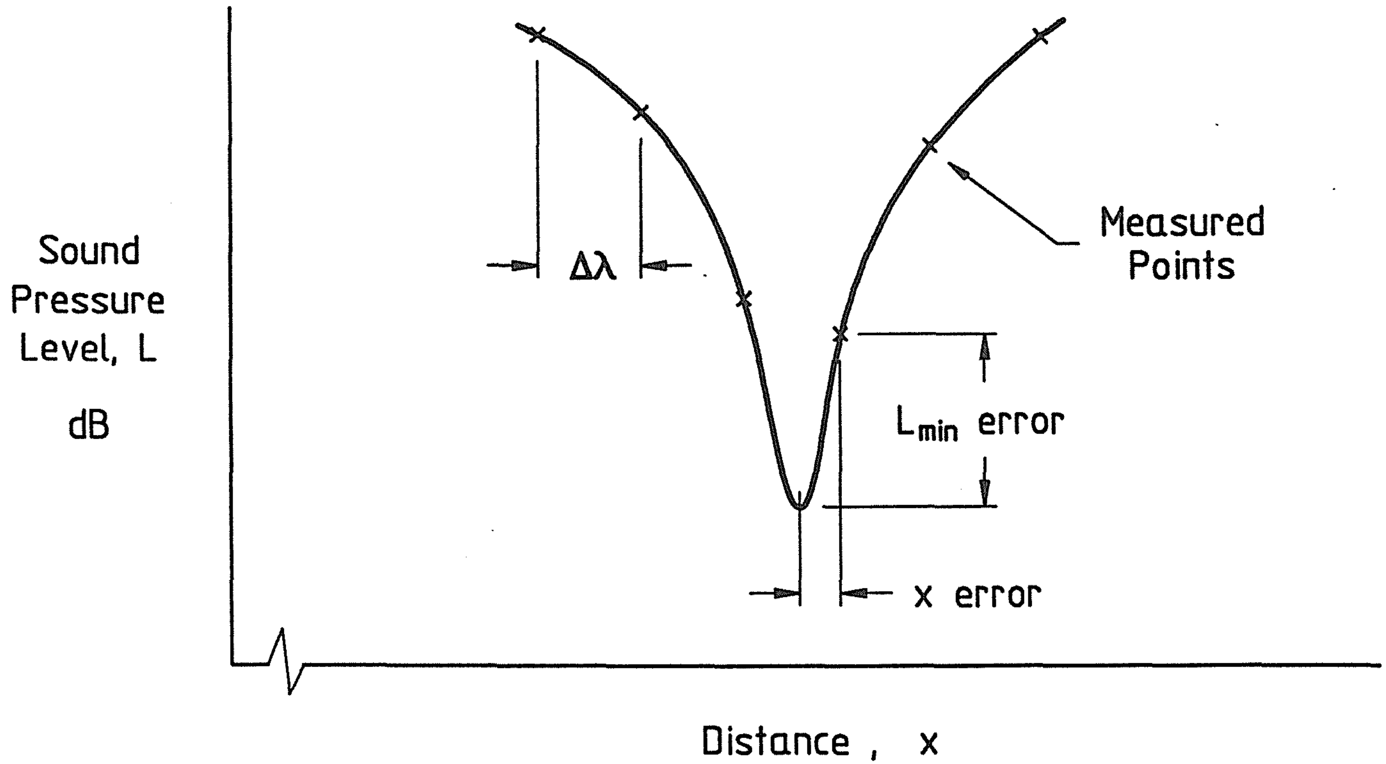

Figure 3. Identificatin of Minimum

OUR NOISY FUTURE

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ABSTRACT

Most justifiable noise complaints received by environmental agencies are a direct result of land use planning decisions that ignored environmental noise. Often these decisions were made many years ago, but there are also many that are being made today that will result in future noise problems.

Most government environment agencies are involved to some extent in land use planning decisions that involve large development but still many planning departments, state development departments, local governments, planning courts and private developers ignore environmental noise in their day to day decisions. It is therefore essential that all people aware of the long term effects of noise actively work to ensure that the future decisions made in Australia do not lead to future noise problems.

This paper will focus on land use planning for noise control. In particular it will look at the relationship between environmental noise control legislation and future planning decisions. It will also look at the need to increase the awareness of all parties involved in planning to recognise the relationship between planning and future environmental noise and will look at reasons why planners dismiss noise as not being a significant planning issue.

The following papers were accepted for presentation at the planned 1989 Conference which was deferred as a result of an airlines dispute. The authors were unable to attend the combined 1989/90 Conference, nor were there alternative presenters available. Given the circumstances, these papers are included in the following section of the proceedings.

ACTUAL AND ACCEPTABLE NOISE LEVELS INSIDE DWELLINGS NEAR MAJOR ROADS AND FREEWAYS - THE CREDIBILITY GAP

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ABSTRACT

When recommended design sound levels for building interiors are compared with expected traffic noise levels it is found that they are not achievable using standard forms of construction for even modest traffic flow rates. Since traffic noise is approximately proportional to the logarithm of traffic flow, it is sensible to concentrate as much traffic as possible onto properly designed major highways and freeways. People living near such highways should be protected as far as possible, by site and building planning and by using special forms of construction - this is costly, but the costs should be included as part of the total road construction package. The benefit to most of the community by discouraging other than local traffic from using local roads is considerable, and should justify the extra expense of attempting to protect those unfortunate enough to be affected by major roads.

INTRODUCTION

Major Australian cities have undergone significant population expansions since World War II, during a time when the conventional perception of Australia was that of an empty continent which could provide unlimited "wide open spaces". This resulted in low-density urban development over ever-widening geographical areas, facilitated by the increasing availability of private motorised transport.

More recently, planners and others have been recommending that the urban sprawl be halted and they are encouraging medium density consolidation. Even if this is successful (despite the "it-is-a-good-idea-but-not-in-my-backyard" syndrome), the widespread existence of low-density development makes viable city-wide public transportation systems unlikely, and thus they will do little to prevent the ever-increasing numbers of private vehicles using the roads.

Not only private motor vehicles are increasing in numbers and usage, the last few decades have also seen a steady erosion of the percentage of freight carried by railways - the balance being taken up by road transportation, with the permitted size and speed of individual goods vehicles also being allowed to increase.

This all adds up to increasing congestion on the urban road system. Innovative traffic management schemes have helped to keep the vehicles moving, but often congestion results in drivers finding new routes and spreading more and more over roads designed for local traffic only. Eventually, the pressure builds up sufficiently for new highways to be constructed. The need for new major roads has usually been anticipated by urban planners and their proposed routes have been determined and publicised many decades previously. New and upgraded roads have the advantage of removing through traffic from what should be local residential streets, and thus the noise problem for some people is alleviated. However, for others accustomed perhaps to living near a "freeway reservation" the actual construction and comissioning of the new road is most unwelcome and it is frequently preceded by vigorous community action and complaints. This is not surprising, since many studies have shown that road traffic is the most prevalent source of excessive noise in urban areas.

TYPICAL ROAD TRAFFIC NOISE LEVELS

There are several well-known methods of describing the noise that will be emitted by streams of traffic. The most common descriptors in use in Australia are $L_{Aeq,T}$ and $L_{A10,T}$, the equivalent A-weighted sound pressure level and the sound pressure level exceeded for 10% of the relevant time period respectively. In most situations where traffic noise is likely to be annoying, the relationship between these descriptors is given by

$$L_{A10,T} = L_{Aeq,T} + 3 \quad \text{dB(A)}$$

thus it is easy to translate from one to the other if the time period, T is the same.

The noise levels that will be emitted by traffic may be predicted if information is available regarding the traffic flow rate, the percentage of heavy vehicles, and the distance of the relevant receiving point from the traffic stream. For more sophisticated prediction methods the vehicle speed, road surface, gradient and the presence of barriers (either topographical or purpose-built) is also taken into account. The road construction authorities in Australia use a method developed by the Department of the Environment in Britain, Calculation of Road Traffic Noise (or the CORTN method)[1]. This method is also referred to in Australian Standard 3761, Acoustics - Road traffic noise intrusion - Building siting and construction.[2]. This Standard gives guidance to planners and building designers regarding the suitability of sites near roads and freeways. It suggests that traffic noise should be considered as a potential problem at various distances from a road, depending on the total traffic flow per 18 hours. (This time period extends from 0600 to 2400 hours, and is chosen to harmonise with the CORTN descriptor, $L_{A10,(18hour)}$). The distances at which potential traffic noise nusiance should be considered range from 50 metres from a road with an 18 hour flow-rate of 1,000 vehicles, up to 1 kilometre from a road with a flow of 20,000 vehicles per 18 hours. At these flow rates and distances, depending on vehicle speeds and the percentage of heavy vehicles, traffic noise levels would be over 50 dB(A), $L_{A10(18hour)}$ and special construction techniques may be required to achieve acceptable sound levels inside many building types.

It should be noted that averaging traffic flows over 18 hours tends to average the traffic noise levels also - typically the hourly flow rate is much higher during the morning and evening peak periods, and it is lower at night - although, as roads become more congested the "peakhours" tend to spread over longer periods. Also, there is no consideration for traffic which uses the roads between 1200 and 0600, and yet, in the Australian context, these are popular times for heavy goods vehicles to travel.

If a typical case is taken of a road with an hourly flow rate of 2000 vehicles, 15% heavy vehicles, a level gradient and an average speed of 70 km/h, the estimated $L_{A10(1hour)}$ level at a distance of 30m from the traffic stream is 73 dB(A). This is aproximately equal to 70 dB(A) L_{Aeq}. All other things being equal, traffic noise levels increase by 3 decibels for each doubling of flow rate, thus if the traffic increased to 4000 vehicles per hour in the above example, the level would be approximately 73 dB(A) $L_{Aeq,T}$ Conversely, if the traffic flow rate decreases from 2000 to 1000 vehicles per hour, the level would be approximately 67 dB(A). (There is also an approximate propagation attenuation relationhip of - 3dB per doubling of distance from the traffic stream, so if the receiving point were only 15m from the source these levels would be 3 dB(A) higher)

This logarithmic relationship between noise level and traffic flow rate means that an additional 200 vehicles per hour on a road already carrying 200 vehicles per hour would increase the level by 3 dB(A). However if the 200 additional vehicles were to travel on a road already carrying 1000 v/h the increase in noise level would be less than half a decibel. This strongly supports the need to concentrate traffic on major roads and highways.

It is interesting to compare these theoretical relationships with actual traffic noise measurements. Table I shows the results of measurements made in 1984/5 at 6 typical sites around the Sydney Metropolitan area.[3] In all cases the microphone was located at 9 metres from the centre of the nearside traffic flow, 1.2 metres above the ground. The traffic flow rates, vehicle mix and sound pressure levels are given as mean values sampled over successive ten minute periods, and rounded to the nearest decibel.

At most of these sites, the nearest dwelling facades were approximately twice the distance from the traffic compared to the microphone position, but the attenuation due to distance is almost negated by the 2.5 dB(A) normally allowed for facade reflections at the standard prediction distance of 1 metre from the facade, thus the levels shown can be assumed to be those to which the dwellings were subjected.

TABLE I

MEASURED TRAFFIC NOISE LEVELS

Road Type	Traffic flow, v/h	% Heavies	L_{Aeq}	L_{A10}
6 lane highway	2,900	9.0	75	79
6 lane highway,10%g	2,250	8.8	75	78
6 lane highway	1,850	18.5	76	79
4 lane highway	2,100	20.0	76	79
2 lane local	490	4.8	66	70
2 lane local	280	3.3	66	69

ACCEPTABLE SOUND LEVELS INSIDE BUILDINGS

The acceptable sound levels inside a building depends very much upon the activity taking place. The most critical situations are bedrooms in residential buildings and special rooms such as concert halls, theatres, broadcast studios, etc. Design sound levels for bedrooms are about 25 dB(A), and for living rooms 30 dB(A), $L_{Aeq,(60s)}$ according to AS 2107, (Acoustics - Recommended design sound levels and reverberation times for building interiors[4]) Equivalent L_{A10} levels would be about 3 dB(A) higher. The highest recommended design sound levels inside buildings are 50 dB(A) for assembly lines in industrial buildings and 45 dB(A) for such spaces as computer rooms, corridors, change rooms, engineering workshops, laboratories, etc. In all cases, levels 5 dB(A) higher than those recommended as satisfactory may be taken as the maxima acceptable.

TYPICAL TRAFFIC NOISE ATTENUATION BY BUILDING ENVELOPES

According to AS 3671, if a building is naturally ventilated, and has up to 10% of its exposed area open, the Traffic Noise Attenuation (TNA) will not exceed 10 dB(A). If it is of standard construction and doors and windows are closed, 25 dB(A) attenuation can be expected. If special construction is used, together with closed doors and windows then somewhere between 25 and 35 dB(A) TNA may be achieved. A method is given in the standard for determining which forms of construction may be suitable to achieve such performance. Thus for naturally ventilated buildings, external noise levels should not exceed the recommended interior levels plus 10 dB(A); if ventilation can be provided by some other means 25 dB(A) may be added and for specially constructed buildings, the acceptable external levels may be 25 to 35 dB(A) higher. If more than 35 dB(A) attenuation is necessary, expert acoustic advice should be sought.

THE CREDIBILITY GAP

The range of acceptable exterior sound levels may therefore be determined by comparing acceptable indoor levels and the attenuation expected to be achieved using different construction categories. Some examples are given in Table II below. (In all cases, recommended maxima are 5 dB(A) higher)

It will be observed that residential buildings, even if built using special construction could not provide acceptable interior sound levels if located near a busy road, i.e. under conditions for which the previous examples of traffic noise were calculated or measured, although it would be possible to achieve satisfactory conditions for offices, transport terminals, etc. with specially chosen construction. These conclusions may be compared with the stated policy of the NSW Road Traffic Authorities, which is only to consider attenuation measures when the $L_{10(18hour)}$ value is above 63 dB(A)[5]

TABLE II

RECOMMENDED EXTERNAL DESIGN SOUND LEVELS, $L_{Aeq,60s}$

Building/Activity Type	Construction Category: Standard	No ventilation	Special
Bedrooms	35	50	50-60
Halls over 250 seats	35	50	50-60
Living areas	40	55	55-65
Private offices	45	60	60-70
General offices	50	65	65-75
Transport terminals	55	70	70-80

If a maximum exterior sound level of 35 dB(A), $L_{Aeq,60s}$ is to be achieved, (and assuming + 3 dB(A) for $L_{A10,T}$) allowing for a naturally ventilated bedroom, located say 20m from the nearside flow, the traffic flow rate should not exceed 1v/hour and there should be no heavy vehicles. The allowable vehicle flow rate increases by about 0.7 for living rooms. Although it could be argued that these results are obtained by using the normal prediction methods, which have not been validated for such low flow conditions, the complete incompatibility of naturally ventilated dwelling rooms and road traffic noise is apparent.

The lowest traffic flow rate given for consideration in AS 3671 is an 18-hour flow rate of 1,000 vehicles this gives a simple average of 55.6 vehicles per hour. At an assumed, realistic, distance of 20m from the flow, with the same gradient, speed and vehicle mix as before, the $L_{A10,1h}$ level would be approximately 55 dB(A), say 52 dB(A) $L_{Aeq,1h}$. (The +2.5 dB(A) for facade reflections has not be included). This would require the use of special noise attenuating construction, and precludes natural ventilation for domestic buildings.

SPECIAL CONSTRUCTION FOR TRAFFIC NOISE ATTENUATION

The Australian Standard gives a number of examples of walls, windows, doors, roofs, etc. that may be used to provide better traffic noise attenuation than normal. However, in all cases it is assumed that there are no openings for ventilation, thus either air must be provided from noise-protected parts of the building or air-conditioning or at least mechanical ventilation must be installed. This all adds up to a very costly excercise, and it is also not usually possible to achieve adequate attenuation by retrofit in typical Australian dwellings. This is because the ubiquitous tiled roof, with overhanging eaves and plasterboard ceiling, forms the major transmission path. Lower storey rooms, with sealed double-glazed windows and protected from exposed doorways may be successfully retrofitted, provided that they obtain ventilation from elsewhere; but rooms directly under a roof are not likely to provide more than about 25 dB(A) attenuation, even with double glazed windows.[6]

CONCLUSION

Road traffic and noise-sensitive buildings are incompatible. By removing as much traffic as possible from residential roads, and concentrating through traffic onto major highways and freeways, the problem can be contained. Wherever possible, highway and freeway routes near residential buildings should be avoided, (and certainly, new dwellings should not be permitted within one kilometre of major freeways unless there are topographical or other effective barriers). Where new roads must be built near existing residential areas, site and building planning and special construction techniques may be used to alleviate their impact. It is essential that the cost of such measures be included in the overall cost of new or upgraded road construction. The community should be prepared to compensate those individuals adversely affected by new roads because of the benefits accruing to the society at large.

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INVESTIGATION OF BARRIER CRACKS WITH IMPULSIVE SOUND

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ABSTRACT

Because acoustic impulses have a wide spectral content, typically 500Hz to over 10kHz, they form a rapid and effective probe for determining barrier attenuation. An advantage of impulse experiments is that the finite duration of the pulse permits the relative energy flowing along different paths to be deduced by time isolation techniques. This idea has been used to quantify the effect of cracks in a barrier, where a readily detectable pre-pulse occurs before the main diffracted impulse. The results show that, while the barrier attenuation is decreased at most frequencies, there are frequencies where interference causes 5dB or more of additional attenuation. This technique can be applied to the leakage of sound in to or out of a room.

INTRODUCTION

The acoustic properties of simple semi-infinite half-planes, wedges and trapezoidal barriers can be predicted using classical theory. However, it is more difficult to predict how real structures with complex surfaces composed of a variety of materials will behave. Another real-world complication is the presence of cracks. In this paper, an impulse technique is discussed which permits the rapid assessment of the effects of complicated barriers in the frequency range from 700Hz to over 10kHz. The measurements have been made on full sized barriers, avoiding the problems of scaling results when models are used. The technique has been used to estimate, quantitatively, the effect of cracks. While discussed in terms of a barrier, it is not difficult to see how the technique can be extended to sound leakage into a room from, say, cracks around doors or windows. Acoustic shielding and effects of machine covers could also be investigated using this technique.

THE EXPERIMENTAL SYSTEM

By discharging a shot-shell primer through a tube, a 0.5ms duration pulse was produced and used as the source in the geometry shown in Fig.1. Because of the symmetry of the sound field, microphone A receives a direct pulse which is identical to the one propagating towards the barrier. Thus a comparison of the waveform recorded at microphone B, behind the barrier, with that captured at A shows the change caused by the barrier.

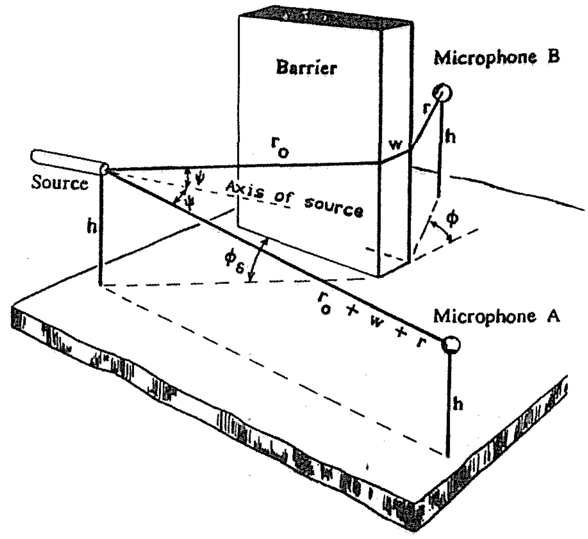

Fig.1: The measurement geometry used in the impulse technique.

Although the effect of the barrier may be apparent on, say, an oscilloscope screen, it is often more revealing to transform the information into the frequency domain. This is achieved by frequency analysing both the captured direct and diffracted waveforms using a fast Fourier transform and then dividing the corresponding frequency components to produce experimental diffraction factors. Often it is convenient to express them as an excess attenuation, EA, such that

$$EA = 20 \log (\text{diffraction factor})$$

$$= 20 \log \left(\frac{\text{diffracted frequency component}}{\text{direct frequency component}} \right)$$

Actually, the diffraction factors are complex numbers so their magnitude is used when taking the logarithm. The presence of relative phase information between frequency components in an impulse is an important attribute of impulse studies. It is essential to know the phase relationships when reconstituting a waveform from its frequency components or when considering theoretical models for impulse diffraction. Further details of the experimental set-up and the analysis techniques are given in an earlier paper by Papadopoulos and Don (1988).

WIDE BARRIERS

Rectangular flat topped barriers constructed from 1cm thick chip-board sheathing a rigid wood frame were investigated experimentally. Two sets of experimental data are presented in Fig.2. They were taken on successive days with the source and measuring microphones being relocated between measurements. Each data set is derived from the average of fifteen individual impulses. The variation between the sets indicates that the reproducibility expected from the impulse technique is generally within ± 1dB over the frequency range of the source.

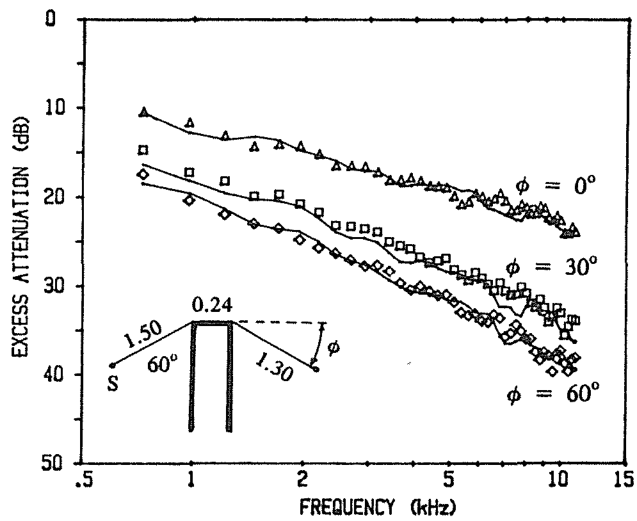

Fig.2: Comparison of two independent data sets obtained using the wide barrier geometry. All distances in metre.

As none of the theoretical treatments available for wide barriers are exact, a comparison was made with predictions based on the suggestion (Maekawa 1968) of replacing a wide barrier with an effective thin barrier, a double application of Maekawa's technique (Kurze 1974) and a more rigorous theory (Pierce 1974). As indicated

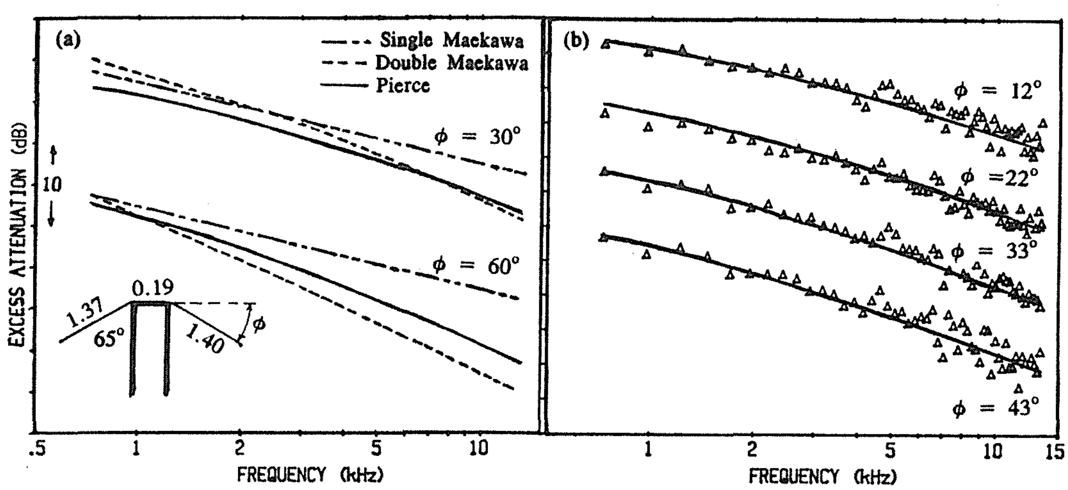

Fig.3: (a) Predictions from three models for estimating the attenuation of a wide barrier and (b) experimental data compared with the Pierce theory.
Note that in these graphs the absolute values of the attenuation have been altered to prevent overlapping of the data at different reception angles.

in Fig.3(a), significantly different predictions are obtained, especially at high frequencies and larger diffraction angles. The best fit with the experimental data is obtained using the Pierce model, as indicated in Fig.3(b), although when the theory is used to predict the diffracted pulse waveform limitations in the phase information inherent in the approximate theory become apparent. Never-the-less, the theory appears capable of predicting the attenuation trend to within 2dB compared to 5dB or more for the commonly used Maekawa chart method.

When the back of the rectangular barrier was removed it left what can be termed an L section barrier. On the reception side, the supporting framework was exposed leaving an irregular surface. Figure 4 indicates the resulting excess attenuation at various reception angles compared to one of the full rectangular barrier data sets from Fig. 2.

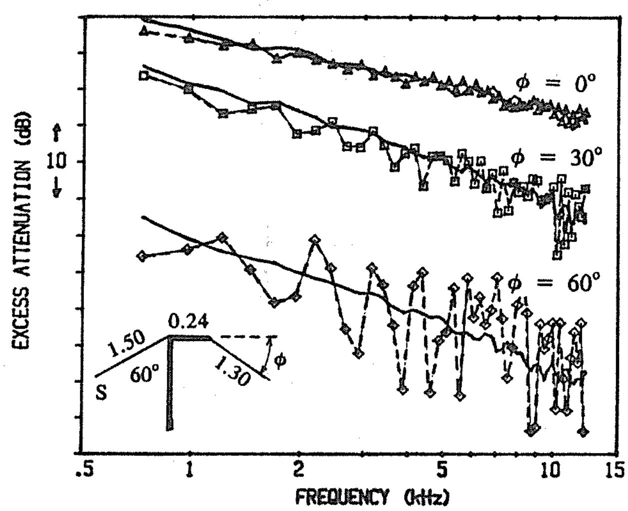

Fig.4: Comparison of complete wide barrier results (continuous line) with results when the back wall of the barrier was removed.

Several aspects of the Fig. 4 data are worthy of comment. Removal of the back of the barrier has often resulted in reduced levels, although the effect is position and frequency dependent. Overall, for sound diffracted through $\phi = 30^0$, measurement of the pulse size indicates that the energy reaching the receiver fell to about 75% of that recorded with the full barrier for the geometry shown in Fig.4. When the diffraction angle is relatively shallow, the variation of excess attenuation with frequency is relatively smooth. However, at reception points closer to the back of the barrier the variations are more marked. This is caused by interference between sound reaching the receiver directly from the edge of the barrier and energy which has been reflected from the supporting framework behind the barrier. Because the latter sound follows a slightly longer path, the additional energy effects the tail of the main diffracted pulse, as is apparent in Fig.5, with reception points close to the barrier being most effected.

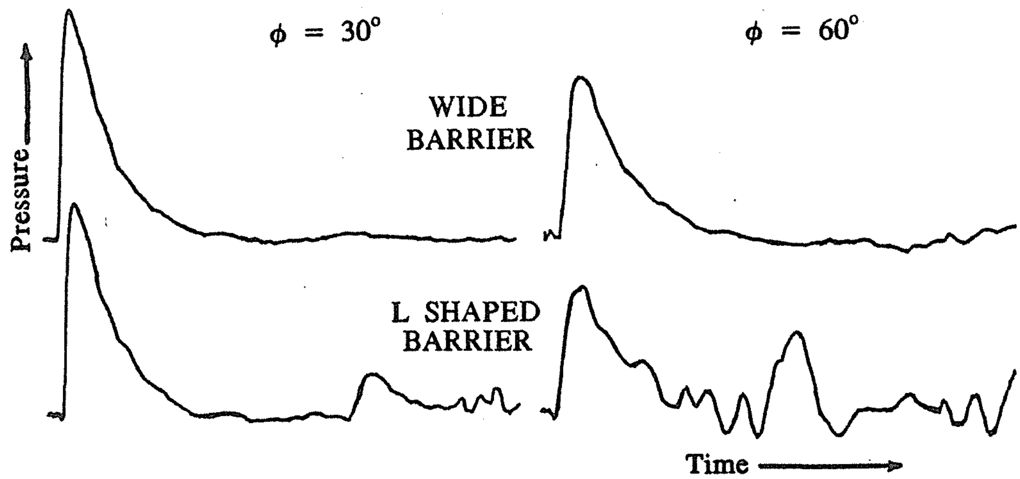

Fig.5: Impulse waveforms recorded at two reception angles for the geometry indicated in Figs. 2 and 4.

EFFECT OF CRACKS

If a crack is introduced into the L shaped barrier, portion of the incident energy will pass through it and produce a pulse preceding the main diffraction, Fig.6(a). With a small crack the main diffraction peak is almost unaltered, however, with a large crack the amplitude of the main peak is reduced. Two effects contribute to this.

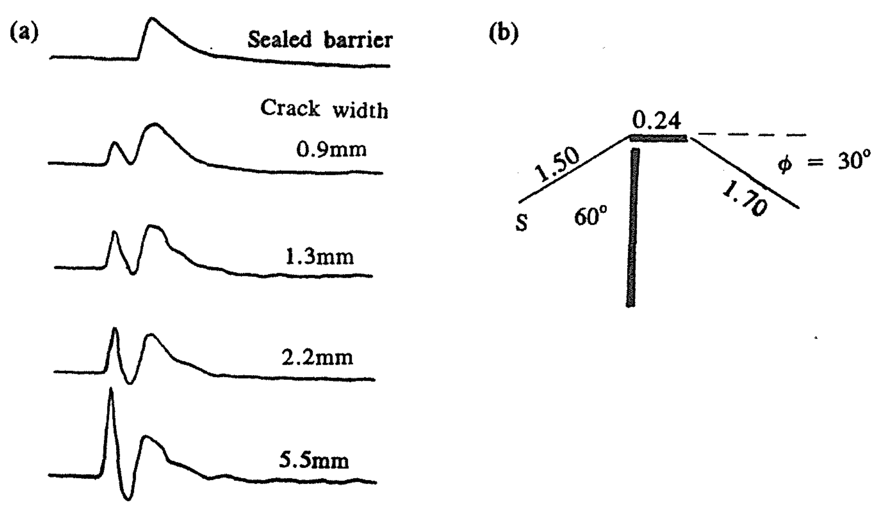

Fig.6: (a) Impulse waveforms recorded for different crack widths and (b) the geometry.

The rarefaction of the leading pulse lies under the main excess pressure peak and also energy may go through the crack rather than around the barrier. As it is difficult to isolate these effects, the data was first analysed by stripping the measured sealed L barrier pulse from the resultant one to obtain the crack pulse with rarefaction. Then its energy content was expressed as a percentage of the diffracted pulse energy for the sealed L barrier. This approach assumes none of the main diffracted energy goes through the slit and so gives an over-estimate of the leaked energy. Alternatively, providing the two peaks are reasonably separated in time, the energies of the two individual peaks can be measured, neglecting the rarefaction of the crack peak. This underestimates the true crack pulse energy but gives an estimate of the fraction of energy diverted through the crack. The following results are for a crack extending along the length of the barrier but of different widths, for the geometry shown in Fig.6(b).

Crackwidth	Technique 1 Through Crack	Technique 1 Around Barrier	Technique 2 Through Crack	Technique 2 Around Barrier
0.9mm	11 ± 2%	100%	9 ± 2%	105 ± 5%
1.3	23 ± 2%	100%	17 ± 2%	95 ± 3%
2.4	35 ± 3%	100%	30 ± 3%	89 ± 2%
5.3	140 ± 4%	100%	130 ± 4%	93 ± 5%

Note that introducing a crack increases the total energy reaching the microphone, so the results are not expected to sum to 100%.

The effect in the frequency domain is shown in Figs. 7(a) and (b) for different slit widths at two reception angles. When $\phi = 30^0$, an interference dip occurs about 10kHz for the narrowest crack - sometimes they are less pronounced than that shown in Fig. 7(a). For wider cracks the dip at 10kHz is reduced while a broad region of increased attenuation is apparent around 2kHz. For this geometry the difference in path length between sound going around the barrier and more directly through the crack is about 2.8cm, corresponding to a delay of 0.082ms, which is consistent with the measured arrival of the pre-pulse relative to the main diffraction. This path difference would suggest an interference minimum around 6kHz, assuming a half wavelength shift was required to produce destructive interference. This disagreement with observation is not unexpected as no account has been taken of the phase shift produced on diffraction around the barrier or through the crack. The situation is even more complicated at $\phi = 60^0$ where additional interference from sound reflected from the back surface of the barrier has an increased significance.

CONCLUSION

Acoustic impulses provide a rapid method of investigating the behaviour of barriers, especially complicated shaped barriers which cannot readily be predicted theoretically. The L shaped barrier considered in this work is an example of a capped barrier where the attenuation is both frequency and position dependent. The introduction of a crack makes the results ever more complicated. In practice, it is unlikely that effects occuring close to the barrier will be important when considering their shielding properties for, say, propagating traffic noise. For that situation it is measurements around the $\phi = 0^\circ$ condition which will be important. However, if information is required about the behaviour of cracks in buildings or machine casings then the higher diffraction angles may be very relevant.

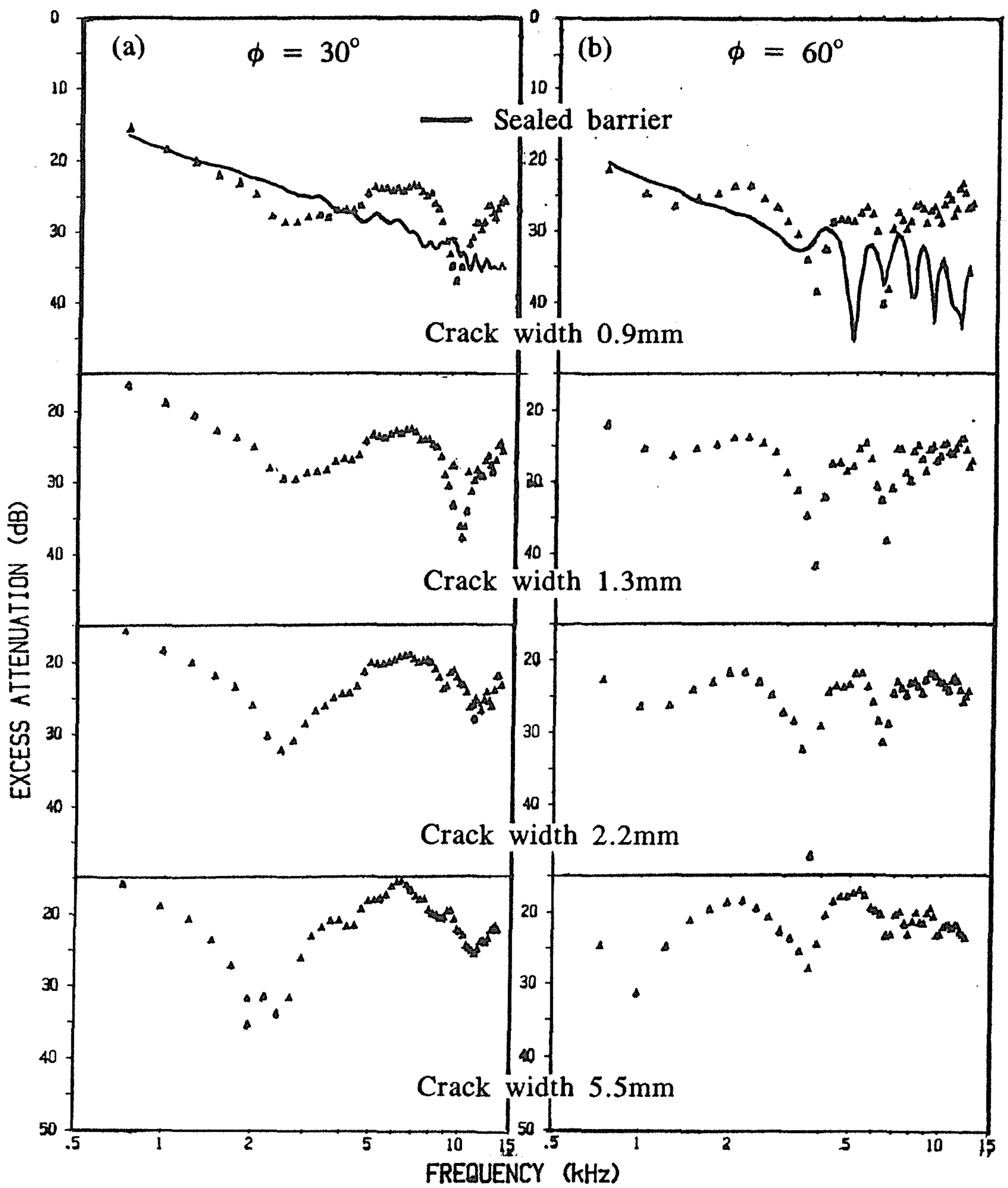

Fig.7: Effect of changing the crack width for two reception angles

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