NOISE INTO THE NINETIES

AUSTRALIAN ACOUSTICAL SOCIETY

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FROM RAY PIESSE

NOISE INTO THE NINETIES

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PROGRAMME

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9.30	Helicopter Noise Assessment – *S. Cooper*
10.00	Eigenfrequency Shifts as a Means of Estimating Duct Dimensions – *W. Qunli & F. Fricke*
	Chairperson – P. Swift
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ROAD TRAFFIC NOISE CONTROL INTO THE NINETIES

by

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and

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Australian Road Research Board

ABSTRACT

The control of traffic noise requires techniques aimed at lowering vehicle noise levels along with complimentary techniques and strategies that deal with the many other elements of the road traffic system. This paper is concerned with techniques for traffic noise control other than those related to reducing vehicle noise emissions. Initially the components of the traffic noise problem in Australia are defined and categorised variously from large volume flows on major road facilities through to general increases in urban area ambient levels. Several noise control techniques are considered and an attempt is made to assess their effectiveness when applied to each particular problem category. The techniques include road design, traffic management, barriers, planning, building design, incentives and promotion campaigns. Since the ultimate effectiveness and acceptance of these techniques depend on how well they are technically and socially implemented, an appropriate strategy for development, coordination and implementation of the techniques is proposed. It is argued that only via such a cohesive strategy can road traffic noise be effectively controlled throughout Australia into the nineties. The paper is based on a recently completed report of the Australian Environment Council's Transportation Noise Subcommittee.

ACKNOWLEDGEMENTS

The AEC Transportation Noise Subcommittee worked long and harmoniously in preparing the material on which this paper is based. Consequently, the authors acknowledge the efforts of their fellow subcommittee members. Particular mention is made of Mr G. Glazier (DMR, NSW), Ms L. Reichelt (DEP, SA), Mr G. Douglas (formerly DEP, NSW) and Mr E. Wrighter (DEP, NSW).

1. BACKGROUND

This paper is concerned with techniques for traffic noise control other than those related to reducing vehicle noise emissions. It identifies a number of these which could be used to reduce levels of traffic noise. The paper defines the components of the traffic noise problem in Australia. It then documents a selection of techniques for noise control and comments on their effectiveness.

Since the ultimate effectiveness and acceptance of these techniques will depend on how well they are technically and socially implemented, a comprehensive strategy for further development, coordination and implementation is proposed. It is argued that such a cohesive strategy for the control of traffic noise is needed for the effective implementation of the traffic noise control techniques identified in the paper. This strategy complements the important continuing control of vehicle noise control emissions and is not intended as an alternative to the emission control program.

The paper is based on the recently completed work program of the Australian Environment Council's Transportation Noise Subcommittee (AEC 1988). This Subcommittee comprised representatives from Environment and Planning Departments, Road Authorities and the Australian Road Research Board. Reporting to the AEC's Environmental Noise Control Committee, it was established to undertake the work which is briefly summarised herein.

2. TRAFFIC NOISE PROBLEMS

The issue of traffic noise has no simple, single solution. It is a multi-faceted problem requiring several solutions that are developed through a structured, cooperative and coordinated effort. For the purpose of developing solutions relevant to Australia, four primary types of traffic noise problems have been defined.

Category 1 Traffic Noise Problems

Large volume traffic flows along major arterials, highways and freeways impacting on residential and other noise sensitive properties.

The residents of houses located next to major roads can experience constantly high noise levels for many hours a day, while those in the second or third rows back are less affected. However this depends on the local topography, the location of the first row of dwellings and the attenuation provided by those dwellings.

Category 2 Traffic Noise Problems

Night-time movement of heavy vehicles along major and minor arterials impacting principally on residential areas.

Noise of any type usually has a greater impact at night than during the day because of the lower background noise at night. Trucks are often a major cause of nighttime traffic noise problems, particularly along major long-distance routes.

Category 3 Traffic Noise Problems

Day and night-time intrusion of light and heavy vehicles in local residential roads and streets.

The noise environment of local residential roads and streets is characterised by lower background noise levels. As a result the occasionally noisy car or truck is readily noticed and may be considered a nuisance. Heavy vehicles moving through rural areas at night create a similar noise nuisance.

Category 4 Traffic Noise Problems

Aggregate increases in ambient noise

Traffic noise constitutes the major component of urban background noise in most areas (OECD 1985). Modern urban and suburban infrastructures are such that increases in traffic noise levels along all roads contribute to what might be termed aggregate increases in ambient noise.

3. TRAFFIC NOISE CONTROL TECHNIQUES

Various techniques are available for the control of traffic noise. These are listed below along with the primary elements in each that provide noise control. For example in Road Design, noise control may be achieved by firstly increasing the distance of the road to an observer and also by suitable selection of the other parameters listed. Space limitations prevent further discussion of these techniques, which are considered in some detail in AEC (1988). It is important to note that there are many interactions between the listed parameters (AEC 1988) and these must be considered when any particular parameter is being selected as a means of traffic noise control. Usually the solution to a traffic noise problem involves application of a suitable combination of several of the listed techniques.

Road Design

Distance
Gradient
Road surface
Road elevation
Road depression
Traffic (flow, speed, composition)

Traffic Management

Channelisation
Local area traffic management
Traffic flow control devices
Capacity improvements
Traffic signal coordination
Restricted use routes
Designated truck routes
After dark restrictions
Flashing yellow traffic lights at night

Planning

Noise tolerant land use
Residential noise impact notice scheme
Development control prescriptions
Residential cluster developments
Separation of noise sensitive land uses
Noise zoning/classification

Building Design

Building use
Building siting
Building construction

Incentives

Improved public transport
Reducing motor vehicle usage
Assistance in purchasing quieter vehicles
Subsidised purchase of noise affected land
Manufacture of low noise vehicles
Restricted use of noisy vehicles
Motor vehicle noise labelling
Community education

Community Awareness

Educational programs for professionals
Public awareness campaigns

Barriers

Including in rew road design
Retrofitting

In Australia to date, all of these techniques except Incentives and Community Awareness have been applied in one form or another. However in most cases the noise control solutions have primarily involved road design, traffic management and barrier techniques adopted by state road authorities. Of these the most visible has been the introduction of barriers alongside existing and new freeways. As yet very few solutions aimed at reducing the noise from existing roads, other than freeways, have been applied in Australia and other developed countries.

4. AN EVALUATION OF THE NOISE CONTROL TECHNIQUES

Each of the seven noise control techniques varies both in effectiveness and applicability to any given traffic noise situation. The practice of traffic noise control is to a large degree concerned with selecting which technique or combination of techniques is best suited to a particular application. Obviously this can only be achieved once the given traffic noise problem has been identified. From there an optimum solution can be put together and implemented as costs and other constraints allow. Consequently an exhaustive and comprehensive evaluation of each technique as applied to the four Traffic Noise Categories was undertaken (AEC 1988). This involved a detailed analysis, the primary approach of which was to cross tabulate the noise control techniques with the problem categories. Where possible, the following eight attributes have been considered in some detail for each element in the cross-tabulation matrix. Furthermore, this consideration has been extended to cover both existing roads or urban infrastructure and new roads or undeveloped (green-field) situations.

Noise Control Technique Attributes

1. Noise control effectiveness
2. Function
3. Indirect benefits and disbenefits
4. Variability of effectiveness
5. Implementation feasibility
6. Dependency on other strategies
7. Cost
8. Time-frame

A summary of the analysis results is documented in Fig. 1. The applicability of each noise control technique is indicated, in a general fashion, for the four problem categories. An overview of the matrix indicates that the effectiveness of any particular technique depends markedly on the nature of the problem to which that technique is applied. (A similar conclusion is also valid for vehicle emission controls.) The matrix analysis demonstrates that only a few of the noise control techniques are highly effective and that none is highly effective over the complete range of problem categories. These highly effective techniques included the road design treatments of varying both source-receiver distance and road surface texture.

The traffic management techniques of traffic channelisation, local area traffic management and restricted use routes were similarly effective. Barriers, on the other hand, were found to be highly effective only along major access controlled arterials, highways and freeways. Noise tolerant land use substitution in existing areas and both residential cluster developments and separation of noise sensitive land uses along new arterials, highways and freeways were observed to be highly effective planning measures. Land classification by noise level was considered to be effective and has added benefits of being easily adapted for all environmental noise problems. Educational programs directed specifically at professional groups were found to be effective in that they encouraged the application of suitable control strategies.

NOISE CONTROL STRATEGY	NEW AREAS Category 1	NEW AREAS Category 2	NEW AREAS Category 3	NEW AREAS Category 4	EXISTING AREAS Category 1	EXISTING AREAS Category 2	EXISTING AREAS Category 3	EXISTING AREAS Category 4
Road Design								
Distance	H	H		S	H	H		S
Gradient	M	M	S	S	M	M		S
Road surface	H	H	S	S	H	H	M	M
Road elevation	S	S						
Road depression	M	M	S					
Traffic (flow, speed, composition)	M	H	M	M	S	S	S	S
Traffic Management								
Channelisation	H	H	H	S	H	H	H	S
Local area traffic management		H	H	M		H	H	M
Traffic flow control devices	S	S	S	S	S	S	S	S
Capacity improvements	M	M	S	S	M	M	S	S
Traffic signal coordination	S	S	S	S	S	S	S	S
Restricted use routes	H	H	H	S	H	H	H	S
Designated truck routes	M	M	M	S	M	M	M	S
After dark restrictions		H	M	S	M	M	M	S
Flashing yellow traffic lights at night	S	S		S	S	S		S
Barriers								
Included in new road design	H	M			M			
Retrofitting					H	S	M	S
Planning								
Noise tolerant lane use substitution					H	H		
Residential noise impact notice scheme	S	S	S		S	S	S	
Development control prescriptions	M	M			S	S		
Residential cluster developments	H	H	H		S	S	S	
Separation of noise sensitive land uses	H	H	M					
Building Design								
Building use	S	S	S		S	S	S	
Building siting	S	S	S		S	S	S	
Building construction					M	M		
Incentives								
Improved public transport	S		S	S	S		S	S
Reducing motor vehicle usage	S	S	S	S	S	S	S	S
Assistance in purchasing quieter vehicles	S	S	S		S	S	S	
Subsidised purchase of noise affected land	S	S			S	S		
Manufacture of low noise vehicles	S	S			S	S		
Restricted use of noisy vehicles		M	S			M	S	
Motor vehicle noise labelling	S	S	S	S	S	S	S	S
Community Education	H	H	M	S	S	S	M	S
Encourage use of quieter vehicles by pricing	S	S	S	S	S	S	S	S
Financial assistance to manufacturers	M	M	M	M	M	M	M	M
Community Awareness								
Public awareness campaigns		S	S	S		S	S	S
Professional Education	H	H	H		S	M	M	

HIGHLY EFFECTIVE (H)

MODERATELY EFFECTIVE (M)

SLIGHTLY EFFECTIVE (S)

NOT APPLICABLE (blank)

Figure 1

EFFECTIVENESS OF NOISE CONTROL TECHNIQUES

It was apparent that in only 14 per cent of the Fig. 1 cases were the noise control techniques rated as highly effective. Furthermore, it was recognised that some techniques were effective immediately after implementation while others, such as planning, were longer term solutions. The effect of any particular technique occurs some time after the technique has been implemented. This time varies and can be significantly influenced by whether the technique is applied to a new or an existing area.

It was also evident in Fig. 1 that a majority of noise control techniques could only be regarded as being of marginal effectiveness when considering all problem categories. For instance, the noise control techniques were considered to be either slightly effective or not applicable in approximately 70 per cent of all applications listed in the matrix. However this observation does not in any way diminish the status of the techniques rated as highly effective. The selection and implementation of these highly effective techniques should be regarded as a high priority for the control of traffic noise. The influence of local conditions needs to be taken into account in the evaluation of techniques rated as either moderately or slightly effective in the matrix. Careful consideration is required prior to the adoption of these techniques. It is recognised that in some cases such techniques are proposed for reasons other than traffic noise control.

There are substantial differences in the costs associated with each of the noise control techniques. Detailed cost estimates were frequently difficult to obtain because of the indirect and hidden nature of some techniques, such as those involving policy changes. For techniques, such as retrofitting barriers, involving direct capital expenditure, some cost estimates were readily available. The costs of the various techniques are borne by different groups. For example, noise barriers would usually be funded by the road construction authority or in some cases the local council. For new housing developments, it may be possible for the cost of the barriers to be incorporated into the total construction cost. On the other hand, the adoption of a planning policy would involve the indirect costs of development and enforcement. Other consequential costs such as the loss or sterilisation of land for buffer zones would be indirectly borne by the community over a long period of time. In this paper only those costs associated with the construction of barriers are considered to place in perspective general costs pertaining to Fig. 1.

Recently, NAASRA (1987) estimated the costs of barriers comprising earth mounds topped with 2 - 3 m fences designed to reduce L10 (18 hour) traffic noise levels at adjacent house facades to 68 dB(A). These ran out to $1.4 M for an 8 km road length. Provision of an additional 400 m of 5 m high cantilevered concrete barrier to protect the upper levels of multi-storied buildings added a further $800,000. Furthermore, reducing the design levels by 3 to 65 dB(A) inflated the original costs by approximately three times to $4 M. In terms of overall construction costs, NAASRA (1987) estimated these barriers would account for around 1.5 per cent for 68 dB(A) and 4.2 per cent for 65 dB(A).

The cost of retrofitting barriers has been estimated to be between $185,000 and $222,000 per kilometre (NAASRA 1987). This corresponds with another independent estimate of $222,000 (Modra 1985). Other estimates of 2 m high timber barriers are $170,000 and $180,000 per lineal metre; a 3 m timber barrier is $190,000 - $200,000. Absorption barriers are more expensive: a barrier on top of a New Jersey kerb (0.8 m high) is $600,000 per lineal metre.

While the associated costs can in some cases be very high, the evaluation has, nevertheless, revealed some highly effective traffic noise control techniques. These should form part of any comprehensive traffic noise control strategy. However, unlike vehicle based controls, the range of non-vehicle based controls is extensive. Some of the noise controls may produce impacts in other areas, such as visual or safety effects, which in some locations may be undesirable. The preferred solution will depend on the particular problem and the specific site being considered. Non-vehicle based controls are also therefore not implicitly amenable to strategies at a National level. Rather these controls need to be applied at state and local levels and require the commitment, cooperation and coordination of road, environment, planning and housing authorities and local government.

5. TRAFFIC NOISE ABATEMENT PLANS

Since effective control of traffic noise using non-vehicular techniques resolves to state and local levels, it is appropriate to have arrangements in place whereby relevant authorities, organisations and individuals at both levels may be readily involved in the selection and implementation of suitable noise control techniques. To this end, Traffic Noise Abatement Plans are proposed for both levels. Development of such plans provides the opportunity to resolve conflicting priorities and incorporate noise control methods that are locally acceptable. Only in this way will traffic noise control be effective into the nineties.

State Level

State Level Plan would primarily address each of the four previously defined problem categories. The area covered by the plan might include the whole of the state or a defined portion such as some roads and adjacent land or the whole of an urban area. Some parts of the plan may involve the introduction of legislation while others would require the development of guidelines, for example on noise barrier construction techniques.

Each State Plan would be assembled by a group of representatives from key organisations involved. These would include road authorities, state planning organisations, government housing groups and the police. Local authorities would be involved so that full integration with Local Level Plans could be achieved. Once agreement was reached about the Plan it could be submitted to public review.

A typical State Level Plan might include the following.

A. Noise classification/zoning techniques for areas adjacent to major road facilities.

B. Freeway barrier treatment guidelines.

C. Planning and development controls (based on Item A) regarding residential development near major road facilities.

D. Road surface texture specifications.

E. Traffic management guidelines.

F. Traffic route restrictions on a time-of-day basis.

G. Development controls on freight depots.

H. Rail transport incentives where appropriate in specific situations.

I. Guidelines for trouble-spot solutions.

J. Promotion of the traffic channelisation concept.

Local Level

A Local Level Plan would address specific local noise problems such as through traffic noise in local streets. It is similar in concept, preparation and operation to the State Level Plan. Controlled by Local Government Authorities, it would be put together in a cooperative fashion by various Local and State Government representatives as appropriate. Local Government Authorities would then prepare the plans, as with the State Plans, and then also submit them to public review. Local Plans would, necessarily, take into account any existing or proposed State Plan.

As an example, the components of a Local Plan might include the following.

A. Local Area Traffic Management Schemes to limit the movement of through vehicles.

B. Promotion of the channelisation of traffic along lower heirarchy roads.

C. Planning, building and development controls to limit traffic noise.

6. CONCLUSIONS

Current traffic noise problems were defined into four categories that ranged from large volume flows on major road facilities through to general increases in urban area ambient levels. There are several techniques available at present for traffic noise control. These include road design, traffic management, barriers, planning, building design, incentives and promotion campaigns. Assessed against the four traffic noise problem categories, the effectiveness of all control techniques was found to be variable. Few control techniques were highly effective for all noise problem categories and applications.

Since effective control of traffic noise resolves to state and local levels, traffic noise control plans at each level were proposed as an appropriate strategy for the nineties. These plans would provide a means whereby relevant authorities, organisations and individuals at both levels may be readily involved in the selection and implementation of suitable noise control techniques.

In the development of State and Local Plans, high priority should be given to those techniques identified as being highly effective in controlling traffic noise. These include the following.

Road design treatments

* Varying the source-receiver distance
* Varying road surface texture

Traffic management techniques

* Traffic channelisation
* Local area traffic management
* Restricted use routes

Noise barriers

* Along freeways
* Along major arterials

Planning

* Land zoning/classification according to noise tolerance
* Noise tolerant land use substitution along existing arterials, highways and freeways
* Residential cluster developments in new areas
* Separation of noise sensitive land uses along new arterials, highways and freeways

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HELICOPTER NOISE ASSESSMENT - S. COOPER

Helicopter Noise Criteria

In December 1982, the State Pollution Control Commission presented a draft set of Helicopter Assessment Guidelines at the AAS Symposium "Aircraft Noise to the Year 2000"

A paper presented by S. McLachlan and G. Mellor to the Symposium detailed the basis upon which the SPCC had determined a residential and commercial noise limit. The first part of the criteria used a maximum level of 82dB(A) fast response for residential receivers (85dB(A) fast response for commercial receivers). The second part of the criteria used an L_{eq},A level of 55dB(A) over the period of 7am - 10pm for residential receivers or 65dB(A) over 24 hours for commercial receivers.

The Commission required all helicopter acoustical assessments to be conducted in accordance with their Guidelines for a trial period to enable the SPCC to monitor response to the Guidelines.

Initial reaction from the Helicopter Industry was (and still is) the criteria if restrictive when compared to similar criteria used overseas (in particular the USA). Extreme difficulty was encounted in achieving the dB(A) maximum criteria where a helicopter landing site was situated in a residential area or in a location accessible to the public.

In a number of appeals for Helipad/Heliport Developments before the NSW Land & Environment Court the Commission's guidelines have been examined at length by the Court and confirmed both the L_{eq}, A and dB(A) maximum criteria to be acceptable in a residential area.

The NSW Noise Control Act 1975 was amended to "schedule" helicopter landing sites under the approval and licence requirements, and as such all helicopter landing sites in NSW (except on Commonwealth property) are controlled by the Commission.

The Commission's guidelines form the basis of the Draft Australian Standard " Acoustics - Assessment of Noise from Helicopter Landing Sites" and include some minor revisions/definitions in order to eliminate possible inconsistencies in the assessment procedures.

The revisions as endorsed by the Commission include elimination of the time period summation in the L_{eq},A Calculation which can lead to an overestimate of the number of movements, use of a 7a - 7pm time period and an increase in the L_{eq},A residential criteria to 60dB(A) or $L_{Aeq,T}$ (Amb) + 10 dB(A) whichever is the lower value.

In addition to the above technical revisions, the Commission developed a noise envelope (footprint) so as to provide a first approximation to the suitability of a site with respect to the Commission's criteria. We have assessed the noise envelope from our database and have found for normal light helicopter types (Jetranger, single/twin Squirrels) the footprint to be conservative in that normal helicopter operations would allow some 12 movements whereas the noise envelope indicates 5 movements per day.

However there are other revisions in the Draft Standard that vary from the Commission's guidelines and require comment.

The Draft Standard stipulates dB(A) Slow response whereas the Commission used dB(A) Fast response. In terms of the L_{AE}/SENEL calculations the difference in Slow versus Fast response are considered to be insignificant for normal helicopter operations Poor or incorrect flying techniques can result in the Fast response SENEL and some 0.5 - 2 dB(A) higher than a Slow response.

With respect to the maximum level measurement, the difference between slow and fast can be quite significant (up to 4dB) and even greater where blade slap occurs or excessive power is used (discussed later).

Our experience has shown that the more critical factor for premises in close proximity to helipads in the L_{Amax} level, and the Commission consider it is essential for a measurement that truly reflects the proposed operation of the helipad and a person's normal response to annoyance be used.

The Commission's continuing involvement in community complaints to helicopter and other noise sources had clearly demonstrated that Fast response is the measurement parameter that should be used for assessing such noise.

The other area of comment on the Draft Standard is the averaging parameter of the individual discrete events to determine the final value of $L_{Amax\ (Hel)}$ for each mode. As a result of the use of decibels for measurements it has been demonstrated (and used for numerous applications) that an energy average is more appropriate and should be used instead of an arithmetic average.

The closing date for comment on the Draft Standard is 30th November, 1988 and your opinion would be most welcome.

Helicopter Noise Abatement

The operational characteristics of a helicopter employ the aerodynamic principle of moving an aerofoil through the air to achieve lift. A fixed wing aircraft utilises forward motion to attain airflow over the aerofoil (wings) for lift while the helicopter rotates the aerofoil (rotor blades) to achieve the same condition.

This permits a helicopter to land and take off from a small area whereas fixed wing aircraft require a runway/long landing area.

A fixed wing aircraft on final approach to a landing normally establishes a constant rate of descent of descent and constant speed until over the runway when the aircraft flares, lands on the runway and then proceeds to reduce speed and taxi off the runway.

A helicopter on final approach to a landing site changes the rate of descent and speed such that the helicopter has zero forward speed when over the landing site. Figure 1 indicates the change in rate of descent and speed for a helicopter maintaining a 6° flight path. During the various permutation of rates of descents and speed, the acoustical signature of the noise emission will vary.

One factor influencing "helicopter noise" is a modulation of sound generated by aerodynamic forces on the main rotor. The modulation that normally attracts the most attention is often referred to as "blade slap". Blade slap can occur in all modes of helicopter operations but is most prominent in landings during partial power descents when a blade intersects its own vortex system or that of another blade. When this happens , the blade experiences locally high velocities and rapid angle-of-attack changes.

Figure 2 shows the typical blade slap region for a Jetranger 206A helicopter. The slap boundary may be larger if the main rotor encounters intermittent wind gusts or the helicopter transition rapidly from one flight regime to another. Noise levels encounted during blade slap in transition phases can be as high as 12 dB(A) using Fast response.

Another potential source of high intermittent increases in noise levels that are more evident when using the Fast response occurs when the helicopter flares for a landing in ground effect with too high a forward speed, requiring additional power to halt forward momentum. A ground effect occurs whenever a helicopter flies or hovers near the ground or other surface (less than a height of half the main rotor diameter) a cushion of denser air is built up between the ground and the helicopter by the air displaced downward by the main rotor.

For receiver locations less than 160 metres from a landing site (normally perpendicular to the flight path) the onset of the ground effect generally produces the maximum level and dependent on the gross weight of the helicopter the difference between Slow and Fast response vary from 0 to 3 dB(A).

Noise abatement landing techniques have been developed to avoid the areas of blade slap and excessive use of power. Depending on the type of helicopter the Noise Abatement flying technique falls into one of the following categories.

- constant airspeed/constant glide slope
- decelerating airspeed/constant glide slope
- decelerating airspeed/variable glide slope

Figures 3 & 4 demonstrate typical and noise abatement profiles we have utilised for the Jetranger 206A helicopter. The maximum levels for positions approximately 160 metres perpendicular to the flight path (on final descent) can be reduced some 3 - 5 dB(A) on Fast response by utilising the noise abatement profile. The SENEL result can be between 0 and 2dB(A) lower using the noise abatement profile.

We note that one side of the helicopter tends to be noisier, corresponding to the respective main rotor's advancing blade. Main rotors of the R22, 500D/E, 206, 109A, BK117, 22A and the S76 turn counterclockwise, as viewed from the top. This results in the right side tending to have slightly higher levels. The main rotor of the 365N turns in a clockwise direction.

Other modes of operation can significantly alter the level of noise emission and include sudden changes in airspeed, angle of bank and turn rate. It has been found that generally there is less noise on inside turns, but where any changes in altitude or power occur during a turn the resultant level of emission increases significantly.

Use of helicopters with more than 2 main rotor blades tends to reduce the overall noise emission , the impulsive characteristic and the incidence of blade slap on landings. Similarly the use of multi-bladed tail rotors/fans has achieved reductions of up to 9 dB(A) for flyover modes of helicopter operations. The Hughes 500 E helicopter with the low noise option utilises a different tail rotor and is purported to achieve a reduction in certification test measurements of between 6 & 9 dB(A) when compared with the standard tail rotor.

For the larger helicopters seating 12 - 15 people, the two bladed main rotor system (Bell 212) is significantly noisier than a four bladed rotor system for the same helicopter (Bell 412). In addition the larger two bladed main rotor helicopters produce a greater low frequency modulation that is clearly audible for a long distance and the smooth control of power and movements is even more important for these types of helicopters.

Use of ICAO Certification data for helicopters has been examined by our office and from field measurements we have found the Certification data to be higher than normal operations for the majority of the light helicopter types, due to the ICAO test procedure requiring use of a flight profile passing through the blade slap region.

The bottom line to reducing helicopter noise emission levels is to get operators to fly responsibly, smoothly and with precision. Cowboy type pilots generate significant blade slap and higher noise for most flight modes. Flying a helicopter in the "limousine" mode, has consistently resulted in lower noise levels, in some cases as much as 10 - 16 dB(A) lower than the show-off type pilot.

FIGURE 1: Glide Slopes

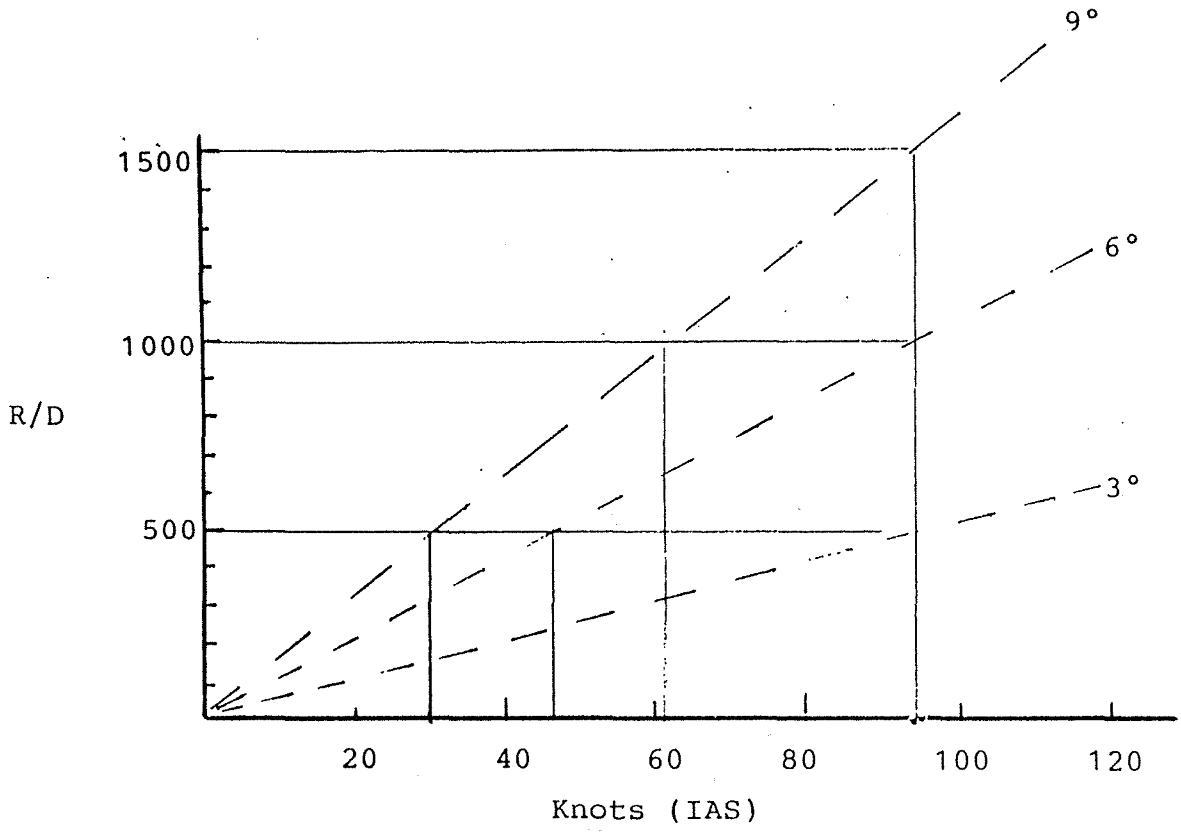


FIGURE 2: Noisy Flight Operations - 206 Jetrangers

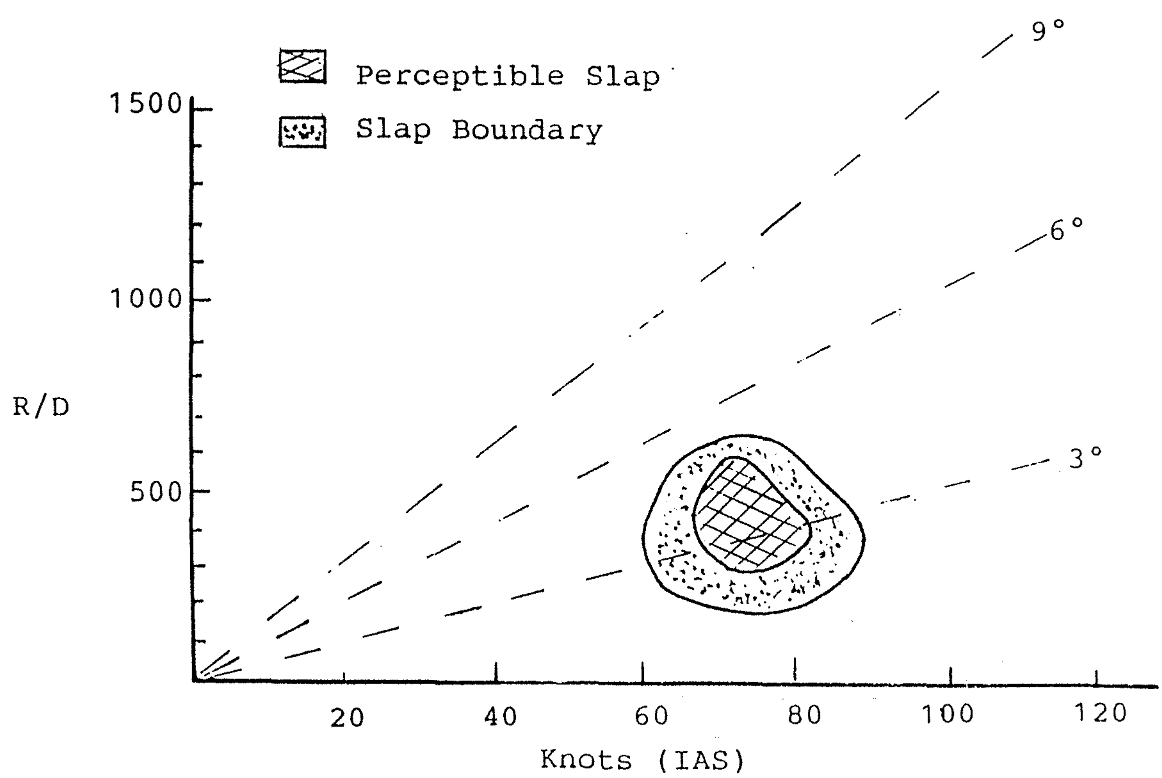


FIGURE 3: Normal Approach 206L - 1

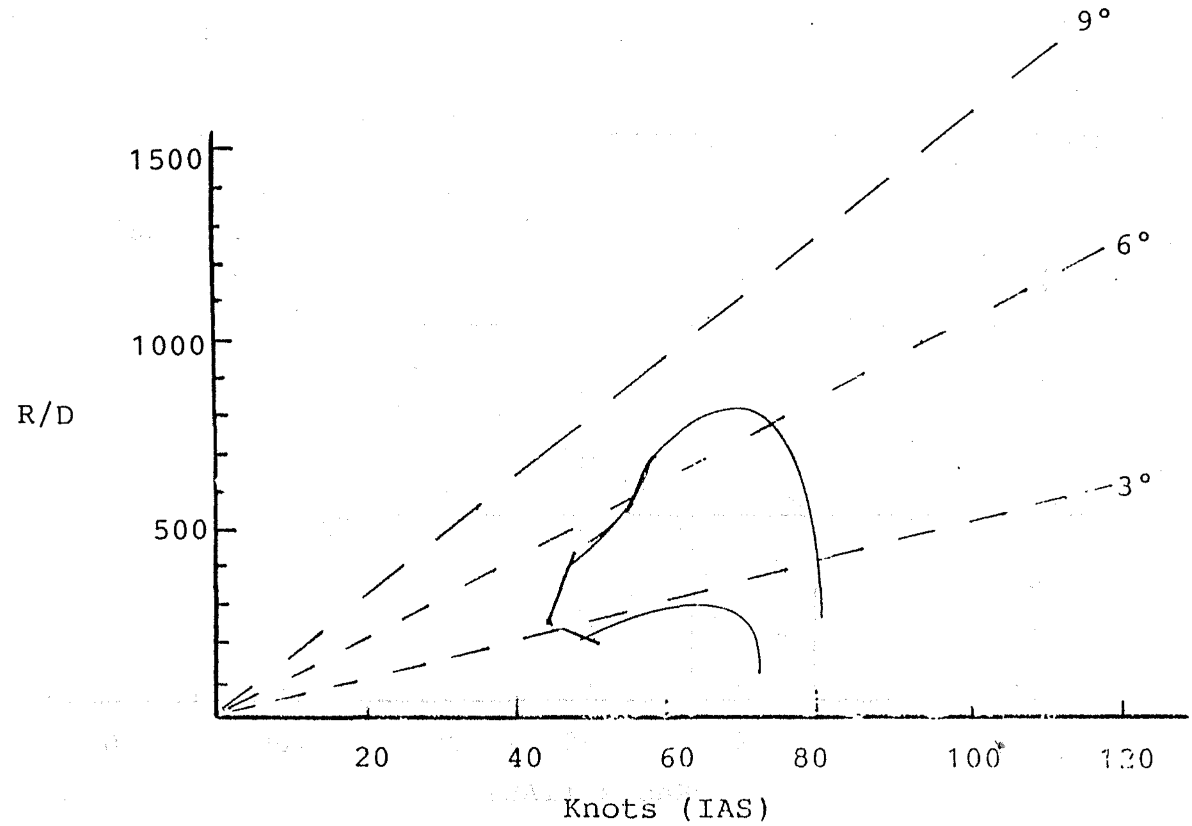
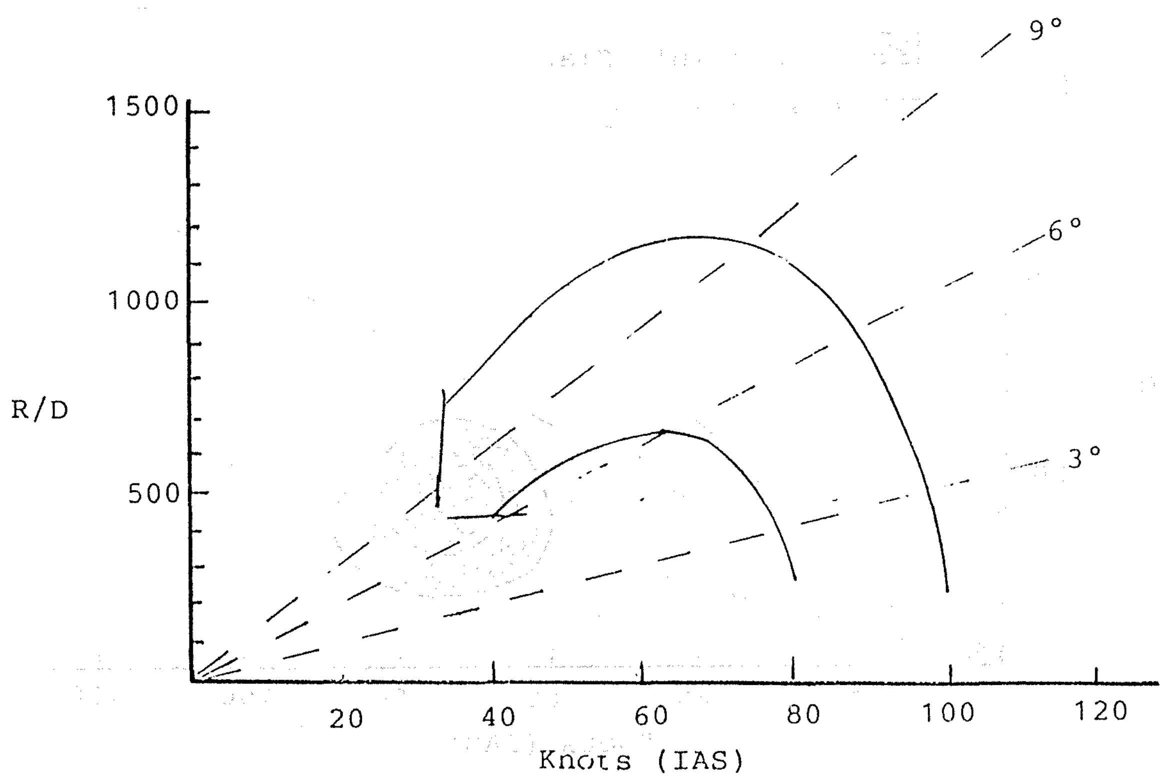


FIGURE 4: Noise Abatement Approach 206L - 1



EIGENFREQUENCY SHIFTS AS A MEANS OF ESTIMATION OF BLOCKAGE DIMENSIONS IN A DUCT

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N. S. W. 2006

ABSTRACT

A new method for estimation of the blockage dimensions in a duct is investigated. The method is based on the eigenfrequency shifts due to the blockage. The study indicates that the eigenfrequency shift pattern is uniquely related to the size and location of the blockage. By analysing the eigenfrequency shift pattern the blockage dimension is estimated. The accurcy of the estimation of the blockage dimensions implies that the eigenfrequency shift method is applicable to some practical situations.

1. INTRODUCTION

For nondestructive measurements of the distribution of blckage in ducts, an approach that has had used is based on the ultrasonic pulse echo method [Tohjima 1988]. However, the ultrasonic pulse echo method requires the expensive ultrasonic sound source, transducer and sophisticated signal analysis systems. Moreover, the ultrasonic method is not suitable to detect the large blockage in the ducts because of its wavelength.

In this paper an effort has been made to develop a new method to estimate the blockage dimensions in a duct. Although the eigenfrequency theory of enclosure has been known for many years, and the techniques for solving the eigenfrequencies of complex enclosure has been developed [Petyt 1976], the application of eigenfrequency theory to estimate the size and shape of the objects in an enclosure has attracted little attention [Schroeder 1967; Gladwell 1987].

In the present investigation, the eigenfrequency shifts were used for the formulation of the problem instead of the eigenfrequencies [Wu and Fricke 1987]. The experimental results show that the eigenfrequency shifts can be used to estimate the blockage dimensions in a duct.

2. EIGENFREQUENCY SHIFT DEPENDENCE ON BLOCKAGE

Consider the one dimensional model of a duct with a blockage in one end, as shown in Figure 1. The duct and blockage cross section areas are defined by A and A_b respectively. The blockage

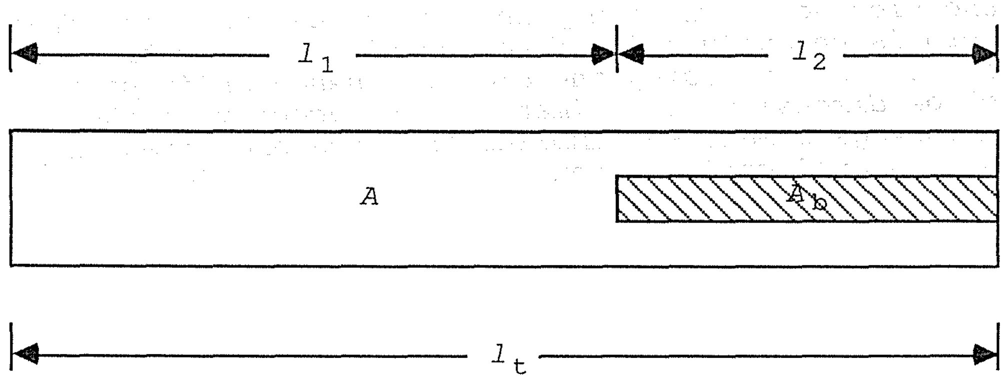

Fig.1 The duct with a blockage in the end

partitions the duct into two segments with length l_1 and l_2, where l_2 is the blockage length. The following nondimensional quantities are defined:

$$\lambda_b = l_2/l_t, \qquad \beta = A_b/A$$

where $l_t=l_1+l_2$

Assuming the duct ends are rigid, the eigenvalue relationship in the duct, derived from the wave equation and the boundary condition is:

$$\sin(k_f(l_1+l_2))-\beta\cos(k_f l_1)\sin(k_f l_2) = 0 \quad (1)$$

where $k_f = 2\pi f_n/c$ is the eigenvalue of the duct, f_n is the eigenfrequency, $\beta = A_b/A$, is the cross section area ratio of the blockage and duct. The second term of Eq.(1) represents the blockage effect on the eigenvalues.

Due to the presence of the blockage, the eigenfrequencies of the duct will shift from the orignal values. The eigenfrequency shift, $\Delta f(n)$, is defined as the eigenfrequency difference with and without the blockage in the duct:

$$\Delta f(n) = f_n - f_n^0 \quad (2)$$

where f_n and f_n^0 are the nth eigenfrequencies of the duct with and without blockage respectively.

The eigenfrequency shift is a discrete function of the mode order n. The pattern of eigenfrequency shift vs mode order of the duct is defined as the eigenfrequency shift pattern. The maximum shift of the pattern is called the amplitude of eigenfrequency shift Δf_{amp}.

2.1 SMALL BLOCKAGE INDUCED EIGENFREQUENCY SHIFTS

The eigenvalue relationship for the blockage in the end of the duct is given Eq.(1). Assuming a blockage cross-section area ratio, $\beta << 1$, corresponding to a small blockage in the duct, the eigenfrequecy solution of Eq.(1) can be expressed as:

$$f_n = f_n^0 + a_1\beta + a_2\beta^2 + \cdots\cdots \quad (3)$$

By neglecting second order terms such as β^2, β^3, $\cdots$, the first order approximate solution of Eq.(1), expressed as eigenfrequency shifts, is:

$$\begin{aligned}\Delta f(n) &= (\beta c/4\pi l_t)\sin(2n\pi\lambda_b) \\ &= -(\beta c/4\pi l_t)\sin(2n\pi(1-\lambda_b)) \quad (4)\end{aligned}$$

Eq.(4) shows that the eigenfrequency shift of the duct is a discrete sine function. This function has the following properties:

(i) The amplitude of the eigenfrequency shift, Δf_{amp}, is proportional to β/l_t.

(ii) The eigenfrequency shift $\Delta f(n)$ is a periodic function of mode order n. The period is:

$$n_T = \begin{cases} l_t/l_2 \ (=1/\lambda_b) & \lambda_b \le 0.5 \\ l_t/l_1 \ (=1/(1-\lambda_b)) & \lambda_b > 0.5 \end{cases} \quad (5)$$

(iii) The eigenfrequency shift, $\Delta f(n)$, is zero when

$$n = \begin{cases} ml_t/2l_2 \ (= m/2\lambda_b) & \lambda_b \le 0.5 \\ ml_t/2l_1 \ (= m/2(1-\lambda_b)) & \lambda_b > 0.5 \end{cases} \quad (6)$$

where m=1,2,3,...

(iv) When $\lambda_b < 0.5$, the first n_1 ($<1/2\lambda_b$) eigenfrequency shifts are positive, i.e. $\Delta f(n) \ge 0$ for $n \le n_1$. In other cases $\lambda_b > 0.5$, the first n_2 ($<1/2(1-\lambda_b)$) eigenfrequency shifts are negative, i.e. $\Delta f(n) \le 0$ for $n \le n_2$.

The study in next section will prove that the properties (ii), (iii) and (iv) are independent of blockage cross section ratio.

In the many cases l_t/l_1 or l_t/l_2 are not integers. Thus n_T and n should be treated as continuous variables in order to estimate the period and locations of zero points.

2.2 LARGE BLOCKAGE INDUCED EIGENFREQUENCY SHIFTS

Eq.(1) shows a nonlinear relation of eigenvalue and physical dimensions of blockage. With increasing blockage cross-section area ratio, β, the eigenfrequency shift pattern of the duct will no longer fit Eq.(4) for small blockages.

A series of numerical studies of Eq.(1) was carried out to demonstrate how the eigenfrequency shift pattern varied with the physical dimensions of the blockage.

The first study was to show that the amplitude of eigenfrequency shift varied with the cross-section ratio, β, for various duct blockage lengths. The results are shown in Figure 2. The results indicate that:

(a) The amplitude of the eigenfrequency shift, Δf_{amp}, is not linearly related to β/l_t.

(b) When $\beta \le 0.55$ $\Delta f_{amp} l_t$ is independent of blockage length. The value of $\Delta f_{amp} l_t$ can be approximately expressed as unique function of β. This function, obtained by regression, is:

$$\Delta f_{amp}\, l_t = \begin{cases} 41.83\beta^{1.18} & \beta \le .55 \\ 59.30\beta^{1.80} & \beta > .55 \end{cases} \quad (7)$$

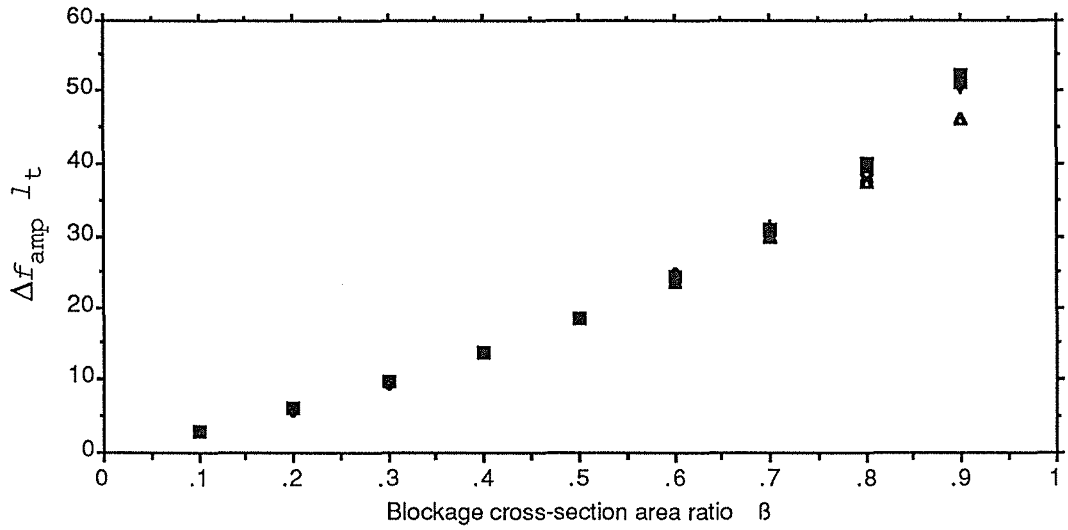

Fig.2 Weighted amplitude of eigenfrequency shifts as function of β

The second study was to see whether the eigenfrequency shift pattern varied with the cross-section ratio, β, for various duct and blockage lengths.

The results indicate that the properties (ii), (iii) and (iv), described in section 2.1, are independent of β.

An explanation of these properties is readily given by reference to standing wave theory. For example, when there is a very short blockage in the duct, the effect of the blockage is equivalent to a shortening of the duct length for the lower mode orders ($\lambda/4>l_2$). This will result in an increase in eigenfrequencies of the blocked duct, and therefore the eigenfrequency shifts will be greater than zero. When the frequency of the standing wave increases and the blockage length, l_2, is between one quarter and one half a wavelength, the effect of the blockage is equivalent to the increase of the duct length. Thus the eigenfrequency shift will be negative. The relationship between the quarter wavelength and the blockage length control the eigenfrequency shift pattern.

3. EXPERIMENTAL RESULTS

3.1 EXPERIMENTAL SET-UP

The experimental studies were conducted in a 104 mm diameter, 2 metre long duct. The duct cut-off frequency is about 2000 Hz. One end of the duct was closed by a 13 mm thick plastic glass cap with a 25 mm driver in the centre. The other end was closed by 150 mm steel piston. The piston can be moved in the duct with and without blockage to change the length of the duct. The eigenfrequencies of the duct were measured by slowly increasing the frequency of the signal and recording the frequencies at which the sound level peaked at the end of the duct. The eigenfrequency shift patterns are obtained by plotting the eigenfrequency shifts versus mode order n.

3.2 ESTIMATION OF THE SIZE AND LOCATION OF THE BLOCKAGE

The reconstruction of the blockage dimension was made on basis of the properties of the eigenfrequency shift pattern discussed in section 2. The procedure of reconstruction is illustrated by the following example.

Example: A rectangular blockage of β=0.456 and l_2=20.3 cm in a 1.85 metre duct.

(a) Eigenfrequency shift pattern obtained from the measurement is shown in Fig.3. The pattern contains four zero points (not including n=0).

(b) The first eigenfrequency shift, $\Delta f(1)$=5 Hz, is greater than zero, so λ_b <0.5

(c) The 4 th, m=4, zero point of eigenfrequency shifts ocurr when the mode order is about, n=10, (from Eq.(6)).

$$\lambda_b = m/2n = 4/36 = 0.11$$

and $$l_2 = 0.11 l_t = 20.4 \quad \text{(cm)}$$

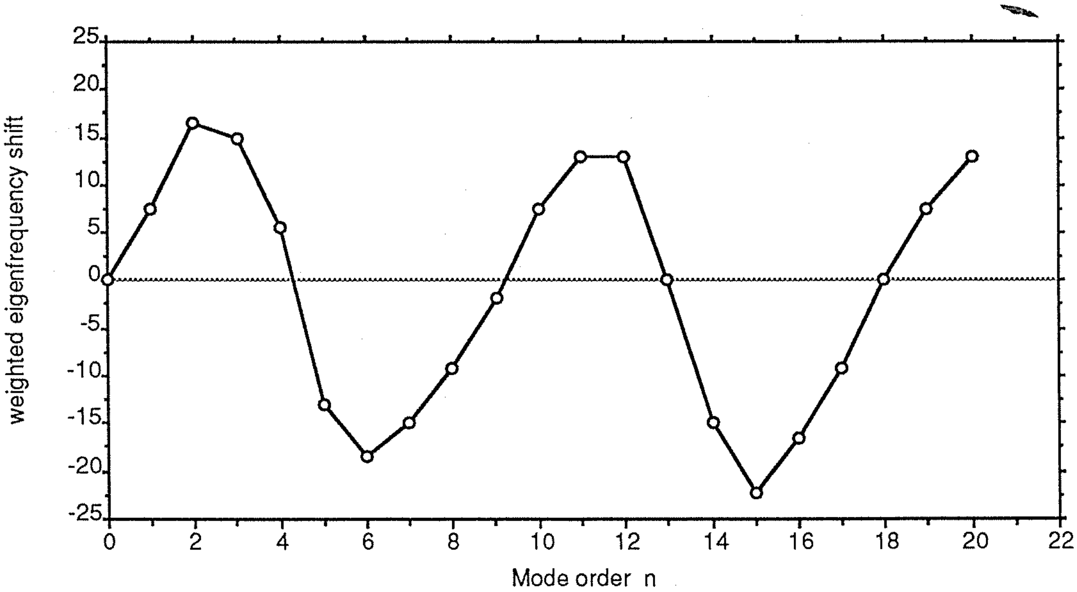

Fig.3 Measured eigenfrequency shift pattern of a rectangular blockage.

Fig.3 shows the periodicity of the eigenfrequency shift. The period is n_T=9, (from Eq.(5)).

$$\lambda_b = 1/n_T = 1/9 = 0.11$$

which is the same result as above.

(d) The amplitude of the weighted eigenfrequency shift, $\Delta f_{amp} l_t$, is 15.7 Hz. From Eq. (7) the value of β may be obtained:

$$\beta = 0.44$$

These results are in excellent agreement with the actual values of l_2 =20.3 cm and β =0.456.

Based on the procedures applied to the measured eigenfrequency shift patterns of various duct lengthes, the blockage dimensions are estimated. The results are given in Table 1.

Table 1 Results of estimated blockage sizes compared with actual values

Duct Length	Estimaed Values		Actual Values	
	l_2 (cm)	β	l_2 (cm)	β
l_t = 1.85 m	20.4	0.44	20.3	0.46
l_t = 1.85 m	50.5	0.59	50.0	0.59
l_t = 1.85 m	50.5	0.46	50.0	0.45
l_t = 1.85 m	50.5	0.21	50.0	0.23
l_t = 1.85 m	6.4	0.33	5.0	0.46
l_t = 2.0 m	50.0	0.59	50.0	0.59
l_t = 1.65 m	49.5	0.46	50.0	0.45

The agreement between the estimated results and actual values of the blockage are good, though it should be noted that when the blockage is small the error may increase.

4. CONCLUSIONS

The properties of the eigenfrequency shift induced by blockage has been studied using a one dimensional model. From the study the following conclusions can be drawn:

(i) The amplitude of eigenfrequency shift can be expressed as a unique function of the blockage cross section area ratio, β.

(ii) The properties of eigenfrequency shift pattern, such as period, node ratio and pattern shapes, are determined by the length of blockage and duct.

These two conclusions led to the method of estimation of blockage dimensions using measured eigenfrequency shifts. The experimental results show that agreement between the estimated values and actual values of the blockages is very good.

The study shows the eigenfrequency shift method can be used to estimate the blockage dimensions in the duct for the special case of a blockage at the end of an enclosed duct. Further studies are being undertaken in order to apply the method to more practical situations.

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The N.A.L. Acoustical Test Facilities

an interim report on performance.

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Facilities Manager

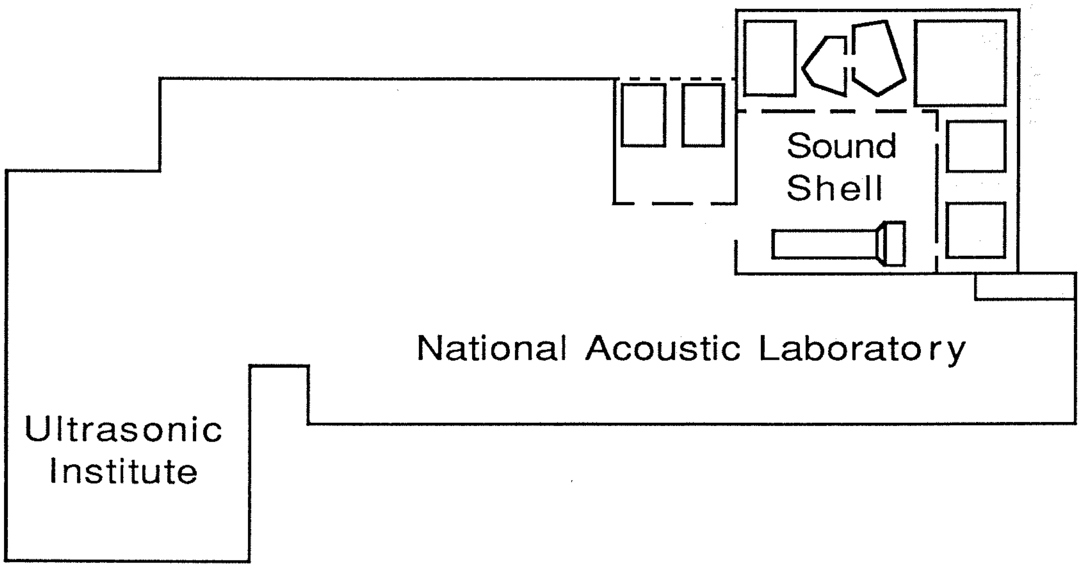
1 The new NAL Laboratories at Chatswood, Sydney.

The Laboratories are located in a wooded valley adjacent to the Lane Cove National Park. The site was selected from a choice of 26 Commonwealth properties around the Sydney metropolitan area because of its low ambient noise and ground vibration levels and remoteness from present or future major highways and aircraft flight paths. These acoustical advantages were enhanced by constructing the building in an excavation of the valley wall such that:-

a. The Facilities could be noise buffered by the valley wall on one side and the main laboratory building on two of the other sides.
b. All NAL internal "noise sensitive" laboratories and test rooms could be noise buffered by staff offices along the outer walls and corridors which serve both laboratories and offices.
c. The whole building was inside the excavation on two sides with the result that entry is on the roof and the whole design blends environmentally with the National Park environment

2. Purpose & Description of the Facilities.

The Commonwealth Department of Health and Community Services Acoustical Test Facilities were originally designed to provide National Acoustic Laboratories (NAL) staff with a means of testing in various acoustical environments covering the full range of frequencies and sound levels spanned by human hearing and to be available for research and investigation purposes by outside research organisations. This intention has been expanded in line with government policy and the Facilities are now available for hire to any person or organisation at reasonable cost-recovery rates.



Plan View of The National Acoustic Laboratories and Ultrasonic Institute

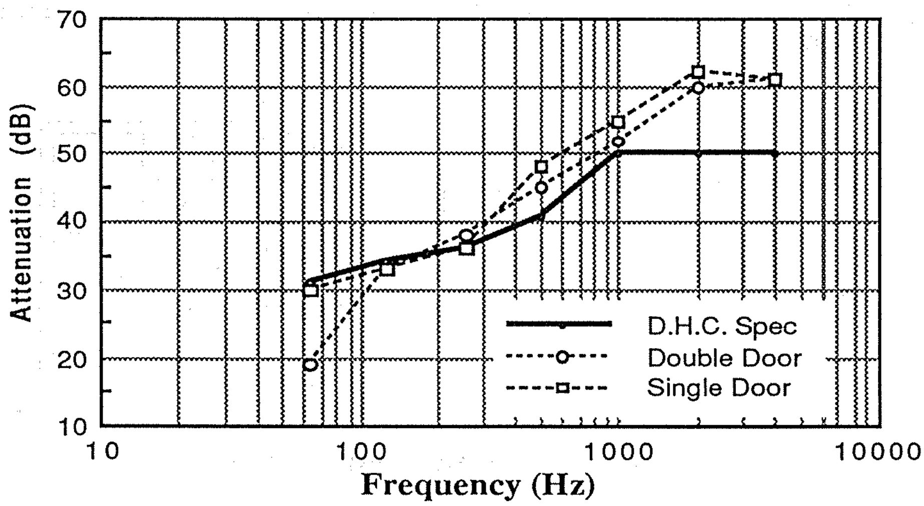
Facilities comprising four anechoic rooms, two adjacent reverberation rooms, a large quiet room (for psycho-acoustic perceptive type tests) and a large horizontally mounted, plane-wave tube are located inside a large enclosure (known as the 'Sound Shell'), which has been structurally isolated from the main N.A.L.laboratory building, to reduce extraneous noise and vibration to minimum levels. These rooms are supplied with air from a separate (to the main building) acoustically treated air conditioning plant and provision has been made to shut off the air to any room if extremely low noise levels are required. Two adjacent high intensity noise rooms, located just outside the entrance to the sound shell to ensure minimum vibration and acoustical coupling to noise sensitive rooms, complete the Acoustical Facilities. Each Test Facility is housed inside a separate room and all walls are 300 mm thick heavily reinforced, poured on site, concrete to ensure maximum possible acoustical isolation from any other Facility.

Each test room has its own control booth and maximum versatility is permitted between them by the provision of sound-treated cableways and entry ports. These cableways and entry ports facilitate the interconnection of cables for signals, data, intercom, CCTV...etc and the entry of special cables as required. It is also possible to direct-link control rooms to the 120 seat NAL theatrette so that tests which could not be carried out in the presence of an audience can be remotely controlled by computer and viewed at the same time.

3. Performance & Methods of measurement.

Door attenuation:

Measurement procedure: Insertion Loss for each door was determined by a noise consultant as the arithmetic mean of Insertion Losses at each of four measurement locations around the door. The procedure comprised recording octave band sound pressure levels (63 Hz to 8 kHz) for all combinations of pink noise source on/off and test door open/closed. The final insertion loss figure being the difference in measurement with the door open and door closed (with appropriate adjustments for any contribution from background noise).



Graph 1. Typical Acoustical Door Attenuation

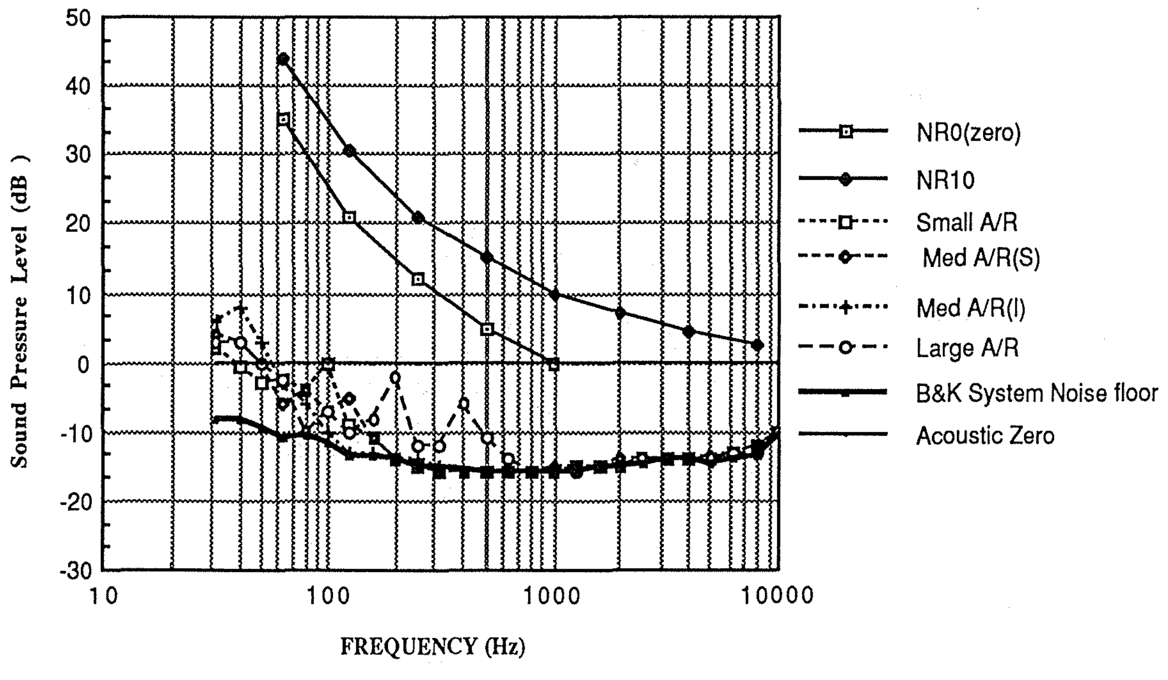
Comment. Door performance in the higher frequencies exceeded Department of Housing and Construction (DHC) specifications, but despite considerable investigations and modification, could not be improved at the lower frequencies. The above measurements were presented by an acoustical consultant employed by the contractors as typical of the performance to be expected for the single leaf and double leaf acoustical doors used throughout the Facilities. These characteristics were finally agreed to as a compromise despite their marginal failure to meet contract specifications in the 63 and 125 Hz Octaves.

4. Free-Field Performance Capability of Facilities.

Frequency response range of the horizontal plane wave tube and four anechoic rooms has been tested as required by construction contract specifications and complies with inverse square law characteristics measured in accordance with the precision methods of ISO 3745. The Large Horizontal Plane wave tube ISO inverse square law compliance capability covers the range 15 Hz to 80 Hz and free field performance of the Large Anechoic Room extends down to 50 Hz. The two Medium Anechoic Rooms are capable of meeting ISO requirements down to 90 Hz and the Small Anechoic Room down to 190 Hz (The Small Anechoic Room is also radio frequency shielded for acoustically testing R.F. sensitive devices). Upper limit testing for ISO compliance was restricted to 12,000 Hz in accordance with contract specifications, but the untested useful upper limit was stated by the testing authority to be well above this figure.

5. Background acoustical noise .

Graph 2 displays measurements of 1/3 Octave noise from 31.5 Hz to 10kHz in all four Anechoic Rooms using standard Bruel and Kjaer equipment driven by a B& K low noise microphone system (Mic. type 4179 and Mic. Preamp type 2660). The results show that specification requirements of NR10 (air conditioning on) and NR0 (air conditioning off) are well exceeded with the air conditioning on. However, the results also indicate that using 1/3 octave band measurement techniques, we cannot measure much below -5dB (S.P.L) at most frequencies and still preserve a 10dB margin to the noise floor (Instrument or acoustical, depending on frequency). If a B& K Slave Filter (narrow band filter) is used instead of a 1/3 octave band filter, meaningful measurements to -10 dB S.P.L. can be obtained at most frequencies.



Graph 2 Ambient Noise in Anechoic Rooms.

A low priority has been placed on extending measurement capability down to the true acoustical noise floors because of the very low levels measured in all test rooms.

6. Vibration mounting performance:

Test results from an acoustical consultant's report are summarised in the following table:

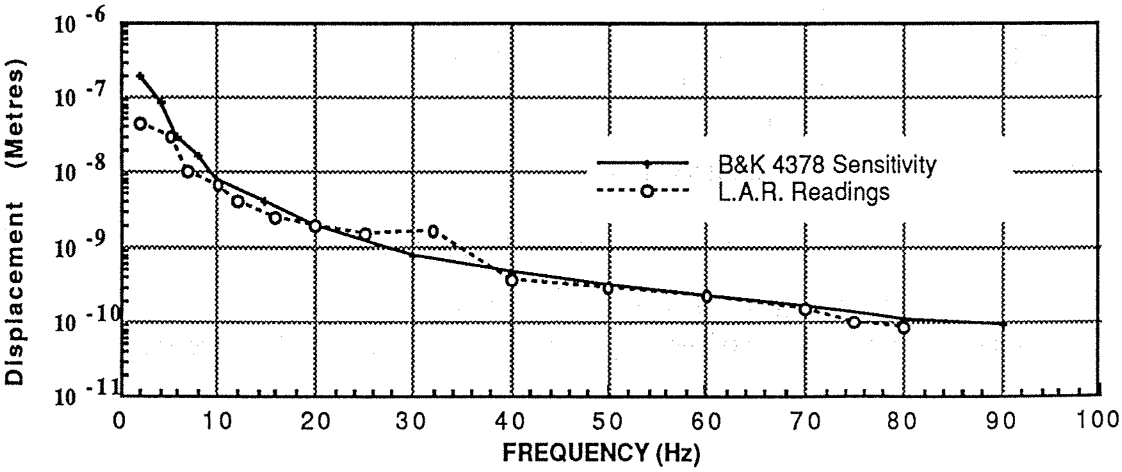
Facility	Mass (kgm)	Resonant Frequency (f_r)	Damping
Large Anechoic Room	670,000	4.7.Hz	3.7%
Medium Anechoic Room 1	192,000	4.5 Hz	3.1%
Medium Anechoic Room 2	184,000	4.8 Hz	3.4%
Small Anechoic Room	44,000	6.7 Hz*	4.0%
Reverberation Room	220,000	5.4 Hz*	2.1%
Diffuse Sound-Field Room	230,000	4.8 Hz	3.9%
High Intensity Sound Room 1	93,000	7.9 Hz*	7.3%
High Intensity Sound Room 2	93,000	7.8 Hz*	5.1%

* Comment: These rooms were outside D.H.C.Specifications (vertical natural frequency, f_r to be in the range 3 Hz to 5 Hz and damping characteristics within 2.5% to 20% of critical damping). They were accepted for the following reasons:-

- **a** reverberation room performance was considered acceptable because it only very marginally exceeded the 5 Hz maximum f_r and 2.5% minimum damping requirement,
- **b** both high intensity noise rooms are placed outside the "sound shell" vibration isolated area and their operation was not considered a vibration transmission problem either to each other, because of the high acoustical noise levels used or any of the other test facilities because of their remote location and,
- **c** the small anechoic room because its lowest operating frequency is 190 Hz.

7. Anechoic Room Vibration Levels.

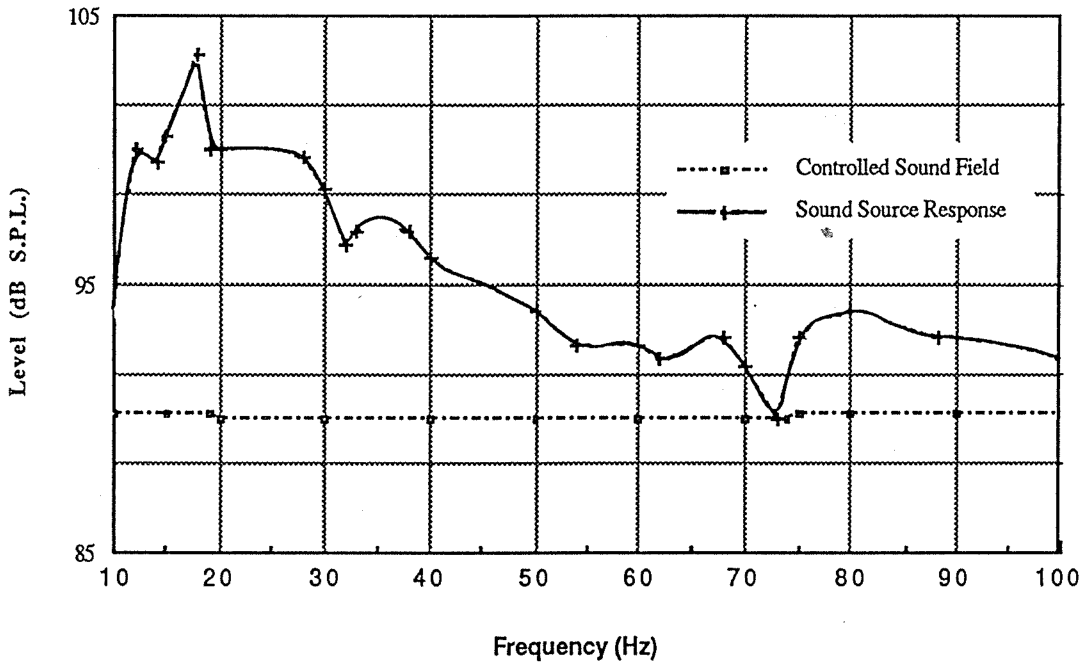
Measurements of displacement levels in the Large Anechoic Room were taken on a single occasion using a Bruel and Kjaer Accelerometer type 4378. The choice of accelerometer, based on published specifications, was made because this unit has the highest 1/3 Octave Band dynamic measurement sensitivity (of all B & K Accelerometers) for a signal-to-noise ratio exceeding 6dB. The results of Graph 3. show that measurements obtained are similar in level to those published for the accelerometer and that accurate determination of the displacement readings cannot be determined by available instrumentation. Other Rooms with spring suspension systems have not been tested to date but similar results are anticipated.



GRAPH 3. Large Anechoic Room Vibration Levels

8. Large Horizontal Plane Wave Tube Performance.

The N.A.L. Horizontal Plane wave tube is 24.5 Metres long and 2 Metres square. It is terminated at one end by an acoustical wedge which is 15 Hz quarter wavelength long and currently is fitted with one vented enclosure (containing a JBL 2245H, 18 inch loudspeaker) mounted in the centre of a 2 Metre square baffle. Performancewise, the tube simulates free field measurement conditions that would be experienced in an anechoic room which has length, breadth and depth dimensions twice the length of the tube but is achieved at a fraction of the cost of such a room. Two measurements only have been recorded to date. These were taken after adjusting the speaker enclosure vented port for optimum low frequency performance and are presented here as Graph 4. The measurements comprised a compressor-controlled frequency response from 10 Hz to 100 Hz (to determine how well the sound field level could be controlled at the measurement location) and a frequency response without compression control over the same frequency range to determine the sound-source frequency response. Seven other vented enclosures of the same design for fitting across the cross sectional area of the tube are currently in assembly so that higher levels of infrasound can be obtained if required.



Graph 4. Plane Wave Tube Performance

9. Reverberation Rooms.

Two 200 cubic Metre side by side reverberation rooms are located in the sound shell between the large anechoic room and the quiet room. The coupling aperture between them is 10 square metres. Boundary surfaces of each room are non-parallel to enhance sound diffusion and each room has a different shape to discourage the possible transfer of energy between them by standing waves if transmission loss testing is required. At present the aperture is sealed and the rooms are configured so that one can carry out sound power and acoustical absorption testing while the other one can be used for duplicating various acoustical environments for device intelligibility testing and for speech communication research.

10. High Intensity Noise Rooms.

These Facilities comprise two spring-mounted high intensity noise rooms located in an enclosed, acoustically isolated space just outside the sound shell. Each is a hard-surfaced, 300mm thick reinforced concrete walled chamber, having internal dimensions of 6m (length) x 5m (width) x 3m (height) and one room is equipped with special ventilation and drainage to allow its use in studies on animal subjects.

11. Large Quiet Room.

A large quiet room with 300mm thick reinforced concrete walls, ceiling and floor and dimensions 10m (length) x 6m (width) x 2.6m (height) has been provided for subjective listening tests. The Room has a low (and marginally variable) reverberation time and is capable of simulating domestic, classroom or other similar type environments.

AWARDS

The Facility anechoic rooms have just won an award for engineering excellence on the basis of design and performance achievement in the 1988 competition for projects of engineering excellence run by the Australian Institution of Engineers and an entry has also been placed in the Australian Acoustical Society 1988 competition for excellence in acoustics.

A.T.F. POLICY.

The aim of Facilities personnel is to assist NAL staff, to satisfy the requirements of any person or organisation through provision of technical advice or hire of Facilities, instrumentation and/or expertise and to anticipate future acoustical activities of the Australian community by an ongoing programme aimed at developing new measurement techniques and improving the performance capability of the Facilities.

AUSTRALIAN ACOUSTICAL SOCIETY

1988 NATIONAL CONFERENCE

VICTOR HARBOUR, S.A.

OPTIMISATION OF VARIABLE LER FOR SPATIAL IMPRESSION

BY

Robert Williamson

School of The Built Environment
(South Australian Institute of Technology)

Abstract

The strength of the lateral energy in a hall was systematically varied while subjective impressions of a music source were investigated using a questionnaire. A number of objective indices were measured from the room impulse responses and an optimum lateral energy condition was established from the subjective impressions. Inter-correlation between subjective and objective measures indicated that the "overall impression" of the hall was determined by only a few independent subjective attributes and a number of physical measures were identified as suitable correlates for these. The use of source width as a measure of spatiousness was confirmed along with the contribution of relative magnitude of early lateral energy to it and all frequencies were found to be subjectively relevant. The strength of the overall sound level at the receiver was also found to be an important contributor to spatial impression but had little effect on other subjective attributes.

OPTIMISATION OF VARIABLE LATERAL ENERGY FOR SPATIAL IMPRESSION IN A HALL

Introduction

Research [1] has emphasised the importance of identifying the physical parameters which contribute to the quality of the musical experience in a concert hall and understanding the complex relationship between these and subjective impression.

Early, lateral, reverberant and overall sound energy contribute both individually and collectively to the subjective music experience for any particular audience location and, although a number of physical correlates of this experience have been established, there is no general agreement over which should be included in concert hall studies.

There is also considerable evidence [2,3] that strong, early lateral reflections produce a desirable feeling of spaciousness and a number of indices [4,5,6,7] have been suggested as suitable objective measures of this.

Spatial impression *(width)* is currently considered one of the most important subjective parameters for listening to music in a concert hall [8].

The primary objective of this study [9] was to investigate the effect of varying the lateral energy ratio in a hall on a number of subjective and objective parameters relevant to room acoustics. Of particular interest was whether a causal relationship existed between variably controlled lateral energy and the subjective impressions gained about the hall when listening to music which would lead to an optimum "overall impression" determined by the lateral energy conditions. Another objective of the study was to rationalise the existing subjective attributes and their physical correlates by studying how they interacted when one of these was varied in a controlled manner. A recent study [18] of a hall with variable acoustics determined that the Centre Time index (t_s) was highly correlated with both the Clarity (C) and the early decay time (T_{10}) and, therefore, not necessarily an improvement on these. By measuring a number of established room-acoustical indices (Table 1) which correlated with the subjective attributes, conclusions were drawn about their relative importance to, or dependence on, the lateral energy conditions and subjective impressions in the hall.

An additional index (St) for stage acoustics was included since it could easily be obtained from the impulse response data and is different from the other indices. The St (Support) index [17], which is characterised by the increase of the early energy received over the direct sound, can be expected to have some similarity to Clarity.

The advantage of controlling a few acoustic variables (e.g.lateral energy and early/late energy) in a hall overcomes some of the disadvantages of studies across different halls caused by the variability of judgement between performances (due to the uncertainty of reliable memory) and the uncontrolled interaction of a large number of variables.

Hall Geometry and Data Acquisition

The objectives were achieved by systematically varying the strength of the early lateral reflections using recorded music and a loud speaker source on stage representing a string quartet (Figure 1). Laterally placed speakers were fed with the original signal, suitably delayed and at various levels relative to the stage source, to provide the increased lateral energy. The loudspeaker source on stage (S) was hidden behind the proscenium curtain for all music listening studies, minimising unrealistic visual clues and a tendency by subjects to focus on a single point. For the same reasons, additional dummy lateral speakers were also included to minimise visual clues to apparent source width.

The stage source (S) was a B+K (4224) sound source which met the need to have separate control and amplification and provided a high frequency spectrum comparable with that for the matched lateral speakers after attenuation by the proscenium curtain. A 10 dB low frequency (125 Hz) deficiency in the stage source spectrum had the advantage of boosting the bass components of the added lateral energy - a factor which is considered important for spatial impression [5] and is significant for the objective correlations reported in this study. There was no evidence of pronounced acoustic deficiencies for the measurement location (R) selected in the hall.

For the objective indices, direct, short duration pulses were used as the room excitation signal - the duration times optimising the available power for the octave bands analysed. The pulses were recorded along with, but on a track separate from, the impulse responses of the room at R.

The output levels of both the direct and the laterally placed speakers were individually varied, so that the overall sound level received at R remained constant. Each lateral speaker signal was delayed relative to S so that the signals received at R represented early lateral reflections. Based on the natural echograms for the hall, the final delay times used included adjustments so that the "reflections" from LL filled a gap in the natural echogram and those from RL reinforced an existing reflection sequence.

In the following discussion test A represents the existing natural hall acoustics. For tests C,D,F and G there was an increase above the natural lateral energy. The effect of an increase in overall sound level at R was invesigated for test B while test E represented an unbalanced lateral energy boost from either side of the hall. The lateral energy conditions in the hall for test A and B were the same since only the overall level at R changes. For each frequency band and for all test conditions the reverberation times of the hall (T_{60}) were 1.5±0.1 secs and were unaffected by the increase in lateral energy or a change from lateral to omni-directional measurement.

For the objective indices investigated (Table 1) most were obtained from the backward integration [13] of the computer stored hall impulse responses. All indices were evaluated for both omni-directional and lateral integrated squared impulse responses.

<u>Subjective Evaluation of the Hall</u>
Subjective impressions were assessed by a questionnaire based on scales of the semantic differential type - the format, length and style being based on similar studies [8,10,11] as well as feed back from a pilot study - and these impressions were related to a lateral energy index [12]. The main subjective determinants used are listed in Table 2 and were assessed by listeners on a continuum between two bi-polar descriptors. Subjects preferred "Blend" to "Reverberance" as a descriptor for the liveness of the space. The scales were considered to be mutually independent although the "overall impression" scale was expected to be dependent on either some or all of the other scales.

For a direct estimate of apparent source width and centre it was appropriate to relate these to the apparent source boundaries and stage centre. On the basis that all subjects interpreted this scale in the same way, direct comparisons could then be made between apparent source broadening and lateral energy [14].

A total of 42 tests were presented in random order for the 7 different test conditions (A to G) to ensure that each test was subconsciously compared with every other test - each test requiring a questionnaire to be sequentially completed. Judgements were carried out by 3 groups of 5, 6 and 3 listeners, respectively, at R - this arrangement allowing all tests to be presented on the same day and ensuring that no subject was more than 1 seat away from R.

Assessments of each parameter were converted to numeric scores based on a scale continuous from 0 at the left hand side to 4 on the right. *"Width"* was estimated from the difference between the extreme source boundary scores and "position" was determined by the averaged sum of these scores.

The music source was a two-track, 30 sec. recorded segment of Mozart's Minuet (K421) - the violin part being on one track and the second violin, viola and cello on the other. Separated by a 15 second interval, 45 repeats of this motif were pre-recorded for continuous presentation to subjects.

<u>Validation of Questionnaire Technique and Assumptions</u>
Figures 2-4 combine the data showing group means, standard deviations and the spread of individual mean scores. Test A represents the natural acoustic field condition for the hall.

The questionnaire technique can be validated in 2 ways - whether subjects are scoring on different scales independently but with "overall impression" as a dependent variable, and whether significant differences are observed on each scale between test conditions. By generating a correlation matrix for the individual scales (Table 3), the independence of judgements can be assessed.

If all scales are independent, no significant correlations will exist between them. Apart from "blend", "clarity" *"width"* and "loudness" have the strongest correlations with "overall impression". Moderate correlations ($r>0.48$) also exist between "loudness-clarity", "clarity-*width*" and *"width*-position. Since there are no redundancies due to significant inter-correlations (i.e. "loudness-*width*" or "loudness-position") then the independence of these scales appears to be a valid assumption.

The difficulty in assessing "blend" is supported both by its dependency on most of the other scales and the indifference of the subjective scores to the test conditions (Figure 3) and, therefore, its contribution in the study was ignored.

The presence of individual differences between test conditions can be assessed by a one-way analysis of variance of the individual mean scores for each scale. The difference between test conditions was highly significant for all scales except "blend" - the order of significance being (in ascending order) "loudness", "position", *"width"*, "overall impression" and "clarity". Both "loudness" and "position" are highly significant because of the influence of test condition B and E respectively.

Although it appears that the subjects are using the scales in a consistent way and that judgements have been influenced by the test conditions this does not mean that the subjects place the same weight on each scale in establishing an overall impression of the hall. The ease with which subjects assessed each test can be seen by the variation in the standard deviation ranges (Figures 2 -4). This was also checked using a chi-square test for variance and the results - in ascending order of consistency - are "clarity", "blend", "overall impression", "loudness", *"width"* and "position". For a specific attribute, some test conditions may have influenced judgements more than others and can be assessed by a t-test to compare each test condition in the hall with test A. Maximum rejection of the null hypothesis (no improvement) occurs for test C for "clarity" and "overall impression" while there is no significant change in source "position".

In applying parametric statistic tests to the data, it is assumed that the frequency distribution of judgements on the individual scales are normally distributed about a mean. By standardising the cumulative frequency distribution of scores for each particular test condition, the normality of the scores can be tested against a corresponding theoretical ogive. The linearity of the subjective scale is determined by the strength of the correlation between the actual and theoretical distribution. For each of the "clarity", "loudness", *"width"* and "overall impression" scales the correlations were better than 0.95.

<u>Characteristics of the Objective Indices</u>

The stored hall impulse responses, from which the objective indices were derived, demonstrated the strong contribution of the bass boosted lateral energy for tests C to G by initial sharp drops within the first 50 msecs. in the 125 Hz band. This characteristic significantly influenced the lateral component values of objective indices based on the early part of the impulse response.

(1) <u>Reverberation Indices</u> (T_5, T_{10}, T_{30})
By linear interpolation over a number of values, the decay times, corresponding to -5dB and -10dB below the direct sound, were estimated from the integrated squared impulse responses. T_{30} was unaffected by an increase in lateral energy or a change from lateral to omni-diectional measurement. The influence of the bass boosted early lateral energy on T_5 and T_{10} can be seen in Figure 5 for tests C to G.

(2) <u>Early Energy Indices</u> (C, t_s, St)
Values for these indices were also calculated from the squared impulse responses for both omni-directional and lateral decays but only the lateral decay indices (Figures 6 and 7) showed significant variation with the test conditions. The values of C for omni-directional responses, when averaged over the frequency range 125 Hz-2 kHz, are within an acceptable range of 0dB to 5dB for all test conditions. The low frequency increase in C contrasts with the minimal high frequency change and results from the strong lateral bass frequencies.

Unlike C, Centre-Time is influenced primarily by any time interval within which strong reflected energy arrives at the receiver - its value decreasing the earlier and more predominant this energy becomes. For the present study, strong, early lateral low frequency energy arrives within the first 50 msecs.

Support is also defined in terms of the arrival of reflected energy at the receiver, but is characterised by the gain of this (within the first 100 msecs.) over the energy of the direct sound. In this study, the increase in early lateral low frequency energy, with a corresponding decrease in direct, increases the value of St in the 125 Hz band.

Although of doubtful subjective significance, these laterally measured indices were more sensitive to a change in lateral energy conditions in the hall than those measured omni-directionally and were included, therefore, in the correlation analysis.

(3) <u>The Lateral Energy Ratio</u> (LER)
The values of LER in Figure 8 demonstrate its broad band frequency sensitivity and, as for the previous indices, the significance of increasing the low frequency lateral energy is reflected by the large spread of values in the 125 Hz band.

(4) <u>Sound Level</u>
The overall sound level at R was determined by analysis of omni-directional and lateral recordings of steady pink noise. For subjective evaluations of the music source, the overall level at R did not exceed 85dB for the relevant test conditions.

<u>Subjective Attribute Assessment</u>
From Figures 2-4, it appears that test B (an 8dB increase in level at R) increases the subjective impression of "loudness", reduces "clarity" and increases the apparent source width relative to Test A. There is only a slight improvement in "blend", however, and the "overall impression" of the hall is worse than for any other test condition. Increasing the lateral energy, relative to the direct sound, clearly increases the apparent source width but its "position" suffers from a localisation shift towards the left lateral speaker as it becomes the dominant source for tests F and G. Adding lateral energy tends to improve "clarity" and the "overall impression" of the hall but appears to have little effect on "blend".

Contribution of Variable LER to Subjective Impression

"Envelopment", spatial impression (SI) and *"width"* imply an apparent spatial spreading of a music source beyond its visual boundaries. In this study, the scale of apparent width used to measure this effect can be linearly related to Barron's [5] spatial impression scale and, therefore, assessed by LER - the recommended room acoustical correlate.

The LER indices for the hall were converted to units of SI which can be assumed to monotonically increase for the high levels of added lateral energy. A strong degree of correlation between *"width"* and LER (r=0.93) confirms the strength of correlation between subjective source width and eqivalent SI in the hall. For values of LER greater than -7dB one degree of SI corresponds to a change in LER of about 1.4dB or 0.3 units of *"width"*. The high degree of correlation between LER (and hence SI) at all frequencies supports the view [14] that all frequencies are subjectively important for SI and should be considered in the practical design of auditoria or sound reinforcement systems.

The subjective scores for "clarity" and "overall impression" (Figures 2-4) suggest an optimum lateral energy condition (test C) which is also statistically significant. Additional support is provided by no significant apparent source shift for test C although this condition could equally result from a modified reflection sequence. The optimum LER of -1.3dB broad spectrum (or -1.0dB low frequency) corresponds to about 5.5 degrees of SI which represents a broad spectrum increase in LER of about 3dB above the natural hall conditions. Increases of this order are probably only likely to occur in small and narrow halls or by electro-acoustic means.

The effect on other subjective parameters when SI was varied can be assessed from the strengths of the inter-correlations given in Table 3. Apart from "position", the strongest subjective correlate with *"width"* is "clarity". This is also optimised for test C by the lateral energy conditions - the value of C at this condition ranging from 0dB at low to mid frequencies to 1.6dB for a broad spectrum average - however, the correlation significance limits this to the single hall study.

Contribution of Sound Level to SI

Research [15,19,5] indicates that the degree of SI is also a function of music sound level - a change of 1.4 degrees of apparent source width corresponding to a change in listening level of 1dB (the difference limen for SI being about 4dB). In this study, an increase of 8dB in the overall level at R (test B) clearly produces a change in SI which is supported by the results in Figure 2 and a t-test which also indicated that *"width"* was the only subjective attribute significantly influenced by the level increase. The 8dB increase, therefore, can be expected to correspond to an apparent source width increase of about 11 degrees or 2 degrees of SI.

The change in apparent source width score between test A and B can be converted to 0.9 degrees of eqivalent SI by considering a score change of 0.22 (on a 0 to 4 scale) corresponding to a linear width change of 0.7 metres between LL and RL in the hall. The increase in overall level, therefore, corresponds to a change in apparent source width of about 6 degrees. At low frequencies, for the same change in score, an 8dB increase is equivalent to a change in SI of 1.6 degrees which agrees reasonably well with the 2 degrees difference limen predicted by Barron.

Since *"width"* and SI are linearly related, it follows that if a 1.6 degrees change in broad spectrum SI had occurred, this would have been equivalent to an 11 degrees change in apparent source width. This appears valid since the degree of SI experienced in the hall was determined mainly by low frequency lateral energy. The lack of any significantly stronger correlation between LER and SI at low frequencies, however, compared to mid to high frequencies, suggests that SI was not determined only by low frequencies.

<u>**Physical Correlates with the Subjective Attributes**</u>

Based on other research [7,17], it was assumed that the LER index is a suitable correlate with the degree of SI in a hall and, with correlations greater than 0.9 for all frequencies, this appears to be valid. Linear regression of each objective measure derived for the hall with the subjective scales provides an assessment of those measures which correlate significantly with the associated subjective attribute. Correlations between the objective measures and the LER index were also included to examine whether any of the other indices could provide a measure of SI equal to that for LER. The highest correlates with the subjective attributes are given in Table 4 with the most significant correlates being indicated. Lateral response and low frequency measures were included because of their relevance to the study conditions. "Blend" was ommitted because of a lack of any meaningful change for the test conditions.

From Table 4, it appears that more than one index may be suitable as a measure of the appropriate attribute. For example, although the "clarity" and *"width"* scales were shown to be independent, the recommended indices, C and LER are, in each case, almost equally highly correlated with the subjective measure. This apparent redundancy may be due to the similar definition of both indices or the contribution of lateral energy to an optimum "clarity" condition in the hall. Because of the contribution of *"width"* and "clarity" to the "overall impression", it is not unexpected that both C and LER also correlate well with it. For SI, the broad spectrum C and LER indices appear to be equally appropriate correlates although the lateral measures of T_5 or T_{10} may also be suitable because of their relative ease of measurement.

By grouping the early energy and reverberation indices separately, the relative importance of their contribution to LER (and hence SI) can be seen in Figure 9. The strength of the inter-correlations between the three groups supports the literature evidence that all of these parameter groups contribute to spatial impression but the results also suggest that no particular group is more important than the other. In general, the strongest correlates are the early energy low frequency lateral measures. Although the reverberation indices all correlate highly with LER, the strongest measures are C and t_s which supports other research evidence [18].

Conclusions

Although limited to a single hall and source/receiver positions, the study has reinforced the importance of achieving an above threshold level of early lateral sound for the satisfactory subjective impression of music and, for the objective correlates which measure this, provided evidence of redundancies.

By varying the degree of SI in the hall, its contribution to an overall impression, relative to other subjective attributes, was investigated for an optimum level of lateral energy. The most favourable overall impression of the hall was determined primarily by an optimum *"width"* (SI) and "clarity" corresponding to test C - LER being -1.3dB when C was 1.6dB. With broad spectrum LER's in Australian and British concert halls being in the range -4dB to -10dB [16], this suggests that there may be considerable scope for improving both spatial and overall impression.

A high degree of correlation was established between SI (*"width"*) and its physical correlate (LER). For a 1 degree change in SI there was a corresponding change in LER of 1.4dB. The importance of low frequency lateral energy to SI was evident from an increase in the difference limen sensitivity.

Increasing the overall level at the receiver also confirmed the importance of "forte" passages in music to the impression of being surrounded by sound but contributed little to the other subjective attributes. An 8dB increase in loudness was equivalent to an apparent source width of about 6 degrees or 1 degree of broad spectrum SI - a comparitively greater increase in apparent source width at low frequencies again reinforcing the importance of these to SI.

Correlations between subjective attributes and objective measures provided evidence of redundancies - particularly for C and LER which were both equally suitable measures of "clarity", *"width"* and "overall impression". A number of physical correlates of SI were found to be statistically significant - probably due to a similar definition or a particular sensitivity to the added lateral energy. Of these, the low frequency lateral measures for T_5 and T_{10} appear to be most suitable because of their relative ease of measurement compared with LER, C or t_s. All of the reverberation, lateral energy and early to late energy parameter groups were found to contribute to SI indicating the importance of these for subjective impression.

Although the study has attempted to rationalise the number of objective measures of importance to SI when it is the main determinant of overall impression, further studies would be worthwhile to establish the influence on SI and overall impression when attributes such as clarity, reverberance and loudness are varied.

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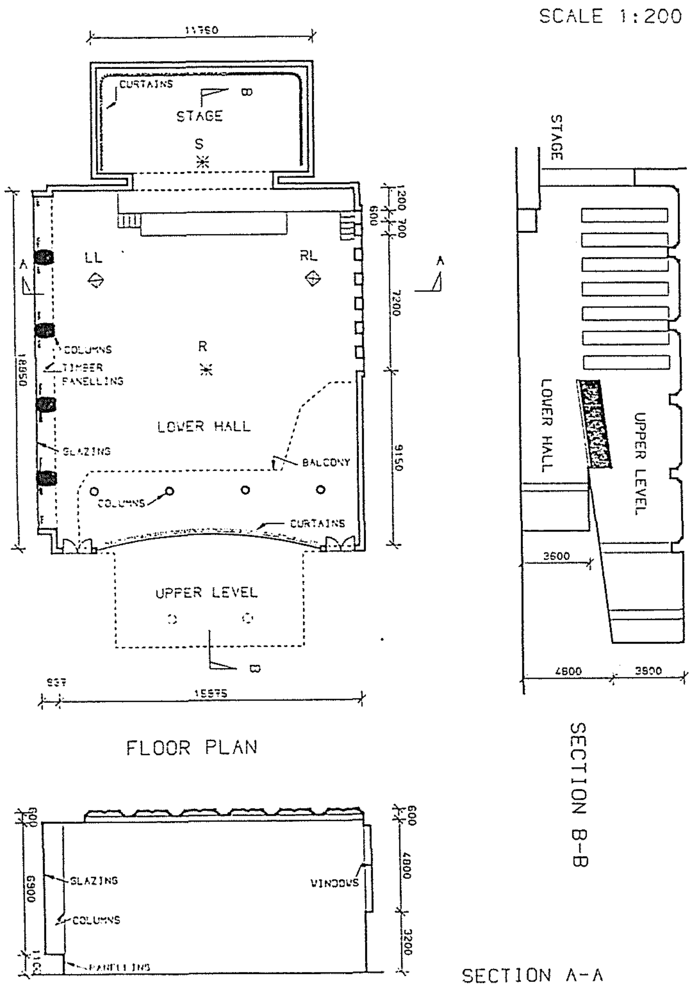


Figure 1 Plan and Section Details of the Albert Thomson Hall (Edinburgh)

Figure 2 Subjective Attributes for the Hall Study

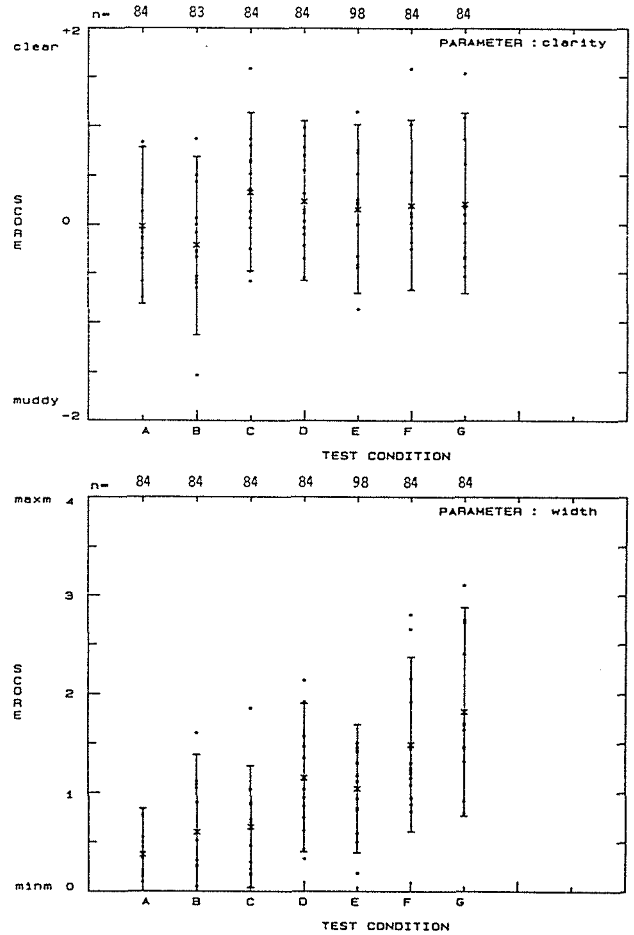


Figure 3 Subjective Attributes for the Hall Study

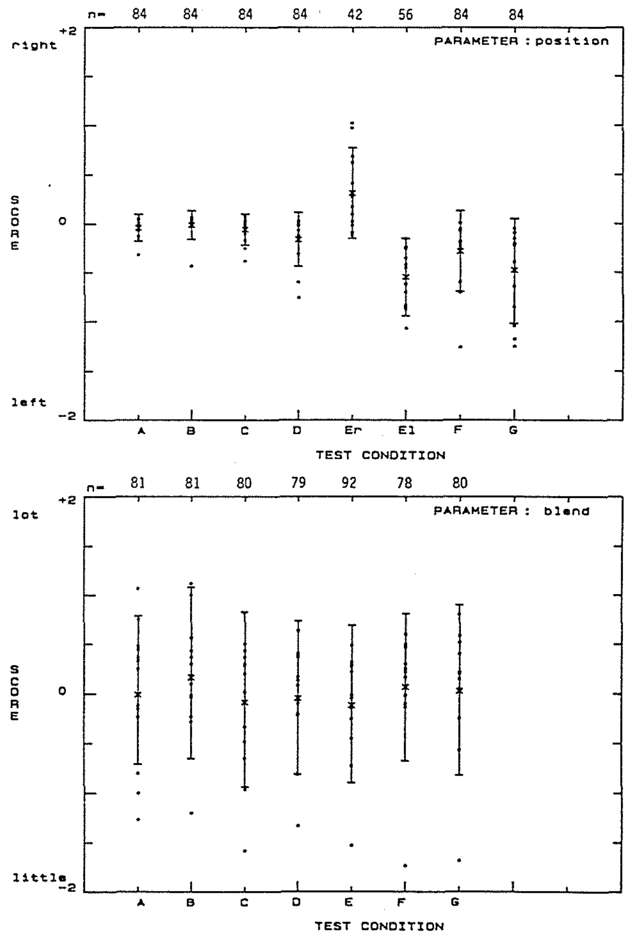


Figure 4 Subjective Attributes for the Hall Study

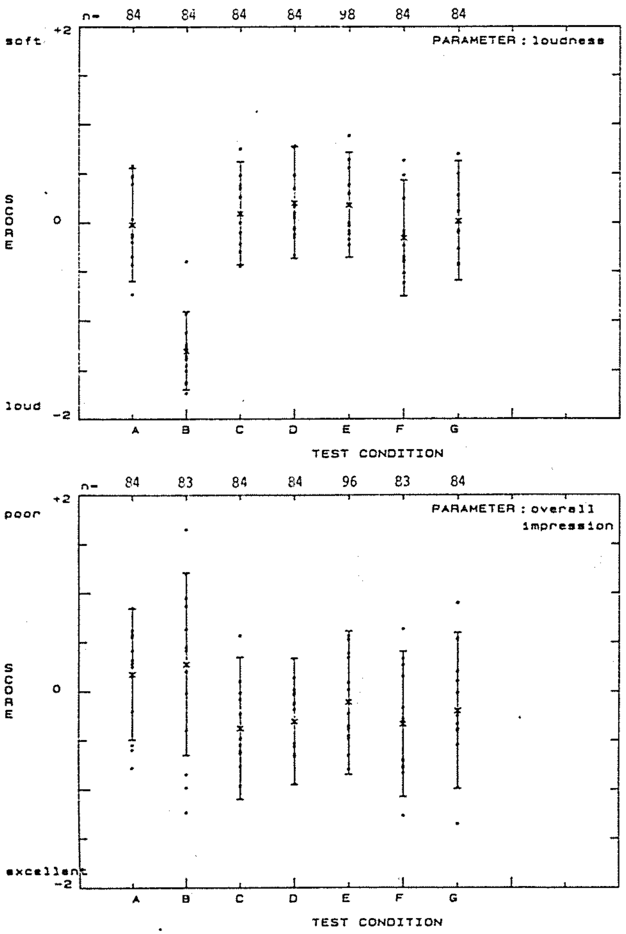


Figure 5 Reverberation Indices for the Hall Study

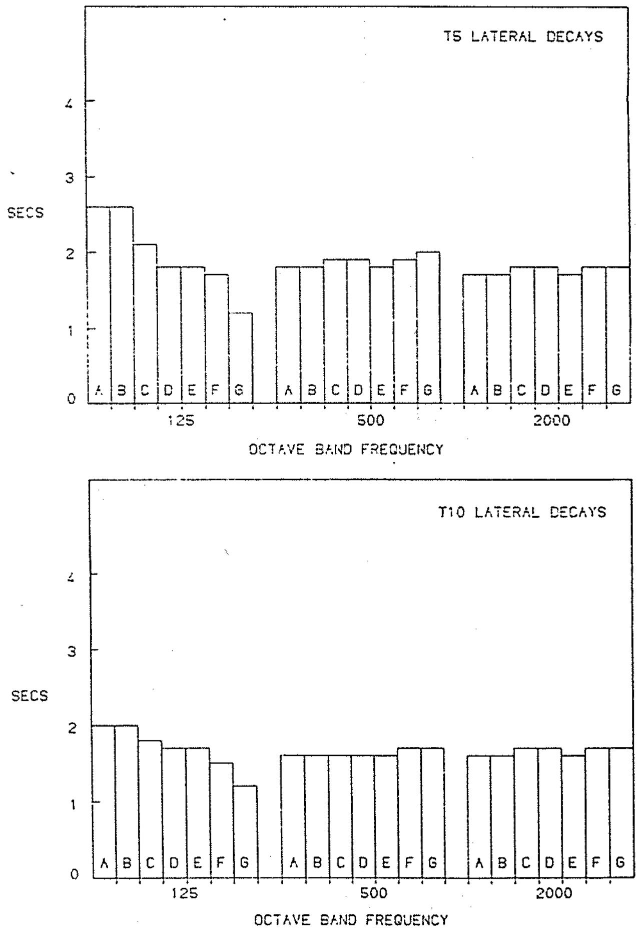
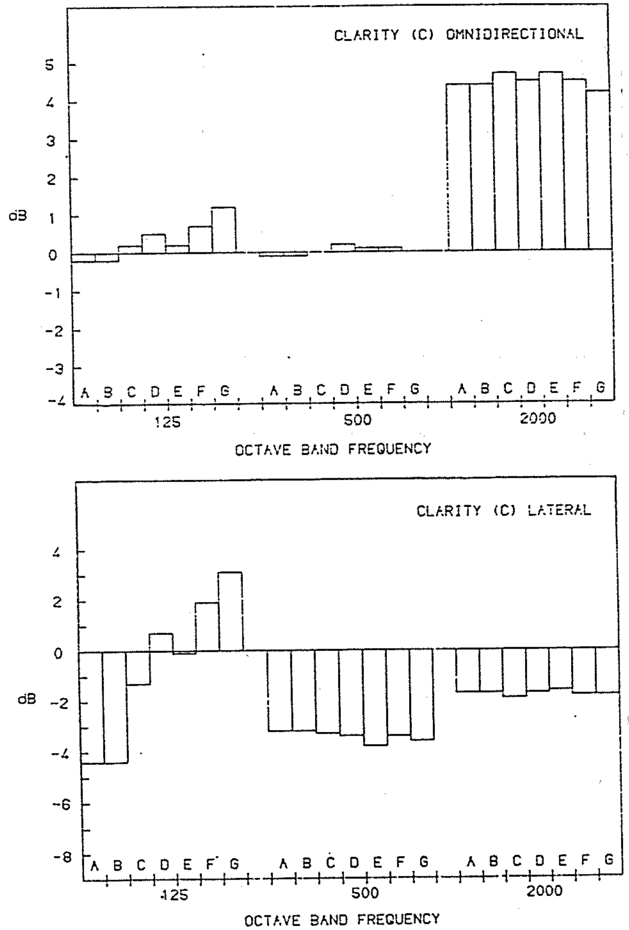
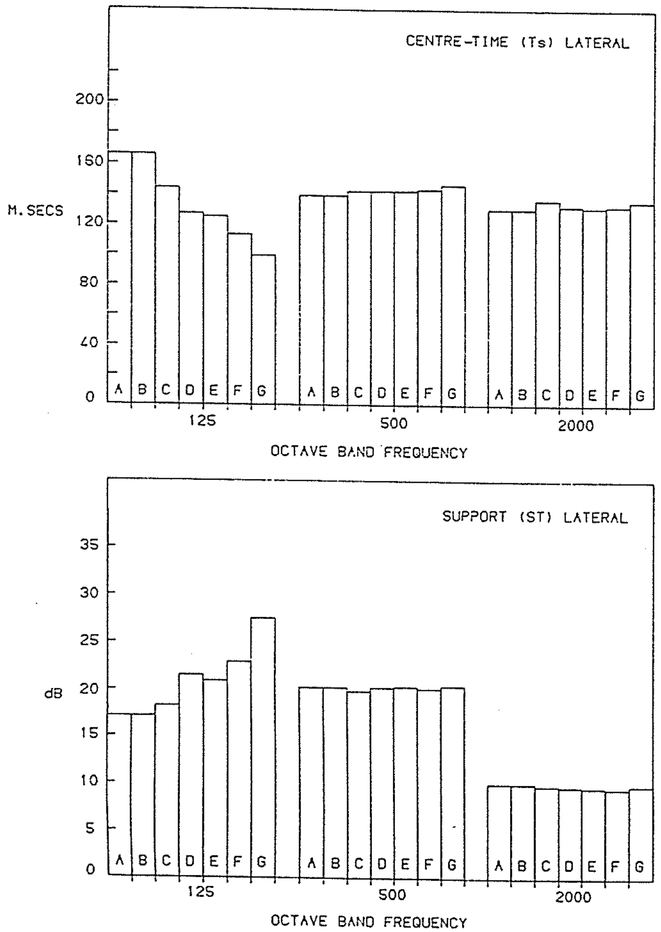


Figure 6 Clarity Indices for the Hall Study



5.12

Figure 7 Centre-Time and Support Indices for the Hall Study



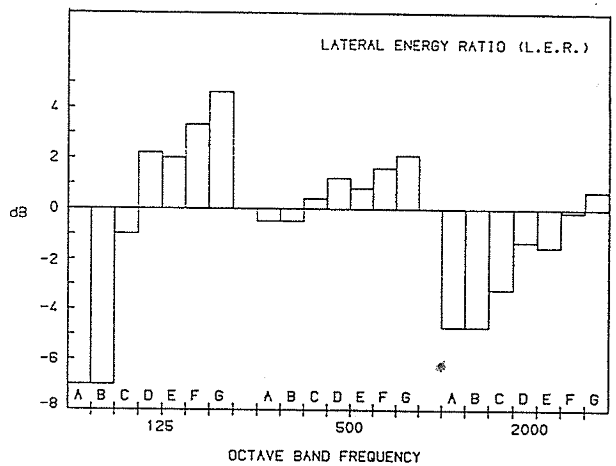
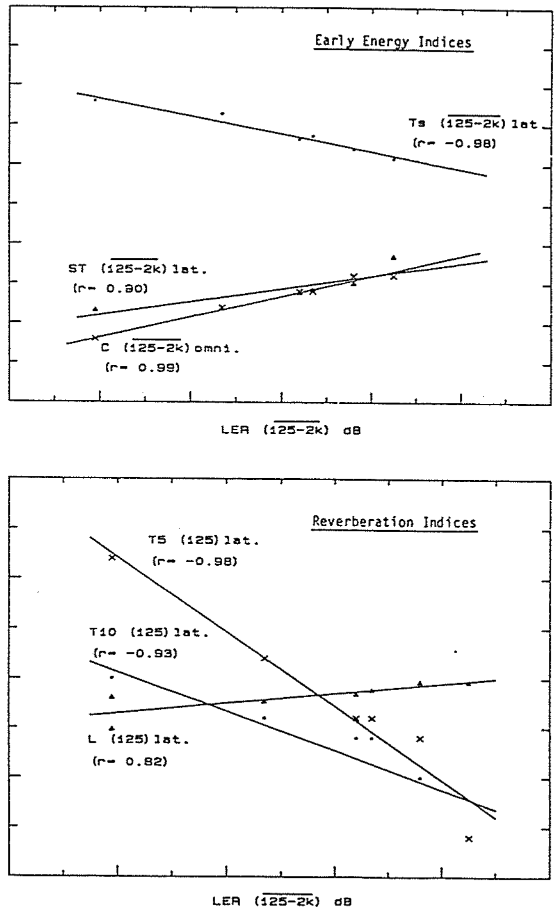


Fig. 8 LER Indices for the Hall Study

Figure 9 Objective Correlates with LER - Hall Study



(all correlations significant at the .01 level)

Table 1 Objective Indices Selected for the Hall Study

Parameter Group	Symbol	Definition	Description
Early Energy	ST	$10 \log_{10} \left[\dfrac{\int^{100} g^2(t).dt}{\int_0^{\infty} g^2(t).dt} \right]$	support; the early energy gain relative to the direct sound
	C	"Klarheitsmass" (Clarity)	Ratio of early to late energy
	t_s	"Schwerpunkzeit" (Centre-Time)	Centre of area in time units of the area below the squared impulse response curve.
Reverb-eration	T_{30}	Reverberation Time (RT)	
	T_{10}	Early decay Time (EDT)	
	T_5	Very early decay time	
Lateral Energy	LER	$10 \log_{10} \left[\dfrac{\int_5^{80} g^2(t).\cos^2\theta.dt}{\int_0^{80} g^2(t).dt} \right]$	Lateral Energy Ratio; the ratio of early lateral energy to total energy
Level	L	Overall sound level	

Table 2 Subjective Attributes Adopted for the Hall Study

Attribute	Bi-Polar Descriptions
Clarity	Muddy - Clear
Width (Envelopment)	None - since apparent width intended to be scaled on plan. (Constricted - Expansive)
Blend (Reverberance)	Little - Lot (Dead - Live)
Loudness	Soft - Loud (Quiet)
Overall Impression	Poor - Excellent

Terms in parentheses are common to other researchers)

Table 3 Inter-Correlation Matrix for Hall Attributes

Scale	Clarity	Width	Position	Blend	Loudness	Overall Impression
Clarity	1.0	-	-	-	-	-
Width	0.48	1.0	-	-	-	-
Position	-0.42	* -0.9	1.0	-	-	-
Blend	-0.78	-0.11	-0.10	1.0	-	-
Loudness	0.83	0.30	-0.26	* -0.90	1.0	-
Overall Impression	* -0.95	-0.56	0.11	0.62	-0.67	1.0

(indices marked * are significant at the .01 level)

Table 4 Highest Correlates for Subjective Attributes

Objective Index	Frequency Range	Measurement Orientation	Correlation Coefficient Attribute or Index: LER 125-2k	Clarity	Width	Loudness	Overall Impression
C	125 125-2k	omni-direct	* 0.93 0.99	0.62 0.75	* 0.96 0.91	0.42 0.56	-0.66 -0.81
	125 125-2k	lateral	* 0.99 0.99	0.72 0.71	* 0.95 0.95	0.54 0.52	-0.78 -0.77
T_5	125 125-2k	lateral	* -0.98 -0.98	-0.69 -0.68	* -0.96 -0.93	-0.54 -0.56	0.71 0.70
T_{10}	125	lateral	* -0.93	-0.58	* -0.96	-0.41	0.60
L	125 125-2k	lateral omni-direct	* 0.82 -0.63	0.40 -0.85	* 0.85 -0.38	0.03 -0.99	-0.58 0.26
t_s	125 125-2k	lateral lateral	* -0.99 -0.98	-0.67 -0.60	* -0.96 -0.97	-0.51 -0.49	0.71 0.65
ST	125 125-2k	lateral lateral	* 0.91 0.87	0.50 0.43	* 0.98 0.96	0.39 0.34	-0.52 -0.45
LER	125 125-2k	–	– –	0.77 0.73	* 0.90 0.93	0.60 0.60	-0.80 -0.80

(– Correlation coefficients very small or meaningless)
(* Correlations significant at the .01 level)

SPEECH AUDITORIA - DETERMINATION OF PREFERRED LOUDSPEAKER PLACEMENT, AIM POINTS AND DIRECTIVITY CHARACTERISTICS TO MAXIMIZE SPEECH INTELLIGIBILITY

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1.0 INTRODUCTION

Conflicting advice from professionals and contractors/suppliers of sound reinforcement systems has led the author to develop a rational procedure for the design of sound systems for speech auditoria. The procedure has been computerized and takes into account the major factors influencing speech intelligibility, namely: auditorium reverberation time, signal to background noise ratio, ratio of direct to reverberant sound pressure levels, speaker locations and aiming points together with their directivity characteristics.

The computerized procedure has been specifically developed to use the type of information readily available from reputable speaker manufacturers together with reverberation time estimates for the auditorium. Speech intelligibility is assessed on the basis of the Percentage Articulation Loss as discussed by Peutz and Klein, with losses less than 10 percent being considered "GOOD" and losses greater than 15 percent being considered "UNACCEPTABLE".

The computer programme can accommodate a variety of loudspeaker types (i.e. different output levels and directivity patterns) and produces contour plots of direct sound pressure levels and percent articulation losses over the entire seated audience head plane and stage area. The results for the stage can be used to decide whether stage foldback is essential and whether microphone feedback problems are likely to be critical.

The procedure is intended primarily for speech auditoria but the general principles for achieving even coverage apply equally well to buildings where music is also important.

2.0 BASIC OBJECTIVES OF A SOUND REINFORCEMENT SYSTEM

A sound reinforcement system can sometimes be used to improve speech intelligibility in an auditorium with otherwise poor acoustic characteristics; but this should not be viewed as its primary function. Sound reinforcement is intended to provide direct sound coverage at listening positions where a talker's natural voice level would not be high enough above the background noise level to ensure adequate intelligibility.

Depending on the quality and realism demanded for a particular application, it may also be a consideration to ensure that the sound appears to emanate roughly from the position of the talking person.

The level of amplification required is primarily a function of the background noise level. There is little to be gained by operating at an average amplified speech level more than 25 dB above the background level. Assuming a fairly loud background level of 50 dB(A) as might occur in a naturally ventilated school hall, then an average operating level of around 75 dB (A) would normally be quite sufficient. For the presence of a sound reinforcement system to be not too obvious, the amount of reinforcement should be kept below 10 dB above the natural voice level. This is not always practical in large auditoria or where the background level is fairly high (eg. normal voice level at 10 metres is typically only 46 dB(A)).

As will be discussed later the production of a fairly uniform direct sound field over the entire seated audience area is necessary to ensure adequate speech intelligibility, typical acceptable level variations over the entire seating area being $\pm$ 3dB. This is seldom easy to check by measurement of sound levels since the reverberant field usually dominates the direct field by typically some 5 dB. Consequently, measurements of sound levels in an auditorium with a pink noise input to the system usually tend to reflect the evenness of the reverberant field rather than give reliable information on the evenness of the direct field coverage.

The important range of frequencies to be amplified include 300 Hz to 5000 Hz for speech and 30 Hz to 15000 Hz for music. Systems for speech auditoria may be deliberately adjusted or selected so as to have attenuated low and high ends of the frequency range to improve speech intelligibility. Attenuating the low frequency end tends to reduce the volume level (and therefore the masking effect) of the vowel sounds relative to the mid to high frequency consonant sounds which are more critical for intelligibility. Attenuating the high frequency end (particularly in the region of 5000 Hz to 10000 Hz) is sometimes necessary to reduce an accentuation of sibilance sounds.

Not only is an overall even direct field coverage essential throughout the audience area, but also, this coverage should appear even at all frequencies within the speech range (ie. flat frequency response). Incorrect loudspeaker placement and system adjustment can result in the sound levels at certain frequencies being severely attenuated due to destructive interference of the sound waves emanating from multiple loudspeakers. This is usually referred to as a "comb filter" effect.

In addition to the above requirements, a sound system should function at the required normal operating sound levels without being liable to "feedback howl" and the sound should be free of any apparent "echo-like" defects.

3.0 SIGNIFICANT OVERALL SYSTEM FACTORS CAPABLE OF BEING QUANTIFIED

3.1 SPEECH INTELLIGIBILITY REQUIREMENTS

One measure used by researchers to assess speech intelligibility is the parameter: Percentage Articulation Loss (%AL). Peutz (1) and Klein (2) have developed suitable relationships between % AL and characteristics of the listening environment. These have been further modified by Davis et al (3) to take account of the directional characteristics of the loudspeakers and the different absorption of audience and room.

The primary equation becomes:

$$\%\ AL_{25} = \frac{200\ D^2\ RT60^2\ (n + 1)}{VQM} \qquad \ldots\ldots(1)$$

Where $\%AL_{25}$ is percentage articulation loss for a signal to background noise ratio of 25 dB or greater.

D is distance from loudspeakers to farthest listener (m)

RT60 is reverberation time (secs)

V is room volume (m^3)

Q is speaker directivity (typically 500 Hz -> 3 KHz)

M is a modifying factor generally taken as 1.0 but defined as:

$$\frac{1 - \bar{a}_{\text{TOTAL ROOM}}}{1 - \bar{a}_{\text{AUDIENCE}}} \cdot \frac{Q_{\text{ACT}}}{Q_{\text{THEOR}}}$$ when speakers are directed at audience exclusively

n + 1 is total number of loudspeakers contributing reverberant sound to the room

n is the number of loudspeakers not contributing direct sound to the listener

$\bar{a}_{\text{TOTAL ROOM}}$ is average absorption coefficient for entire room

$\bar{a}_{\text{AUDIENCE}}$ is average absorption coefficient for audience area

Q_{ACT} is average directivity over audience area

Q_{THEOR} is the theoretical directivity which would apply if the loudspeaker radiated uniformly only within a solid angle which covered the audience area and equals:

$$\pi / \sin^{-1} (\sin (\theta/2) \cdot \sin (\emptyset/2))$$

θ is angle subtended horizontally at loudspeaker by audience area

$\emptyset$ is angle subtended vertically at loudspeaker by audience area

A secondary equation is used to provide a maximum limit on $\%AL_{25}$ where the reverberant field dominates.

$$\%\ AL_{25}\ (MAX) = 9 \times RT60 \qquad ....(2)$$

This applies for $D > D_L$ where D_L is a limiting distance corresponding to the point where the direct sound field is more than 11.5 dB below the reverberant field. Equation (1) can be expressed in an alternative form involving the difference between the reverberant and direct sound levels.

$$\text{i.e. } \%\ AL_{25} = 0.641\ RT60 \times 10^{\left(\frac{REVSPL\ -\ DIRSPL}{10}\right)} \qquad ....(3)$$

Now equations (1), (2), (3) apply only if the signal to background noise ratio (S/N ratio) is greater than 25 dB. Above 25 dB, no improvement in intelligibility results, but for lesser values of S/N, the %AL increases.

The %AL at lesser S/N ratios where the reverberant field dominates can be calculated from:

$$\%AL = 10^{\left[\left\{ \log_{10} (\%\ AL_{25}) - 2 \right\} \left\{ \frac{S/N + 10}{35} \right\} + 2 \right]} \qquad ....(4)$$

This relationship is based on a 100% AL when the S/N ratio equals -10 dB. This is a plausible assumption when it is considered that the programme level (i.e. signal level) is defined as the average level and some voice peaks may be as much as 10 dB higher than average.

Thus it can be seen that provided the total reinforced sound level (i.e. reverberant plus direct sound) is greater than 25 dB above the background level, then % AL increases as the direct sound level reduces relative to the reverberant level, reaching a maximum value of 9 x RT60 when the direct sound level is more than 11.5 dB below the reverberant level. If the reinforced sound level is less than 25 dB above the background level, then the % AL also increases.

According to Peutz (1) losses less than 10 percent are considered "GOOD", and losses greater than 15 percent are considered "UNACCEPTABLE". American experience has shown that if an empty auditorium is designed to have less than 15% articulation loss, then the articulation loss is usually less than 10% when the auditorium has a full audience.

3.2 ECHO-LIKE DEFECTS

Perceived echoes in auditoria are normally due to reflections off some acoustically hard surface within the auditorium. One possible problem area in a large auditorium with an acoustically hard rear wall is the front few rows of seats. Here, the direct sound travels the shortest distance to any of the seats whereas reflections off the back wall travel the longest. If the difference in transit time is sufficiently long and the level of the reflection is high relative to the direct sound, this will be perceived as an echo. Consequently, loudspeakers should not be aimed directly toward acoustically hard distant walls if echoes are to be avoided.

Another poor situation can occur in a large auditorium of exceptional width where even sound coverage is achieved by spatially separating the loudspeakers across the width of the hall (see Figure 1). In this instance it is possible to have a significant difference in transit times from two loudspeakers to a listener position immediately in front of one of the loudspeakers. Consequently, the contribution from the second more distant loudspeaker may appear more like an echo than a reinforcing contribution to the dominant level component from the nearest loudspeaker. Figure 2 shows indicative levels of reflected sound at various delay times with respect to the direct sound which may be perceived as an echo if exceeded. These data are attributable to Doak and Bolt and represent 10% annoyance.

Where loudspeakers are provided along the length of an auditorium, electronic delays should automatically be incorporated so that the sound from the frontmost loudspeakers generally arrives first. If this is done with the delay time increased an additional 18-25msec over and above the correction necessary to compensate for the path difference, then according to the Haas effect, the sound from all loudspeakers will be perceived as coming from the front of the auditorium rather than from the loudspeakers located further to the rear. Provided the delays are correctly determined, the contributions from all loudspeakers will constructively reinforce each other without any adverse "echo-like" effects.

3.3 LOUDSPEAKER DIRECTIVITY CHARACTERISTICS

Most loudspeakers have directionality characteristics such that at a given distance from the loudspeaker the sound level does not change dramatically with angular variation in the horizontal plane. However, in the vertical plane, dramatic changes in sound level can occur for relatively small angular changes depending on the loudspeaker being considered. Figure 3 shows typical directively patterns for two loudspeakers of different manufacture, both having 100 watt capacity, and both regularly used in general purpose auditoria of schools.

Figure 3 has been developed from manufacturers' test data by averaging results (on an energy basis, not arithmetically) for 500 Hz, 1000 Hz, 2000 Hz and 4000 Hz. It can be seen that loudspeaker A has a total vertical dispersion angle (at 6 dB down) of approximately 60 degrees (ie. $\pm$ 30 degrees) whereas loudspeaker B has a total vertical dispersion angle of 100 degrees (ie. $\pm$ 50 degrees). Surprisingly, these two loudspeakers have a similar mean directivity factor on axis of about 8.5 dB. The reason for this becomes apparent when it is observed that loudspeaker B exhibits a greater variation in directivity with changing angle in the horizontal plane than loudspeaker A.

Where test data at various frequencies are limited and where only approximate calculations are to be made, American experience has shown that use of the 2000 Hz data gives a good indication as to overall behaviour. This is due to the fact that the 2000 Hz octave band includes the major contribution of any octave band to overall intelligibility.

Unfortunately, manufacturers tend only to present directivity information for vertical and horizontal planes passing through the main axis of the speaker. For sound propagation directions other than in these planes, it is necessary to obtain a representative value by interpolation. The complete directional bahaviour of a loudspeaker is not uniquely defined by the standard test data in the horizontal and vertical planes and any interpolation relationship used must of necessity be approximate. A reasonable relationship which matches the limiting values given by manufacturers for the horizontal and vertical planes is given by:

$$DB(\emptyset) = DBVER \cdot Sin^2 \emptyset + DBHOR \cdot Cos^2 \emptyset$$

Where DB(Ø) is required directivity relative to on-axis value of 0 dB

DBVER is manufacturer's directivity relative to on-axis value of 0 dB for vertical plane at angle θ

DBHOR is manufacturer's directivity relative to on-axis value of 0 dB for horizontal plane at angle θ

Ø is the angle obtained by projecting the receptor point onto a plane normal to main axis of loudspeaker (refer Figure 4)

θ is the angle defined by the intersection of either the horizontal or vertical plane through the main loudspeaker axis; the spherical surface through the receptor point; and a plane through the receptor point normal to the main loudspeaker axis (refer Figure 4)

3.4 FEEDBACK LIMITATIONS ON TALKER TO MICROPHONE DISTANCE

"Feedback howl" occurs when the overall acoustic gain exceeds unity. That is to say, the amplification system "runs wild" when it is capable of generating a noise level at the microphone which exceeds the source noise level being amplified. If the amplified noise level at the microphone is dominated by the direct field, then use of a microphone with a special directional characteristic (eg. unidirectional, cardiod etc) may be of assistance. However, with these microphones, "feedback howl" may simply be initiated at a lower frequency than previously. This is due to the fact that lower frequencies bend around corners (diffract) with much less attenuation than higher frequencies. If the reverberant field dominates, certainly the problem may not readily be improved by a change of microphone. Another factor is the reflection off the talker's face as he moves about. This can increase the overall acoustic gain and bring it closer to unity.

To avoid "feedback howl" in a system which has been equalized it is recommended that the average amplified programme level be kept at least 6 dB below the voice level at the microphone. For an unequalized system, the margin needs to be 12 dB. An average voice level varies with distance from the talker approximately as:

$$\text{Voice level (dB (A))} = 66 - 20 \log_{10} [\text{distance (metres)}]$$

Consequently, the maximum distance an average talker can stand back from a microphone and still have his voice amplified to the target programme level (typically "background + 25 dB(A)") can be calculated as:

$$\text{Maximum distance from microphone (metres)} = 10^{\left[\frac{60 - \text{Programme Level dB(A)}}{20}\right]}$$

If more than one microphone is switched on simultaneously, the system gain must be reduced by a factor 10 $\log_{10}$ (No. of microphones), to prevent "feedback howl". The maximum distance from talker to microphone must therefore be reduced accordingly.

4.0 LOUDSPEAKER PLACEMENT AND AIMING

4.1 CHOICE OF LOUDSPEAKER CONFIGURATION

For maximum realism, loudspeakers should be placed as near as possible to the source of sound. For a basically rectangular or fan shaped auditorium the options usually resolve themselves into:

(a) Louspeakers located on either side of the stage proscenium with each loudspeaker generally directed to the diagonally opposite corner of the auditorium (refer Figure 5A-1). If the auditorium is quite long, additional loudspeakers may be mounted off the side walls (refer Figure 5A-2).

(b) Loudspeakers located on the centre line of the auditorium at ceiling level, in front of and above the stage proscenium (refer Figure 5B). If the auditorium is quite long, additional loudspeakers may be mounted at ceiling level on the centre line of the auditorium at various positions along the length of the auditorium.

(c) Distributed array of ceiling mounted loudspeakers aimed vertically downwards (refer Figure 5C).

For a stereo system, Arrangement (a) with loudspeakers located either side of the stage proscenium and off the side walls is the most appropriate. While this arrangement can also be used quite effectively with a mono sound system, it is not the recommended approach for a variety of reasons. The most common objection is that phase cancellation can occur at different frequencies for different seat positions, thereby producing a very non-uniform frequency response at most seats (ie. comb filter effect). This is a very valid objection, but can be alleviated to a large extent if the loudspeakers on either side of the auditorium have slightly different gains (a 3 dB difference produces a remarkable reduction in the comb filter effect while a 6 dB difference may produce an even greater improvement - some experimentation may be necessary on site to optimize). It has been the experience of this author that a significant problem can also be the achieving of adequate sound coverage in the zone immediately in front of the stage (shown shaded in Figure 5A-2) if the frontmost speakers are located too close to the side walls rather than immediately to the sides of the stage. The choice of this loudspeaker arrangement may be favoured for auditoria where side walls are highly reflective, the ceiling to floor depth is large and the reverberation time is high (eg. greater than 2 seconds). In such cases it can be desirable to place the loudspeakers as close as possible to the audience so as to maximize the direct sound field level while contributing as little as possible to the overall reverberant room level. However, with Arrangement (a), there may be a tendency for listeners in seats nearest the side walls to be more aware of the sound as coming from the nearest loudspeaker rather than from the actual speaking person.

Arrangement (b) is to be generally preferred for most auditoria since the direct field coverage of the audience area tends to be more uniform. For an auditorium with a large ceiling height, acoustically reflective side walls and long reverberation time, the arrangement may be less advantageous than Arrangement (a) if it results in substantially more sound energy contributing to the overall reverberant room level. A major advantage of Arrangement (b) is that for all seat locations the result appears more realistic, and sound is more likely to be perceived as emanating from the actual talking person rather than from the loudspeakers. This will only be so, of course, if the loudspeaker to listener distance is greater than the distance from the actual speaking person to the listener. The other significant advantage with Arrangement (b) is that phase cancellation or "comb filter" effects which would otherwise adversely affect the frequency response at different seats within the audience area can be eliminated if the loudspeakers are correctly adjusted electrically. This adjustment is termed alignment and involves the insertion of delays in the order of microseconds to compensate for errors in physical alignment of speakers forming part of a cluster.

Arrangement (c) with a distributed array of ceiling mounted speakers generally lacks realism, because of the different sound path distances between listener to the actual speaking person and listener to nearest loudspeaker. Because most of the sound is directed primarily onto the audience which is highly absorptive, the contribution to the reverberant field is generally quite small and speech intelligibility is very good. The arrangement may be particularly attractive for highly reverberant auditoria where ceiling height is fairly low and it is more important to ensure that speech is understood than to achieve realism. Some phase cancellation effects would seem to be inevitable with this arrangement.

4.2 TRIAL LOUDSPEAKER POSITIONS AND AIMING POINTS

For either of the arrangements with loudspeakers mounted off the side walls or overhead on the centre line, the initial conceptual design can be based on the individual loudspeakers being positioned and aimed so that the ratio of distances from loudspeaker to farthest point of coverage and loudspeaker to closest point of coverage is less than three (refer Figure 6). This recommendation relies on the assumption that there will be a directivity difference of about 5 dB between the two different travel directions and that this will be offset by a 5 dB reduction in noise level due to the difference in travel distances, thereby resulting in a fairly even direct sound field coverage. The suitability of the initial selection of loudspeaker positions and aiming points can be checked for the actual loudspeakers proposed using a computerized procedure discussed later.

If a distributed array of ceiling mounted speakers is proposed it is essential that sufficient speakers be used. The number of speakers required increases if the ceiling height is lowered or if the total dispersion angle of the speakers (6 dB down point) is reduced. For speakers with 90 degree total dispersion angle, the maximum centre-to-centre distance of speakers should be 2(H-h) metres where "H" is floor to ceiling height and "h" is head height above floor level (usually 1.8 metres for standing audience; 1.2 metres for seated audience). Figure 7 illustrates the basis for positioning such a ceiling mounted speaker array.

In all cases, loudspeakers should be located and aimed so as not to produce excessive levels at microphone positions otherwise "feedback howl" may become a problem. Placement should preferably be such that microphones will be in the reverberant field of the room, not the direct field of any one loudspeaker.

Where loudspeakers are separated laterally to give added coverage, the spacing should be limited to avoid generation of any "echo-like" defects.

5.0 REFINEMENT OF LOUDSPEAKER POSITIONS AND AIMING POINTS

Loudspeaker positions and aiming points can be refined once the direct sound field has been calculated for every audience seat location as well as at representative locations on stage and near microphones. The author has developed a computerized procedure to do this. Basically, the input information required includes:

(a) Number of loudspeakers of a particular type, their sensitivity (dB at 1 metre for 1 watt input), power per speaker, and directional characteristics in the horizontal and vertical plane for each different loudspeaker type.

(b) X, Y, Z co-ordinates for all loudspeakers and their aim points.
(c) Information on the orientation of the vertical axis of the loudspeaker.
(d) Average midfrequency reverberation time for the auditorium.
(e) Room volume and maximum background noise level in dB(A).
(f) Co-ordinates of corners of seated head planes of interest (including on stage and in galleries).
(g) Size of grid mesh defining seat locations at which direct sound field is to be calculated.
(h) X, Y, Z co-ordinates of proposed microphone locations.
(i) X, Y, Z co-ordinates of locations to be checked for possible "echo-like" defects due to spatial separation of loudspeakers.

The output of the computer analysis takes the form of:

(1) Contour plots of direct sound field noise levels on each of the seated head planes nominated.
(2) Contour plots of Percentage Articulation Loss on each of the seated head planes nominated.
(3) Tables of overall noise levels due to direct and reverberant sound fields at each microphone location together with maximum microphone to talker distances to avoid "feedback howl".
(4) Tables of transit times and relative direct sound level contributions from all loudspeakers together with a check for possible "echo-like" defects and recommended delay times to be incorporated as electronic delays.

Manual adjustments to the loudspeaker selections, locations and aiming points can be performed quite easily until the calculated results satisfy the criteria of acceptability outlined previously. Typical results for an optimized general purpose school hall arrangement are shown in Figure 8. As can be seen, the direct sound field in the audience area is within $\pm$ 3dB and the % AL is everywhere considerably better than 10%. The calculations also suggest that the % AL on the stage is significantly greater than in the auditorium seating area and stage foldback speakers would be advisable.

6.0 CONCLUSION

A rational procedure for determining preferred loudspeaker directionality characteristics, placement and aiming points has been developed and incorporated into a computer programme for rapid analysis of a particular design. Manual refinement of the loudspeaker layout can be done very effectively once all the required information relating to evenness of the direct sound field, % AL, potential for "feedback howl" and "echo-like" defects are available. An example result for a particular general purpose school hall shows that evenness of the direct sound field coverage can be obtained to within $\pm$ 3dB over the entire seated audience area and that a % AL of better than 10% is readily achievable for such halls.

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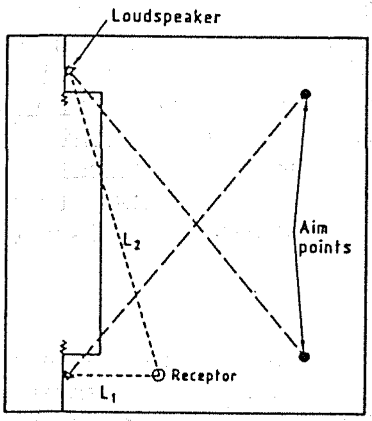

Fig. 1. ILLUSTRATION OF EFFECT OF WIDE HALL ON TRANSIT DISTANCES FOR WIDELY SEPARATED LOUDSPEAKERS. ($L_2 \gg L_1$)

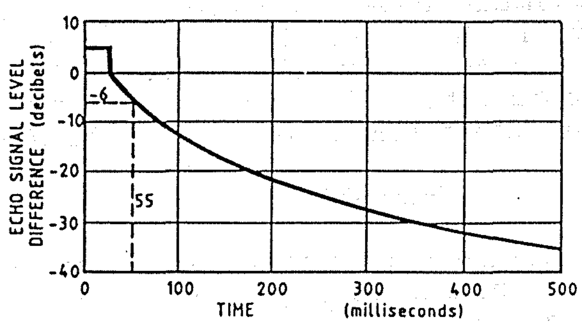

Fig. 2. RELATIONSHIP BETWEEN LEVEL OF REFLECTED SOUND AND TRANSIT TIME TO AVOID ANNOYING "ECHOES".

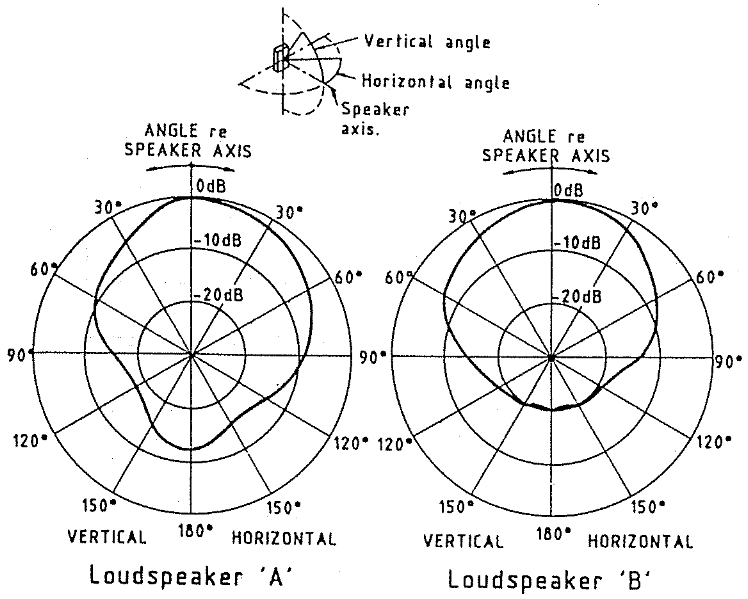

Fig. 3. TYPICAL DIRECTIVITY PATTERNS FOR TWO LOUDSPEAKERS HAVING ABOUT 100 WATTS CAPACITY.

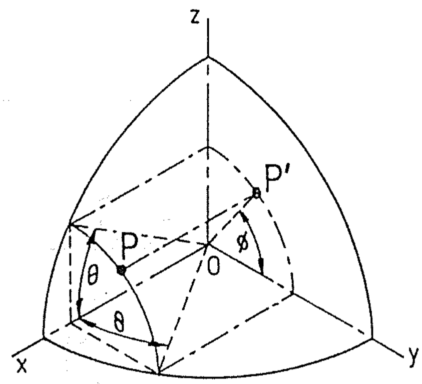

Fig. 4. DEFINITIONS OF ANGLES θ, ϕ FOR LOUDSPEAKER DIRECTIVITY TOWARDS P RELATIVE TO MAIN SPEAKER AXIS.
θ IS AN ANGLE IN THE x-z AND x-y PLANES. P' IS PROJECTION OF P ONTO y-z PLANE AND ϕ IS IN y-z PLANE.
(O-x = Main speaker axis)

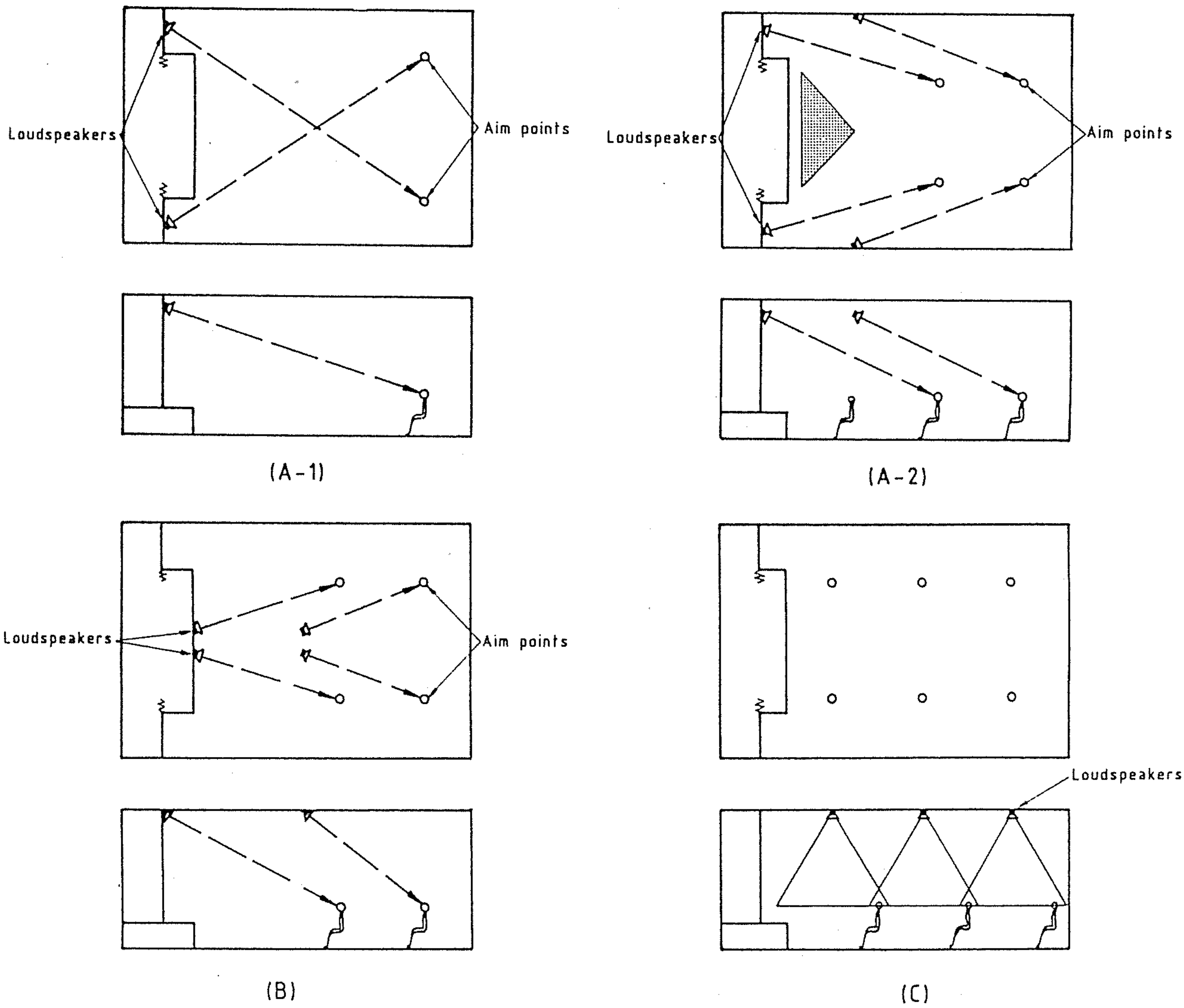

Fig. 5 (A-1). LOUDSPEAKERS ON EITHER SIDE OF STAGE PROSCENIUM
(A-2) LOUDSPEAKERS ON SIDE WALLS OF AUDITORIUM (SHADED AREA MAY HAVE POOR COVERAGE)
(B) OVERHEAD LOUDSPEAKERS ON CENTRELINE OF AUDITORIUM
(C) DISTRIBUTED ARRAY OF CEILING SPEAKERS.

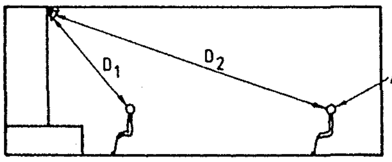

Fig. 6. BASIS FOR INITIAL LOUDSPEAKER POSITIONING ($D_2/D_1 \le 3$)

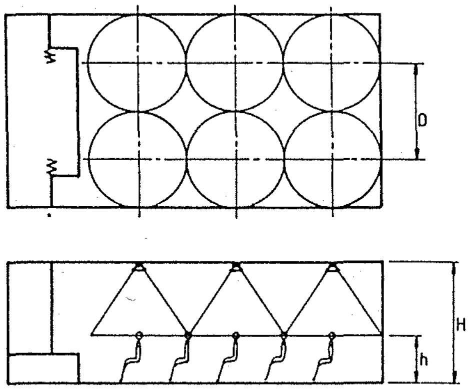

Fig. 7. BASIS FOR LAYOUT OF DISTRIBUTED CEILING SPEAKERS. ($D \le 2(H - h)$

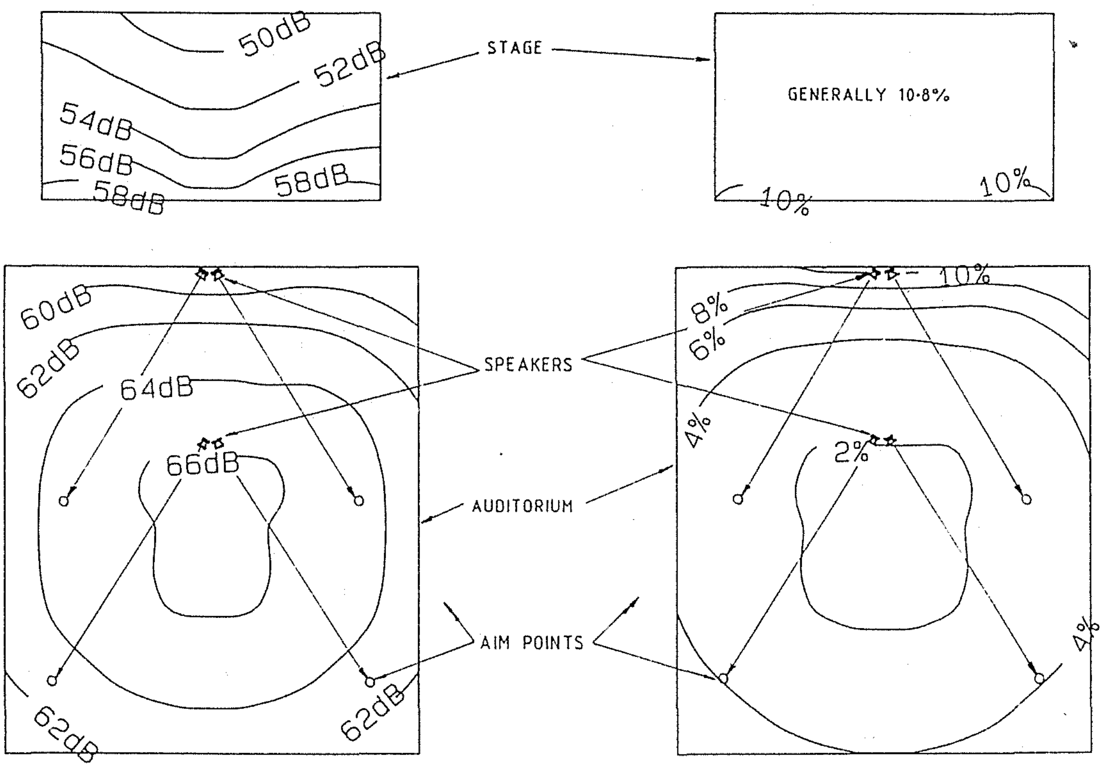

Fig. 8. DIRECT SOUND LEVELS AND PERCENTAGE ARTICULATION LOSSES FOR A TYPICAL HIGH SCHOOL AUDITORIUM WITH 4 × 100 WATT SPEAKERS AND 40dB(A) BACKGROUND LEVEL. (S/N ratio ≥ 25 dB)

BARRIER ATTENUATION OF IMPULSE SOUND

by

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A number of approximate and supposedly exact models for the diffraction of sound around a simple straight-edged barrier have been programmed. A two microphone technique has been used to experimentally determine the effects of barriers on a 1ms impulse, which can be time-isolated from secondary reflections or diffractions. A comparison of impulse waveforms will be used to discuss the effect of changing the geometry or barrier profile, while the relative computer time necessary to calculate the waveforms will be indicated. These results will then be applied to more complex examples, such as castellated diffraction edges and the effect of cracks, which frequently occur in practical situations.

INTRODUCTION:

This work originally sought to determine the relative noise contributions to the sound level behind an extended barrier due to sound passing over and around the barrier, through openings, reflections from buildings and from vegetation adjacent to the barrier. Real-world barriers, such as residential fences, were to be investigated by using an impulse source on one side of the barrier. The differing times required for the sound to reach the receiver would identify the various contributions and hence indicate the relative importance of cracks, buildings and vegetation in a way not possible with a continuous wave source. Attempts to utilize this technique indicated that the signals were complex and not easily decipherable. One problem was the need to know the actual shape of an impulse after being diffracted: hence the following laboratory-based experiments.

THE EXPERIMENTAL SYSTEM:

The impulse source consisted of shotshell primers discharged through a 30mm diam. horizontal plastic tube. An acoustic pulse, of about 1ms duration and reaching 140 dB at 1m, originates from a point source at the mouth of the tube. Because of the conical symmetry of the sound source, the two microphone system described in Fig. 1(a) was used. This allows a comparison of the diffracted and direct waveforms of two, originally identical, impulses which have travelled the same distance. This comparative technique eliminates any effects due to atmospheric absorption or distance considerations. Thus the ratio of the two peak amplitudes leads directly to the excess attenuation caused by the barrier and does not include any attenuation due to inverse square law spreading.

As the experimental measurements were to be made on a reasonably full scale barrier, it was necessary to rotate the

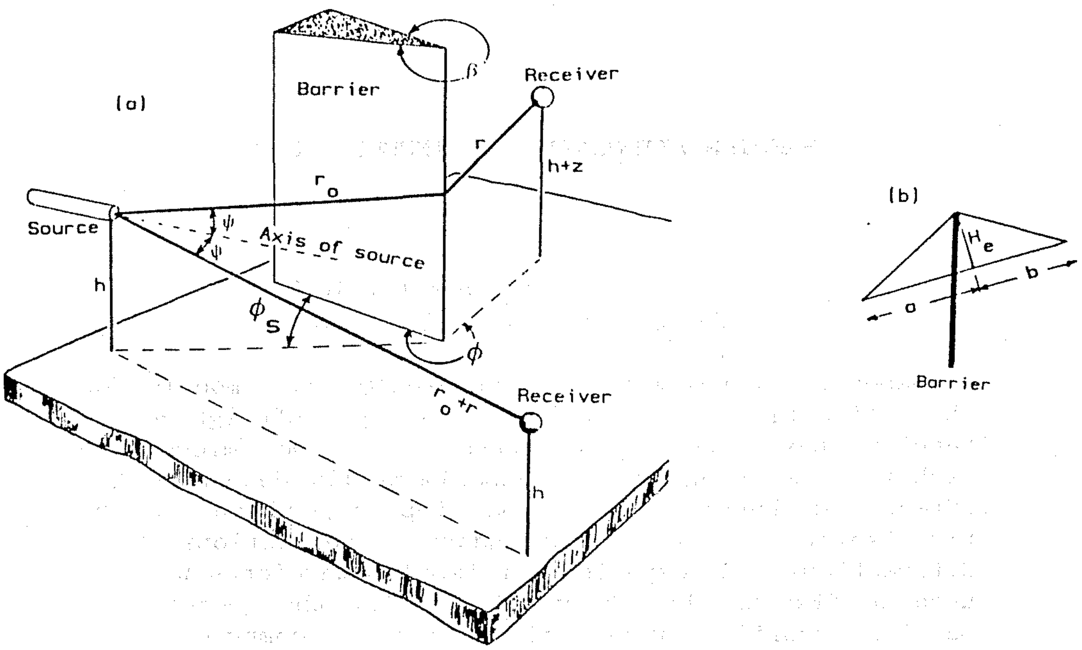

Fig. 1: The measurement geometry.

complete barrier structure through 90°, so that the diffracting edge was vertical. By having the source and microphones near a plane midway between the floor and the ceiling, unwanted reflections were sufficiently delayed to be unimportant, providing the distances from the edge to the source and receiver were less than 2m. Generally, barriers were constructed of 12mm particle board mounted on a rigid backing frame. One exception was the wedge experiment where a metal sheath (2.5mm thick) was constructed to fit over the wooden wedge. The offset between source and receiver, distance z in Fig 1(a), was set to zero for all measurements reported here.

Two sound level meters (B & K, type 2218) with 1/4" microphones were used, the signal being connected directly to a dual-trace digital storage CRO (Tektronix 5223B), permitting both the direct and diffracted impulse waveforms to be captured simultaneously. An interface between the CRO and an Apple IIe computer enabled all captured traces to be stored on a floppy disk for later analysis on a Prime computer.

THEORETICAL MODELLING AND RESULTS:

Prior to discussing the theoretical models, it is advantageous to indicate how single-frequency, continuous-wave theories can be applied to pulse work. A direct, un-diffracted impulse waveform is separated into its individual frequency components by using an FFT algorithm. Typically, waveforms were recorded using a 2μs sampling rate on the CRO, resulting in frequency components in 244Hz intervals. Each of these components is then multiplied by a diffraction factor, as determined by the appropriate theoretical model. The modified components are then re-constituted using an inverse FFT to produce the theoretical diffracted impulse. Note that this requires diffraction factors to be calculated for each frequency component and, preferably, these should be complex as phase is important in determining the final pulse shape.

Diffraction by a straight edge is described, simplistically, by Maekawa[1] using Kirchhoff-Fresnel diffraction theory. At a wavelength, λ, a real diffraction factor for a semi-infinite screen is given by:

$$DF = \frac{1}{2}\left[\left\{\frac{1}{2} - C_{(V)}\right\}^2 + \left\{\frac{1}{2} - S_{(V)}\right\}^2\right] \qquad (1)$$

where $C_{(v)}$ and $S_{(v)}$ are the Fresnel integrals for the variable v, defined through Fig.1(b) as:

$$v = H_e \sqrt{\frac{2}{\lambda}\left[\frac{1}{a} + \frac{1}{b}\right]} \qquad (2)$$

Experimentally, impulses incident upon a straight edged barrier at 90°, and diffracted through an angle of 10° or 40° result in the waveforms shown in Fig.2(a) and (b), where they are compared with predictions from the Kirchhoff-Fresnel theory. The peak value is under-predicted, largely because the higher frequencies are not faithfully reproduced, as evidenced by the premature rounding of the leading edge.

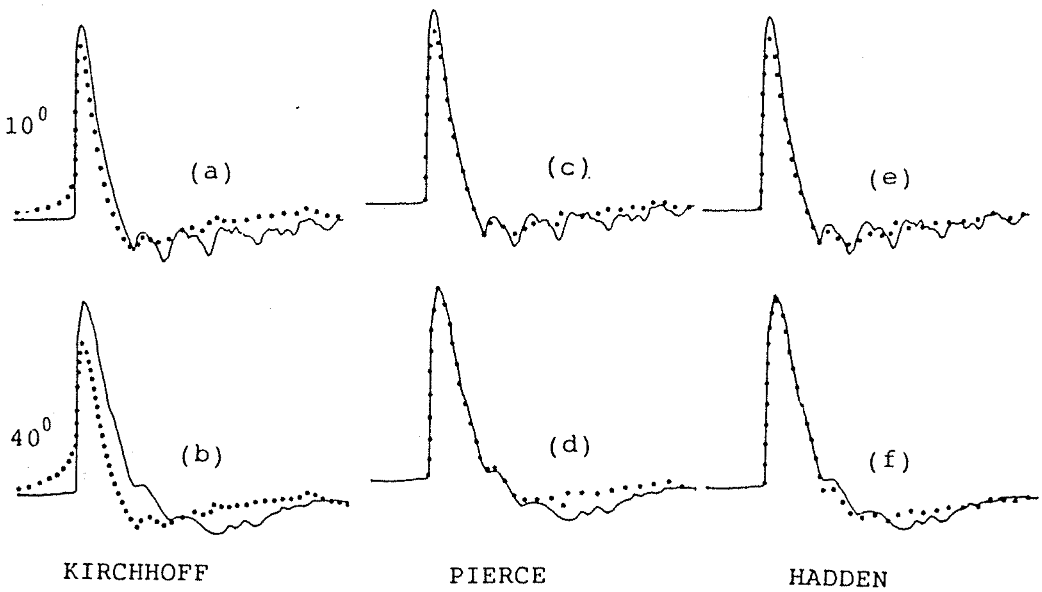

Fig.2: Experimental (solid) and theoretical (dotted) waveforms for the geometry in Fig.4.

A more exact approach, originally due to Oberhettinger[2] and used by Ambaud & Bergassoli[3] and Pierce[4] also includes diffraction by wedges. For any wavenumber, k, the diffraction factor is defined by:

$$DF = -\frac{1}{2\beta}\int_0^\infty \frac{e^{-ikR}}{R} \sum_{n=1}^{4} \frac{\sin \nu x_n}{\cosh \nu s - \cos \nu x_n}\, ds \qquad (3)$$

where x_n represents the four combinations of $\pi \pm \phi \pm \phi_s$, R is an effective path length and $\nu = \pi/\beta$ is the wedge index. The integral is performed along the length of the barrier.

To perform the numerical integration, 6-point Gaussian Quadrature was used, with finite limits replacing the theoretically infinite ones. Because of the highly oscillatory nature of the integrand it is necessary to perform the integration in a step-wise nature with an interval size of 0.01 steps from 0 to 20. The summation technique is a very time-consuming procedure, as it is necessary to evaluate the integral for all the significant frequency components in the original pulse: typically 250 times. As shown in Fig.2(c) and (d), the resulting pulse shapes are in close agreement with those observed experimentally.

The lengthy computational time can be reduced by using a reformulation of the theory by Hadden and Pierce[5]. This avoids difficulties caused by the infinite integration limits, the oscillatory nature of the integrand and its attendant slow convergence, especially at small angles of θ. The diffraction factors are:

$$DF = -\frac{1}{\pi} \sum_{n=1}^{4} A(\zeta_n) \left[\frac{e^{ikL}}{L} \right] F_\nu \tag{4}$$

where $A(\zeta_n)$ is a function[5] of the angles β, ϕ and ϕ_s, L is a path length and the term F_ν is given by:

$$F_\nu = \int_0^\infty e^{-y} \, q \left[\frac{k\,L}{kL + iy} \right] \left[1 + \frac{i}{(kL + iy)} \right] dy \tag{5}$$

where $q = \frac{1}{|A|} \tan^{-1}\left[\tan |A| \tanh \nu X \right]$,

with $\sinh X = \left[\frac{yi}{2\alpha} - \frac{y^2}{4\alpha kL} \right]^{1/2}$ and $\alpha = krr_0/L$.

This expression is of the standard Gauss-Laguerre form and may be evaluated numerically by replacing the integration variable, y, with successive Laguerre roots. The integral is dominated by the e^{-y} factor, is non-oscillatory and has a magnitude bounded by unity.

INTEGRATION METHOD USED.	APPROX. TIME REQUIRED.	AGREEMENT WITH EXPERIMENT.	PEAK VALUE (dB)			
			at 10°		at 40°	
			OBS.	CALC.	OBS.	CALC.
KIRCHHOFF	2-3 min	POOR	9.8	11.2	17.6	19.6
PIERCE	0.5 hrs	GOOD	9.8	10.6	17.6	17.5
HADDEN	1 min	GOOD	9.8	10.8	17.6	17.9

As indicated in the table, the reformulation reduces the calculation time by a factor of 30 without a significant reduction in agreement with the observed results, as is apparent in Fig.2(e) and (f). The effect of changing the angle of diffraction on the peak value is plotted in Fig.3, where it is apparent that the modified formulation of Hadden and Pierce produces results that follow closely the observed attenuation curves. The Kirchhoff-Fresnel method, however, over-estimates the observed attenuation by as much as 3dB.

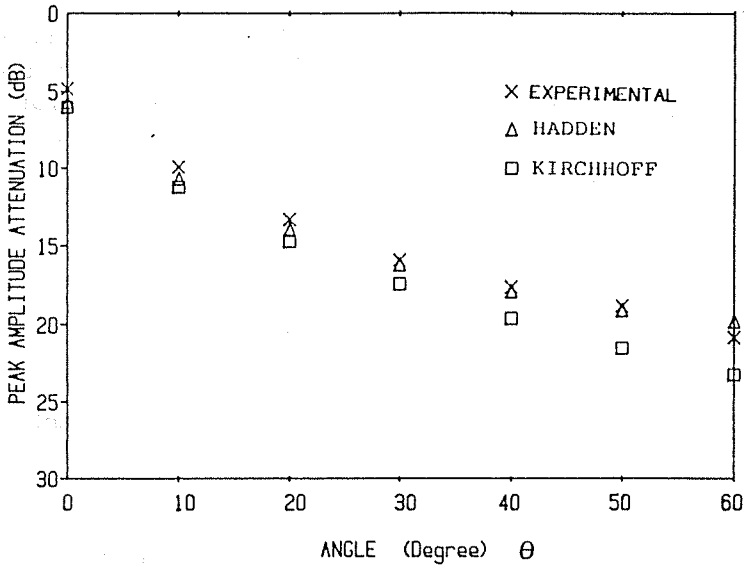

Fig.3: Comparison of peak values as a function of angle θ defined in Fig.4.

To further illustrate the agreement between theory and experiment, the excess attenuations experienced by different frequency components in the diffracted pulse are compared with theoretical predictions in Fig.4. Although the general trend is correctly predicted there is a tendency for the theory to overpredict the attenuation at high frequencies.

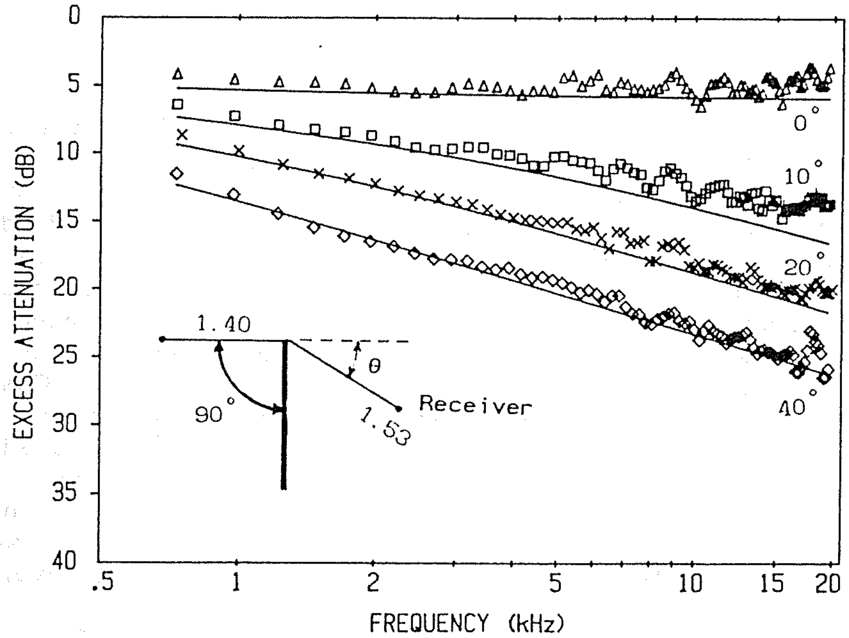

Fig.4: Frequency dependance of excess attenuation compared with prediction of Hadden and Pierce.

Wedged barriers have also been investigated. The agreement between experiment and theory is similar to that obtained for thin half-planes.

An extension of the work involved wide, flat-topped barriers, using a similar measuring technique to that employed for single-edge diffraction. Levels predicted by a theory devised by Pierce[6] for double diffraction are in close agreement with overall observed attenuations as a function of frequency, as shown in Fig.5. This theory does not allow the accurate prediction of the pulse-shape under double diffraction conditions as there is no phase information.

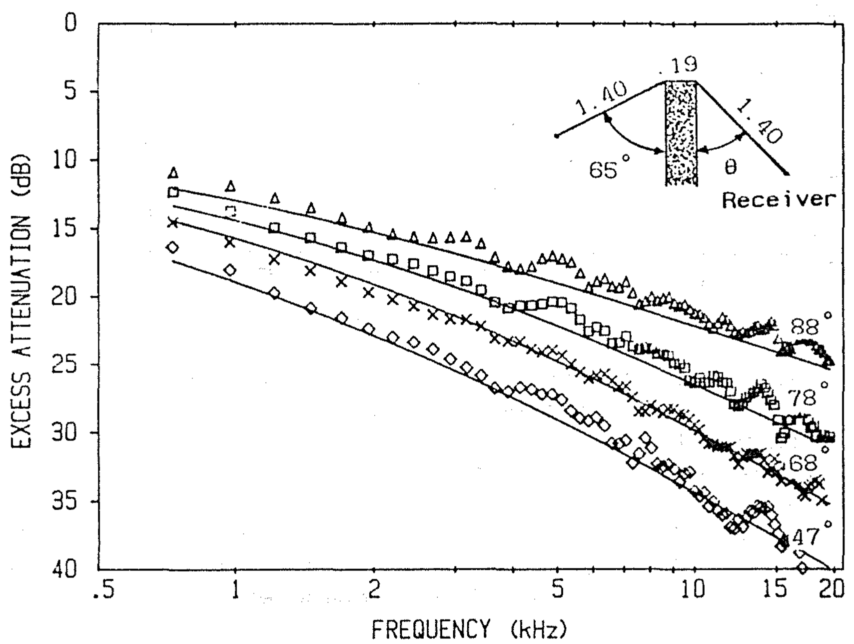

Fig.5: Results for a thick barrier compared with predictions of Pierce.

As part of these studies of standard barrier geometries, two closely related problems have been investigated.

Often normal suburban paling fences can, simplistically, be considered as a thin half-plane, topped by a castellated edge. To investigate whether or not a prediction technique could be devised to model the diffraction over such an edge, the number of castellations were minimized while their size was exaggerated. In one experiment, the problem was simplified to a 2-slot castellation, where each slot was 65mm wide and 65mm deep, Fig.6(a). The resulting pulse shape is shown in Fig.6(b), along with the results of a simplistic attempt to model the situation (dotted curve). Two separate diffractions are assumed to take place, one from the the top of the castellations and the other from the bottom. After applying an appropriate time-shift to allow for the different path lengths, two diffracted waveforms of half the original amplitude have been added together to try and duplicate the measured waveform. Experimentally, the first peak is narrower than predicted, possibly because of diffraction from the sides of the slots. Measured waveforms for 5 and 20 slot situations are also presented in Fig.6.

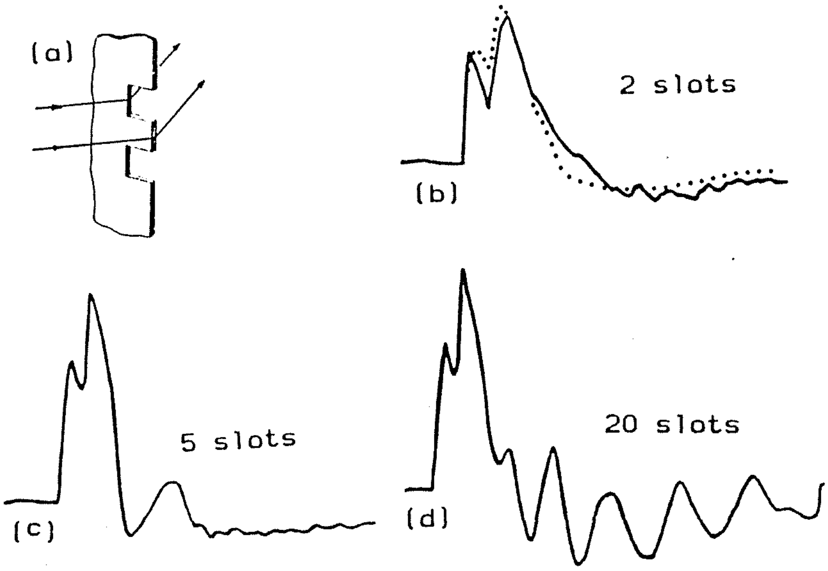

Fig.6: Geometry and waveforms from castellated diffraction.

The other area being investigated is that of leakage due to cracks. As shown in Fig.7, a small "pre-pulse" occurs prior to the beginning of the main diffracted impulse due to leakage of the sound through the join between the two pieces of timber. Trials undertaken using a variety of controlled crack widths, indicate that a close relationship exists between the size (either lateral or longitudinal) of the crack and the area of the pre-pulse created. It is anticipated that this technique will permit an accurate estimate of the contribution to the sound level behind a barrier arising from a crack.

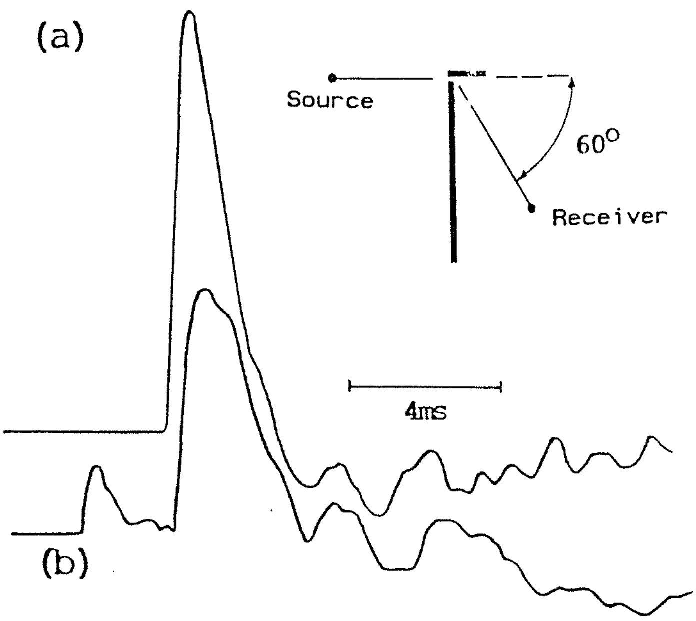

Fig.7: Effect of a crack: (a) diffraction around sealed corner, (b) is with crack present, showing pre-pulse.

CONCLUSION:

Impulses provide a rigorous test of theoretical diffraction models and the measurements are in reasonable agreement with existing models. Difficulties still exist in predicting correct attenuation levels and pulse shapes at high reception angles, a matter currently under review.

Castellated tops and the effects of cracks in barriers are both topics with practical implications. These areas are currently being pursued in an attempt to quantify the effects.

ACKNOWLEDGEMENTS:

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ECONOMIC EVALUATION OF NOISE REDUCTION TREATMENTS

A. PREAMBLE

In recent years, the community has become more circumspect regarding environmentally beneficial projects which are detrimental in other areas, such as employment. This appears to have coincided with the perception that Australia may be not so well off economically, as it once thought.

Increasingly and of necessity, governments require that expenditures, including those incurred on noise abatement treatments, should be justifiable on a benefit/cost basis, as occurs in the US (Ref 3).

The cost of meeting environmental criteria is not a trivial matter. Below are some quotes from a paper which addresses this issue, presented two years ago in Toowoomba, namely "The Effect of Buffer Zones Around a Coal Mine." by Allan Lawson & Colin Tickell (Ref 1):-

(a) "The intention of buffer zones around Saxonvale Mine ... and other coal mines ... is to physically separate their associated potential impacts from the surrounding community. Such effects as dust fallout, water quality, ground vibratio & noise.(4.1,#1)....."Some mines have spent 10-15% of their capital budget on buffer zones..the point has been reached where the mining companies have to employ farmers and agricultural consultants." (4.1,#8).

(b) "Costs of noise reduction engineering to mining equipment ma be around $20,000 per machine" eg to reduce haul trucks from 90-95dBA at 15m to 87dBA(4.1,#6;5.0,#5,6).

(c) The predominant sentiment of the paper was one of concern that the cost of some environmental requirements (and particularly the acoustical requirements) was much greater than the anticipated benefits, given that the surrounding area was "essentially agricultural, with cattle grazing, vineyards,dairies and hobby farms. Occupiers of hobby farms had heightened environmental awareness and consciousness of their "rights"." (2.0,#5,6)

Since that paper was written, economic conditions re coal export and in the local industry have deteriorated markedly. With the benefit of 20/20 hindsight, and given the highly competitive coal export market, it can be seen that the avoidable capital costs of noise control, at Saxonvale and possibly at other mines, together with a likely multiplier effect, (due to consequent lost turnover) may have contributed significantly to the problems in this industry, which in turn, have had a significant economic effect on the whole community.

The Electrical Power Industry in NSW is another industry whose financial standing has caused recent concern, where noise treatments costing millions of dollars have been required, sometimes where it is understood, only a few houses were affected. My employer, the DMR in NSW has found similar with respect to construction noise from freeway sites in rural areas.

B. THE NEED FOR BENEFIT/COST ANALYSIS OF NOISE TREATMENTS

At present warrants for noise mitigation treatments are normally based on criterion levels. In N.S.W., both my employer, the

D.M.R. and the S.P.C.C. have followed this practice. However this does not take into account vital and elementary factors such as the number of affected properties. Much of the resistance to the requirements of environmental authorities is due to concern along the lines that potential expenditures on noise treatments would achieve greater community benefits in other areas such as, for the DMR, to reduce traffic delays or to improve road safety, ie to improve quality of life, reduce air pollution, preserve fossil fuel and to save human lives.

An accepted method of quantifying the benefits of a noise treatment would have the following benefits:-

(a) Treatments which are not worthwhile, will be rejected.
(b) For noise treatments, whose likely benefits exceed the estimated cost, decision makers would be assured that expenditures were achieving worthwhile benefits. allaying resistance to expenditure on these projects.
(c) The funds which are expended on noise treatments will be generally directed to achieve the greatest benefit.
(d) Borderline anomalies will be minimised.

C. PURPOSE OF PAPER

The purpose of this paper is as follows:-

(a) To summarize present knowledge in this area.
(b) To describe a methodology for estimating the benefit/cost ratio of a noise mitigation treatment, suitable for use by a sub-professional designer. The model will be capable of rejecting projects which are not worthwhile, and also of indicating the most economic degree of treatment.
(c) Whilst being presented in the context of traffic noise, the principles are of more general application.

D. THE ECONOMIC VALUE OF A NOISE LEVEL VARIATION

This has been the principal area of difficulty. What is the value of a noise level rise or fall?

The present position of knowledge worldwide is well summarized by Ariel Alexander & Jean-Philippe Barde in Ref 2. Their principal conclusions are:-

(a) That the hedonic prices method is the most widely used for evaluating the social cost of noise. (By this method, multiple regression analysis of property prices is used to evaluate the value the market place places on various attributes of properties such as a third bedroom,a swimming pool etc.)
(b) "As a rule of thumb, 0.5% of house value depreciation per decibel constitutes a reasonable guide and is based upon a substantial number of different studies" applicable above say 50dBA (Leq,24hrs).
(c) "It might also be that the unit percentage of depreciation increases with (i) the noise level and (ii) the value of the house. These points merit further consideration."

My own reading of several relevant references, (nb Refs 8, 12, & 13) has lead to a similar conclusion to (c), that several studies indicate evidence of (i) but that there may be insufficient data to be quantitative with confidence. Significantly, Holsman & Bradley (Ref 8, p90) find a three times greater value of G (See Appendix B) along main roads than along nearby local streets and cite Authors 11, 13 and C.J. Langley as having reported similarly. Conclusion(ii) is supported by A.A.Walters in Ref 4.

To further discussion in this area, a simple non-linear relationship is proposed in this paper, however a linear model such as (b) above could also be used to estimate economic benefits.

E. THE BASIS OF A PROPOSED NON-LINEAR RELATIONSHIP BETWEEN REAL ESTATE VALUES AND TRAFFIC NOISE LEVELS (IN DBA)

To date evaluations of the value of a variation due to the traffic noise level at a property has been based on percent of property value per DECIBEL rise or fall. An alternative unit is the SONE, or a similar unit, the NOY.

One sone is the perceived loudness of 40dB at 1000Hz. Two sones is perceived as being "twice as loud" and occurs at 49dB, four sones at 58dB etc. Earlier versions doubled the number of sones for each 10dB rise.

The noy unit was developed to assess perceived annoyance, particularly of aircraft noise. Again, one noy occurs at 40dB at 1000Hz,but with 2noy at 50dB, 4 noy at 60dB etc.

The logic of both these units is that if there is a sound, noise, or noise problem of magnitude 1.0 at 40dB, it will be of approximate magnitude 2.0 at 50dB, and of 4.0 at 60dB etc, other factors being equal.

A most significant difference between the dBA and the sone or noy is that whereas the dBA is a logarithmic expression of noise intensity, the SONE and the NOY are SUBJECTIVE, ie they are based on HUMAN REACTIONS to various noise levels. For this reason, Eberhard Zwicker argues for the general use of sones, in preference to decibels in Ref 5, on the basis that the "man in the street" will appreciate twice as many noise units being associated with a noise which is heard as being twice as loud.

This concept is also supported by von Gierke in Ref 6 in which the following weighting curve is stated as being widely accepted for environmental evaluation within the US EPA.:-

$$W = 5.37^{((L-65.4)/10)}/(1 + 10^{((L-62.9)/20)})$$

This curve is tabulated in the table below, is similarly non-linear to the sone or noy described below.

Ldn.	40	45	50	55	60	65	70	75	80	85
W.	.013	.029	.061	.124	.235	.412	.666	1.00	1.43	1.96
5dBA incr.	.016	.033	.063	.111	.177	.248	.334	.43	.53	
10dB growth (M)*			4.0*	3.4	2.8	2.2	1.9	1.7	1.6	

* Growth over 10dBA in the effect of 1.0dBA variation.

It is commonly accepted that a ten-fold increase in the number of vehicles will increase noise levels by 10dBA. By the above logic, this also represents a doubling of the loudness, and of the noise problem, for each tenfold increase of the traffic volume. I invite you all to consider, subjectively whether this is reasonable. In Appendix A, noise level averaging based on sones is compared to other methods.

Use of the sone for economic analysis, in preference to decibels, will give relatively greater weight to properties subjected to high levels of traffic noise and less weight to those at moderate levels, given the same decibel reduction. The principle of diminishing returns would apply as the noise level falls, whether due to distance or to an increasing level of treatment.

Let me give an important example where this approach could apply. A draft standard is now being circulated dealing with "Building Siting and Construction" in relation to traffic noise, which if applied rigorously, would require all house construction or modification within 700m and visible from a major arterial road to be evaluated by noise measurement and analysis. Other criteria apply to lesser arterial roads, all based on 55dBA (L10,18hrs). Most houses in Sydney could be affected, and a significant cost burden placed on the public. By considering sones, rather than dB, and benefit/cost rather than criterion levels, attention would be concentrated to houses nearest to a major road. It is my judgement that few properties not in the first two rows of houses from a major road have a significant traffic noise problem from that road.

F. EXACT FORM OF RELATIONSHIP

Generally, there appears to be insufficient data in the various research papers on this topic to develop a new relationship, however an approximate value of the economic value of each sone reduction can be obtained as follows:-

(a) In this paper I will adopt a unit I will call the sone/noy (SN).One SN occurs at 40dBA (L10,18hrs) of traffic noise, 2SN at 50dBA etc. My purpose is to emphasize that this unit is not equal to a sone or a noy, but is similar to both.

(b) The effect of noise is twofold:-
  (i) Its own intrinsic annoyance.
  (ii) Masking of other sounds affecting the "acoustical flavour" of an environment.

  Because of (ii) I have adopted 10dB/SN doubling in preference to 9dB. This will be easier to visualize mentally, and is a slightly lower degree of non-linearity. This should be refined by further research.

(c) Let A be the economic value of a noise level reduction from 70dBA to 65dBA, let B be the same, 10dB lower ie 60dBA to 55dBA, and let M = A/B.Then:-
  (i) Under the model proposed in this paper, M = 2.0, ie A is twice the value of B.
  (ii) Under Para C(b), the "linear" relationship, M = 1.0 thus 5dBA reduction has the same economic value, no matter the initial noise level.
  (iii) For industrial / hearing loss / noise dose type assessments, M = 10 approximately, on the basis that a 10dBA rise will reduce the allowable exposure time to approximately one tenth or increase the noise dose tenfold.
  (iv) For 9dB per doubling of SN, M = 2.2.
  (v) If acoustic pressure were considered to be the determining factor, then M = 3.2.
  (vi) The US EPA curve above puts M at 1.6 - 4.0.

These are listed to demonstrate that doubling of the estimated economic benefits each 10dB rise (M = 2) is moderate when compared with most other areas of acoustic investigation.

(d) The effect of frequency weightings and noise level distributions are deemed to be sufficiently constant as to not significantly affect the gist of this basic logic.

(e) Let us assume that the many studies on this topic considered by Alexander, generally lie between 57dBA and 77dBA (L10,18hrs or Leq,24hrs), averaging 67dBA for the "before" and "after" levels. At 0.5% per dB, this would represent 10.0% of property value.

57dBA and 77dBA represent 3.2SN and 13.0SN respectively (See Appendix A), a difference of 10 SN. On this basis, 1.0SN is approximately equal to 1.0% of property value. This value could be refined by further research, and could vary somewhat with the property value, or the nature of th neighbourhood. "Better" properties may be more sensitive, and "poorer" properties less sensitive according to Walter in Ref 4.

(f) Below 50 dBA, the estimated benefit per decibel becomes quite low (See Fig 1), and Alexandre in Ref 2 concludes that no significant economic effect below this level has been discerned. Nevertheless reversion to a linear relationship, eg 0.1-.15% of PV per dBA may be appropriate below 50dBA.

(g) Derivation of formulae is set out in Appendix B.

G. METHOD OF VERIFICATION OF A SONE TYPE RELATIONSHIP

The present linear relationship requires only one constant for definition (0.5%/dBA), and needed many years of research to be evaluated. Any nonlinear relationship would require at least two defining constants, and could be more difficult to verify with confidence. The following methods are possible:-

(i) Re-analysis of data already obtained. Much of the variability in these studies may be due to sones rather than dB being the principal causative variable.
(ii) Walters describes a method, originally undertaken with misgivings, whereby estate agents and similar were asked to place values on various noise environments. This method was later verified as being most effective
(iii) Similar studies as in the past.

H. OTHER FACTORS

I have avoided some areas of economic modelling, which may need to be taken into account. These include the following:-

(i) That capital expenditure by a Government should return a greater benefit.
(ii) That various weightings may be given to different properties to take account of factors such whether the road project was public knowledge for many years.

In fact, there are many other factors which are major constraints on noise barrier construction including:-

(a) Access may be made more difficult.
(b) An earth mound 2m high is likely to occupy 10m, approximately half the width of a house block. Often there will be more valuable uses for such land.
(c) Loss of sunlight.
(d) Concerns regarding security /concealment
(e) Loss of view, particularly by the motorist. I wish to consider this aspect in some detail, as the roadside scenery is an important part of a road.

J. THE VALUE OF ROADSIDE SCENERY

The roadside scenery is one of many reasons for road travel. To assess its value, let us assume the cost of driving a car is over $20/hr, ($5/hr for petrol, $8/hr for depreciation eg 10c/km at 80kph, $3/hr for repairs, plus provision for the value of the occupants' time).

For a few motorists, the "Sunday motorists", the scenery is the principal purpose for their journey. For "tourists", scenery is an important secondary reason, and for most motorists, scenery adds significantly to the pleasure of their journey.

Whilst "urban scenery" is sometimes not as pretty as rural scenery, it is usually more interesting, and for many is equally attractive, particularly at night.

The value of roadside scenery appears to have been entirely unresearched. In the light of the above, I have put it at $2 per hour. $1 per hour, which is only 60c/hr per occupant appears to be inadequate, but may be comparable with 0.5%/dBA. To give roadside scenery no value is unreasonable in my view.

At 40,000 vehicles per day, 80 kph, and 82% of daylight traffic, this would represent $300 per road metre per annum, or $4000 per road metre at 7.5% discount rate. This is much greater than the cost to construct noise barriers, which is about $120/sq m for opaque barriers.

Clear noise barriers cost $500 - $1000 /sq m and, based on the above, appear to be economic, at least at locations of high scenic interest, where the motorists' view would otherwise be lost. They could attract graffitti.

In Appendix C, I have tried to illustrate the implications of the above along a typical length of new road.

K. CONCLUSIONS

This paper describes a method for modelling a traffic noise barrier, to assess its economic viability. Other principal conclusions are that:-

(1) The value of a decibel rise or fall approximately doubles for each 10dBA rise in the initial noise level.
(2) The value of any loss of the motorists' view should be taken into account, as this is usually much greater than the cost of providing the barrier.

L. DISCLAIMER

The author wishes to thank the Commissioner of Main Roads, NSW for his support in preparing this paper. The views expressed therein are those of the author alone.

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APPENDIX A. COMPARISON OF NOISE AVERAGING PROCEDURES

Noise Levels(dBA)	60 +70.	61 + 70.	60 + 71.	61 + 71.
Arithmetic.	65.00	65.50, .50	65.50, .50	66.00, 1.00
Logarithmic(Leq).	67.41	67.40, .09	68.32, .91	68.41, 1.00
Sone/Noy.	65.85*	66.19, .34	66.52, .67	66.85, 1.00

* S60 = 4.0 S70 = 8.0 Sav = 40 + 10*log((8+4)/2)/log2 = 66

APPENDIX B. DERIVATION OF FORMULAE AND EXAMPLE

Let Lo be the noise level before the project, Lt the noise level due to the project and L1 the total "after" noise level, all in the same units, eg L10(18hrs) or Leq(24hrs).
Then $L1 = 10*\log(10^{(.1Lo)} + 10^{(.1Lt)}) = 4.34*\ln(e^{.23Lo} + e^{.23Lt})$

Let S1 be the number of SN corresponding to L1. Then:-
$S1 = 2^{((L1-40)/10)} = 10^{(.3(L1-40)/10)} = e^{(\ln 2(L1-40)/10)}$

If each sone equals 1% of property value (PV), the value of one decibel reduction in the project noise level is equal to:-

$$
\begin{aligned}
G &= d(S1)/d(Lt) = d(S1)/d(L1) * d(L1)/d(Lt) \\
&= \ln 2/10 * S1 * 4.34 * .23 * e^{.23Lt} / (e^{.23Lt} + e^{.23Lo}) \\
&= \ln 2 * 10^{-1} * 10^{-1.2} * (10^{.1L1})^{.3} * 10^{.1Lt} / 10^{.1L1} \\
&= 10^{-2.36} * 10^{(.03Lt)} / (1 + 10^{.1(Lo-Lt)})^{-.7} \\
G &= 10^{.03(Lt-78.6)} / (1 + 10^{.1(Lo-Lt)})^{.7} \\
&= 2^{.1(Lt-78.6)} / (1 + 10^{.1(Lo-Lt)})^{.7} = Go * F \qquad (3)
\end{aligned}
$$

eg Where Lo is low, G = 1% of PV per dBA at Lt = 78.6dBA., 0.5%/dBA at 68.6dBA, 0.25% at 58.6dBA, 0.2% at 55.3dBA, 0.15% at 51.1dBA and 0.1% of PV per dBA at 45.3dBA.

Tabulated values of these formulae are set out below.
Number of SN at a given noise level, ie $S = 2^{(.1(L-40))}$

dBA	0	+2	+4	+6	+8	+10
30	.5	.57	.66	.76	.87	1.0
40	1.0	1.15	1.3	1.52	1.74	2.0
50	2.0	2.3	2.6	3.0	3.5	4.0
60	4.0	4.6	5.3	6.1	7.0	8.0
70	8.0	9.2	10.6	12.2	13.9	16.0
80	16.0	18.4	21.1	24.3	27.9	32.0

The value of each dBA reduction is as follows:-
Go = d(S1) / d(L1) = d(S1) / d(Lt), for Lo low.
$= 10^{.3(Lt-78.6)} = 2^{.1(Lt-78.6)}$ SN/dBA.

dBA	0	+2	+4	+6	+8	+10
40	.07	.08	.09	.10	.12	.14
50	.14	.16	.18	.21	.24	.28
60	.28	.32	.36	.42	.48	.55
70	.55	.64	.73	.84	.96	1.10
80	1.10	1.28	1.46	1.67	1.92	2.20

Correction Factor (F) for the effect of background noise.
$F = (1 + 10^{.1(Lo-Lt)})^{-.7}$, G = Go * F.

Abs(Lo-Lt)	20	15	10	8	6	4	2	1	0.5	0
Lo<Lt.	.99	.98	.94	.90	.85	.79	.71	.66	.64	.62
Lo>Lt	.04	.09	.19	.25	.33	.42	.52	.57	.59	.62

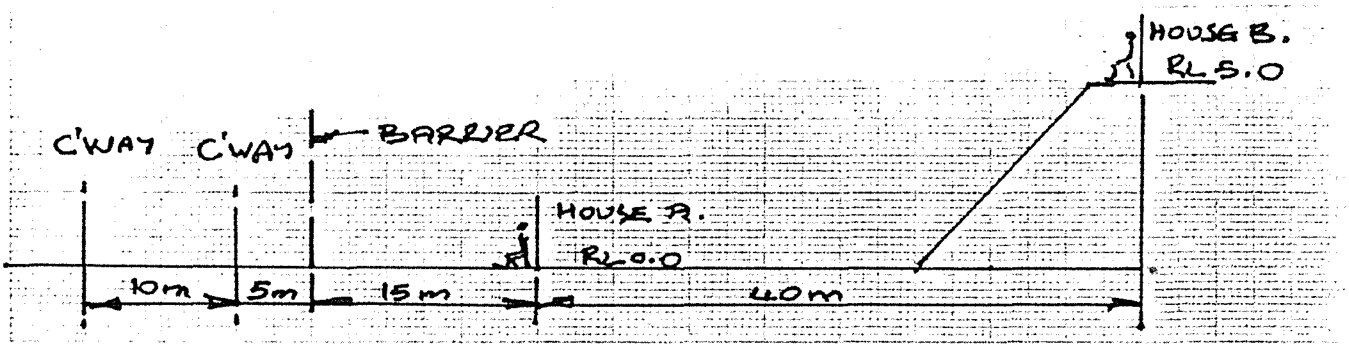
Example. The noise level at a house worth $200,0000 will be increased from 60dBA (Lo) to 70dBA (L_1), the level due to a nearby road project (L_t) having been estimated to be 69.6dBA. Calculate (a) the economic loss, and (b) the value of a 2.3dBA reduction in the traffic noise level.

ANSWER. 67.3dBA (69.6 - 2.3) + 60dBA = 68.0dBA. At 60dBA, there are 4 SN, at 70dBA 8 SN, and at 68dBA 7.0 SN. For (a) the loss is 4SN * 1%/SN * $200K = $8,000.

For (b) 1%/SN * $200K * (1 = 8-7)SN = $2000.
Alternatively, for (b), at L_T = Av of (67.3+69.6) = 68.5
G = Go * F = .49 * .91 = .44.
The benefit B = .44 * 2.3% * $200K = $2000 as above.

APPENDIX C. EXAMPLE.

The site details are as set out in the sketch below, the houses shown being repeated at 20m intervals. All noise units are L10(18hrs) in dBA, and modelling is by the CORTN procedure. There are 20,000 vpd per carriageway, at 80kph, with 10% heavy vehicles, and no gradient. 82% travel during daylight. The background noise level is 50dBA. The house value is $160,000 each. The cost of barrier construction is $160 per sq metre for opaque material, and $800 for clear. 7.5% discount rate applies. Scenery is deemed to lie between 0 and 3m of barrier height. Find the optimal barrier height, based on the value of the roadside scenery being (i) Zero. (ii) $1 /vehicle hour, (iii) $2/hr.



SOLUTION. At $2/hr the disbenefit due to total loss of view is estimated to be:-

D = 40,000vpd *.82 * $2 * 365 / 80,000metres/hr
= $300/road metre/annum = $2000/metre of barrier
= $670 per sq m of barrier = $340/sq m at $1/hr.

Estimated noise levels and reductions in SN levels are:-

BARRIER	NEAR HOUSE.		FAR HOUSE		TOTAL SN	INCR. SN	INCR.	AVERAGE
TOP RL.	Lt	S1	Lt	S1	REDUCT'N.	REDUCT'N.	RED/m.	.RED/m.
-.15	73.7	10.31	69.6	7.52	NA	NA	NA	NA.
.20	73.2	10.00	68.6	7.52	.32	.32	.91	.91
.60	71.3	8.80	68.8	7.47	1.57	1.25	3.10	2.27
1.0	69.9	7.96	67.9	7.05	2.83	1.26	3.1	2.46
1.5	67.5	6.77	66.1	6.24	4.83	2.00	4.0	2.93
2.0	65.8	6.04	64.6	5.67	6.13	1.30	2.6	2.85
2.5	64.4	5.48	63.0	5.06	7.30	1.17	2.3	2.75
3.0	63.1	5.03	61.6	4.65	8.16	.86	1.9	2.59
3.5	62.0	4.67	60.5	4.32	8.85	.69	1.4	2.42
4.0	60.9	4.36	59.4	4.04	9.44	.59	1.2	2.27
4.5	59.9	4.10	58.4	4.82	9.92	.48	1.0	2.13
5.0	59.0*	3.88*	57.6	3.62	10.35	.43	0.9	2.01

* 59.0dBA + 50dBA = 59.5dBA = 3.9 SN.

The value of 1 SN reduction is 1% of $160,000 = $1600 or $80 per sq m, ie each metre height of barrier should achieve 2SN reduction to be economic, if the value of roadside scenery is zero. The most economic height becomes 2.5m. If the houses were replaced with town houses at 10m average spacing, and of similar value, the optimal height would rise to 4.5m.

However if account is taken of the value of the scenery, then any barrier becomes quite uneconomic.

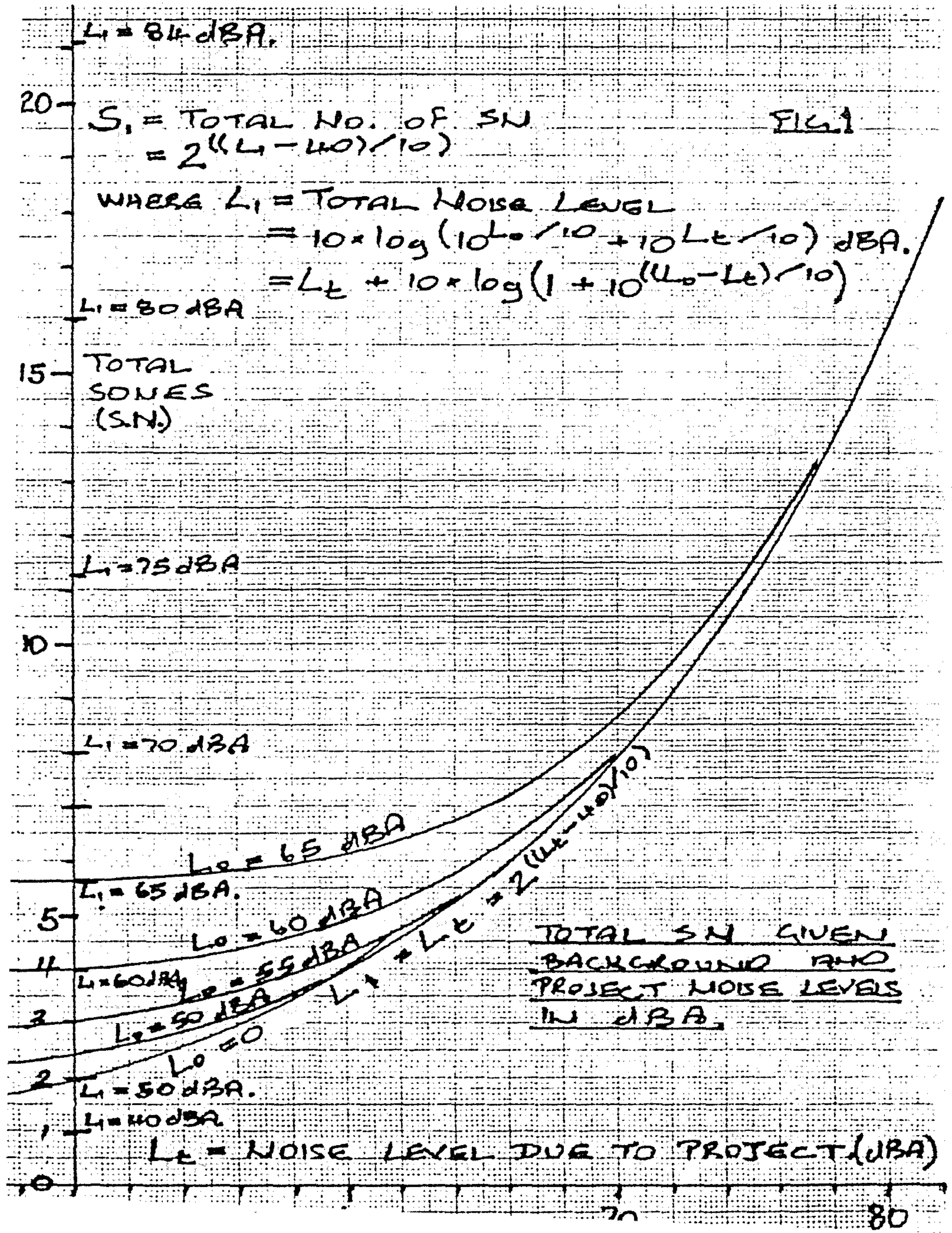


FIG 1
S_1 = TOTAL NO. OF SN
$= 2^{((L_1-40)/10)}$
WHERE L_1 = TOTAL NOISE LEVEL
$= 10 \times \log(10^{L_o/10} + 10^{L_t/10})$ dBA.
$= L_t + 10 \times \log(1 + 10^{((L_o-L_t)/10)})$
TOTAL SONES (SN)
L_1 = 84 dBA.
L_1 = 80 dBA
L_1 = 75 dBA
L_1 = 70 dBA
L_1 = 65 dBA.
L_1 = 60 dBA
L_1 = 50 dBA.
L_1 = 40 dBA
L_o = 65 dBA
L_o = 60 dBA
L_o = 55 dBA
L_o = 50 dBA
$L_o = 0$
$L_1 = L_t = 2^{((L_t-40)/10)}$
TOTAL SN GIVEN BACKGROUND AND PROJECT NOISE LEVELS IN dBA.
L_t = NOISE LEVEL DUE TO PROJECT (dBA)
0
1
2
3
4
5
10
15
20
70
80

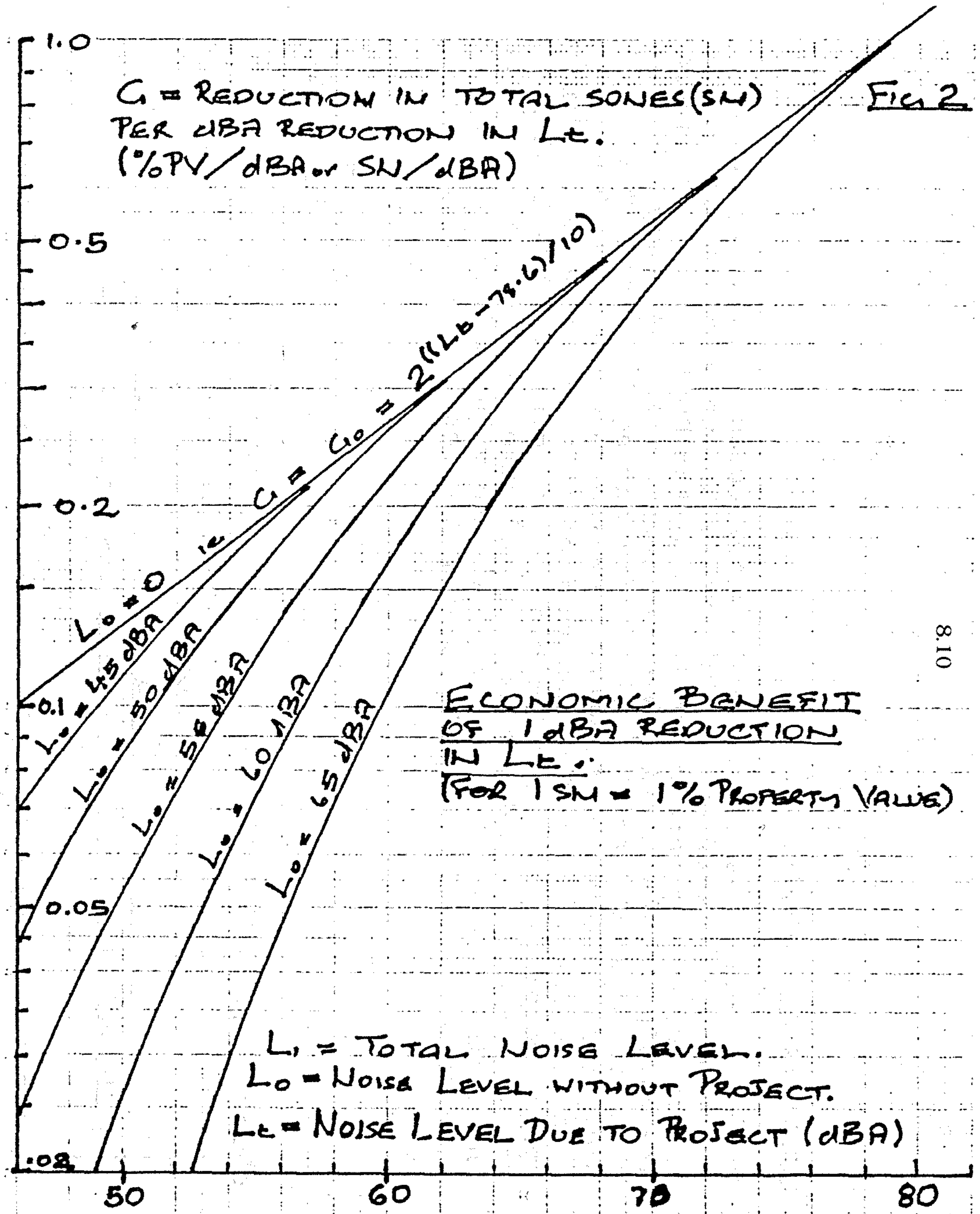
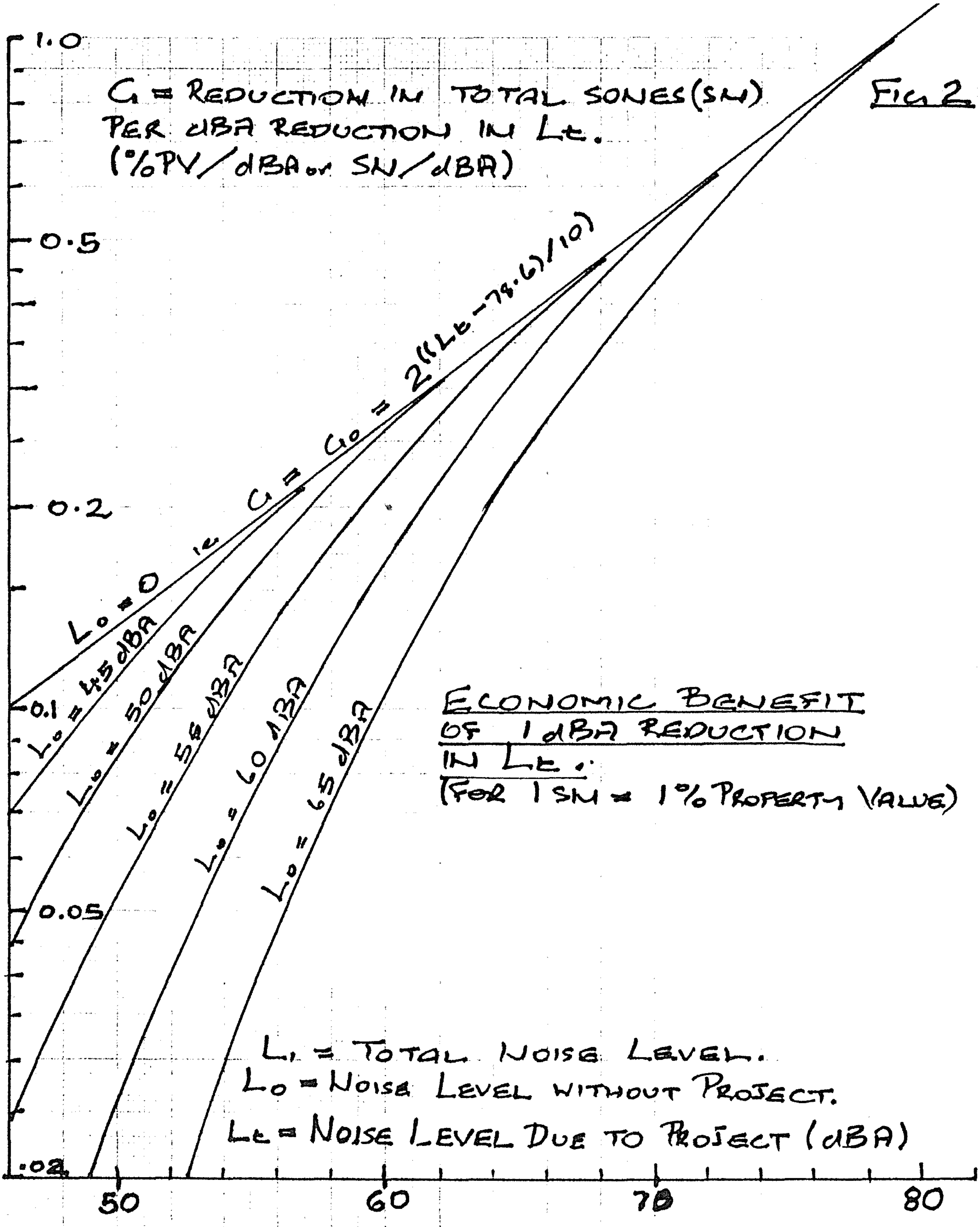
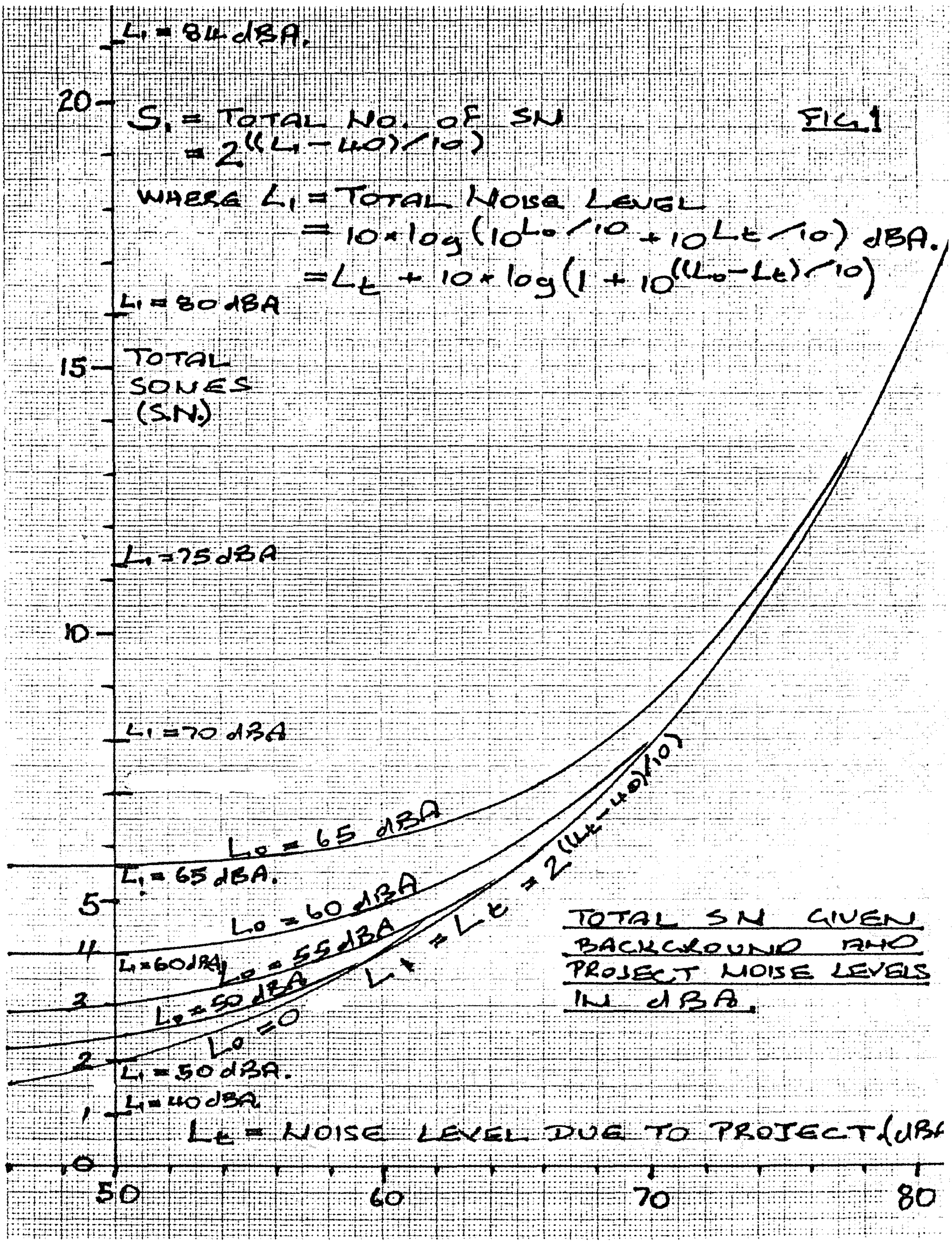


FIG 2
G = REDUCTION IN TOTAL SONES (SN) PER dBA REDUCTION IN L_t.
(%PV/dBA or SN/dBA)
$L_o = 0$ ie $G = G_o = 2^{((L_t-78.6)/10)}$
L_o = 45 dBA
L_o = 50 dBA
L_o = 55 dBA
L_o = 60 dBA
L_o = 65 dBA
ECONOMIC BENEFIT OF 1 dBA REDUCTION IN L_t.
(FOR 1 SN = 1% PROPERTY VALUE)
L_1 = TOTAL NOISE LEVEL.
L_o = NOISE LEVEL WITHOUT PROJECT.
L_t = NOISE LEVEL DUE TO PROJECT (dBA)
1.0
0.5
0.2
0.1
0.05
.02
50
60
70
80



1.0
0.5
0.2
0.1
0.05
.02
50
60
70
80
G = REDUCTION IN TOTAL SONES (SN) PER dBA REDUCTION IN Lt.
(%PV/dBA or SN/dBA)
Fig 2
Lo = 0 ie G = Go = 2((Lt-78.6)/10)
Lo = 45 dBA
Lo = 50 dBA
Lo = 55 dBA
Lo = 60 dBA
Lo = 65 dBA
ECONOMIC BENEFIT OF 1 dBA REDUCTION IN Lt.
(FOR 1 SN = 1% PROPERTY VALUE)
L1 = TOTAL NOISE LEVEL.
Lo = NOISE LEVEL WITHOUT PROJECT.
Lt = NOISE LEVEL DUE TO PROJECT (dBA)



$L_1 = 84$ dBA.
S_1 = TOTAL No. OF SN
$= 2^{((L_1 - 40)/10)}$
FIG. 1
WHERE L_1 = TOTAL NOISE LEVEL
$= 10 * \log(10^{L_o/10} + 10^{L_t/10})$ dBA.
$= L_t + 10 * \log(1 + 10^{((L_o - L_t)/10)})$
$L_1 = 80$ dBA
TOTAL SONES (S.N.)
$L_1 = 75$ dBA
$L_1 = 70$ dBA
$L_o = 65$ dBA
$L_1 = 65$ dBA.
$L_o = 60$ dBA
$L_1 = 60$ dBA
$L_o = 55$ dBA
$L_o = 50$ dBA
$L_o = 0$
$L_1 = L_t = 2^{((L_t - 40)/10)}$
TOTAL SN GIVEN BACKGROUND AND PROJECT NOISE LEVELS IN dBA.
$L_1 = 50$ dBA.
$L_1 = 40$ dBA
L_t = NOISE LEVEL DUE TO PROJECT (dBA
20
15
10
5
4
3
2
1
0
50
60
70
80

NOISE BARRIER RETROFITTING PROGRAM

ATTITUDINAL SURVEY OF NEARBY RESIDENTS

A Paper

to be presented

at the

Australian Acoustical Society

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By

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August, 1988.

1.0 BACKGROUND AND OBJECTIVES

In 1985, the Road Construction Authority of Victoria (RCA) commenced a $7 million (1988 $'s) program of retrofitting noise barriers to its older freeways, that is, those constituted prior to mid 1979. To date, some $2¼ million has been spent on the Project, with timber barriers being erected along sections of the Tullamarine, South Eastern and West Gate Freeways. The barriers are erected where the noise levels 1 metre from the facade of residences adjacent to the freeway exceed the Environment Protection Authority's (EPA) interim guideline level of 68 dB(A) L_{10} (18hr). Since installation noise has been reduced by as little as 2dB(A) in some cases but up to 8dB(A) in other cases.

In order to ascertain resident perceptions of the noise attenuation barriers and the resultant change in noise levels, the RCA in association with the Ministry for Planning and Environment (MPE) decided to conduct a survey of residents who live adjacent to the barriers and who live beyond the immediate area of influence. In the light of this objective, Wilson Sayer Core Pty. Ltd., planning and development consultants, were commissioned to undertake the Survey.

2.0 METHOD

Sample Distribution

An attempt was made to gain 300 interviews, a 100% sample of representatives from each of the adjacent residences (approximately 173) and the remainder from residences beyond (127) in fifteen locations. A residence adjacent to the freeway was taken as one where one boundary of the allotment was coincident with the boundary of the freeway, or where there was no development between the residence and the freeway.

Every effort was made to gain an effective interview from those residences drawn in the original or primary sample. For those adjacent residences, it was not until three callbacks were made to a residence, that is, three unsuccessful attempts were made to gain an interview that a residence from the original or primary sample was substituted by a residence from the secondary sample. This sample consisted of residences from three additional locations.

In the case of residences in those areas beyond the primary area of influence, generally only one attempt was made to gain an effective interview. If this was unsuccessful, then that residence was substituted by the one next door.

In all, some 280 effective interviews were achieved, 129 or 57.8% of all the adjacent residences and some 151 beyond.

Data Analysis

A questionnaire was administered to each respondent by skilled interviewers and the results of the interviews were processed in tabular form, analysed, and where possible, correlated with noise measurements taken at various sites along the freeways before and after installation of the noise attenuation barriers.

3.0 FINDINGS

General Attitudes to the Local Area

In order to seek an appreciation of residents' awareness of freeway traffic and the associated noise levels in their local areas, residents were first asked several questions about their living environment. They were asked to rate their area as a place to live, and to state their particular likes and dislikes about living in that area and any changes that have occurred over recent years.

Some 111 or 50% of all residents who participated in the Survey rated their area as a 'very good' place to live. Another 41% rated their area as 'good'. Whilst overall the majority of survey respondents have lived in the area more than ten years, the results showed little variation in attitude by length of residence or between residents who live in areas adjacent to the freeways or beyond. However, when the mean scores were calculated and the results further examined on a locational basis, that is, at those locations where noise attenuation barriers have been installed, some marginal variations were noticed. Locations one (adjacent to Tullamarine Freeway) and eighteen (West Gate Freeway near Williamstown Road exit) were the two locations given the highest ratings overall, 9.2 and 8.9 respectively. Conversely, locations eleven and thirteen, both adjacent to the West Gate Freeway near Williamstown Road, received the lowest ratings of 7.4 and 7.2 respectively.

When respondents were asked about the things they liked and disliked about the area, the likes of the area were more easily articulated than the dislikes. For example:

Likes	%
Close to shopping/shops	43
Quiet/peaceful	36
Close to public transport	36
Pleasant friendly people	32
Close to City	30

Only one dislike of any significance was expressed, 'freeway noise', by 29% of respondents. Furthermore, this negative feature was more noticeable among the residents interviewed at locations five, seven and thirteen. At locations five and seven adjacent to the Tullamarine Freeway near Bell Street, about half the respondents raised freeway noise quite spontaneously, and consistent with its lower rating, some 85 % of residents interviewed at location thirteen near the West Gate Freeway near Melbourne Road, named such. The noise levels measured at sites within these locations were not higher than at other locations. In fact, in most cases they did not exceed the 68dB(A) level post installation of the noise attenuation barriers. Hence, there was no clear explanation for this attitude.

It is also noteworthy that only 45% of residents had noticed any change in noise levels over recent years. Moreover, not all perceived the change for the better.

To illustrate:

Reasons for change in Noise Level	Total N=93	Adjacent N=41 %	Beyond N=52 %
More traffic/freeway traffic/ noise	43	32	52
Barrier has cut noise level	39	61	21

Those residing in the areas beyond the immediate area of influence stated the noise had worsened. Conversely, those residing adjacent to the freeway were more positive, with some recognition of the installation of the noise attenuation barriers as being the reason for the change, the reduction in noise levels.

Attitudes to the Noise Attenuation Barriers

After establishing residents' general feelings toward their living environment and spontaneous reactions to any change in noise levels and any awareness of the noise attenuation barriers, residents were questioned specifically on the barriers to test their knowledge further and so gain more detailed information on their attitudes towards and perceptions of the barriers. At this time, more positive responses were more forthcoming.

In spite of only 36 or 12.9% of respondents noticing a change in noise levels and attributing such to the installation of the noise attenuation barriers, some 70% of respondents liked the noise barriers. Moreover, the feeling was more prevalent among those residing in the adjacent residential areas. Of the 129 respondents from these areas, 79% stated they liked the noise barriers. When compared with the areas beyond, this figure is quite significant. 62% stated they liked the noise barriers in these areas.

Overall reasons given for liking the barriers generally related to:

	N=196 %
- quieter/stops noise	59;
- more privacy/security/stops burglars	13;
- looks good	11;
- very safe/can't climb	10;
- keeps area tidy/neat/clean/cuts pollution	7;
- safer/thicker/stronger/stops cars	4; and
- better than before	5.

Of all the persons interviewed, only forty seven respondents or 17% stated they disliked the barriers. They still complained of the noise (36%) and some perceived the barriers as ugly and unattractive (19%) or too high. If the latter were the case, views were said to be blocked and/or they felt closed in (17%). As can be seen from the above, the positive comments far outweigh the negative. Moreover, no location could be specifically identified as being still troublesome or noisy post installation of the noise barriers. In other words, the negativism that exists is scattered a little among the respondents of all locations and cannot be linked to one area.

Changes to Activity Patterns Since Installation

A number of activities, for example, sleep, conversation in the home, use of yard, reading, relaxation and so on were put to the respondents to find out whether the noise level from the freeway traffic affected these activities and whether their pursuit of such had improved since the installation of the barriers.

Prior to installation of the barriers a number of the residents' daily activities, were said to be affected 'frequently' or 'all the time'. These were:

		%
-	use of yard	21
-	sleep	19
-	general peace	16
-	relaxation	13

Since installation, residents, particularly those in the adjacent residential areas, have noticed marked improvements to their daily pattern of activities and the general environment. The most noticeable improvements related to:

		Adjacent %	Beyond %
-	general peacefulness	41	21
-	use of yard	36	15
-	sleep	32	21
-	relaxation	31	16

Time between installation of the noise attenuation barriers and the conduct of the interviews was quite considerable, in some cases up to three years. Hence, it is not surprising that any change in noise levels is perceived at all, and if it is, that it is not immediately associated with the installation of the noise attenuation barriers. It would appear that residents have become accustomed to the traffic and the associated noise, it is part of their living environment, thus for many, change may go unnoticed or is soon forgotten.

The noisy engines of accelerating vehicles were still expressed as 'extremely annoying' features of the environment. Some 23% of respondents noted this, and other problems associated with traffic for example, incessant noise and vibrations were also noted but to a much less degree. In spite of this, it is noteworthy that more than one third of the total sample population do not find such features annoying at all, and a further 20% only a little annoying. Hence, it can be said that whilst no significant changes have been experienced, there are some 50% who are reasonably satisfied.

4.0 CONCLUSION

It can be concluded that the noise attenuation barriers have had a positive effect on the living environment of many residents, even though residents did not all perceive any reduction in noise levels and/or did not automatically attribute any reductions to the noise attenuation barriers.

In light of the above, it has therefore been recommended that the program of fitting the noise attenuation barriers continue. However, in order that a more accurate picture of the effects be gained, it is suggested that a more structured approach be taken, that is, research be undertaken with the affected residents pre and post installation of the noise barriers, so that the perceived changes can be correlated more accurately with the calculated or actual changes in noise levels.

5.0 LIMITATIONS OF THE STUDY

Considerable time had lapsed since the installation of the noise attenuation barriers, hence any immediate impact or reaction to the barriers by residents adjacent or nearby was difficult to measure. Since installation, residents, particularly those living adjacent to the freeways, had time to become reconditioned to the changed noise levels. Moreover, their recall of the situation prior to installation had dimmed somewhat. In other words, the extent and magnitude of the effect as indicated by the survey responses is likely to be an under-representation of the perceived effect.

In order to gain an even more accurate and in turn more beneficial response, a more controlled way to conduct the Study would have been preferable. An interview survey of residents pre and post installation of the noise attenuation barriers could have been undertaken. The responses or perceptions of the noise levels could then have been correlated with the actual or calculated noise levels before and immediately after installation.

In this instance, only the after situation can really be examined, and in some cases, noise levels were not calculated before and/or after installation, hence there are some gaps in the data base and limitations on the depth of the interpretation of the findings. In spite of this, this Study was worthwhile. The survey revealed that the retrofitting program is beneficial, but at the same time, recommended that in Stage 2, a more structured approach, including qualitative and quantitative research methods, be adopted to allow for a more extensive and in turn accurate assessment.

Measurement of Facade Insulation Using Insitu Intensity Technique

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2. Danish Acoustical Institute (DAI), Denmark

ABSTRACT

Although quite a lot of work has been reported on the application of the intensity technique in sound insulation measurement, facade sound insulation measurement in situ has been less investigated. Measurements were made in a newly built house in Copenhagen, and the results show the advantages of the intensity approach in diagnosing the main transmission path and determining the contributions from each part of the facade. Good agreement was found between results of sound intensity measurement and conventional sound pressure level measurements. For the situation of stationary background noise, a reverse method (noise source in room) was tested. The results indicate that this method can be recommended.

I INTRODUCTION

The sound insulating effect of panels, windows, walls, outdoor air intakes and any facade part can be measured using sound pressure approaches both under laboratory conditions and in situ. [1,2] Earlier work on the application of the two-room intensity approach in sound insulation measurement [3,4] showed the advantages which were considered to be that two special rooms were not needed. Cauberg [5] reviewed some aspects of field measurements of facade sound reduction. The following has been suggested:

a) sound absorption should be added in the receiving room in order to keep the reactivity as small as possible [5]. This has been shown, however, among others in [6] not to be too important;

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b) the distance between measuring surface and facade can be 0.1 - 0.2 m [7,8];

c) if detailed information about the sound transmission path is needed, the distance between two measurement points is recommended to be so small that there are at lease 2 measurement points per bending wave [5];

d) a scanning technique is recommended to smoothen the effect of bending waves; if scanning is preferred, the scanning speed may be up to o.3 m/sec. [5,7];

e) in practice an averaging time of 8 sec. is a suitable starting value [8].

Cooperative research projects have been carried out by the researchers of BMILP and DAI for the purpose of establishing some guidelines for facade sound insulation measurements using the intensity technique in situ. Bruel & Kjaer kindly provided an engineer to assist in the work and supplied some instruments and software to support the measurements.

Detailed results from the measurements with sound source outdoors can be found in DAI Reports [9,10]. This paper thus focusses on a new method, i.e. with sound source indoors.

II FIELD MEASUREMENT 1
SOURCE OUTDOORS

A field measurement was carried out in a newly built two-storey building near a quiet street in Hvidovre, Copenhagen, in 1987. Conventional measurements were carried out according to the proposed Nordtest-method [2] and the sound reduction index of facades was calculated from Equation (1):

$$R_{45^\circ} = L_{p1}' - L_{p2} + 10\mathrm{Lg}\ (S/A) - 1.5\ [\mathrm{dB}] \qquad (1)$$

Intensity indoors was measured with two sets of Bruel & Kjaer's intensity analyser 3360 which were controlled by Bruel & Kjaer's software package WW9078 and DAI software BE/BJ-8701 respectively. The sound reduction index of the entire facade was calculated from Equation (2):

$$R_\theta = L_{p1} - 6\mathrm{dB} + 10\ \mathrm{Lg}\ \cos\theta - L_{W2} + 10\ \mathrm{lg}\ S \qquad (2)$$

The results are shown in Figure 1. A good agreement was obtained between the two methods. Moreover, the intensity method gave more information. For example Figure 2 illustrates the A-weighted intensity distribution on the windows and air intake part of the facade.

III FIELD MEASUREMENT 2, SOURCE INDOORS

3.1 Low Background Noise Case

The idea to put the source indoors and to measure intensity outdoors is based on the following:

a) The incident sound field on the source side should be uniform or as diffuse as possible.

b) The reactivity of the sound field on the receiver side should be as small as possible.

c) The effect of stationary background noise can be cancelled according to the Gauss theorem when surface sound absorption is negligible.

When background noise is low and/or rather steady, this method may have outstanding advantages when measuring sound insulation of building facades. Among others, they are:

a) No absorption is needed in the room.

b) Measuring on the source side is very easy.

c) Usually the outdoor sound field is nearly a free field, so the reactivity may be very low.

The principle of the method is illustrated in Figure 3. The sound reduction index R of the facade can be calculated using Equation (3) when this method is used.

$$R = L_{p1} - 6 - L_{I2} \quad [dB] \qquad (3)$$

where

L_{p1} = space-average sound pressure level in the room. It can be measured with a microphone on a rotating boom or in some randomly chosen microphone positions [dB]

L_{I2} = average intensity level outdoor [dB]

Test measurements were carried out in one office on the ground floor of DAI. For comparison, a conventional measurement was made, too. The sound reduction index of the window was calculated by means of Equations (1) and (3). The results are shown in Figure 4. A good agreement between the results from the two methods can be seen.

3.2 Reactivity Tests

The reactivity of the outdoor sound field was tested by measuring in 20 points (5 rows x 4 columns) placed 10 cm outside the window pane. The pressure-intensity index $\delta pI = L_p - L_I$ varied from 0 to 4 dB.

For comparison, the reactivity of the indoor sound field obtained from the measurement in Chapter II's case is shown in Table 1. Although 2 foam mats and 5 mineral wool absorbers were mounted in the room, the reactivity indoors is much worse than that obtained in the case of source indoors.

Table 1 Example of Results of Reactivity Test (source outdoors)

No.	1	2	3	4	5	6
Location	Radiator	Window (right)	Window (left)	Above window(r)	Above window(l)	Air intake
$\delta pI_{(dB)}$	10.4	3.9	0.9	5.9	17.0	0.4

3.3 Scanning

In this experiment three different scanning methods, i.e. 4x5, 3x3, 2x2 scanning subareas, respectively, were used to replace 42 fixed measurement points. The scanning track was "square S-like" and the scanning speed was about 0.5m/s. The distance between scanning lines was about 5 cm. The results obtained using various scanning methods agree very well with those obtained with fixed points, cf Figure 5.

3.4 Effect of Background Noise and Its Elimination

In practical cases, e.g. roadside cases, background noise cannot be as low as above mentioned. Some experiments were made with a loudspeaker as a source placed outdoors, 5m from the facade with 45° angle of incidence, cf Figure 3. The level of its artificial background noise was adjustable. The same scanning method of 4 subareas were used to measure the sound power transmitted through the window from the sound source indoor under different background noise levels. Table 2 shows some examples. Even though the measured intensity on the 4 subareas was changed while the background noise increased, say, from case 1 to case 3, the measured total sound power remained constant.

However, if the background noise becomes too high intensity measurement will be more and more meaningless because of the reactivity.

To resolve this problem, a portable screen, which was designed as a mineral-lined square-funnel with a mineral wedge in the centre, was tested. A typical result is shown in Table 2. The screen obviously reduced the reactivity (δpI was 3 -5 dB), but at the same time led to an increase in power level. Pulling out the probe from the screen for 10cm did not help. This indicates that the increase in power level is not due to the focusing effect of the screen. The reason can be the effect of the screen on the value of the Gauss integral. That is, a screen moving with the probe violates Gauss' law. It is believed that a stationary or portable screen with a floppy edge, which can be attached to the facade so as to reduce background noise, will prove helpful. Future work is expected.

Table 2 Intensity levels and sound power level of window with various background noise conditions

Case No.	Intensity distribution (dB)				Total sound power level dB (Lin)
	Subarea 1	Subarea 2	Subarea 3	Subarea 4	
1	64.8	68.1	65.3	69.1	69.0
2	66.7	69.3	47.7	68.2	69.0
3	-69.0	73.6	-70.6	72.4	68.9
3 (with screen)	70.1	72.2	70.1	72.2	72.0

IV CONCLUSIONS

1) The field intensity measurement can give the facade sound reduction index without obvious deviations in results from the conventional pressure approach and more detailed results for different facade parts than the pressure approach.

However, the sound reduction of the good sound insulating parts, e.g. the wall, can not be measured by means of this intensity approach.

2) Intensity measurements can be made using the source-in-room method if the background noise is low or reasonably steady.

The main advantages of this approach are:

- Neighbours are less disturbed by the noise from the loudspeaker.
- It is easy to mount the source.
- It is not necessary to mount sound absorbers in the room.
- It is easy to obtain a diffuse sound field on one side - although problems are inherent in small rooms at low frequencies - and a free field on the other side.

The main problems are:

- It is hard to control fluctuating background noise.
- The source has to be much more powerful than when an outdoor source is used, especially at high frequencies.
- Difficult access to the outdoor measurement surface can give some limitation in practice. This problem also exists e.g. in the Nordtest proposal [2].

3) When high background noise levels occur, a stationary or portable screen may be helpful. Further work has to be done.

V REFERENCES

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[7] ELAB Report STF 44 A86166, 1986.

[8] T.G. Nielsen, "Intensity Measurements in Building Acoustics". B&K application notes.

[9] J. Kragh et al, DAI Technical Note No. LI 326/87, 1987.

[10] Q. Chen et al, DAI Technical Report No. LI 302/87, 1987.

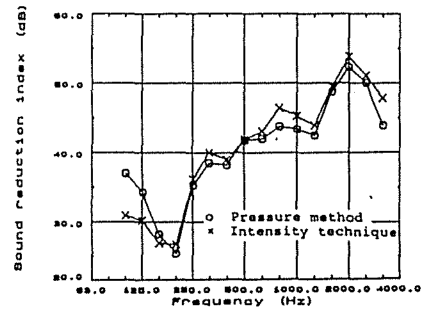

Figure 1 Sound reduction index of the facade, i.e. windows, open air intake and wall measured in different ways.

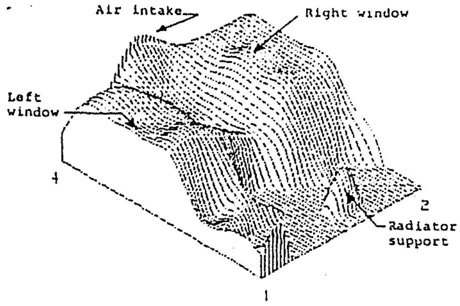

Figure 2 3-dimensional plot of overall A-weighted intensity distribution on the main part of the facade

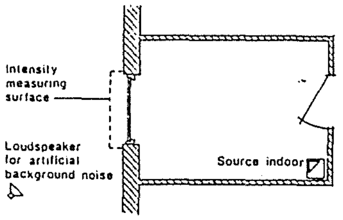

Figure 3 Sketch showing the principles of measuring sound reduction index with a noise source indoors. Plan view.

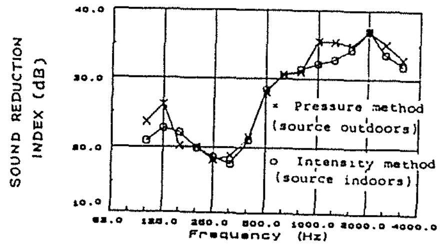

Figure 4 Comparison between results obtained by the tw approaches. Sound reduction index of window i DAI-office in Lyngby.

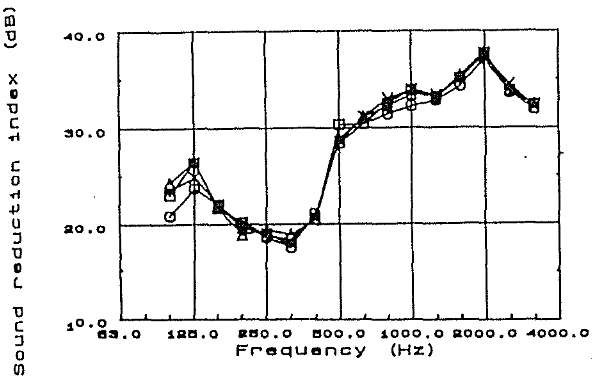

Figure 5 Sound reduction index obtained in various ways using the intensity technique.
o 42 fixed measurement points
× 4 × 5 scanning subareas
Δ 3 × 3 scanning subareas
□ 2 × 2 scanning subareas

HEARING LOSS: IS DAMAGE RISK RELATED TO SOUND PRESSURE SQUARED OR SOUND PRESSURE ?

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Standards and legislation in Europe and Australia refelect the opinion that hearing damage is proportional to pressure squared or the amount of energy experienced by the ear. This opinion is represented in standards as a 3 dB(A) trade off between noise level and exposure duration; that is, for a fixed noise exposure, the noise level must be reduced by 3 dB(A) if the exposure duration is doubled. The United States, on the other hand, uses a 5 dB(A) trade off rule which das been justified on the basis of the recuperative powers of the ear. However, there are presently moves afoot there to change this 5 dB(A) trade off to 3 dB(A). In this paper we present evidence and a consistent hearing loss model in support of the conclusion that hearing damage is proportional to pressure rather than pressure squared; that is, there is a 6 dB trade–off between hearing loss and noise exposure.

1. INTRODUCTION

It has long been recognized that exposure to excessive noise can cause temporary loss of hearing which may become permanent if the exposure is prolonged or particularly intense. The advent of the industrial revolution subjected whole groups of people to excessive noise exposure and consequent loss of hearing. Modern industry has continued that process. However, with the development of the sound level meter and audiometer it became possible to begin quantitative investigation of the problem.

2. HEARING LOSS OR THRESHOLD SHIFT

In 1952 in the United States a working sub–committee was set up to determine, if possible, what quantitative relation might exist between exposure to excessive noise and permanent threshold shift [1]. Over the intervening years very much effort has been expended in this exercise. The International Standards Organisation draft standard ISO 1999.2–1985 summarizes the best information available on the relationship between noise exposure and hearing loss [2]. The latter draft standard provides means for calculating expected threshold shift on a statistical basis at specified frequencies in the audio frequency range. Thus for each specified frequency and fractile (fraction of the total sample population whose threshold shift or total hearing loss exceeds a specified value), the draft standard provides the following relation,

$$H_s' = H_s(T_A) + N(L,T_N) - \frac{1}{120}\left[H_s(T_A)N(L,T_N)\right] \quad \text{(dB)} \qquad (1)$$

In equation (1) the quantities are all defined for a specified frequency and fractile and have the following meanings:

H_s' is the total threshold shift (hearing loss or hearing level) which is dependent upon the sex of the population. s is set equal to M for males and F for females.

$H_s(T_A)$ is a term due to presbycusis. It accounts for aging and is sex specific. It is determined by reference to the hearing levels of a non–noise exposed control population.

$N(L,T_N)$ is the noise induced portion of the threshold shift, defined so that it is the same for both sexes. It is dependent upon the equivalent A–weighted sound pressure level, L dB(A), of the excessive noise and the duration, T_N (usually years), of the exposure to that noise.

3. PRESBYCUSIS

The investigation of presbycusis seems to be well advanced and the referenced standard provides means for estimating $H_S(T_A)$ as a function of age, T_A, and sex of the population for the various specified audiometric frequencies and fractiles describing the statistical distributions at those several frequencies.

Alternatively the draft standard suggests that alternative estimates for the presbycusis term $H_S(T_A)$ may be provided by the user. For example, some evidence has been published which strongly suggests that this term may also be race specific. In studies conducted in the United States black people were found to have suffered less hearing loss than a comparable group of white people [3].

4. NOISE INDUCED HEARING LOSS

The term $N(L,T_N)$ of equation (1) as been empirically determined based upon a preponderance of data of steady state A-weighted sound pressure levels, L_A, and various durations T_N of exposure to steady noise. The data for such studies have been provided by retrospective investigation of large numbers of workers in a wide range of industries. It is of importance to note that the data base relies upon steady state exposures and that extrapolation for use with non-steady state exposures is critically dependent upon the assumed relationship between that base and the definition of exposure. This matter will be the concern of the remainder of this paper.

Following the referenced draft ISO standard the following definitions will be of importance. The A-weighted sound exposure $E_{A,T}$ is defined as follows:

$$E_{A,T} = \int_o^T p_A^2(t)dt \qquad (2)$$

The quantity $P_A(t)$ is the time varying A-weighted sound pressure of the excessive noise.

An exposure level may be defined conveniently in terms of the sound exposure $E_{A,T}$ by the introduction of a suitable reference exposure, E_o. The exposure level is defined as follows:

$$L_{EX,T} = 10 \log_{10}(E_{A,T}/E_o) \qquad (3)$$

Yet another convenient quantity may be defined which is an equivalent steady state A-weighted level for a time varying sound pressure. An equivalent A-weighted level L_{Aeq} is defined as

$$L_{Aeq} = 10\log_{10}\left[\frac{1}{T}\int_o^T p_A^2(t)dt\right] + \text{constant} \qquad (4)$$

We now depart somewhat from the ISO draft standard and propose the following generalization of equation (4). Let

$$L'_{Aeq} = 10\log_{10}\left[\frac{1}{T}\int_o^T (p_A^2(t))^n dt\right] + \text{constant} \qquad (5)$$

The reference quantities of equations (4) and (5) may be chosen so that the constants of the latter two equations vanish. In equation (5) the integer n has been introduced. For example if n = 1 then equation (4) is obtained. However if n = ½ then it may be shown that

$$L_{Aeq} \geq L'_{Aeq} \qquad (6)$$

Introduction of equation (5) into equation (3) leads to the following expression for the exposure level.

$$L_{EX,T} = L'_{Aeq} + 10\log_{10}(Y/Y_o) \qquad (7)$$

In equation (7) reference quantities have been chosen so that the additive constant is zero. In the equation Y is the specified duration of the exposure to the equivalent A-weighted sound pressure level L'_{Aeq} in suitable units of time, for example in years and Y_o is then one year. Note that equation (7) becomes the equivalent expression given by the draft ISO standard when n of equation (5) is set equal to 1.

For the case of steady state levels and if $p^2{}_A(t)$ of equation (5) is a constant, equation (7), which follows from it, becomes

$$L_{EX,T} = n\ L_{Aeq} + 10\log_{10}(Y/Y_o) \qquad (8)$$

It is of interest to note that Australia and Europe follow the practice of setting n = 1 while the United States sets n = 3/5 in their various respective noise control regulations. The United States does not fully accept equation (4) but relies upon an equation of the form of equation (5). This practice is justified using an argument which may be questioned, and takes account of the apparent recuperative powers of the ear in an intermittant sound field.

With the introduction of the generalized expression of equation (8) the noise induced hearing loss is assumed to be a function of the single variable $L_{EX,T}$. Thus

$$N(L,T_N) \equiv N\ (L_{EX,T}) \qquad (9)$$

Introduction of equation (9) into equation (1) gives an expression which for a specified frequency, fractile and sex is a function of the variables as follows.

$$H'_s = f_s(X) \qquad (10)$$

where $$X = n\ L_{Aeq} + 10\log_{10}(Y/Y_o) \qquad (11)$$

The choice of the index n is reflected in the following trading rules. In Europe and Australia it is assumed that for a fixed sound exposure, the noise level must be reduced by 3 dB(A) (n=1) for each doubling of total duration of exposure while in the United States the reduction is 5 dB(A) (n=0.6). As shown here a 6 dB(A) (n=½) reduction would be more in accord with existing data.

6. <u>REVIEW OF THRESHOLD SHIFT DATA</u>

Burns and Robinson describe a British investigation of the relation between noise exposure and hearing loss [4]. Robinson concludes in Appendix 10 of the latter book that noise exposure and hearing loss are directly related and in effect he sets n of equation (11) equal to 1. However, review of Figure 10.1 of the latter appendix leads to a different conclusion [5].

The latter figure has been reproduced as Figure 1 for purposes of illustration. Referring to the figure a value of n = 1 in equation (11) would lead to the expectation that observed threshold shifts (median hearing levels in this case) at 3.15 years of exposure, A_1, A_2, & A_3, would be the same as observed threshold shifts at 31.5 years of exposure and 20 dB lower sound levels, C_1, C_2, & C_3, respectively. On the other hand, a value of n = ½ in equation (11) would lead to the expectation that observed threshold shifts at 10 years of exposure B_1, B_2, & B_3, would be the same as observed at 31.5 years of exposure. The figure shows quite clearly that the choice n = ½ is more in accord with the data than is the choice n = 1.

This observation, with reference to equations (10) and (11), strongly suggests that the data of Burns and Robinson would scale much better on X of equation (11) with n set equal to ½ rather than to 1. Reference to Figures 2 and 3 confirms this expectation. In the latter figures Burns and Robinson's empirical equation describing their data has been plotted against X with n = 1 in Figure 2 and n = ½ in Figure 3. Burns and Robinson provided only the empirical equation and no data. A range of possible values of n were investigated before deciding that n = ½ gave best fit.

The data of Burns and Robinson might be faulted on the basis that their measuring equipment for determining noise exposure might not be up to present day standards, but in any case it is important to consider the best available data, and reference will again be made to the draft ISO standard for this purpose. The latter draft standard and annexes has been used to calculate the predicted threshold shifts for 1, 2 and 4 kHz tones as shown in Figures 4 to 9. Data are shown only for men but data for women follow similar trends.

Review of Figures 4 to 9 shows quite clearly that n = ½ gives a better collapse of the data on the parameter X than does a value of n = 1 for the 1 and 2 kHz frequencies. Although the case for 4 kHz is not as clear cut, the choice of n = ½ still gives slightly better collapse of the data.

7. REVIEW OF PERCENTAGE RISK DATA

The percentage risk of developing a hearing handicap (or percentage of a given population who will suffer a hearing handicap) may be determined for any specified handicap and for any specified combination of audio frequencies [2]. Generally, the handicap usually specified is a hearing loss of 25 dB and the characterizing frequencies are usually some combination or average of 0.5, 1, 2, & 4kHz. The total percentage risk is considered to consist of a component due to noise and a component due to presbycusis [2,6].

Glorig [6] has provided some early estimates of total percentage risk and noise induced percentage risk of developing a hearing handicap (25 dB loss averaged over 0.5, 1, & 2kHz). Reference to Table 17.1 in [6] shows the following for total probability of sustaining a hearing handicap;

30 years of exposure at 80dB(A), total loss 7.7%
10 years of exposure at 90dB(A), total loss 7.9%

30 years of exposure at 90dB(A), total loss 23.3%
10 years of exposure at 100dB(A), total loss 22.0%

30 years of exposure at 100dB(A), total loss 48.5%
10 years of exposure at 110db(A), total loss 47.5%

The above result strongly suggests a value of n = ½ according to equations (10) and (11).

The data contained in the referenced table have been used to construct Figures 10 and 11. The remarkable collapse of most of Glorig's data on to a single well defined empirical curve in Figure 11 certainly supports the assumption that the best choice for n is ½.

The following is to be noted. In Figure 11 the data points which lie below the line of best fit correspond to relatively short duration high sound pressure level exposures while the data points which generally lie above the line of best fit correspond to relatively low level and long duration exposures. With the exception of the latter two cases the greater majority of Glorig's data lie on a single empirically determined line when plotted as a function of $\frac{1}{2} L_{AEQ} + 10\log_{10}(Y/Y_o)$. An integral of pressure amplitude with time dependence is strongly implied.

The percentage risk of developing a hearing handicap (defined here as a hearing loss in excess of 25 dB) has been calculated according to the procedures outlined in the referenced ISO draft standard using an average of the loss at 1, 2 & 4 kHz. Results for men are shown in Figures 12 and 13 (results for women follow similar trends). Review of these figures shows quite clearly that $n = \frac{1}{2}$ is a much better choice than $n = 1$, suggesting again that the important measure of noise exposure is an integral of pressure amplitude with time.

8. SUGGESTED MECHANISMS

The results reviewed above tend to support the view that hearing threshold shift is more accurately described as a function of exposure based upon an integral of pressure amplitude with time rather than the entrenched concept of energy input. This result suggests failure due to cyclic stress. Consequently the following mechanisms may be considered.

The hair cell derives its name from the fact that each cell possesses a small external bundle of tiny hair like structures called stereocilia. The cell may thus be considered to consist of two essential parts which are the body of the cell and its external stereocilia.

Excessive noise might be imagined as over stressing the hair cell stereocilia. These are the tiny rigid fibers that sit on the top of the hair cell between the reticular lamina and the tectorial membrane [7]. During any excitation they are subjected to shear forces which cause them to produce a substantial analog voltage. Over stressing might be thought of as inducing permanent stresses which eventually will cause a fatigue failure of the stereocilia. As they are not replaced, then the sensing which they formerly supplied is permanently lost.

The same over stressing might also cause trauma to the body of the hair cell. This would probably result in a temporary loss of sensitivity of the cell resulting in a temporary threshold shift. In the proposed model temporary threshold shift is to be associated with a temporary failure of the body of the hair cell while permanent threshold shift due to noise exposure is to be associated with stress induced failure of the stereocilia of the hair cell. The implication is that, if presbycusis is associated with deterioration of function of the body of the hair cell while noise induced hearing loss is associated with failure of the stereocilia, then the two effects are not additive as commonly assumed, since they are associated with different mechanisms of deterioration.

As a matter of interest the proposed model is in accord with the observation that permanent threshold shift and temporary threshold shift are uncorrelated. That is, in spite of much effort expended, no correlation has ever been demonstrated.

9. CONCLUSIONS

It has been shown that hearing loss correlates better with the integral of pressure with time than it does with the entrenched and widely assumed integral of pressure squared with time. This conclusion suggests that noise exposure in a varying noise level environment will be over estimated (see equation (6)) when determined according to the 3 dB(A) trading rule which implies that an increase in sound level of 3 dB(A) provides the same increase in noise exposure as an increase in the exposure time by a factor of two. A careful study of musicians might shed light upon this matter.

An integral of pressure amplitude with time would allow inclusion of impulsive noise of very large amplitude as simply an intense exposure. The anomolous treatment of intense impulsive noise and long duration noise of less intensity as separate kinds of phenomena as presently done would no longer be necessary.

Finally, if presbycusis is associated with cell body failure whereas noise induced hearing loss is associated with stereocilia failure then the two effects are probably not additive as currently assumed.

10. REFERENCES

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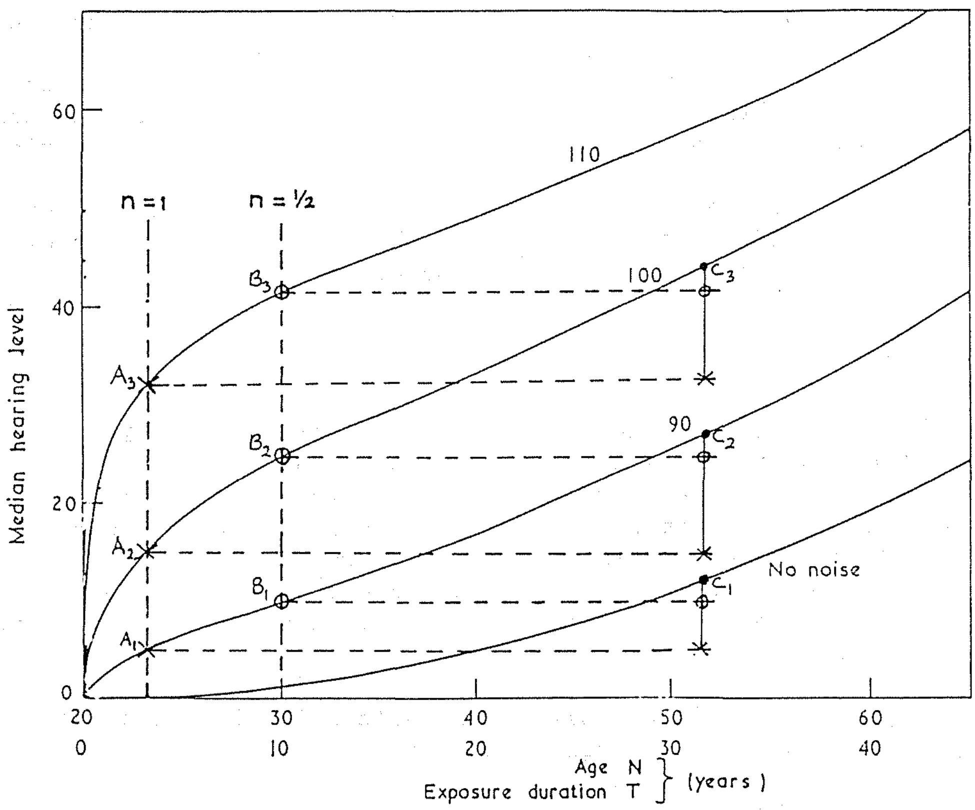
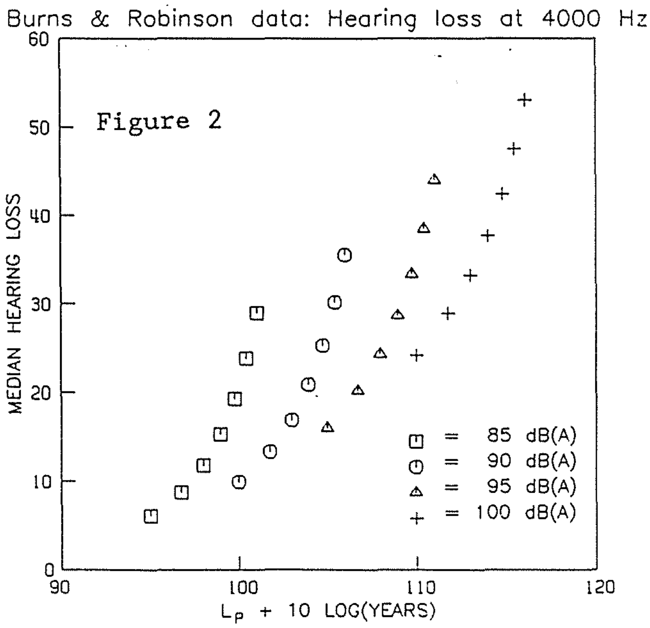
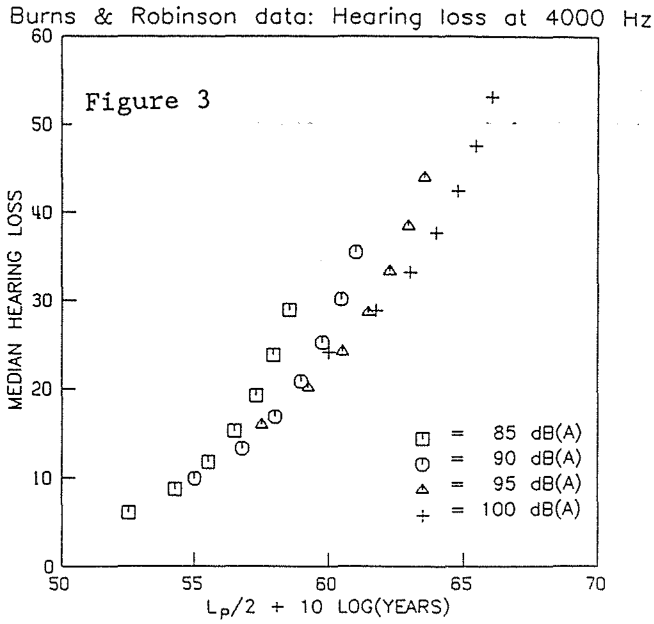
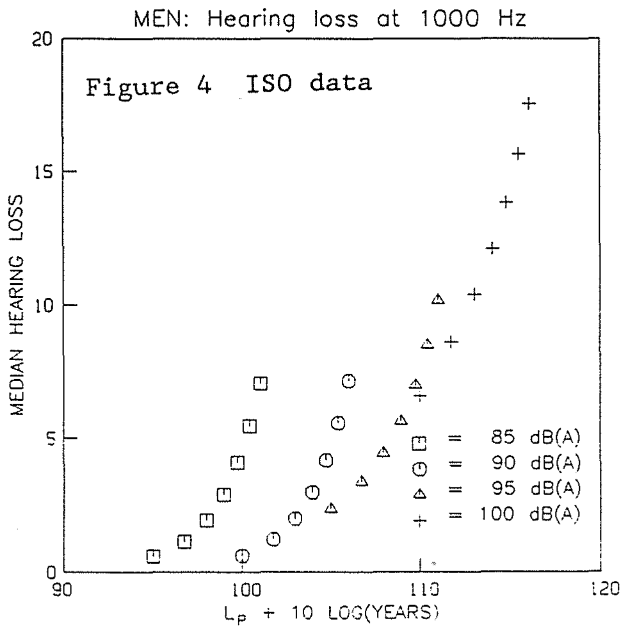
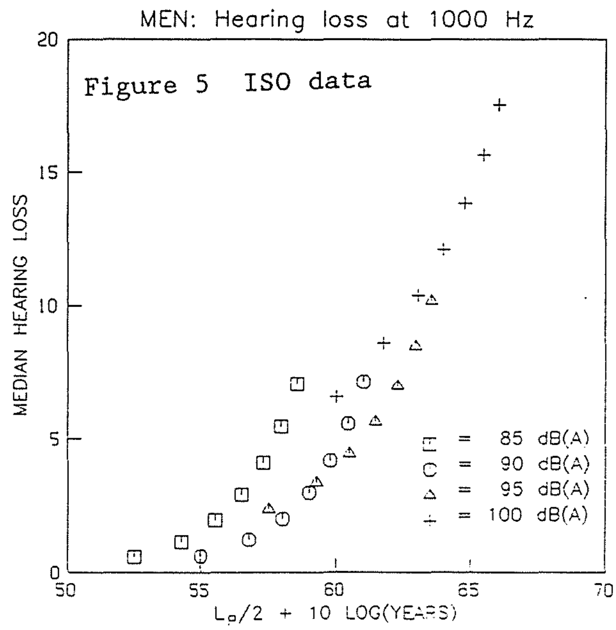


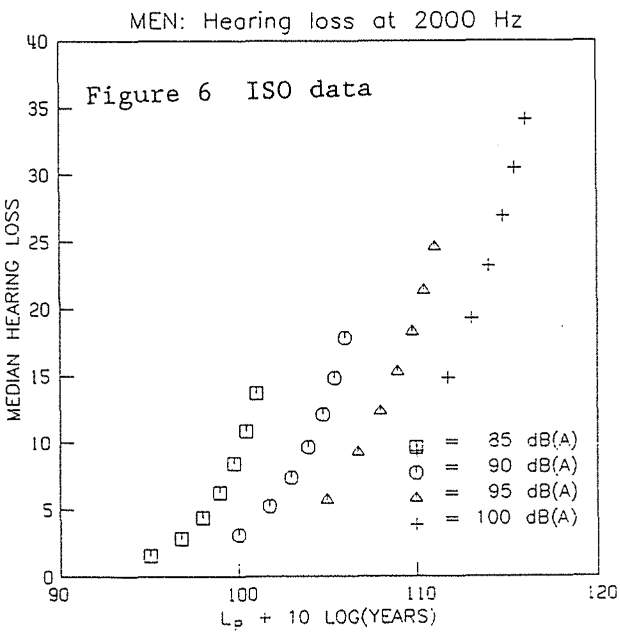
Figure 1. Illustration of the better fit of the 6 dB (n=½) rule over the 3 dB (n=1) rule for Burns and Robinson's data [4].

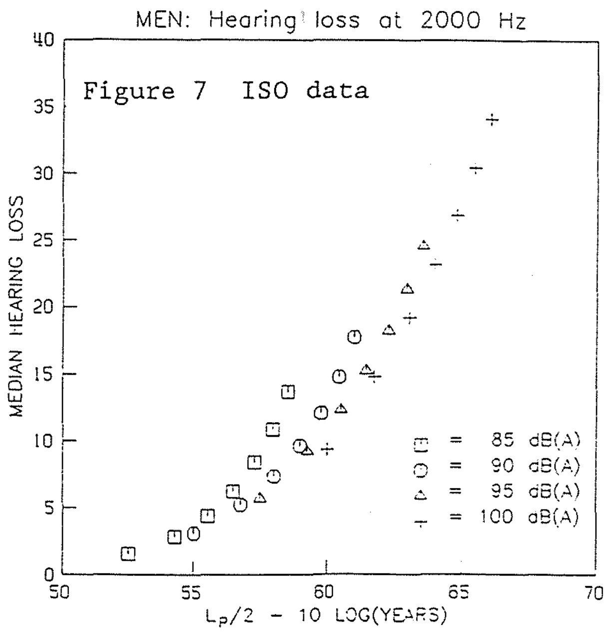

Burns & Robinson data: Hearing loss at 4000 Hz
Figure 2
MEDIAN HEARING LOSS
L_p + 10 LOG(YEARS)
□ = 85 dB(A)
○ = 90 dB(A)
△ = 95 dB(A)
+ = 100 dB(A)

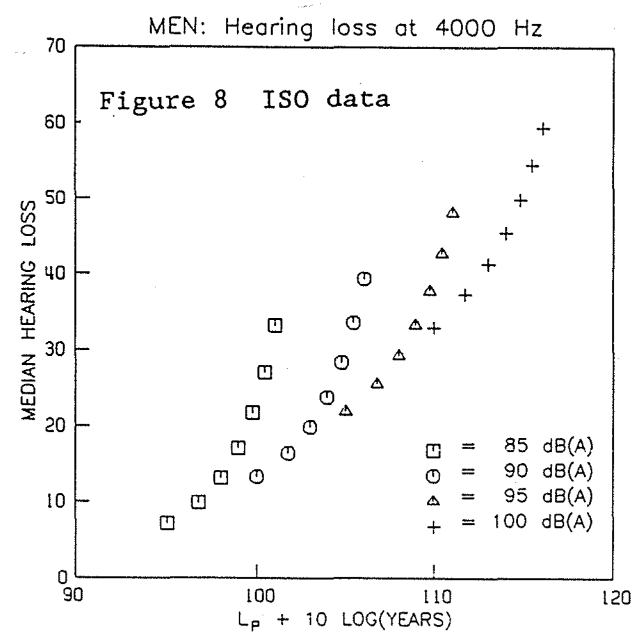

Burns & Robinson data: Hearing loss at 4000 Hz
Figure 3
MEDIAN HEARING LOSS
$L_p/2$ + 10 LOG(YEARS)
□ = 85 dB(A)
○ = 90 dB(A)
△ = 95 dB(A)
+ = 100 dB(A)

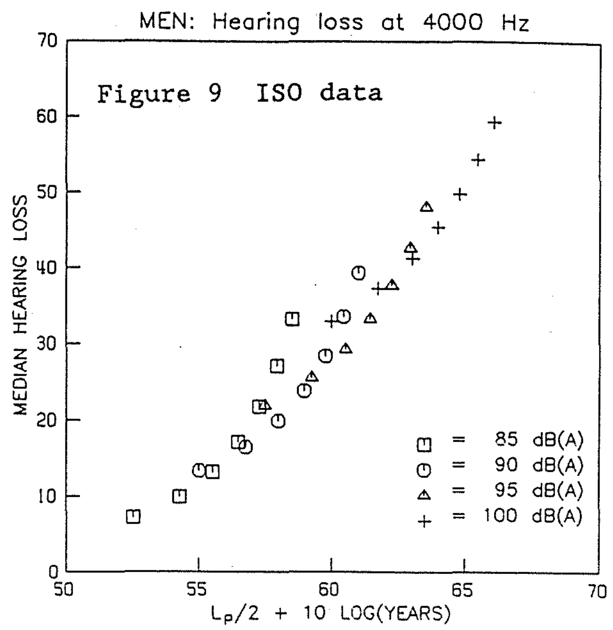

MEN: Hearing loss at 1000 Hz
Figure 4 ISO data
MEDIAN HEARING LOSS
L_p + 10 LOG(YEARS)
□ = 85 dB(A)
○ = 90 dB(A)
△ = 95 dB(A)
+ = 100 dB(A)

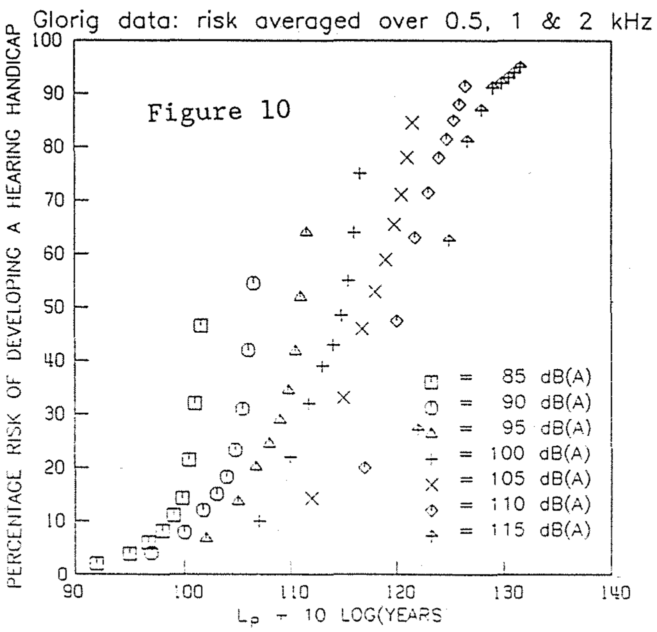

MEN: Hearing loss at 1000 Hz
Figure 5 ISO data
MEDIAN HEARING LOSS
$L_p/2$ + 10 LOG(YEARS)
□ = 85 dB(A)
○ = 90 dB(A)
△ = 95 dB(A)
+ = 100 dB(A)

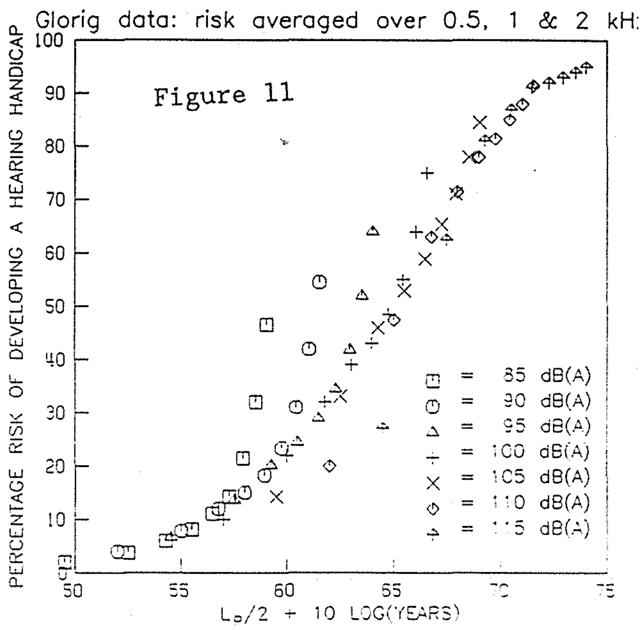

MEN: Hearing loss at 2000 Hz
Figure 6 ISO data
MEDIAN HEARING LOSS
L_p + 10 LOG(YEARS)
□ = 85 dB(A)
○ = 90 dB(A)
△ = 95 dB(A)
+ = 100 dB(A)

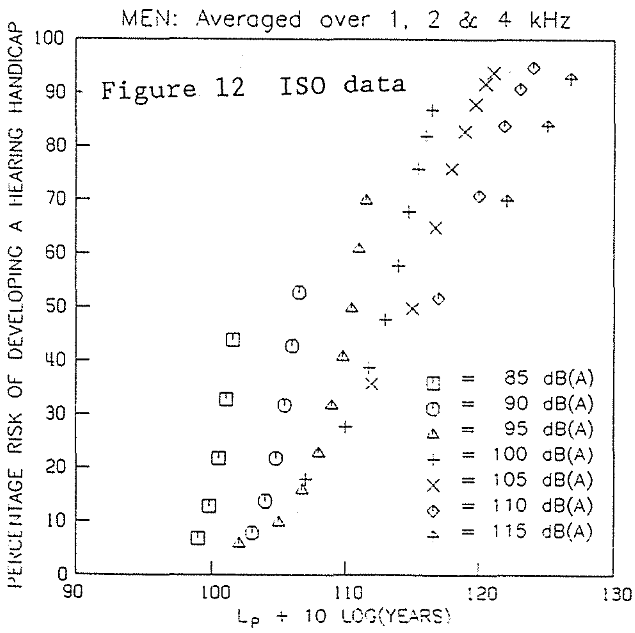

MEN: Hearing loss at 2000 Hz
Figure 7 ISO data
MEDIAN HEARING LOSS
$L_p/2$ - 10 LOG(YEARS)
□ = 85 dB(A)
○ = 90 dB(A)
△ = 95 dB(A)
+ = 100 dB(A)

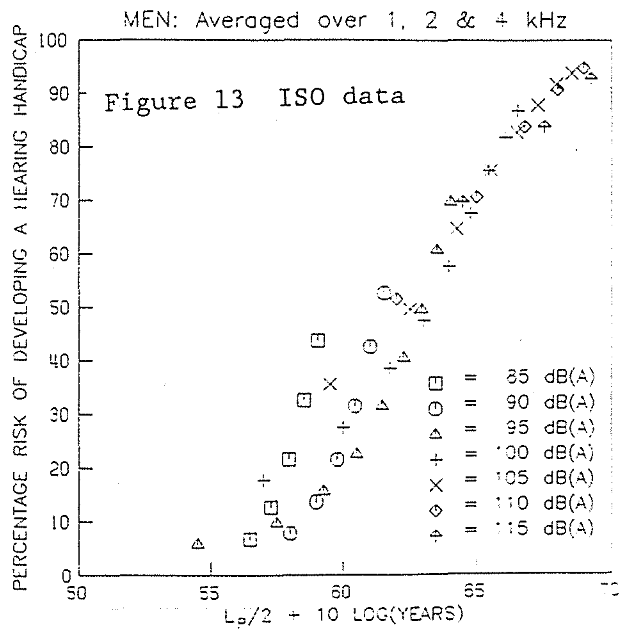

MEN: Hearing loss at 4000 Hz
Figure 8 ISO data
MEDIAN HEARING LOSS
L_p + 10 LOG(YEARS)
= 85 dB(A)
= 90 dB(A)
= 95 dB(A)
= 100 dB(A)


MEN: Hearing loss at 4000 Hz
Figure 9 ISO data
MEDIAN HEARING LOSS
$L_p/2$ + 10 LOG(YEARS)
= 85 dB(A)
= 90 dB(A)
= 95 dB(A)
= 100 dB(A)


Glorig data: risk averaged over 0.5, 1 & 2 kHz
Figure 10
PERCENTAGE RISK OF DEVELOPING A HEARING HANDICAP
L_p + 10 LOG(YEARS)
= 85 dB(A)
= 90 dB(A)
= 95 dB(A)
= 100 dB(A)
= 105 dB(A)
= 110 dB(A)
= 115 dB(A)


Glorig data: risk averaged over 0.5, 1 & 2 kH:
Figure 11
PERCENTAGE RISK OF DEVELOPING A HEARING HANDICAP
$L_p/2$ + 10 LOG(YEARS)
= 85 dB(A)
= 90 dB(A)
= 95 dB(A)
= 100 dB(A)
= 105 dB(A)
= 110 dB(A)
= 115 dB(A)


MEN: Averaged over 1, 2 & 4 kHz
Figure 12 ISO data
PERCENTAGE RISK OF DEVELOPING A HEARING HANDICAP
L_p + 10 LOG(YEARS)
= 85 dB(A)
= 90 dB(A)
= 95 dB(A)
= 100 dB(A)
= 105 dB(A)
= 110 dB(A)
= 115 dB(A)


MEN: Averaged over 1, 2 & 4 kHz
Figure 13 ISO data
PERCENTAGE RISK OF DEVELOPING A HEARING HANDICAP
$L_p/2$ + 10 LOG(YEARS)
= 85 dB(A)
= 90 dB(A)
= 95 dB(A)
= 100 dB(A)
= 105 dB(A)
= 110 dB(A)
= 115 dB(A)

STATISTICS AND ACOUSTICS

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ABSTRACT

All calculations involve error which can jeopardise the validity of results and calculations. An awareness of error and the methods of dealing with it are essential to the confidence in acoustic prediction techniques for both acousticians and end users.

1. INTRODUCTION

Noise is not a simple quantity or quality of sound. Noise is a response to a quantity of sound measured by approximations. These approximations indicate and predict quite well the subjective response of a wide range of individuals.

Sound level is measured as the "average" of the pressure fluctuations around a mean pressure which itself varies with time. The level about which it fluctuates and the degree of that fluctuation is judged by a receiver as acceptable or unacceptable depending on the receiver's mood, social standing and view of the world. From the turn of the century the world of acoustics has spawned a multitude of descriptors and acceptability criteria to match those descriptors. These criteria tend to be laid down as hard and fast limits to be achieved in the very grey area of public acceptance.

Acoustics is not an exact science and cannot be so as we are dealing with the likes and dislikes of the vast range of people who make up the "General Public" When criteria limits have been set every effort is required to stay within those limits. It is not enough to work "to-the-limit" we must be sure we're working within the limit. How large a margin is becomes a matter of issue as the cost of the project and of rectification of an error increases.

In any measurement and calculation, error will creep in and we must be aware of them and deal with them as best we can to minimise their effect.

2. ERROR

Standard texts list errors as illegitimate, random and systematic.

Illegitimate error covers

- i. chaotic
- ii. computational
- iii. blunders

Random error covers

- i. judgement
- ii. definition
- iii. variation of conditions

Systematic error covers

- i. calibration
- ii. human
- iii. experimental
- iv. technique
- v. loading

Illegitimate errors are self explanatory. Blunders cover all those situations we'd rather forget ranging from forgotten microphone covers on sound meters through to careful measurements of the wrong source.

Computational errors cover such things as wandering decimal points, arithmetic slip ups and vanished or replicated calculation steps. I have even seen calculation procedures pick up steps from pages away in different sections.

Chaotic covers all those unforeseeable circumstances that render readings unusable. These would include such things as, suspicious constables checking the acoustician at 3 a.m. in the morning, crickets invading the microphone cover and the almost inevitable dog yelping every time the meter is switched on.

With care these errors can be avoided or deleted as appropriate. Judgement is required as to what is a legitimate editing and what is a doctoring of results. In outdoor measurements we are looking for the "typical events" not one off anomalies (usually).

This leads onto random errors. Judgement errors are subjective decisions based on available information. Similarly definition errors are based on personal assessment of each project. Only experience and exposure will reduce these. The third area of random error is variation of conditions. This area bedevils all areas of commercial acoustics.

For example, in architectural acoustics there is no such thing as the precise transmission loss of any specific structure. Variations in material properties, construction technique and aging all attribute to unpredictable changes which we must attempt to foresee and compensate for.

In industrial acoustics there is no such thing as the typical machine. Variations will occur in operating conditions, tolerances and state-of-repair. We can calculate reasonably well for a well-maintained machine under the "typical" load but what happens when components are reaching the limit of their life and the machine is being driven to the limit. An example of this was measurement of an overburden drill I once did. Pity the gearbox collapsed on the day of my visit.

In environmental acoustics, the less said the better. A major variant in this area is meteorology. Present day fluid dynamics is able to accurately model this beast down to a scale of 50 km. A cray computer can predict the meanderings of a cyclone with reasonable accuracy over several days before reality deviates too far. Below this scale meteorologists resort to previous empirical records and accepted trends. The environmental acoustician deals with the world on a much finer scale than the meteorologist and with limited data of widely spaced weather stations, to predict variations in the local microclimate. Local information gained by short term weather tests using radiosondes are of limited value. Save for distances and the elevation of structures and contour lines, nothing is certain in environmental acoustics. A common fault in this area is to claim accuracy in terms of dB for a particular model. Accuracy is site, distance and time specific. Levels vary by time of day, time of year at the whim of the weather. Studies over a long period will always show a diverging scatter of results with distance.

Illegitimate error is avoidable through care and planning. Random error is uncontrollable but experience can provide estimates of error. Systematic error is the only area where we can define precise bounds.

Systematic error covers errors of calibration, human effects, experimental errors, technique and loading. Calibration error assumes that the instrument can be calibrated to within half its smallest division. In the field, accuracy beyond this is not possible. Calibration occurs to one frequency and assumes a linearity in the microphone, and meter performance. It is not possible to carry a complete calibration lab into the field nor does industry recognise its necessity.

AS1259 gives the following tolerances on microphone performance.

(Hz)	±30 degrees Tolerance(dB)		±90 degrees Tolerance(dB)	
31.5 - 1000	+0.5	-0.5	+1.0	-1.0
1000 - 2000	+0.5	-0.5	+1.0	-2.0
2000 - 4000	+0.5	-1.0	+1.0	-3.0
4000 - 8000	+0.5	-1.5	+1.0	-6.0
8000 - 12500	+0.5	-2.0	+1.0	-10.0

If your meter and microphone are within these tolerances in the lab, the 90 degrees tolerances should be used. In the majority of situation the sound field is unpredictable and surrounds the measurement point. There is a way in which a paddock full of crickets are going to align themselves in front of your microphone. Unless you are aware of the directivity pattern and response of your microphone, readings will fall, somewhere in the 90 degree tolerance range, exactly where is pure speculation.

Fortunately most spectra dominate in the 250 Hz to 1000 Hz region. This down-grades this sources of error to within the ± 1 dB range. In the other frequencies the results are swamped by the 250 Hz to 1000 Hz region (but not always).

Human error includes parallax error, and individual idiosyncrasies. With the increasing use of digital equipment with standard "statistical" packages this aspect should be modified and standardised. Any bias in the measurement function will at less be uniform across the industry.

Experimental error covers the multitude of sins related to the equipment used. The industry is being infested by digital sound meters and PC's. The computer is only as good as the original programmer. All computers work to a certain number of decimal places and round-off error is inherent in all their calculations. Because a computer prints out to 16 places of decimal it doesn't mean that the answer is correct to 16 places of decimal. It is only as good as the original input data. Electronic wizardary cannot move back in time and improve the accuracy of the original data.

Technique error is the misapplication of measurement and calculation techniques. If the sound field is understood this should not be a source of error. Many view the sound field as uniform right back to the imaginary acoustic centre and are unaware of near field, far field and the transition "between" the two. This simplistic view glosses over many acoustic problems in the design phases. It is not valid to assume free field conditions simply because the designers drawing the surroundings fail to show. At one stage it was suggested that hearing conservation tests for underground mining equipment be conducted in an open area on the surface.

The sound field is vastly different 500 m below ground in a roadway development with the roof one metre overhead and the roadway walls press in to within half a metre on either side.

Loading error, the last on the list, is derived from the operator. The presence of the operator will effect the recording of sound. Physically the operator can modify the sound field of the sound meter hence biasing results in the 300 Hz to 1000 Hz range.The operators body is a physical object introduced into the sound field. The operators presence can also effect measurement by attracting unwanted attention from inquisitive bystanders. How many acousticians have had clients start a discussion as soon as the meter is turned on.

3. ERROR INTERPRETATION

A final area which is seldom dealt with is the interpretation of readings and the interpretation of results. Acoustics is riddled with "mean values", "average values" and "standard deviations". These are terms misused and abused. These terms apply to absolute values, not logrithmic terms.

In standard engineering we deal with absolute values. Statistics is able to deal with these quite well. In acoustics, logrithmic terms are treated in the same way (unfortunately). Sound is measured as the logrithm of a ratio and acoustic calculations manipulate these logrithms expressed as the unit the dB. Standard statistics does not cover this area and hence the standard techniques for dealing with error does not apply here. I once discussed this with a Professor of Mathematics who was surprised and amused by this application of statistics. His conclusion was that in acoustics the calculation of error can only be dealt with by the duplication of each acoustic calculation. For each calculation, two others must be done. Using the relative error of each input the maximum positive and negative derivations should be calculated. For example, say that we're calculating hemispherical divergence. Over a hundred metres from a base sound level at five metres, the resultant distance attenuation would be either 26.02, 26.94 or 25.19 or rounding off 26dB, minus 0.8, plus 0.9. Similarly, over twenty metres the attenuation becomes 12 dB minus 1.0 plus 1.2.

As the calculations progress the errors accumulate. Error in the original measurement adds to error in internal propagation error, adds to transmission loss, adds to external propagation error. A five step calculation with $\pm$ 1 dB at each step leads to an answer $\pm$ 5 dB. To these must be added the error due to uncontrollable variables. If guarantees are involved these errors must be understood by both parties and compensated for by an agreed safety margin. It is not sufficient to design to the limit hoping the errors will cancel.

4. UNCONTROLLABLE ERROR

Minimising error in any acoustic calculation requires great precision in the input data. Unfortunately limitations to measurement techniques and time restrictions force certain areas to be rather imprecise.

Two of these areas are the transmission loss of built structures and atmospheric effects. Transmission loss data is published for one off specimens and over a limited frequency range. The frequency limitations are due to the physical limitations of the measurement facilities. The measurement of low frequency transmission losses requires large volume chambers but structural limitations and source power restricts chamber size for high frequency measurement. Measurements are limited to the 125Hz third octave band through to the 4KHz third octave band. The high and low frequency ends must be extrapolated to cover the entire auditory spectrum. In the high frequency end the typical noise source tends to fall away while the transmission loss either stabilises or continues to increase. At the low frequency end the situation becomes unpredictable. It is also rare for a laboratory measured "construction" to precisely match the field installed construction. The laboratory data must always be adapted for the field installation. This problem may diminish with recent developments in sound field measurement.

With atmospheric attenuation the available data is limited to the basic data available for weather prediction. Very little is available on temperature/humidity swings, on temperature profiles or local wind directions and velocity profiles. They are presented as independent variables not as interrelated functions suitable for acoustic calculations. Detailed radiosonale data is available but only for limited locations and specific times.

5. SUMMARY

Error is an integral part of any measurement or calculation. We should be aware of errors and strive to control them. To the academic, error control is important for scientific integrity and credibility. To the engineer error control is important for quality assurance and cost optimisation. On a large project it is possible for the cost of noise control to be measured in hundred of thousands of dollars per dB and for rectification to measure in the order of millions of dollars. As part of that process we must be sure of the techniques being used to estimate error and their limitations.

Many of the techniques currently used are inappropriate for acoustics; their use represents a misapplication of techniques from statistics. This would indicate a fundamental lack of understanding of both acoustic theory and statistical theory. It should be understood that many of our acoustic terms are given credence by convention rather than having any basis in mathematics. In some areas error cannot be controlled. We should be aware of those areas and their potential impact. It is not sufficient to plead ignorance when a project fails to meet criteria.

AN EVALUATION OF NOISE IMPACT STATEMENTS

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SUMMARY

The paper reviews the requirements for noise impact statements and attempts to evaluate the usefulness of these statements. Some information on noise impact statements produced in NSW is presented and a discussion of ways to improve the value of noise impact statements is given, together with some alternatives to them.

1. INTRODUCTION

Noise Impact Statements are required as separate entities for developments such as Church Halls or air conditioning units or, more commonly, they are part of an Environmental Impact Statement for developments such as mines, heliports and industrial premises.

There have been a number of publications reviewing the effectiveness of environmental impact statements eg. [1] [2] [3] but none, as far as we are aware, specifically looks at the efficacy of noise impact statements. Environmental Impact Legislation has been in force for nearly a decade in NSW and it seems appropriate to review its effectiveness in ensuring a satisfactory aural environment.

Firstly let us consider what is required of an Environmental Impact Statement. An E.I.S. is required for certain prescribed activities and for activities which are likely to affect the environment significantly. An E.I.S. however is not required for proposed legislation or developments or activities which occur under special legislation and so major developments, such as that in Darling Harbour, can be undertaken without the need for an E.I.S. A lot of noise has been generated by proposed changes to the Education Act. This has not been the subject of an EIS either.

An E.I.S. sets out the environmental implications of a proposed project. It helps the developer plan an economically and environmentally sound project and provides the decision-making authorities with data for making well-informed decisions.

In N.S.W. the following aspects shall be included in an E.I.S. under the Environmental Planning and Assessment Act, 1980:

(a) A full description of the proposed development/activity
(b) A statement of the objectives of the proposed development.
(c) A full description of the existing environment likely to be affected by the proposed development, if carried out,
(d) An identification and analysis of the likely environmental interactions between the proposed development and the environment.
(e) An analysis of the likely environmental impacts or consequences of carrying out the proposed development.
(f) A justification of the proposed development in terms of environmental, economic and social considerations.
(g) Measures to be taken in conjunction with the proposed development to protect the environment and an assessment of the likely effectiveness of those measures.
(h) Any feasible alternatives to the carrying out of the proposed development and reasons for choosing the latter.
(i) Consequences of not carrying out the proposed development.

The State Pollution Control Commission, NSW requires that a pollution control approval be obtained for any new development and any changes to premises scheduled by the Noise Control ct, 1975. The application for approval is required to be supported by an EIS. The Commission, in its Environmental Noise Control Manual specifies the information to be provided in the Noise Impact Statement. In particular information is required concerning the site (plan, zoning,etc.), plan of the development, topography, proposed times of operation, existing background noise levels, proposed noise control measures, noise level predictions and the assessment of the noise impact.

Canter [1] includes a further requirement for U.S. Draft Environmental Impact Statements: an indication of what other interests and considerations are thought to offset the adverse environmental effects of the proposed action. The exclusion of this requirement in the NSW legislation is significant as it suggests that one environmental consideration cannot be compensated for by another and that each environmental consideration must be treated on its own. It also suggests, as a consequence, that each environmental aspect must conform with a given criterion.

As all developments must comply, when built, with the State Pollution Control Commission's Clean Waters, Clean Air and Noise Control and other Acts there may seem to be little point in requiring E.I.S.'s to cover these aspects of a proposed development. The argument we have seen to counter the suggestion that there is no need for noise to be covered in EIS's was contained in reference [2]. The argument ran that the Noise Control Act can prohibit a noise from continuing whereas an EIS could prevent the noise from happening in the first place.

A better justification would be based on the difficulty of predicting resident reaction to noise on the basis of any noise measure [4]. This argument though might cause difficulties in justifying sections of the Noise Control Act and Regulations under the Act. A Noise Impact Statement is also a means of gathering information about background noise levels which the regulating authorities
do not have but need in order to assess environmental impacts and to plan future activities. This and other general issues need to be addressed but the aim of this paper is to discuss more specific issues.

Unfortunately it is very difficult to treat rigorously many of the issues to be raised. There is very little information available and, to be realistic, it is unlikely that information will become available unless there are some very serious underestimates of the noise from developments.

As a consequence this paper is very much a discussion paper based on a review of some Noise Impact Statements, questionnaire returns from Consultants involved in producing NIS's, some first hand experience and information and comments from Local Councils.

By way of introduction it is also worth stating that it is fairly easy to be critical of the way NIS's are prepared and assessed though it is much more difficult to improve these processes.

2. QUESTIONNAIRE RETURNS FROM CONSULTANTS

A Noise Impact Statement questionnaire was distributed to organizations who were thought to be involved in the preparation of Noise Impact Statements. Seven completed questionnaires were returned. The answers to the questions cannot be dealt with in any sophisticated fashion but there are some answers which are both interesting and informative.

The questionnaire asked about the number of Noise Impact Statements or noise sections of Environmental Impact Statements prepared in the last year. The number of statements range from 6 to 120 with an average of about 50. The cost of producing an NIS ranged from $300 to $120,000 with a median value of approximately $3000. Assuming that the non-respondents to the questionnaire have similar profiles to those who did respond, the Noise Impact Statement business in NSW is worth well over $\$2 \times 10^6$ per year. NIS work forms on average, about 10-15% of the total cost of producing an E.I.S.

Limited information came from a question concerning the types of activity that NIS's were prepared for. One respondent considered that this information was confidential however it does seem that the respondents have their speciality areas and that the construction industry is the greatest source of N.I.S. work. Of the responses obtained NIS's were prepared for the following:

Construction	26%	Mines/quarrying	8%
Manufacturing	22%	Domestic	7%
Other	17%	Heliports	3%
Commercial	15%	Open Air Concerts	1%
		Motor racing	1%

A further question inquired about the cost of noise control measures. There were fewer responses to this question because costs were, in some cases, unknown. The highest maximum cost of noise control measures was $5 million and the lowest cost was, not surprisingly, nil. A median cost would seem to be somewhere between $1000 and $100,000.

Information was also sought on the hours of operation of developments or activities. The median number of operating hours was about 12 with a minimum of 2 and a maximum of 24 hours.

More interesting are the answers concerning the difficulties in producing an N.I.S. and the most likely sources of error in noise level predictions. The most commonly cited difficulty was in obtaining sound power data for machinery and equipment. It seems that such data is often not available from manufacturers, that similar equipment is not available to make measurements on and that the proponent either misleads consultants as to the equipment to be used, at the time of preparation of the N.I.S., or does not know what equipment will be used.

Another factor cited was concerned with the criteria used to judge whether noise was acceptable or not, especially for impact noise. There was also a comment about inconsistent interpretation of the SPCC's Noise Control Manual by the Land & Environment Court which also caused difficulties in producing N.I.S.'s.

A major factor given as a source of error was geophysical effects: predicting the influence of terrain and barriers of wind and temperature gradients. Despite these uncertainties the median value of the estimate of the accuracy of predictions was $\leq \pm$ 3dB(A) in five of the seven responses with $\pm$ 1dB(A) cited by one respondent. Of the other two respondents one said it was not possible to estimate the accuracy. The other respondent indicated that for distances at less than 100m the predictions were better than $\pm$3dB(A) but for larger distances the errors would be larger.

As the percentage of cases where the predictions are checked with measurements is fairly small (averaging about 15% of the cases for which N.I.S.'s were produced) the above estimates of accuracy will be based on relatively few cases. Another limitation of these estimates of errors is that, even in cases where confirmatory measurements are made, these measurements would be made over a limited time and hence under a limited range of atmospheric conditions favourable for making measurements.

The final question was about what percentage of the total work of respondents was concerned with Noise Impact Statements. N.I.S.'s averaged about 30% of the respondent's work with a range of from 2 to 70%.

3. INFORMATION FROM THE SPCC & LOCAL COUNCILS

Noise impact statements may be required by either the State Pollution Control Comission or Local Governments. In 1986 the total number of applications for pollution approval received by the SPCC was 465. Of these 230 required noise control conditions. In 1987 the number of applications was down to 382 and of the 185 required noise control conditions.

An examination of the applications received in 1985 by the SPCC was made. What is immediately apparent from these applications accompanied by NIS's is that very few of the NIS's present the methodologies, calculations and assumptions used in arriving at sound level estimates. Although many applications were approved, with the condition of a compliance check on levels, no compliance check had been made. It appears that about 50% of statements are accepted without modification.

Fourteen of the 39 local councils in the Sydney Metropolitan Area were questioned about Noise Impact Statements. The number of NIS's varies greatly (0-70 per year) with an average of about 15 per year. There are obviously very different attitudes to the NIS's in Councils. Most of Councils had not refused a D.A. on the grounds of noise. In the municipality receiving most NIS's about 70% were accepted without modification, 20% needed additional information and 10% were rejected because of unsatisfactory or misleading information.

One comment was made that NIS's were not necessary because the Health Surveyor would set the criteria to be complied with.

4. DISCUSSION

The preparation of Noise Impact Statements and the noise control work associated with them is worth many millions of dollars a year in NSW. The question we set out to answer was whether this money was well spent? We are no nearer an answer to that than when we started but some issues have arisen which may make planning for noise and more efficient, equitable and accurate exercise. These issues are outlined below:

1. There is an obvious need for a source sound power database. Such databases exist in at least a crude form elsewhere (see Fig. 1). Should a more comprehensive database be prepared it should be possible to extrapolate this information, based on factors such as the power output or usage, type of power source, speed, purpose for which it is used etc. to predict accurately the sound power output of equipment for which data is not available. The preparation of such a database should be considered a high priority. The Australian Environment Council has agreed to develop such a database, nationally together with a database on noise control measures.

2. It would seem impossible to carry out an objective assessment of the value of Noise Impact Statements unless a control group of developments were allowed to proceed without the preparation of NIS's. In the case of the control group it would probably be necessary to specifiy noise criteria to be met. In theory it should not be necessary to present an NIS to a determining authority where known noise criteria must be complied with. In practice however the production of an NIS may ensure better acoustic design and closer compliance with the noise criteria specified.

3. Compliance with the noise conditions set out in Development Applications approvals was rarely checked before 1985.* The accuracy of noise level predictions in N.I.S.'s is also rarely checked. What is also important besides compliance checks is that cases of overdesign are also investigated. Checking predicted levels is fundamental to improving prediction techniques. If all developments cannot be investigated then a random sample of them should be taken.

4. Inherent in the concept of an Environmental Impact Statement is the investigation of alternative methods of achieving the required result. Alternative methods of controlling noise are noticeably missing from Noise Impact Statements though the SPCC has a policy of accepting "the best practical means" of achieving noise control. Also missing is the concept of "compensation". If the noise environment of an area will unavoidably deteriorate because of a development, can this deterioration be compensated for in other ways, eg. by improvement of the cultural environment of the area?

5. Determining appropriate environmental noise quality objectives is important. At present there is little room for negotiation; the noise criteria to be used are predetermined. Social surveys of residents around similar developments elsewhere should be an acceptable alternative method of investigating the impact of a development. Other alternatives to sound level predictions of resident responses need to be investigated as noise level methods are notoriously poor and social surveys are notoriously expensive. The Delphi Technique may be one such alternative.

6. There is a need for more objectivity in Environmental Impact Statements. We have yet to read an E.I.S. which did not support the proponent's case and we have yet to read a criticism of an EIS which did not seriously question the validity of the E.I.S. In particular the practice of measuring the background level on one day for about 10 minutes is open to abuse. Perhaps a database of environmental noise levels is also necessary. Loose wording such as, "The majority of operatives will conform to the environmental criteria", is not helpful even though, because of wind and temperature variations, it is not possible to determine sound levels with the degree of certainty that environmental criteria appear to require.

* Since late 1985 the SPCC has required a "certificate of compliance with Pollution Control Approval" which means that all developments are now compliance checked.

Could Noise Impact Statements be produced by an independent organization financed by proponents? Alternatively should proponents be required to lodge a sum of money, worth say 20% of the cost of the project, with the local council before the commencement of the project. If the project does not meet the council's requirements and criteria, including noise, part or all of this money could be withheld from the proponent. This procedure is already used in some European countries. Another alternative would be to introduce tax incentives for the purchase of quiet equipment, as is done in the Netherlands.

The support for these alternatives comes from the fact that most Noise Impact Statements are accepted with little or no modification, that noise criteria to be complied with already exist, that the proposal, when implemented, must comply with the Noise Control Act, and that many proposals never get past the planning stage.

7. Require Environmental Impact Statements to be prepared for any proposed legislation.

8. In most Environmental Impact Statements the method of arriving at noise level predictions is not given. It would seem important that assumptions are stated and that calculation procedures are at least outlined.

9. Not only do noise levels need to be monitored and compared with predictions but the reactions of residents need to be monitored also. This would be a more realistic way of obtaining improved noise criteria than undertaking major surveys such as those by Hede & Bullen [5] & Tracor [6].

10. Greater justification of proposals, more consideration of alternatives and the consequences of not carrying out the proposal should be given. Indirect effects of proposed developments should be more closely monitored. For example it is only the noise in the immediate vicinity of a proposed heliport which is considered. New heliports will create extra helicopter traffic on preferred routes which will have an effect on residents well away from the heliport.

The only conclusion that can be reached is that there has been an improvement in consideration of the noise environment since the introduction of Noise Impact Statements and there has almost certainly been better planning for noise control but further improvements, could be made and alternative methods of obtaining these improvements can certainly be thought of.

ACKNOWLEDGEMENTS

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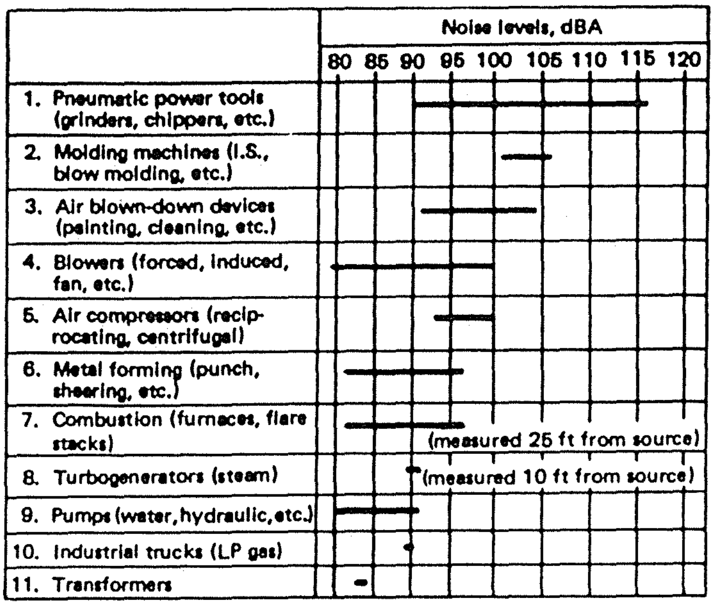
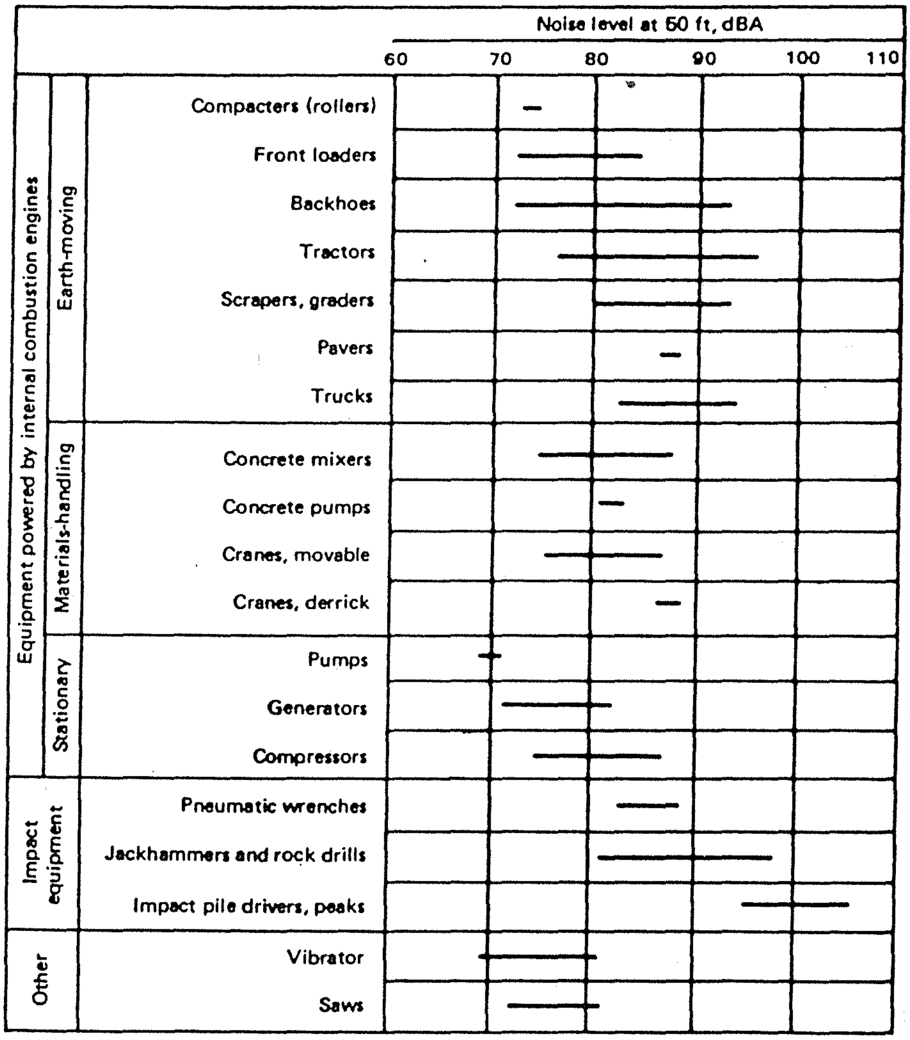


*Measured at operator positions, except for 7 and 8.



Note: Based on limited available data samples.

Fig 1. Equipment Noise Levels
(from Canter) [1])

SOUND PROPAGATION IN A REFRACTING ATMOSPHERE

by

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The effect of wind and temperature gradients on the propagation of outdoor sound has been investigated using acoustic impulses. Such gradients cause the sound to follow curved ray paths and thus effect the time delay between direct and reflected components. Under certain conditions a meteorological shadow boundary is formed into which no simple direct or reflected sound may penetrate. Inside this shadow zone a marked reduction in sound level is observed. Predictions using ray bending models above the shadow zone and creeping wave theory inside the shadow zone are discussed and compared with experimental impulse results.

INTRODUCTION:

Both wind speed and temperature vary with height close to the ground producing a vertical sound speed gradient. Under such conditions sound follows curved ray paths, altering the expected levels relative to those predicted in a neutral atmosphere where no sound speed gradient exists. When propagating upwind, or in the presence of a dominant temperature gradient which decreases with height, the sound is refracted upwards producing a shadow zone into which no simple direct or reflected sound may penetrate.

The actual ray paths depend upon the nature of the gradient. A simple linear gradient results in circular paths: the radius of curvature depending upon the magnitude of the sound speed gradient. The ray which just grazes the ground forms the shadow boundary, ray B in Fig. 1(a). In most outdoor situations the gradients tend to be non-linear with a much larger gradient existing close to the ground. Under such conditions, sound rays cross the one which grazes the ground, ray A in Fig. 1(b), so that the effective shadow boundary becomes the envelope of rays which have been refracted upwards, indicated by B in Fig. 1(b). When the boundary is a caustic, no sound follows this path, however, it can be considered as an effective limiting ray.

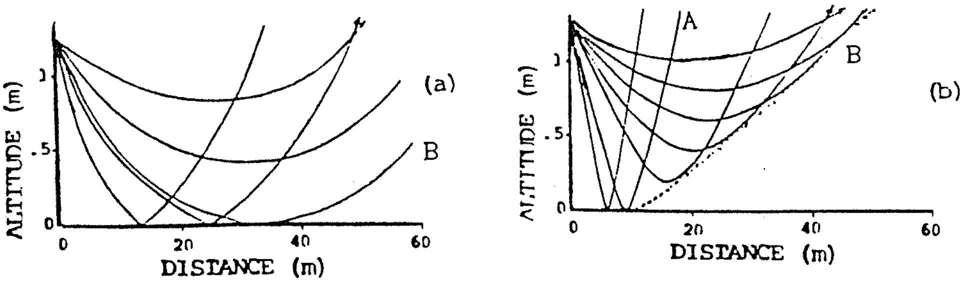


Fig. 1 Ray paths in (a) linear gradient and
(b) a non-linear gradient.

Refraction will change the relative path lengths of the direct and reflected rays and hence the delay between these components reaching a receiver. The angle of incidence of the reflected ray is also effected, altering the phase change upon reflection.

For the case of a linear sound speed gradient, the differential equations developed by Thompson[1] have been used to calculate the travel-times and angle of reflection. Although slightly different paths are predicted for the case of a temperature gradient and those produced in a moving medium by a wind gradient, for the conditions in this work the predicted pulse shapes and amplitudes were essentially equivalent.

Ray models only allow predictions above the shadow boundary. At greater distances, a more sophisticated wave approach called creeping wave theory[2] is necessary to describe sound diffraction into the shadow zone. Predictions from both theoretical approaches will now be compared with experimental results taken over grassland.

PRIOR TO THE SHADOW ZONE:

This investigation used the acoustic impulses produced by the discharge of a 22 Hornet blank cartridge. Each data point represents the ensemble average of at least 10 individual shots, a process which effectively eliminates the minor fluctuations in waveform produced by turbulence. The capture and analysis system has been detailed earlier[3]. All waveforms shown have been scaled to remove differences in peak level for ease of shape comparison. Their relative sizes can be deduced from the corresponding excess attenuation graphs.

Theoretical waveforms were generated by frequency analysing an experimental reference impulse obtained at 2m from the source, appropriately modifying the components and reconstituting the pulse using an inverse DFT as detailed earlier[3]. In the case of the ray bending predictions this essentially means allowing for the delay between direct and reflected components and for spherical wave reflection through the image strength $Q = Rp + (1-Rp)F(w)$, the latter depending upon the complement of the angle of incidence, ψ, and the ground impedance. In the image strength, the term Rp corresponds to the specular reflection while the second term is usually refered to as the ground wave.

A. ATTENUATION AND RAY BENDING PREDICTIONS:

Fig. 2(a) presents a set of experimental results obtained over grassland in the upwind direction for a source-receiver height of 1.25 m. The ground impedance was measured and found to be consistent with an effective flow resistivity of $\sigma_E = 300$ cgs rayls. The experimental results follow the inverse square law prediction out to a distance of around 30m after which an excess attenuation is observed. It was originally thought that the onset of excess attentuation corresponded to the location of the shadow boundary, however, we now beleive some of the initial excess attenuation is a result of the curved ray paths.

As indicated by curve A in Fig. 2(a), in a neutral atmosphere, excess attenuation is not predicted until around 100m and the attenuation can be calculated at any distance as there is no shadow boundary. When a linear sound speed gradient is present, curve B indicates that excess attenuation occurs closer to the source, although the curve is discontinuous as the model is limited by the shadow boundary.

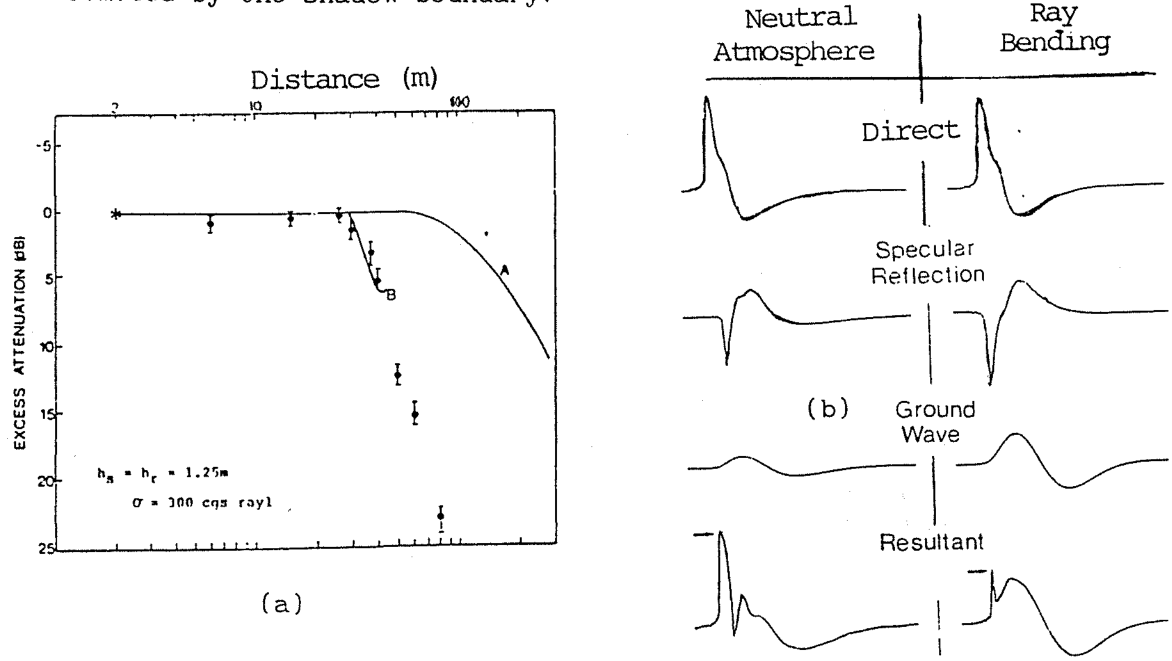


Fig.2 (a) Experimental data compared to values calculated assuming (A) neutral atmosphere theory, and (B) a linear sound speed gradient.
(b) Individual components and resultant impulse predicted for neutral atmosphere and linear gradient.

The reason for the earlier excess attenuation in the presence of a sound speed gradient can be explained by comparing the individual components which constitute the resultant impulse, as indicated in Fig. 2(b) for both a neutral atmosphere and linear gradient case. In the latter situation, the reduced delay allows the inverted specularly reflected impulse to partially cancel the main peak of the direct impulse. The decrease in the angle ψ also causes greater inversion of the specularly reflected impulse which aids this cancellation and increases the ground wave amplitude.
The residue of the direct peak is shown arrowed in the resultant impulse and should be compared with the uneffected peak predicted in a neutral atmosphere. Changing the value of the linear sound speed gradient will move the position of the shadow boundary but will not alter the above ideas.
Thus the presence of the sharp-rise time of the direct impulse provides a method of determining the location of the effective shadow boundary. Fig. 3 shows experimental impulse waveforms measured at various distances from the source and it is apparent, as indicated by the arrows, that the residue of the direct impulse peak is reduced as the shadow boudary is approached. In fact the ground wave component is the dominant term close to the shadow boundary. If the measured waveform possesses some residue of the direct impulse, indicated either by a small spike or the rapid

rise time of the direct impulse, then the receiver is not inside the shadow zone. Once the measured signal makes the transition to the much more rounded impulse with the relatively long rise-time, the receiver must be very close to or inside the shadow zone.

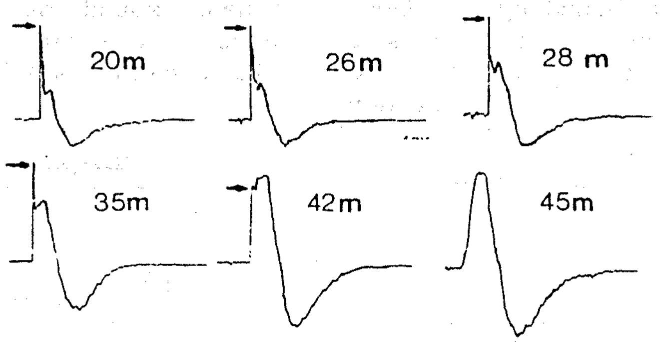


Fig.3 Impulse waveforms measured at various distances from the source.

Based on experimental rise times, the data of Fig. 2(a). indicates a shadow boundary at 45m. This corresponds to a linear sound speed gradient of 1.6s^{-1}. This value was used to calculate curve B in Fig. 2(a) and it is in good agreement with the experimental excess attenuation. Furthermore, experimental and theoretical pulse shapes shown in Fig. 4, at various distances before the shadow boundary, are also in excellent agreement.

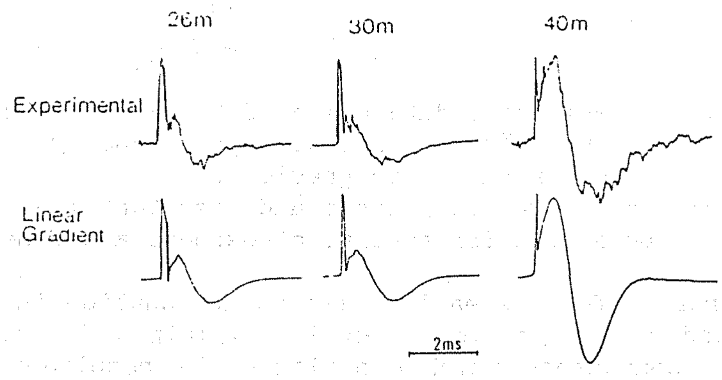


Fig. 4 Comparision of measured and calculated pulse shapes.

B: PROFILE OF THE SHADOW BOUNDARY:

By using the method of impulse rise-times to locate the shadow boundary it is possible to plot the boundary profile by taking results at different receiver heights for a given source height. A rig supporting three microphones was placed at various distances from the source, permitting pulses to be recorded simultaneously at three receiver heights for a given source height. By repeating measurements at three different source heights nine source-receiver geometries were investigated which enables the downward as well as the upward leg of the effective limiting ray to be plotted.

The profile obtained from a series of such measurements is plotted in Fig. 5. Assuming an uncertainty of ± 2m in the location of the boundary, there is excellent agreement between the data points. Although the profile is symmetric, which is consistent with a linear gradient model, it is not circular suggesting a non-linear gradient.

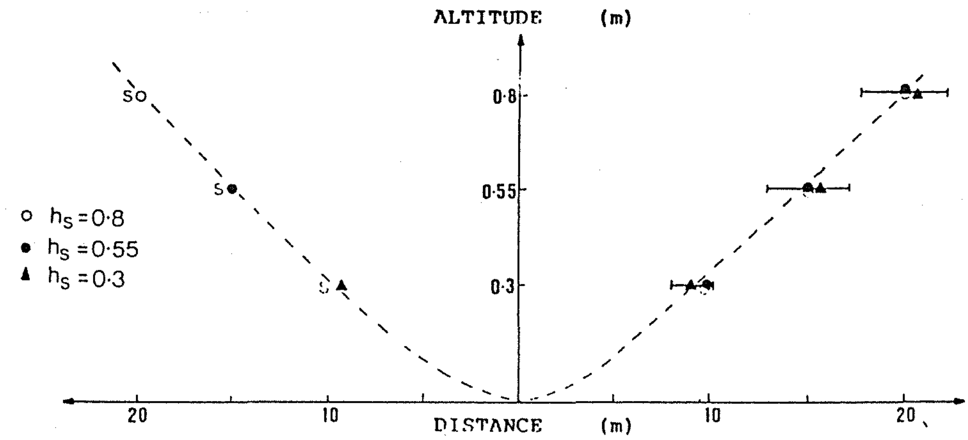


Fig. 5 Profile of the effective limiting ray.

CREEPING WAVE CALCULATIONS:

In contrast to the ray models, the creeping wave theory can be applied inside the shadow zone. Based on the diffraction of sound waves, this theory envisages sound entering the shadow zone by following the limiting ray from the source down to the ground and then creeping along the ground, Fig.6.

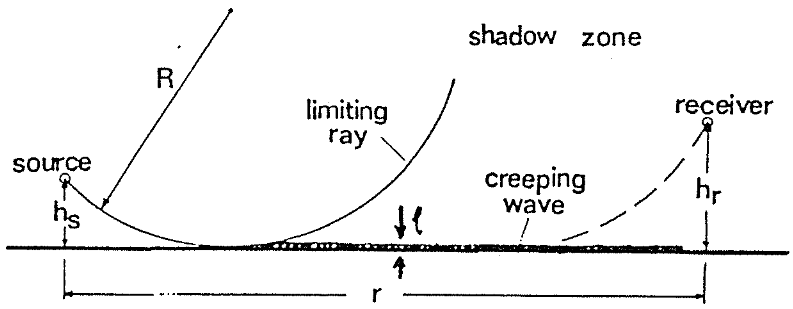


Fig.6 Formation of a shadow zone in a linear sound speed gradient.

As the wave propagates along the ground, energy is continuously shed up into the medium. In earlier treatments[4] the theory was restricted to receiver heights above a wavelength depedent height l, a criterion diffucult to satisfy when using impulses and thus relatively poor agreement was obtained. More rigorous treatments remove this assumption and has provided significant improvement over the earlier attempts, although discrepancies still existed[5]. A more recent version of the theory has been reported by Berry and Daigle[6], which can be applied above the shadow boundary as well as inside the shadow zone although the equations still assume a linear sound speed gradient. For the creeping wave calculations the Fourier components of the direct impulse were modified by complex co-efficients which depend upon ground impedance and the magnitude of the sound speed gradient. For a receiver located at a height h_s and a distance r away from a source at a height h_s above ground with an impedance Z_2, the complex coefficient at the angular frequency ω is given by

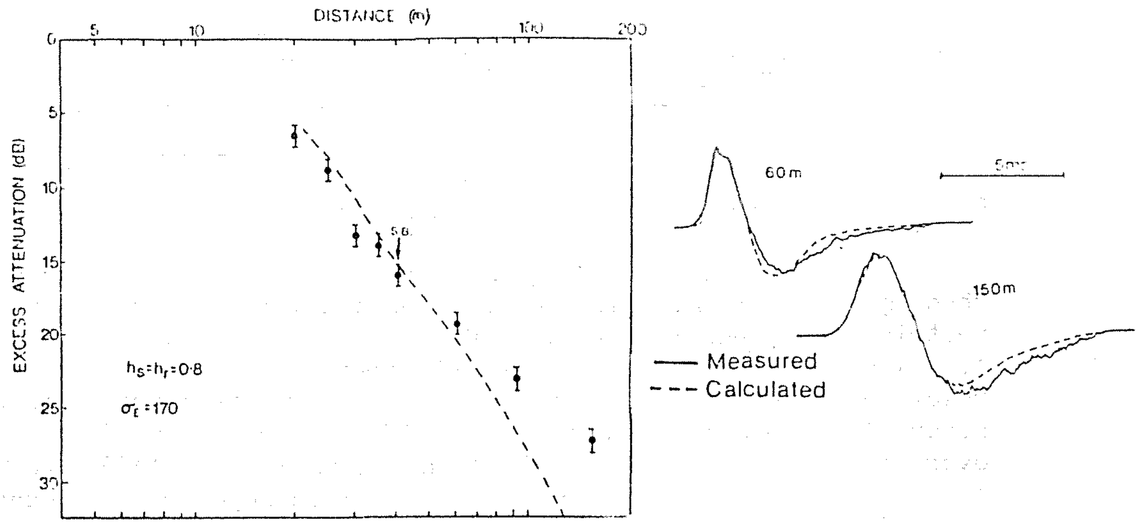
$$\frac{r\, e^{(i\pi/6)}}{e^{(i\omega r)}} \sum_n \frac{H_0^{(1)}(k_n r)\; Ai[b_n-(h_s/l)e^{2i\pi/3}]\; Ai[b_n-(h_r/l)e^{2i\pi/3}]}{[Ai'(b_n)]^2 - b_n[Ai(b_n)]^2} \qquad (1)$$

where $H_0^{(1)}$ is a Hankel function of the first kind and of order zero and Ai represent Airy functions. The ground impedance enters through the variable $q = ik_0 l Z_1/Z_2$ where l depends on the radius of curvature $l = (R/2k_0^2)^{1/3}$ and $b_n = (k_n^2-k_0^2)l^2\exp(i2\pi/3)$ are the roots of the equation

$$Ai'(b_n) + q[\exp(i\pi/3)]\; Ai(b_n) = 0 \qquad (2)$$

This differential equation must be solved at each frequency for b_n and the value substituted into Eq.(1).

Fig. 7(a) presents a set of results taken for a source-receiver height of 0.8m. The shadow boundary determined from impulse rise times was at 40m, which corresponds to a linear sound speed gradient of $1.4s^{-1}$. The creeping wave prediction is shown as the dashed line in this figure. Prior to the shadow boundary, the predicted levels and pulse shapes agree well with measurements and are essentially identical with those calculated in this region using the ray tracing technique. Thus under these conditions the more sophisticated wave theory does not offer any advantages over the simpler ray treatment, in fact it is difficult to get creeping wave calculations to converge as the receiver approaches the source.



(a) (b)

Fig.7 (a) Experimental results and --- creeping wave prediction.
(b) Comparision of measured and calculated pulse shapes.

Inside the shadow zone, the creeping wave predictions initially are in excellent agreement with the experimental pulse amplitudes, although well inside the shadow zone the levels are under-predicted. Fig. 7(b) contrasts experimental and theoretical waveforms inside the shadow zone, where it can be seen that there is excellent pulse shape agreement even when the level is under predicted.

Despite the above agreement, it should be noted that the measured sound speed gradient was not $1.4s^{-1}$ but around $4s^{-1}$. This strong gradient would move the shadow boundary, and hence the onset of excess attenuation, closer to the source. Similarly, the gradient choosen in Fig. 2(a) was $1.6s^{-1}$ where in practice, the gradient was non-linear with the average gradient being around $3ms^{-1}$. Thus although a linear gradient can be chosen to fit the data it does not correlate with the measured gradient, especially as the actual gradient is non-linear.

CONCLUSION:

Calculations based on a ray bending model have been found to be in good agreement with experimental results, both in pulse shape and amplitude, in the region prior to the shadow zone. The more sophisticated creeping wave theory provided similar agreement in this region. Inside the shadow zone the creeping wave theory offers good shape agreement at all distances investigated although it underpredicts the level at the larger distances. Other experimental results have been taken at various source-receiver heights up to 1.25m and over ground with a different impedance. The theories correctly allow for such changes and similar agreement to that above is obtained. Despite this agreement the theories use a linear sound speed gradient which is in conflict with the actual non-linear gradient observed when taking the measurements. Furthermore the experimentally determined effective boundary profile is non-circular, implying a non-linear gradient. Thus an inconsistency has appeared. The waveforms and excess attenuation measurements can be largely predicted using the assumption of a linear gradient chosen to fit the data, however, this does not correlate with the measured gradients.

ACKNOWLEDGEMENTS:

The authors acknowledge support from the Australian Research Grants Scheme.

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THE QUIET HOUSE - FRONT LINE DEFENCE AGAINST TRAFFIC NOISE

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Introduction

Traffic noise is a major environmental problem in urban and rural areas. Over one million people in Australia are exposed to levels of traffic noise above OECD criteria. Increasing private and commercial vehicle usage are continuing to contribute to a growing problem that needs to be addressed, now, before the social, environmental and economic impacts become even greater than at present.

While emission controls on vehicles are providing an amount of protection and will continue to be the foundation of traffic noise control programs, it is clear that other measures are needed. These alternatives are termed non-vehicular controls and include measures such as noise barriers, treatment of road surfaces, planning and development controls and better housing design.

Housing design can play an important part in creating acceptable indoor acoustical environments. The Quiet House is designed to protect its inhabitants from traffic noise. The house was constructed this year jointly by the New South Wales State Pollution Control Commission and the Department of Housing with the aim of promoting the benefits of certain design features in houses to be located near busy roads. The design was selected from over a hundred entries in a competition held by the Commission. The winner was a Sydney architect, Mr Geoffrey Le Sueur. Figure 1 is a sketch of the house and Figure 2 is a plan with side elevations. Figure 3 is a copy of Mr Le Sueur's winning entry in the competition.

Inadequacy of Existing Housing

Almost all housing adjacent to busy roads was designed without regard to the effects of traffic noise on inhabitants. Project homes are designed for universal use with some allowance for modifications depending on lot sizes and slopes. These houses are best suited for quiet residential areas with low traffic flows.

Existing project home designs tend to have noise sensitive rooms at the front when they should be at the rear. These rooms often have acoustically weak large front facing windows with air gaps and inadequate thickness of glass. External front facing doors are also a problem because of poor sealing and in some cases, a lack of door mass causes noise transmission. Considerable amounts of noise pass through eaves which have materials with inadequate mass and air gaps. Noise is also transmitted through the considerable air gaps in tiled rooves.

Home units and town houses, buildings that are mostly designed for specific sites, rarely have features that control traffic noise intrusion. Large windows and noise sensitive rooms are usually nearest the street, little use is made of barrier walls either at the street frontage or incorporated into the building structure. Home units are built with balconies that reflect noise from the undersides towards the windows of the units below.

Older dwellings, constructed in days before the mass use of motor vehicles, are generally closer to roads and consequently more exposed to road traffic noise.

To be effective in controlling traffic noise, dwellings require special attention during construction and the builder or project supervisor need to know of the importance of reducing noise paths and the noise control effectiveness of various grades of materials.

Broader Importance of the Quiet House Concept

As urban areas spread, the cost of providing transport and service infrustructures increases. This is particularly important in Sydney where the 1/4 acre block has become accepted by home buyers as the traditional minimum size for housing. The solution proposed by planners is urban consolidation, an approach which involves the construction of higher density accommodation in the form of home units, town houses or villas.

If dwellings of the future are to be built on smaller lots and buffer zones become too costly, then designs like the Quiet House are the only feasible options.

Location of the House

The house is located adjacent to extremely busy Pennant Hills Road at North Parramatta, approximately 25km west of the Sydney CBD. Pennant Hills Road is a major connecting route between western Sydney and the northbound Pacific Highway and carries a considerable amount of heavy vehicle traffic. Weekday traffic is of the order of 16,500 vehicles over a 24 hour period.

Noise Control Features of the Quiet House

One of the aims of the project was to use conventional and easily obtained materials as much as possible rather than special acoustical materials. The Quiet House achieves its objective using design and construction features. The house is approximately 17 squares, three bedrooms, living and dining areas, kitchen, bathroom, family room, laundry and double garage. Bedrooms are at the rear away from noise and utility rooms such as the kitchen, garage and laundry are at the front. The bedrooms and living areas are separated, as shown on the plan, to provide an effective amount of household noise control. A door between the these areas isolates bedrooms from frequently noisy locations.

Two double brick walls are used at the front as noise barriers to shield internal spaces. The front solid core door is fitted with acoustic seals along all sides.

A flat roof helps to reduce noise and two brick walls which encapsulate the bedrooms and living areas provide a substantial shielding barrier. The roof incorporates sound absorbing materials and a heavier gauge metal steel decking.

Measuring the Noise Control Effectiveness of the House

At the time of writing this paper, the house not finished, some windows had not been installed and carpets and other sound absorbing materials had not been fitted. A few measurements were taken to assess the performance of the house to this level of completion.

The measurement descriptor used was Leq, taken over a short representative period of time during daylight hours. At the front of the house next to the garage opening the Leq of the traffic noise was 76dB(A). The Leq measurement taken in one of the bedrooms (nearest the car park) was 37dB(A). During the latter measurement, the rear-facing door of the bedroom was partially open so some further reduction would be expected with the door closed.

A program of noise measurement is to be carried out when the house is completed and fully furnished using microphones located in living areas, bedrooms and at the front.

Promoting the Quiet House

The house will be evaluated in terms of its technical effectiveness prior to the official opening. But the real test of its effectivess will be in its ability to challenge and influence community attitudes to the design of houses. The result hopefully will be houses that fit our noisy environment in a more effective way.

The house will be open for inspection for about a year.

Acknowledgements

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Norbrik Bricks
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Stramit
Swanes Nursery
Philips
Caroma
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Australian Gas Light Company
Boral
Invicta Carpets
Pot-Porri Fabrics
Sebel Furniture
MLA Holdings

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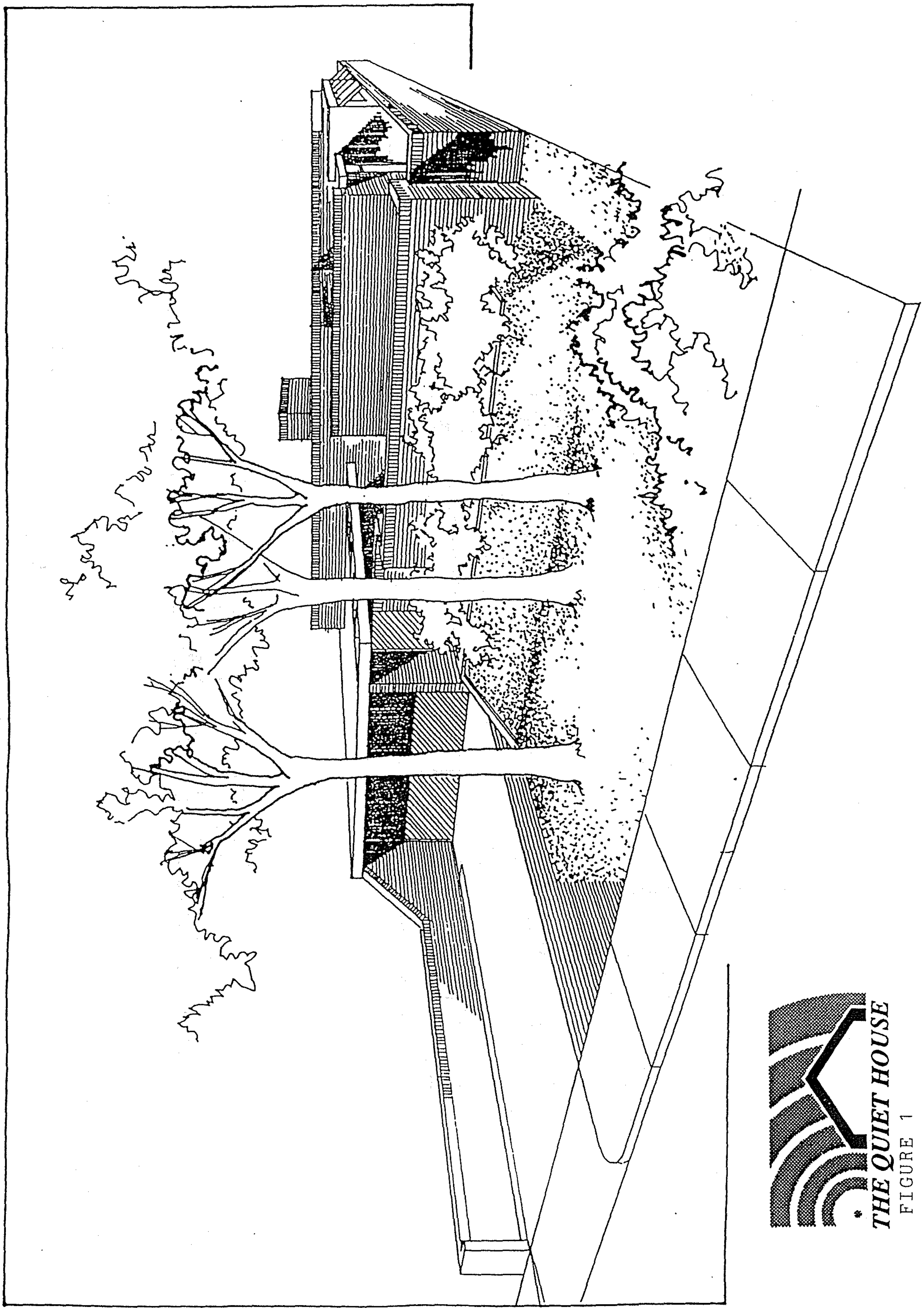


THE QUIET HOUSE
FIGURE 1

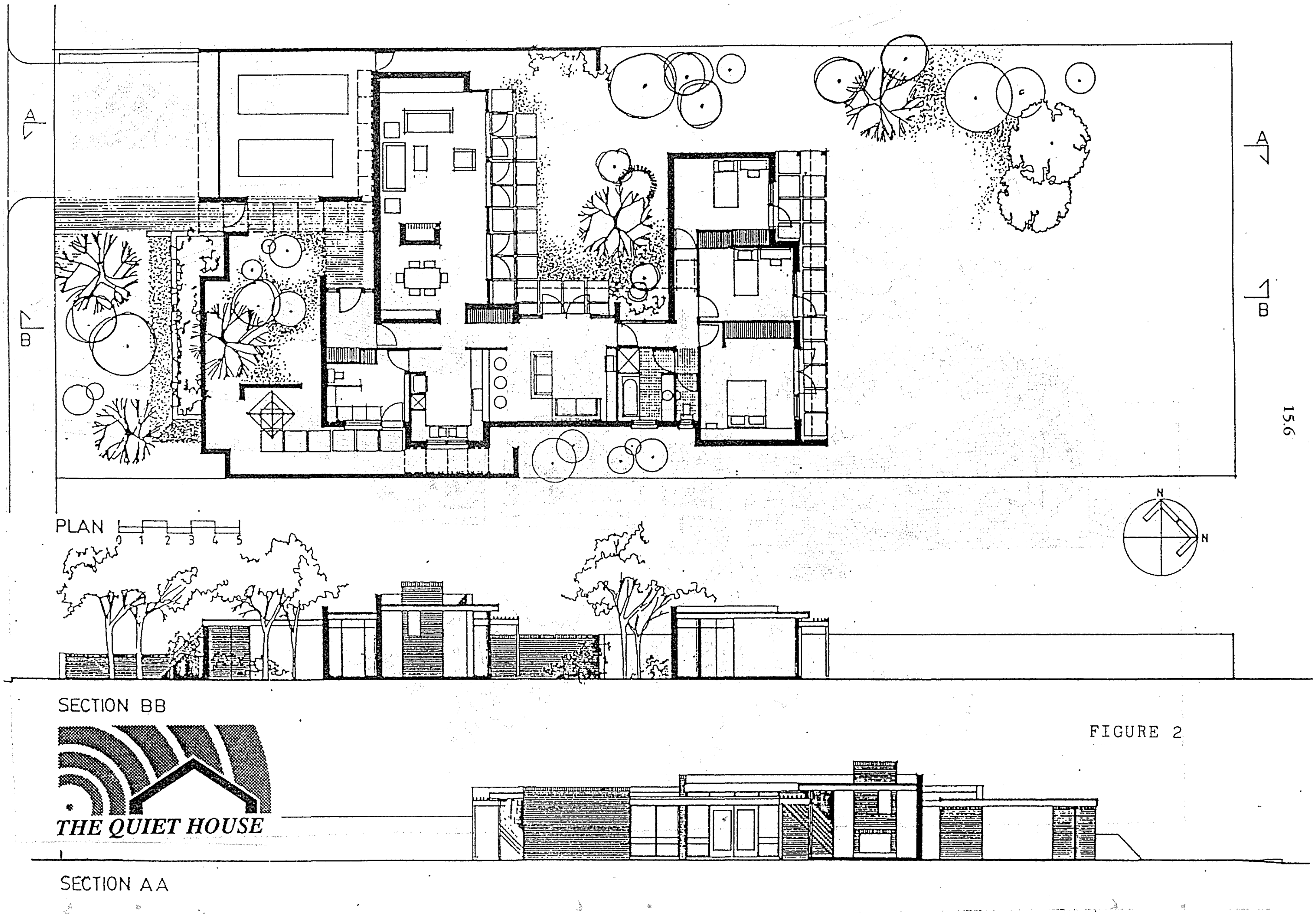
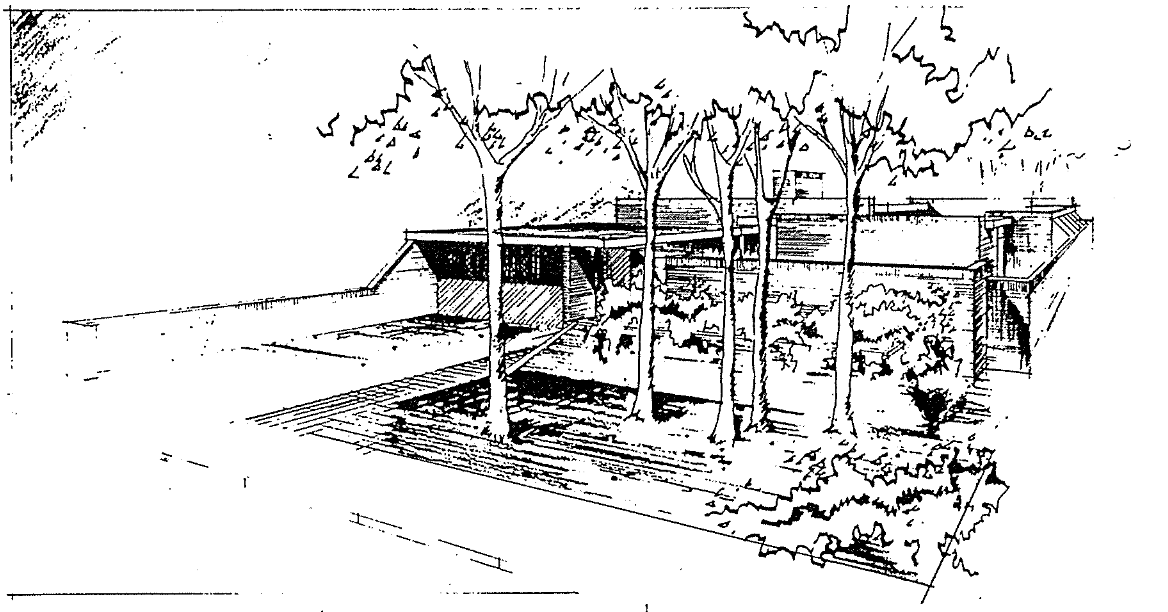
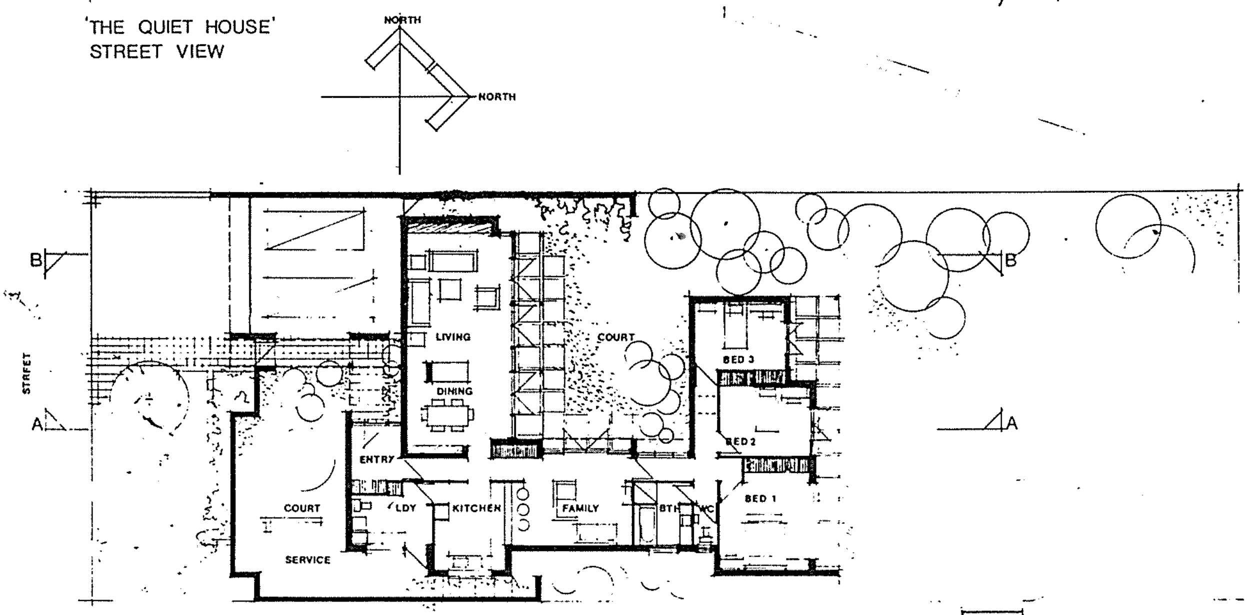


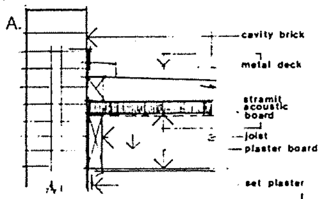
FIGURE 2



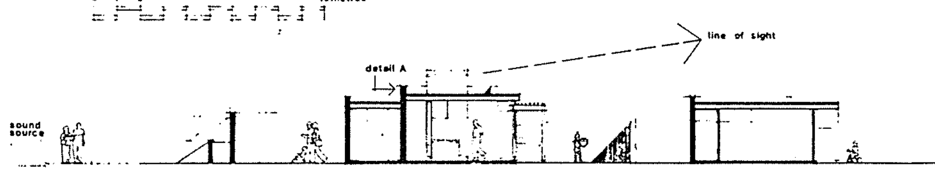
'THE QUIET HOUSE'
STREET VIEW



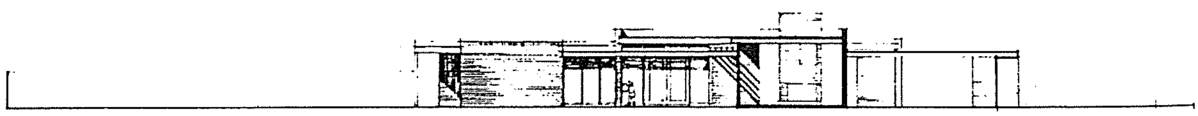
PLAN



DETAIL A.



SECTION AA



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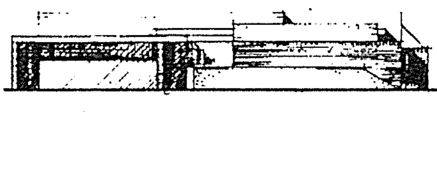

STREET ELEVATION

QUIET HOUSE
COMPETITION

THE QUIET HOUS

FIGURE 3

PREDICTION AND MEASUREMENT BREAKOUT NOISE FROM RECTANGULAR DUCTWORK

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1. INTRODUCTION

The accurate calculation of sound power levels emitted by the walls of air conditioning ductwork is of considerable interest to the mechanical services industry. Over estimation of sound power levels can result in unnecessary specification of expensive noise control measures, while under estimation can result in a system which is unacceptable to the customer and for which retrofit of noise control equipment is difficult and costly. At this point in time three prediction schemes [1], [2], [3] which give quite different results are in common usage. The purpose of the work described here was to determine which prediction scheme is most appropriate. This was done by experimentally determining duct internal and emitted sound power levels (and hence duct wall transmission loss) for a large duct section in the size range in common use in the air handling industry, and comparing the measured data with calculations made using the three prediction schemes. Tests were repeated for three different duct wall thicknesses which covered the range expected to be found in practice.

Modifications to the commonly used analytical prediction schemes to take into account a decreasing sound intensity within the duct as distance from the source increases (due to duct internal losses as well as sound radiated through the duct walls), are also discussed. These modifications are important if the ductwork is lined on the inside or if long lengths of unlined ductwork are involved.

2. OUTLINE OF CURRENT BREAKOUT PREDICTION SCHEMES FOR RECTANGULAR DUCTS

2.1 1984 ASHRAE scheme

The current ASHRAE prediction scheme [1] for the estimation of breakout sound power for rectangular ductwork is based upon a Bolt, Beranek and Newman report [4], which in turn is based primarily upon work by Cummings [5], [6], [7], [8] and [9]. The sound power radiated by the ductwork to its surroundings is given by [4]

$$PWL_o = PWL_i - TL_{out} + 10\ \mathrm{Log}_{10}\left[\frac{P\ell}{S}\right] + C\ \text{(dB)} \tag{1}$$

In the equation, PWL is the sound power level of the sound field propagating down the inside of the duct (PWL_i = cross sectional area S(m^2) multipled by the average sound intensity over the cross section), TL_{out} is the transmission loss (dB) of the duct walls, P is the cross sectional perimeter of the duct (m), ℓ is the duct length (m) and C is a correction factor to allow for gradually decreasing values of PWL_i as the distance from the sound source inside the duct increases. For short lengths of unlined duct (1-2m in length) C is usually small enough to ignore. For unlined ducts longer than 2m or any lined ducts, C can be estimated using [4]

$$C = 10\log_{10}\left[\frac{1 - e^{-(\tau+\beta)\ell}}{(\tau+\beta)\ell}\right] \qquad (2)$$

where β = $\Delta/4.34$
Δ = sound attenuation (dB/m) due to interior losses
= 0.1 dB/m for unlined ducts
(do not use tabulated values in [1] as these include losses due to breakout)
$\tau = \frac{P}{S}\, 10^{-TL/10}$

The current ASHRAE prediction scheme does not include the quantity C (which is always a negative number) of Equation (2) above, although it is included in [4] and is important for internally lined ductwork or long unlined ductwork.

The quantity TL_{out} may be calculated (according to ASHRAE [1]) using the following procedure. First of all the cross-over frequency from plane wave response to multimodal response is calculated using

$$f_{cr} = 612/(ab)^{1/2} \quad \text{(Hz)} \qquad (3)$$

where a is the larger and b the smaller duct cross sectional dimensions in metres.

At frequencies below f_{cr}, the quantity TL_{out} may be calculated using

$$TL_{out} = 10\log_{10}\left[\frac{f\, m^2}{(a+b)}\right] - 13 \text{ (dB)} \qquad (4)$$

At frequencies above f_{cr},

$$TL_{out} = 20\log_{10}(fm) - 45 \text{ (dB)} \qquad (5)$$

where m is the mass/unit area of the duct walls in kg/m^2 and
f is the frequency in Hz.

The minimum allowed value for TL_{out} is given by

$$TL_{out} = 10\log\left[\frac{P\ell}{S}\right] \text{ (dB)} \qquad (6)$$

The maximum allowed value for TL_{out} is 45 dB.

2.2 Mass law prediction

The mass law prediction scheme is outlined in many text books [2] and is generally found to give very conservative results, especially at low frequencies where the predictions often indicate that noise control is necessary when in fact it isn't. Essentially this scheme uses equation

(1) above (with the quantity C excluded) and uses either calculated or measured field incidence mass law transmission loss values for a flat panel of the same material as the duct wall.

In the absence of measured data, the following expression may be used for calculating the mass law transmission loss [10].

$$TL_{out} = 20\log_{10}[fm] - 48 \quad (dB) \qquad (7)$$

Values calculated using Equation (7) are applied over the whole frequency range of interest.

2.3 Modified mass law prediction

Users of the prediction scheme outlined 2.2 above have consistently found it advisable to modify the low frequency predictions to give less conservative results. One such modification scheme [3] is as follows.

First of all calculate the frequency at which higher order modes begin to propagate in the duct.

$$f_o = 343/2a \qquad (8)$$

where a is the largest duct cross sectional dimension.
At frequencies of $2f_o$ and above use Equation (7) to calculate TL_{out}.
At f_o, add 4dB to result calculated using Equation (7).
At $f_o/2$ add 8dB to result calculated using Equation (7).
At $f_o/4$ add 12dB to result calculated using Equation (7).
Interpolate for intermediate frequencies using a straight line on a log-frequency plot.

2.4 Frequency range limitations

The above-mentioned prediction schemes are limited in their application to frequencies between 1.5 times the fundamental duct wall resonance frequency and half the critical frequency of a flat panel equal in thickness to the duct wall.

In most cases the fundamental duct wall resonance frequency is well below the frequency range of interest and can be ignored. If this is not the case it may be calculated [8] and the transmission loss for the third octave frequency bands adjacent to and including f_o should be reduced by 5dB from that calculated using one of the above-mentioned prediction schemes.

In most cases the duct wall critical frequency is well above the frequency range of interest. If this is in doubt the critical frequency may be calculated [10] and then the transmission loss predictions for a flat panel [10] may be used at frequencies above half the duct wall critical frequency.

3. OTHER DUCT SHAPES AND BREAKIN TRANSMISSION LOSS

The transmission loss for circular and oval ducts is difficult to predict accurately with an analytical model. It is recommended that the guidelines outlined in [1] for the estimation of these quantities, be followed closely.

4. BREAK-IN SOUND TRANSMISSION PREDICTION

The sound power entering into a duct from a noisy area and propagating in one axial direction in the duct is given by

$$PWL_i = PWL_o - TL_{in} - 3 \quad (dB) \tag{9}$$

For frequencies less than f_o, TL_{in} is the larger of the following two quantities [1]:

$$TL_{out} - 4 - 10 \log(a/b) + 20 \log(f/f_o) \quad \text{or} \quad 10 \log(\ell/a + \ell/b) \tag{10}$$

For $f>f_o$,

$$TL_{in} = TL_{out} - 3 \quad dB \tag{11}$$

PWL_o is the sound power incident upon the exterior of the entire length of ductwork. The 3dB in Equation (9) allows for the incoming power to be split equally into each of the two opposing axial directions.

5. LABORATORY TEST PROCEDURE

Tests were carried out to determine the TL_{out} for a duct test section of 1.15m x 0.7m in cross section and 1.8m length, mounted in a 180m^3 reverberant room containing a rotating diffuser suspended from the ceiling. The test section was attached to double wall wooden ductwork at both ends as shown in Figure 1. Sound was introduced at one end using a Bruel and Kjaer type 4211 sound source. After passing through the test section, the sound was absorbed by a conical wedge. The effectiveness of the wedge was determined by measuring the standing wave in the test section and upstream ductwork, and was found to be satisfactory.

The net power flow into the ductwork was determined by measuring (with a Bruel and Kjaer type 3360 Intensity System) the average sound intensity over a cross sectional surface adjacent to where the sound was introduced (see Figure 1). The sound power propagating down the inside of the duct was also determined by using a simple average pressure measurement made with a microphone traversing down the centre of the duct test section. Measurements taken using this method generally yielded internal sound power levels between 1 and 2 dB higher than measurements taken using the sound intensity method for one third octave bands below and including the 250 Hz one third octave band. The agreement at frequencies above 149 Hz is surprising since this is the frequency at which the first higher order duct mode begins to propagate. Other higher order modes begin to propagate progressively at 245Hz, 287Hz, 298Hz, 386, 490, 512 and 574Hz.

For the 315Hz one third octave band and higher, pressure measurements and intensity measurements gave sound power results differing by up to 10dB, which is hardly surprising considering the presence of a number of higher order propagating modes.

The net power flow out of the duct was determined by measuring the space average sound pressure level in the reverberant room (using 20 discrete microphone locations) and the average room reverberation time as described in [10]. Sound emitted from the wooden ductwork on either side of the steel test section was measured by removing the test section and connecting the two wooden sections together, taking care to provide adequate sealing at the joints. Sound levels due to the wooden ductwork only, were generally 10-20dB less than those due to the metal ductwork, except in the 100Hz one third octave band where the difference ranged between 3 and 10dB, the 125Hz and 800Hz one third octave bands where the difference ranged between 5 and 10dB, and the 1000Hz one third octave band where the difference ranged between 5 and 8dB depending upon measurement location and duct wall thickness. Measurements were carried out using one third octave bands of noise with centre frequencies ranging from 100Hz to 1000Hz.

Tests were carried out for duct sections with three different wall thicknesses, (1.2mm, 1.0mm and 0.8mm) which are commonly used in the commercial air handling industry.

6.0 RESULTS AND DISCUSSION

Results of measurements carried out as described in the previous section are presented in figures 2 to 4. It is clear that the ASHRAE prediction scheme agrees best with the measured data. However it seems that even the ASHRAE scheme is a little conservative at low frequencies, underpredicting measured TL_{out} values by up to 6dB. The reason for the large reduction of TL_{out} in the 315Hz one third octave band is unclear, although similar dips have been reported elsewhere [4] (see figure 5).

As an aside the measured transmission loss for the double walled wooden duct with no test section interposed was compared with predictions made by assuming the duct walls behave as simply supported panels, as described in [10]. Figure 6 indicates good agreement between prediction and theory. Note that a high panel loss factor of 0.1 was achieved for the wooden duct wall panels by sandwiching two 16 mm thick panels together with viscoelastic material in between. The wooden wall construction consisted of two of these sandwich panels separated by a 0.1m air gap which contained a 50mm thick glass fibre blanket. The panels were connected at points approximately 1.5m apart on average.

7. CONCLUSIONS

The current ASHRAE prediction procedure yields results within +6, -2dB of measured data for breakout sound power levels of rectangular ductwork, provided the correction term C of Equation (1) (which accounts for a decreasing sound intensity in the duct as distance from the sound source is increased) is included. In general, especially at low frequencies, the prediction scheme is conservative, usually underestimating duct transmission loss (thus overestimating "break-out" sound power) by up to 6dB.

ACKNOWLEDGEMENTS

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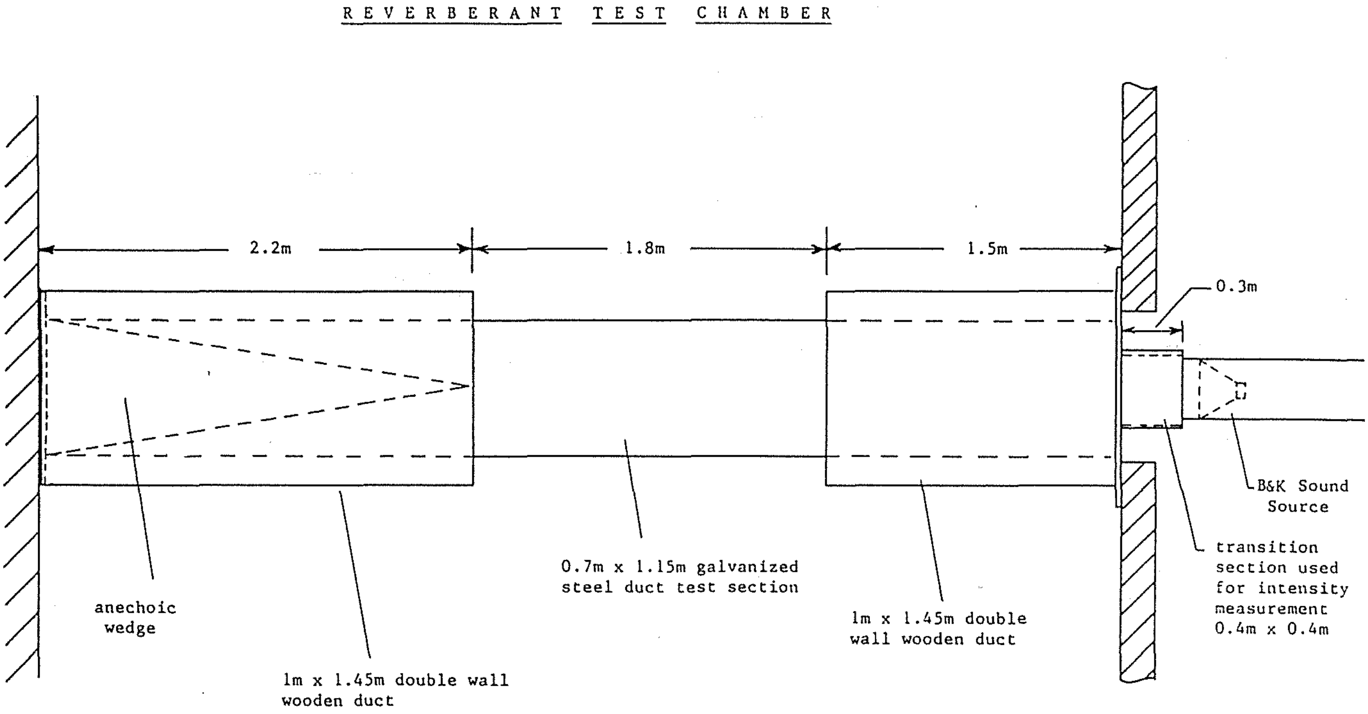

Figure 1. Experimental arrangement.

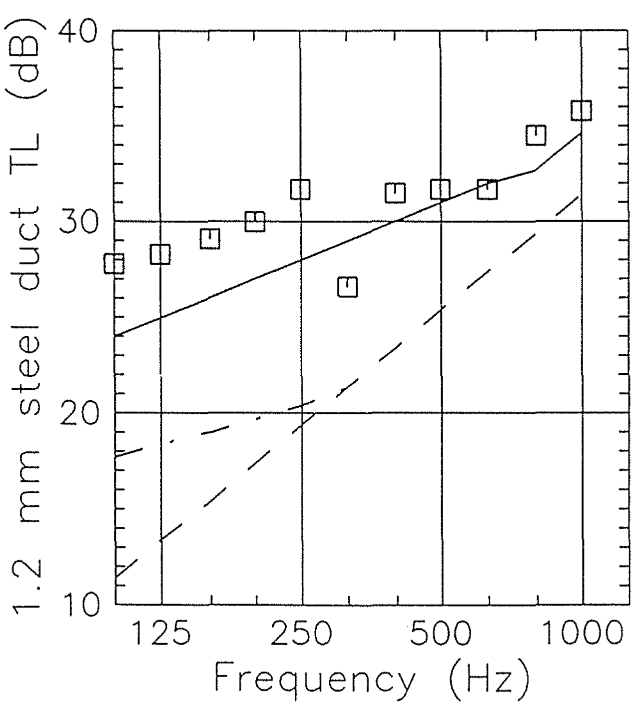

Figure 2. TL data for a steel duct of 1.15 x 0.7m cross section and 1.2mm wall thickness.

□ = measured data
—— = ASHRAE theory predictions
— — = Mass Law predictions
—·— = Modified Mass Law predictions

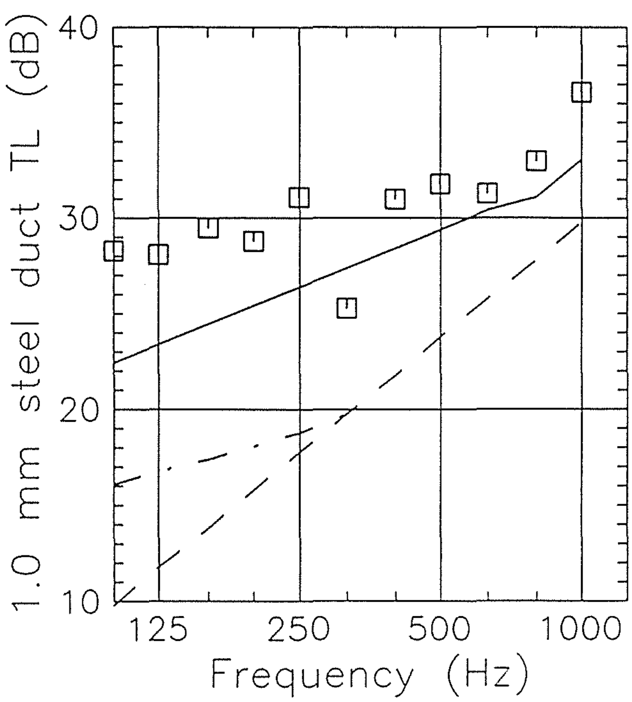

Figure 3. TL data for a steel duct of 1.15 x 0.7m cross section and 1.0mm wall thickness.

□ = measured data
—— = ASHRAE theory predictions
— — = Mass Law predictions
—·— = Modified Mass Law predictions

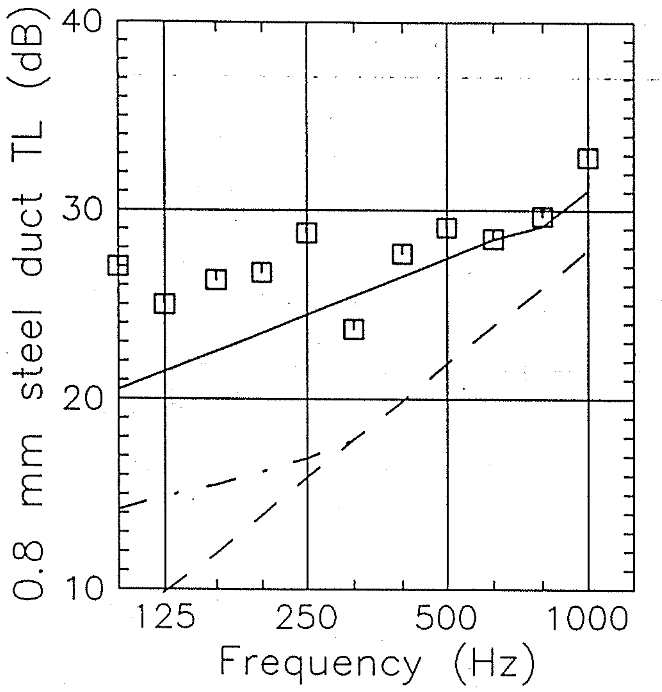

Figure 4. TL data for a steel duct of 1.15 x 0.7m cross section and 0.8mm wall thickness.

□ = measured data
—— = ASHRAE theory predictions
— — = Mass Law predictions
—·— = Modified Mass Law predictions

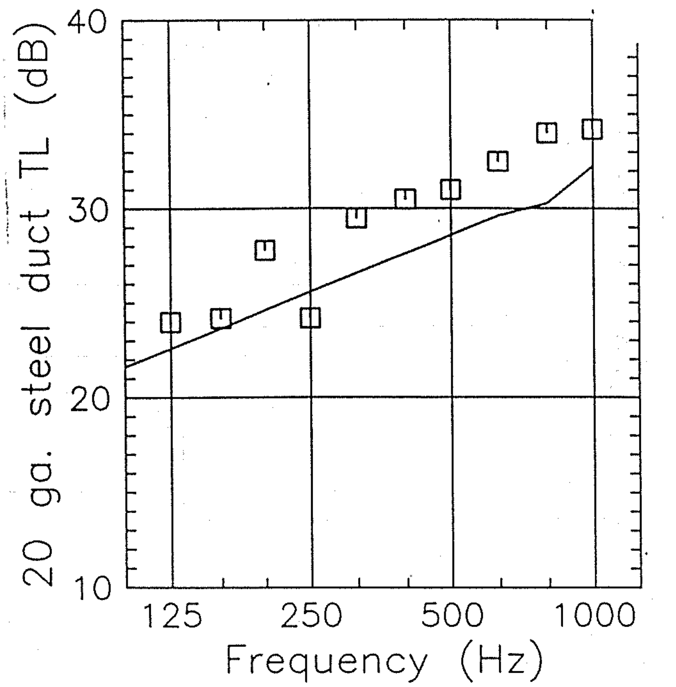

Figure 5. TL data for 20 ga. steel duct, of cross section 914 x 914 mm (from [4]).

□ = measured data
—— = ASHRAE theory predictions

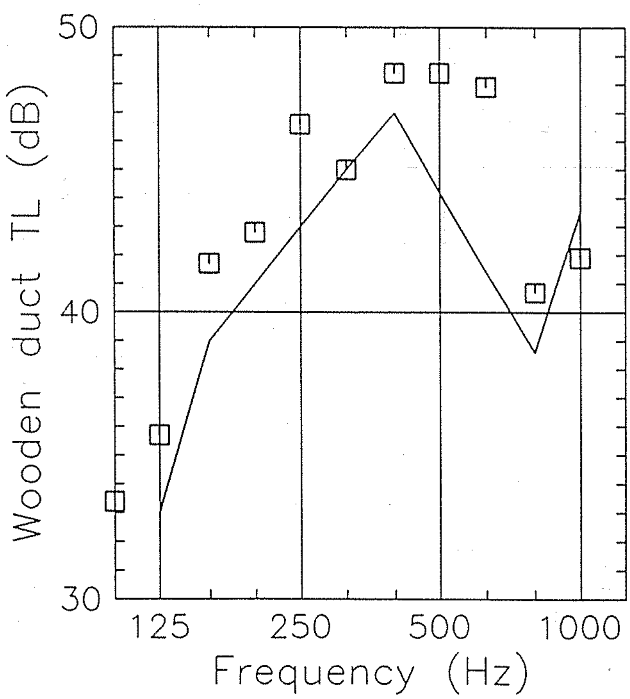

Figure 6. TL data for double wall wooden ductwork used to terminate and connect the test section to the sound source.

□ = measured data
—— = theoretical predictions for a double flat panel [10]

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ON NOISE GENERATION MECHANISMS IN IDLING CIRCULAR SAWS

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1. INTRODUCTION

Noise resulting from the use of circular saws has been the subject of investigation within the Department of Mechanical Engineering at the University of Adelaide for several years and forms part of an extensive involvement with the problems of industrial noise control reaching back to the mid-sixties. This early work resulted in several published papers [1-5]*, one Master's thesis [6], and four patents for a noise suppressing guard [7].

The aerodynamic noise which characterises the well-damped blade constitutes the base level of an idling saw and sets the minimum ambient level to which a worker continuously using, for example, a trim saw is exposed. It is therefore a major factor in determining the hearing damage risk to which the worker is subjected. Various investigators have considered the problem of aerodynamic noise generation by an idling circular saw. Kanapathipillai [6] provides a comprehensive review of relevant literature up to 1982. Stewart [8] was one of the first investigators to suggest vortex shedding from the saw teeth, and the consequent lift and drag fluctuations on each tooth, as the probable noise source mechanism for a damped idling circular saw. Cho and Mote [9] in their measurements of force fluctuations on individual teeth pointed to the wake flow generated by the teeth as the major source of aerodynamic noise, but assumed that it consisted of irregular turbulent eddies. In an investigation of whistling instability in undamped idling saws Mote and Leu [10] again proposed vortex shedding as the initiating mechanism, although they were unable to confirm that this was the case. In hot-wire measurements of the wake velocity behind a single tooth on a rigid rotating disc Price and Mote [11] observed a flow structure characteristic of periodic vortex shedding. The flow structure persisted as the number of teeth was increased although the strength of the vorticity was much diminished. In noise measurements and flow visualisation experiments on a single tooth mounted on the periphery of a plain disc Leu and Mote [12] concluded that vortex shedding was the primary noise generating mechanism. They also deduced that it was the flow separation at the leading edges of the tooth that had the dominant effect and that the vortex shedding from the tooth trailing edges was only important to the extent that it influenced this leading edge separation. They make the observation that the radiated sound is produced by the pressure fluctuation on the lateral faces of each tooth. It might be expected therefore that the fluctuating vortex wake of a tooth will not only influence the leading edge separation on that tooth and thus the magnitude of the pressure fluctuation on its lateral surfaces, but will also, by impinging on the downstream tooth, influence the leading edge separation there and thus the pressure fluctuation on this surface also.

The series of tests reported in this paper are part of a more extensive study intended to investigate further the explanation of the noise generation mechanism proposed by Leu and Mote.

2. EXPERIMENTAL APPARATUS

2.1 Rotating Blade Rig

2.1.1 Model saw blades

The model saw blades used in the rotating blade rig were made from standard saw steel blanks 350mm in diameter and between 3.3 and 3.5mm thick. Model teeth were formed by grinding radial slots in the periphery of the disc. The sides of the slots were parallel and the root rounded, as shown in figure 2.1, producing a tooth with a slightly flared profile. The teeth were not given any set.

The blades used by Kanapathipillai [6] (one of which was used in these tests) had the full complement of 40 teeth and gullets equally spaced around the periphery. In the present tests blades with 5, 4, 3, 2, 1 and 0 gullets were also used, in which the gullet shape and spacing was essentially the

* Numbers in square brackets refer to references listed at the end of the paper.

same as those on the 40 tooth blade. In these blades the gullets occupied only a short arc of the blade periphery. The actual dimensions of the blades used are given in Table 2.1. A blade with a single tooth was also used. This was machined from a 350mm diameter blade blank to the dimensions shown in figure 2.2. The blades were carefully balanced and damped with shim steel discs [5] to prevent resonant vibration.

2.1.2 Drive System

The saw blade was mounted on the cantilever end of a 44 mm diameter shaft supported in two self-aligning bearings. The shaft was driven by a 2.2 kW AC motor via a rubber cross type flexible coupling. Both motor and bearing pedestal were rigidly mounted on a steel sub-frame which was in turn mounted on rubber vibration isolation mounts on a cast-steel table. The motor/shaft unit was enclosed in a fibreglass lined chipboard box through which the shaft carrying the saw blade protruded. Motor speed was controlled by a frequency inverter drive unit and measured by a shaft mounted sensor.

2.1.3. Instrumentation

Sound power measurements were conducted in a calibrated reverberation room of volume 106m^3. Sound pressure measurements in the reverberant field were made by traversing a Bruel & Kjaer 4134 1/2" microphone across an oblique diagonal of the room, the sound pressure level being averaged over two complete traverses. Microphone output was processed on a Bruel & Kjaer Type 2134 Sound Intensity Analyser, displayed in 1/3-octave bands on a Bruel & Kjaer Type 4175 Display Unit and recorded on a Bruel & Kjaer Type 2313 Graphics Recorder.

2.2 Wind tunnel tests

2.2.1 Flow visualisation tunnel

In order to investigate details of the flow around models of saw teeth a flow visualisation wind tunnel has been constructed. A forced draft configuration is used with a 14.5 kW, variable speed (up to 1800 rpm.) centrifugal fan. The AC drive-motor speed is controlled by a frequency inverter drive unit. The fan discharges to a rectangular working section via a diffuser, a settling chamber with screens and honeycombs, and a contraction designed according to a method described by Mikhail [13]. The 3m long working section has a gradually diverging rectangular section approximately 760 mm wide x 340 mm high and has clear perspex walls on two adjacent sides (the bottom and one vertical side) to allow observation of the flow. The vertical perspex wall is openable along the 3m length to allow access to the models. Instrumentation such as hot wire probes may be mounted on, and inserted through, removable panels in the top wall.

2.2.2 Saw tooth models

Initially the tunnel was used to investigate the flow over geometrically similar models of the saw tooth shapes used on the experimental rotating blades. However the short aspect ratio of the teeth in these linear cascades and the consequently highly three-dimensional nature of the flow around them made interpretation of the visualised flow difficult.

In order to remove the secondary three-dimensional flow phenomena that were obscuring the basic flow behaviour a simplified model has been used in the experiments reported here. The model consisted of a stream-wise cascade of rectangular cross-section bars completely spanning the 760 mm width of the working section of the wind-tunnel. The chord (c = 116mm) and thickness (t = 19mm) of each of these bars was kept constant giving a thickness to chord ratio (t/c) of 0.16. The separation (g) between adjacent bars could be varied. The first bar in the cascade had an elliptically faired leading edge, to permit boundary layer growth more akin to that experienced in the outwardly spiralling flow over the face of a rotating saw blade. Each bar represents an individual saw tooth and the gap between adjacent bars represents the saw blade gullet. The general layout is shown in figure 2.3.

2.2.3 Instrumentation

Apart from a system for flow visualisation the only instrumentation used in these tests was a single hot-wire probe used to measure boundary layer thickness on the model teeth or wake thickness in the gullet area. It was also used to measure the characteristic wake velocity fluctation frequency in order to determine the Strouhal number of the wake flow. The probe was mounted on a carriage allowing it to be traversed manually both up- and downstream parallel to the plane of the cascade of model teeth and transversely perpendicular to that plane. The wire of 5 μm diameter and 1.0 mm active length was positioned perpendicular to the main flow direction with its axis parallel to the plane of the model teeth. The hot wire signal was low-pass filtered (100 Hz) and then processed digitally using an RTI-800 data collection card in a PC-XT computer. Two thousand samples were collected at 200 Hz and the mean value of the signal calculated.

2.2.4 Flow visualisation

The flow is made visible by an ammonia gas/sulphur dioxide system. The two gases are introduced into the flow through a pair of small holes in the surface of a model tooth, one hole being placed downstream of the other (see fig. 2.3). Upon mixing in air in the presence of moisture the two gases produce a thick white cloud consisting of fine particles of ammonium sulphite. Because of the toxic and corrosive nature of the gases it is important for them to be injected in the correct proportions for complete reaction. Flow control is effected by a pair of needle valves and monitored by a pair of flow meters. Solenoid operated shut-off valves come into action automatically when the tunnel is shut down.

The flow patterns have been observed using still photography, a 'Hycam' high-speed movie camera and a video system. In order to visually 'freeze' the flow, in the same way as can be done with strobed illumination, the video camera was used in conjunction with a mechanical shutter. This consisted of a 590mm diameter disc with two diametrically opposed sectors of 60 degrees removed and mounted on a spindle driven by a variable speed motor. The flow was viewed by the video camera through the apertures in the disc whose rotational speed was adjusted until the video picture was apparently steady.

Illumination is always a problem in flow visualisation studies. In this case a carbon arc light source from a drive-in movie projector was used. The source is mounted on a carriage beneath the tunnel and the light reflected upwards into the working section by a polished metal concave mirror mounted on a separate carriage. Streamwise strips of illumination were produced by appropriate masking.

3. EXPERIMENTAL RESULTS

3.1. Effect of tooth spacing on the flow

The work of Kanapathipillai [6] on rotating blades showed that at a given peripheral speed, provided that the gullet width to blade thickness ratio (g/t) is greater than two, the noise generated by a model circular saw blade is principally dependent on tooth area. For gullet width to blade thickness ratios less than two, the noise generation appeared to be linearly dependent on this ratio, as well as on tooth area. Kanapathipillai postulated a change in the character of the flow at a g/t of 2 to account for this change in acoustic radiation behaviour.

This hypothesis was tested in the flow visualisation tunnel using the linear streamwise cascade of rectangular bars described in Section 2.2.2. The spacing between the bars was varied over a range of g/t ratio from 1 to 10. The flow visualisation gases were normally injected into the flow from the upper surface of the first round-nosed bar at mid-span and the flow observed in the first and second gaps between the bars. On some occasions the gases were injected on the upper surface only, or on both surfaces simultaneously. The tunnel free-stream velocity was 1 m/s for gap to thickness ratios between 1 and 2, and 2 m/s for gap to thickness ratios of 2.5 and greater. This gave a Reynolds number based on bar thickness t and free stream velocity U_∞ of either 1.26×10^3 or 2.52×10^3. This is lower than the equivalent Reynolds number based on saw disc thickness t and peripheral velocity at tooth mid-height of about 12×10^3 on a model saw blade rotating at 3000 rpm, but was a consequence of the need to obtain sufficient smoke density for good filming and photography and because of limitations of the available video equipment.

At the smallest gap sizes (where g/t was 1 and 1.5) the flow in the first gap consisted of two stable contra-rotating vortices. The vortex originating from the lower rear edge of the first bar was much larger than the other, almost filling the space between the adjacent bars. The smaller vortex originating from the upper edge remained immediately adjacent to the upper part of the rear face of the first bar; on occasions it seemed to disappear altogether. This flow asymmetry in the gap was ascribed to bouyancy effects arising from the exothermic reaction between the flow visualisation gases.

At a gap of g/t = 2 a similar flow was observed in the first gap although it was rather less stable than at the smaller gaps. Again one eddy seemed dominant (that orginating from the underside of the model where the flow visualisation gases were injected) but its upstream face moved up- and downstream erratically in the gap, apparently under the influence of the second eddy orginating at the upper rear corner of the model tooth. However the external stream and the flow over the surface of the downstream bar was not greatly disturbed by this flow behaviour in the gap.

When the gap was increased to g/t = 2.5 the flow in the first gap remained similar in character but the up- and downstream motion became much more regular and pronounced. Brief bursts of fluid were seen to leave the upper and lower sides of the downstream portion of the gap alternately and flow downstream over the faces of the following bar.

Increasing the gap to g/t = 3 produced a marked change in the flow, with vortices forming and being shed alternately from the rear corners of the upstream bar. The shed vortices moved downstream in the gap until they interacted with the downstream bar. There they spilled out of the gap and the residual disturbance was convected along the lateral face of the bar. It seemed that the major portion of the vortex flow spilled out of the gap on the same side as it was generated, producing a regular pulsing of fluid from alternate sides over the downstream 1/2 or 1/3 of the gap. The vortex formation and the out-spilling of the flow was very regular and periodic. This spilling of the flow out of the gap just upstream of the perpendicular face of the downstream bar produced an order of magnitude increase in the thickness of the disturbed shear layer flowing over the lateral faces of the downstream bar.

Increasing the gap beyond a g/t of 3 produced little change in the flow pattern in the first gap or over the downstream bar, apart from increasing the number of shed vortices contained within the gap region at any given time.

The same behaviour was observed in tests with free-stream velocities of 4, 6 and 8 m/s, corresponding to Reynolds numbers of 5.03×10^3, 7.55×10^3 and 10.07×10^3 respectively, and to rotational speeds of a rotating saw blade of 1253, 1880 and 2506 rpm respectively. At these higher speeds and a g/t of 2 the smoke filament exhibited some unsteadiness immediately upstream of the downstream bar but the flow pattern in the gap itself was unchanged.

The flow in the second gap in the array was less clearly defined than that in the first gap, principally because the smoke filament had become well diffused by this point. However in all cases it seemed that vortex formation was less pronounced than in the first gap, and that the flow in the second gap was less regularly periodic.

These wind tunnel experiments confirmed an earlier series of tests conducted by Oakeshott [14] in a water tunnel using dye filaments. Although the model teeth used in those experiments did not span the tunnel working section and had an aspect ratio (h/c) of only 0.83 and a thickness to chord ratio (t/c) of 0.035 the flow exhibited the same basic characteristics as were observed in the wind tunnel, with regular periodic vortex shedding only becoming established at a gap to thickness ratio (g/t) greater than 2.5.

3.2 Effect of vortex shedding on boundary layer thickness

The flow visualisation studies on the wind tunnel model had indicated that the vortex shedding from the trailing edges of the first tooth and the impact of these vortices on the edges of the leading face of the following tooth produced a very rapid thickening of the shear layer in which the latter tooth and succeeding teeth were immersed.

The thickness of the shear layer on the first and second teeth was measured by traversing a calibrated hot-wire probe perpendicular to the mean flow direction and through the shear layer from the free stream into the region immediately downstream the rear face of the tooth. The traverse path passed 0.5 mm behind the rear face of each tooth (see figure 2.3).

Typical mean velocity profiles are shown in figure 3.1 for a free-stream velocity of 4m/s and a gap to thickness ratio of 3. In these plots u is the velocity at a distance y from the model tooth upper surface and x is the distance of the traverse downstream of the nose of the first model tooth. It can be seen that the vortex disturbance in the first gullet has produced an order of magnitude increase in thickness of the shear layer between the first tooth location and the next.

3.3 Effect of gullet number on sound power radiated

Reverberation chamber measurements were made of the sound power radiated by a blade with a full complement of gullets or notches (40) equally spaced around the periphery, and by blades having 5, 4, 3, 2 and 1 gullets. A blank blade having no gullets was also tested. The spacing between adjacent gullets and the gullet size and shape were approximately the same in each model (see figure 2.1 and table 2.1). Tests were run at blade rotational speeds betwen 1500 rpm and 5500 rpm in increments of 500 rpm.

The sound power level produced by the various blades at 2000, 3000, 4000 and 5000 rpm is shown in figure 3.2. The dramatic difference in sound power between an un-notched blade and the blade with a single gullet can be clearly seen - about 25 dB at the highest rotational speed. Also obvious is the more gradual increase in sound power as the number of gullets is increased. Initially each additional notch seems to add about 1.1 dB to the radiated sound power. This rate of increase clearly diminishes since the sound power radiated by a 40 notch blade is only about 12 dB above the sound power of a blade with a single notch.

3.4 Comparison of toothed and notched blades

Comparative tests were conducted in order to establish whether the significant noise source originates in the separation of the flow from the rear face of a tooth or in the interaction of the separated flow with the following tooth. To do this the sound power radiated by a blade having a single notch in its periphery was compared with the sound power radiated by a blade having a single protruding tooth on its periphery. The variation of sound power level with rotational speed is shown for these two models in figure 3.3, together with data for a blade having the full complement of 40 notches. The dependence of sound power on speed is the same in each case except that the single-notched blade has a radiated power level some 8 dB higher than that for the single-tooth blade.

4. DISCUSSION

The flow visualisation studies on simplified saw tooth models in a flow visualisation tunnel show that, provided the gap size to model thickness ratio is 3 or greater, a periodic vortex wake is shed from the trailing edges of the tooth forming the upstream boundary of the gap. This wake traverses the length of the gap and interacts strongly with the leading face of the following tooth, spilling out of the gap on the same side as that on which it was formed. This violent lateral motion, forced on the wake flow by the front face of the downstream tooth, will inevitably produce a region of strong separation near the front of the tooth with an accompanying reduction in the local pressure. As the remains of the vortex flow are swept downstream over the surface of the tooth the pattern will be repeated on the opposite side of the tooth, and so on alternately. In addition to producing a greater pressure amplitude in the separated flow region originating at the sharp upstream edges of the tooth, the interaction is likely to increase the strength of the vortices shed from its downstream edges. A tooth subjected to this flow is likely to be a stronger noise source than is an isolated tooth.

That this is so is borne out by the comparison of the sound power radiated by a rotating blade with a single notch and one with a single isolated tooth. The eddy shedding from the trailing edge of the single tooth will certainly induce fluctuating pressures on each lateral face, 180° out of phase with each other, so that the tooth will act as a dipole-like noise source. However the much greater radiation from the single-notched blade indicates that the interaction between the shed vortex wake and the leading edges of the blade surface immediately downstream of the notch produces a much stronger, if similar, pressure fluctuation and is therefore a much stronger noise source. The effective extent of the 'tooth' up- and downstream of the single notch is indeterminate, which would indicate that it is probably that part of the blade surface just downstream of the notch that is acting as the major radiator of sound. Leu and Mote [12] demonstrated the important role of the leading edge flow in noise generation. Measurements of this pressure fluctuation on a wind tunnel model are planned to test this inference. It should be recognised however that this initial gullet is peculiar to the experimental configuration and does not occur on a real saw blade having equally spaced gullets around the entire periphery. In this latter case each blade is subjected to identical flow conditions. The actual flow around the teeth and gullets of a rotating saw blade is probably much better modelled by the flow in the second and subsequent gaps of the wind tunnel model.

The violent interaction of the vortex wake with the downstream tooth was observed in the wind tunnel to produce a very rapid thickening of the 'boundary layer' over the second tooth and this seemed to have the effect of reducing the regularity of the vortex shedding in the second gap. This flow condition over the second (and subsequent) bars corresponds to the stable flow state on the teeth of a rotating saw blade, where every tooth is submerged in the wake of the teeth upstream. In measurements of the boundary layer growth on a rotating circular saw blade, as the flow spirals outwards from the axis of rotation, Bull et al [16] have observed a very sudden thickening of the layer in the immediate vicinity of the teeth.

As noted in section 3.1 the flow in the second gap of the wind tunnel model was not well-defined under the conditions of the tests reported here, and further investigation of the flow in this region is required. However the apparent reduction in the regularity with which vortices are shed in this region could account for the effect shown in figure 3.2, where the downstream teeth seem to contribute less to the radiated power than does the area of the blade in the vicinity of the first gullet. An analysis of this data by Bies et al [15] indicates that good correlation is obtained by using the equation:

$$L_W = 50 \log_{10} N + 10 \log_{10} (n+1) + K$$

where L_W is the sound power radiated, N is the blade rotational speed in rev/min, n is the number of gullets and K is a constant. This correlation implies that the ill-defined tooth formed by a lone gullet has twice the acoustic power output of any additional tooth.

Bies et al [15] suggest that the radiation from the saw teeth is incoherent. If this is so it is a simple matter to estimate the sound power radiated from a single tooth in an array of teeth. The result of this calculation is shown in figure 3.3 as the curve labelled 'individual notches'. This curve was obtained from the noise radiation data for blades with 2,3,4,5 and 40 notches. In the last case the radiation from a single tooth was obtained by assuming that each of the 40 notches contributes equally to the measured radiation. Note that in this context the term 'notch' includes the up- and downstream half-teeth that define the notch boundaries; the 40 notch blade therefore consists of 40 half-tooth/gullet combinations. For the short arrays of notches extending over only part of the blade periphery the measured sound power of a single notched blade was subtracted from the measured power of the 2,3,4 and 5 notched blades respectively, and the residual acoustic power divided by the number of remaining notches. The individual data points are not shown in figure 3.3, but the spread of the data is indicated. The sound power radiated by an individual notch in an array of notches is some 3 to 5 dB less than that radiated by a single notch. This indicates clearly that the isolated notch is a more efficient noise generator than the notches submerged in the thickened boundary layer. Even so, these individual teeth are more effective radiators than the isolated tooth. It is probable that the vortex interaction with the leading edge of a tooth intensifies the vortex shedding process at the trailing edge, compared with that occurrring on an isolated tooth, and thus produces larger fluctuating pressure amplitudes on the tooth surface. However this intensification is not sufficient to compensate completely for the effect of the thickened boundary layer, which seems to diminish the shed vortex strength below that occurring in an isolated notch.

5. CONCLUSIONS

The combined evidence of flow visualisation studies in wind and water tunnels, and sound power measurements from rotating model circular saw blades indicates that the dominant noise source mechanism in idling circular saws is not simply the vortex shedding process from individual teeth, but is rather the interaction of the vortex wake so formed with the leading edge of the following tooth. The noise generation mechanism seems to be a complex interaction between the vortices shed from the rear of one tooth and the flow around the leading edge of the following tooth, modulated by the effect of the thickened shear layer on the vortex shedding process. Although it is the individual teeth on the saw blade, and the fluctuating pressures acting on their lateral faces, that are the noise sources for an idling saw blade it seems more appropriate, when considering the flow mechanisms that give rise to these pressure fluctuations, to think of the source mechanism as being formed by a gullet and the two adjacent half-teeth.

In designing a quieter circular saw both factors, blade lateral area and gullet flow, must be taken into account. However attention to the latter aspect is of primary importance if significant improvement is to be gained.

ACKNOWLEDGEMENT

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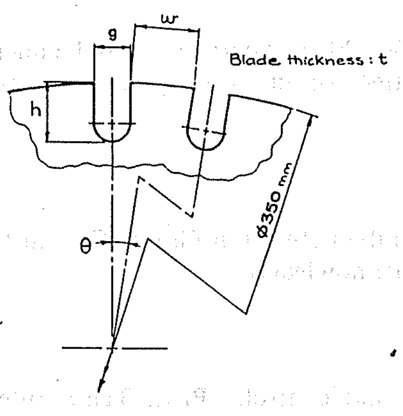

Fig. 2.1: Rotating Blade Model Tooth Geometry

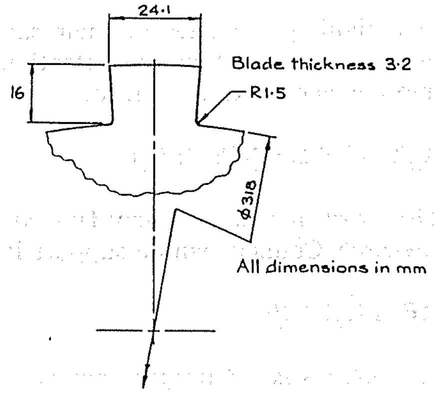

Fig. 2.2: Rotating Blade Single Tooth Model Dimensions

No.	GULLET Width g(mm)	Depth h(mm)	TOOTH Tip Width w(mm)	Mean Chord* c(mm)	Angular Separation θ(deg.)	BLADE Thickness t(mm)	g/t	h/c	t/c
0	–	–	–	–	–	3.4	–	–	–
1	9.0	16.3	–	–	–	3.4	2.65	–	–
2	9.0	16.3	18.5	17.2	9.0	3.4	2.65	0.95	0.20
3	8.9	16.4	18.5	17.3	9.0	3.5	2.54	0.95	0.20
4	8.9	16.4	18.6	17.3	9.0	3.3	2.70	0.95	0.19
5	8.9	16.3	18.5	17.3	9.0	3.3	2.70	0.94	0.19
40	10.2	17.6	17.3	15.9	9.0	3.3	3.09	1.11	0.21

* At mean tooth height

Table 2.1 : Model Saw Blade Dimensions

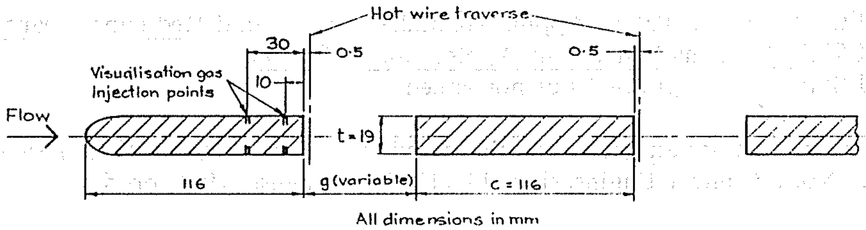

Fig. 2.3: Cross-section of Array of Model Saw Teeth in Wind-tunnel

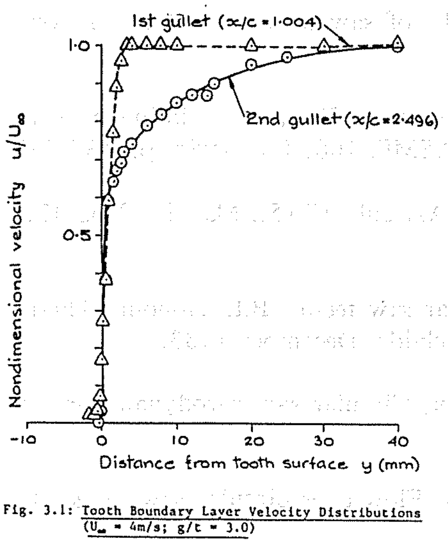

Fig. 3.1: Tooth Boundary Layer Velocity Distributions (U_∞ = 4m/s; g/t = 3.0)

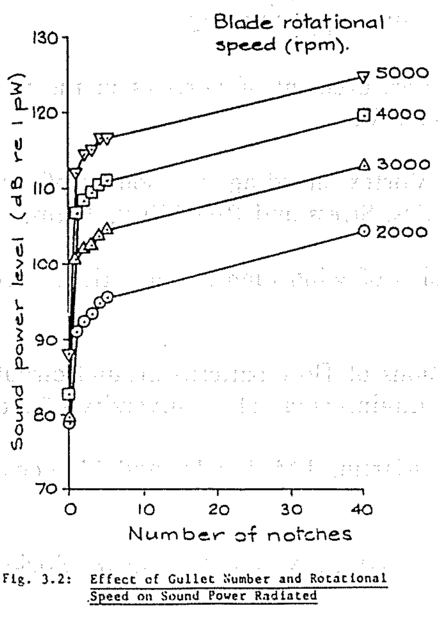

Fig. 3.2: Effect of Gullet Number and Rotational Speed on Sound Power Radiated

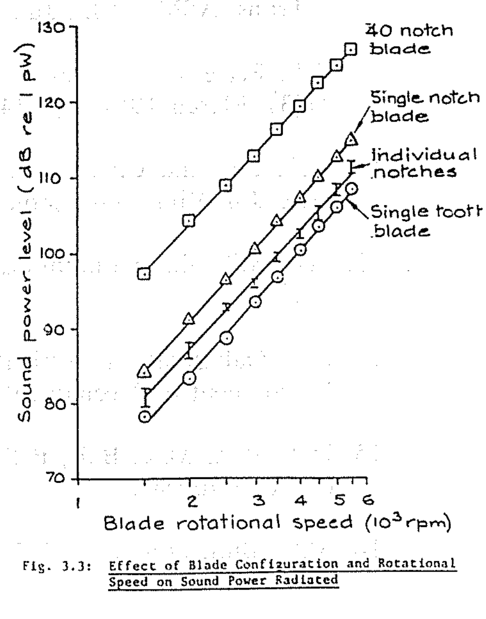

Fig. 3.3: Effect of Blade Configuration and Rotational Speed on Sound Power Radiated

PRACTICAL ENGINEERING APPROXIMATIONS FOR DETERMINING ATTENUATIONS IN AIRCONDITIONING DUCT SYSTEMS

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1. INTRODUCTION

The ability to reliably predict attenuations due to various elements in an airconditioning duct system is of considerable concern to any acoustic consultant or building services engineer. The literature abounds in a variety of predictive schemes for determining attenuations due to individual components such as lined and unlined ducts, lined and unlined mitre bends, radiussed bends, mitre bends with turning vanes, filters, cooling coils and plenum chambers. Unfortunately, not all of these components are included in any one reference, and furthermore, some of the prediction schemes, whether in the form of equations or tabulations are significantly at variance with one another. In this paper, predictive schemes which reflect the authors' best judgement of available methods are given for each of the major attenuating elements occurring in airconditioning ductwork.

Considerable effort has been expended in developing a simple generalized scheme for predicting attenuations due to internally lined ducts having lining thicknesses in the range 25mm to 150mm. The resulting prediction accuracy for duct sizes and linings tested by two of Australia's insulation manufacturers as well as for various attenuator configurations has been found to be very good compared with alternative predictive methods.

The method proposed for bends involves use of standard curves showing attenuations normalized against (D/λ). Many of the curves correspond closely with tabulated data from the various references. In some instances, more recent test data have suggested that approximations used in the past may not be sufficiently conservative and the relevant curves have been adjusted accordingly.

It should be recognized that the predictive schemes proposed are largely empirical, and the opinions and test data on which they are based exhibit a considerable scatter. The methods proposed in this paper tend toward the conservative end of the scatter band, but are by no means absolutely conservative. The manner of normalizing the attenuation data has tended to follow the general direction suggested by relevant theory, but empirical adjustments have been made where necessary to improve agreement with measurement. The various predictive schemes have been formulated with ease of computerization in mind, particularly simplicity of interpolation for a wide range of duct sizes, lining thicknesses, etc.

2. ATTENUATIONS DUE TO BARE AND EXTERNALLY LAGGED DUCTS

The procedure presented here for bare and externally lagged ducts has been developed from the general recommendations and test data presented by Dr I.L. Ver (1) together with data from Woods (2). The main variations from Ver's proposal include an allowance for the attenuation to reduce for increasing duct size as indicated by Woods (2) and a revised adjustment procedure for attenuation at frequencies above 4000 Hz. The calculation procedure for bare and externally lagged ducts is as follows:-

RECTANGULAR DUCTS

The attenuation of bare rectangular steel ducts may be determined using Figure 1. First the resonance frequency is determined from:-

$$f_{RES} = \frac{c}{\pi}\left(\frac{\rho}{\sigma D_{eq}}\right)^{1/2} \tag{1}$$

Where :- c = velocity of sound in air (346 m/s at 25°C)
ρ = density of air (1.2 kg/m^3)
σ = surface weight of the duct (kg/m^2)
= density of steel x duct wall thickness
= 7.85 x t when t is in mm
D_{eq} = 4 A/P (m)
A = cross sectional area of duct (m^2)
P = perimeter of duct (m)

At the resonance frequency the attenuation is 1.0dB/metre for unlagged bare ducts and 2.0dB/metre for bare externally lagged ducts. Below the resonance frequency the attenuation can be determined from:-

$$A_B = \left(\frac{f}{f_{RES}}\right)^2 \times A_{RES} \tag{2}$$

Where :-
A_B = attenuation of bare duct (dB/m)
A_{RES} = attenuation of bare duct at resonance (dB/m)

Between twice the resonance frequency and 4000 Hz the attenuation can be determined from:-

$$A_B = 0.09/D_{eq} \text{ (dB/m)} \tag{3}$$

When D_{eq} is less than 0.3 metres, equation (3) tends to overestimate the attenuation. Hence for ducts with D_{eq} less than 0.3 metres a limiting value of 0.3 dB/m should be used.

Above 4000 Hz the attenuation can be determined from:-

$$A_B = Y \times \frac{f}{4000} \text{ (dB/m)} \tag{4}$$

Where :-
$Y = 0.09/D_{eq} \leqslant 0.3$

CIRCULAR AND HEAVY RECTANGULAR DUCTS

The attenuation of bare circular steel ducts or heavy rectangular ducts (e.g. concrete ducts) may be determined using Figure 2. The resonance frequency is again determined from equation (1) and the cut off frequency from:-

$$f_c = c/D \qquad (5)$$

Where :-
c = velocity of sound in air (346 m/s at 25^{o}C)
D = duct diameter (m)

At the resonance frequency, the attenuation may be determined from:-

$$A_{RES} = 0.018/D \qquad (6)$$

Below the resonance frequency, equation (2) can be used to determine the attenuation.

Above the resonance frequency the attenuation rises as indicated to the plateau level where equation (3) (with D replacing D_{eq}) applies up until 4000 Hz. Above 4000 Hz equation (4) may again be used.

Typical results for some specific sizes are shown in Tables 1 and 2.

TABLE 1 ATTENUATION (dB/m) OF SQUARE BARE DUCTS WITH A WALL THICKNESS OF 0.6mm

Height/ Width	63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz	P/A (m/m^2)	RES.F (Hz)
200	0.26	0.99	0.30	0.30	0.30	0.30	0.30	0.60	20.0	124
400	0.51	0.60	0.22	0.22	0.22	0.22	0.22	0.45	10.0	87
600	0.77	0.31	0.15	0.15	0.15	0.15	0.15	0.30	6.7	71
800	0.98	0.11	0.11	0.11	0.11	0.11	0.11	0.22	5.0	62
1600	0.51	0.06	0.06	0.06	0.06	0.06	0.06	0.11	2.5	44
3200	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.06	1.2	31

TABLE 2 ATTENUATION (dB/m) OF CIRCULAR BARE DUCTS WITH A WALL THICKNESS OF 0.6mm

Diameter	63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz	P/A (m/m^2)	RES.F (Hz)
200	0.02	0.10	0.14	0.18	0.24	0.30	0.30	0.60	20.0	124
400	0.02	0.07	0.09	0.12	0.17	0.22	0.22	0.45	10.0	87
600	0.02	0.05	0.08	0.11	0.14	0.15	0.15	0.30	6.7	71
800	0.02	0.04	0.06	0.08	0.11	0.11	0.11	0.22	5.0	62
1600	0.02	0.03	0.04	0.06	0.06	0.06	0.06	0.11	2.5	44
1200	0.01	0.02	0.03	0.03	0.03	0.03	0.03	0.06	1.3	31

3. ATTENUATIONS DUE TO INTERNALLY LINED DUCTS

RECTANGULAR DUCTS

The procedure presented herein represents a refinement of that presented by Dr I.L. Ver (1) in so far as a number of empirical coefficients have been derived so as to achieve better prediction accuracy for a wide range of lining types and thicknesses applicable to the Australian market.

Before the test data could be normalized in such a way as to permit interpolation for different duct sizes, the following key points had to be identified:

(1) For ducts with the narrowest airway dimension approaching the thickness of the duct lining, the effective height of the duct for determining the mode of sound propagation along the duct needs to be increased by 25% of the duct lining thickness (i.e. effective height equals narrowest duct dimension plus 25% of duct lining thickness).

This dimension determines the effective cut off frequency above which the wavelength of the sound will be less than the effective narrowest dimension of the duct. At frequencies much greater than the cut off frequency, very little attenuation is achieved.

(2) The normalizing of the attenuation per unit length of duct should be in terms of dB per characteristic dimension where the characteristic dimension (D) is defined as 4A/P (A is the effective internal cross sectional area of the duct and P is the corresponding perimeter. In calculating A and P, the effective internal dimensions of the duct should be based on the clear internal airway sizes plus 25% of the lining thickness).

(3) The attenuation due to lined duct can be represented for a particular lining thickness by a unique function of frequency below 0.65 times the cut off frequency (refer Figure 3) and by a unique function of the ratio of frequency to cut off frequency above 0.65 times the cut off frequency (refer Figure 4).

(4) The maximum theoretical value of dB x D/L equal to 12.6 is seldom achieved on an octave band basis and a practical upper limit of 11.0 should be used in the calculations.

(5) Where there are significant components in the incident sound wave which are higher order modes, these will be attenuated at a faster rate than the plane wave mode. Generally, the additional attenuation amounts to about 5dB for frequencies above twice the cutoff frequency, diminishing to zero at the cut off frequency. For very short duct lengths (eg. less than 2xD) the 5 dB allowance needs to be reduced.

(6) There are only minor differences in acoustic performance between the commercially available fibreglass and rockwool lining materials provided they have the same types of facings. The linings covered by this calculation procedure include fibreglasses having a density in the range 28-48 kg/m^3 and rockwools having a density in the range 55-96 kg/m^3.

Flow resistivity should be typically in the range 10,000-40,000 MKS rayls/metre.

With the above factors in mind, the prediction scheme is as follows:-

STEP 1 Determine the cut off frequency from:-
$f_c = 1000.c/d_e$

Where :-
f_c = cut off frequency (Hz)
c = speed of sound in air (346 m/s at 25°C)
$d_e = d + t/4$ = effective duct dimension for determining cut off frequency
d = smallest duct clear airway dimension (mm)
t = lining thickness (mm)

STEP 2 Determine the normalized base insertion loss IL_B as follows:-

(i) For frequencies below 0.65 times the cut off frequency extract the value of IL_B from Figure 3 for the particular thickness of lining material.

(ii) For frequencies above 0.65 times the cut off frequency:-

$IL_B = 12.6\ (0.65\ f_c/f)^n$ (plotted in Figure 4)

Where :-
IL_B = normalized base insertion loss (dB)
f_c = cut off frequency (Hz)
f = octave band centre frequency being considered (Hz)
n = 1.445 for 25mm perf. faced linings
1.328 for 50mm perf. faced linings
0.776 for 100-150mm perf. faced linings
1.231 for 25mm tissue faced linings
1.052 for 50mm tissue faced linings

(iii) For frequencies above twice the cut off frequency, limit IL_B in accordance with:-

$IL_B \leqslant 12.6\ (2f_c/f)^2$ (plotted in Figure 4)

(iv) Truncate IL_B to 11.0 if numerically greater than 11.0.

STEP 3 Divide the values obtained from STEP 2 by the characteristic dimension (i.e. D = 4A/P) of the duct in metres. Use the corrected dimensions including the t/4 correction for duct lining thickness in determining "D". The result is the plane wave insertion loss per metre (IL).

STEP 4 Flow correction:-

(a) When the sound propagates against the flow there is no correction for flow.

(b) For velocities above 10m/s and when the sound propagates with the flow, the insertion loss figures are multiplied by 0.9. For velocities less than 10 m/s the multiplying factor is varied linearly toward 1.0 for the no flow condition.

STEP 5 The total attenuation in the straight length of duct including the attenuation due to bare unlined duct is then calculated as:-

$$A_L = (IL + A_B) \times L$$

Where :-

A_L = total duct attenuation (dB)
IL = duct insertion loss (dB/m) determined from Steps 1 to 5
A_B = attenuation of equivalent bare duct (dB/m) from Section 2
L = the length of the straight duct (m)

STEP 6 Entrance loss:-

If the sound field in the duct upstream of the duct segment being considered contains random (higher order) modes, add a correction of 5dB to the attenuation calculated for all frequencies above twice the cut off frequency. Between the cut off frequency and twice the cut off frequency, add a correction linearly varying with log (frequency) from 0 to 5 dB. If the duct segment length (L) is less than twice the characteristic dimension (i.e. < 2D), adjust the 5 dB higher order mode correction in accordance with:-

$$\text{Correction (dB)} = (L/2D)^2 \times 5$$

STEP 7 Large values of calculated attenuation should be treated with caution as flanking through the duct walls and self generated noise, even in low velocity systems, will impose an upper limit on the amount of attenuation that can be achieved. In practice calculated values of attenuation should be truncated to 45 dB unless special efforts have been made to ensure the greater degree of attenuation calculated can be achieved.

It is difficult to demonstrate the prediction accuracy of the above scheme in a simple manner, however Table 3 shows a comparison of predictions and manufacturers' test results for a number of selected situations.

ROUND DUCTS

The approach discussed for rectangular ducts can also be applied to round ducts with a tolerable degree of accuracy. There is a tendency to underestimate the total attenuation at low frequencies below the cut off frequency for ducts similar to round duct silencers of length equal to one times (1D) and twice (2D) internal diameter. On the other hand, the predictions for smaller diameter ducts of considerable length appear to be higher than typical values given in ASHRAE (3) for mid to high frequencies.

TABLE 3 COMPARISON BETWEEN PREDICTED ATTENUATION FOR INTERNALLY LINED DUCTS AND MANUFACTURERS' TEST RESULTS⊖

Duct Details	63	125	250	500	1000	2000	4000	8000
	Attenuation (dB) at frequency (Hz)							
25mm LININGS (PERFORATED FACED)								
279H x 330W x 3000L								
Predicted	1.2	3.4	9.9	29.1	45.0	35.2	16.7	9.3
Manufacturer	2.0	3.5	11.5	24.5	49.0	31.5	16.5	13.0
432H x 838W x 3000L								
Predicted	0.6	1.8	5.3	15.5	26.3	14.2	8.4	6.2
Manufacturer	0.0	2.0	6.0	16.5	36.5	15.0	9.5	6.0
635H x 940W x 3000L								
Predicted	0.5	1.4	4.0	11.7	15.1	9.0	6.5	5.5
Manufacturer	1.5	1.0	3.5	13.5	21.0	9.0	7.0	5.5
50mm LININGS (PERFORATED FACED)								
229H x 279W X 3000L								
Predicted	3.4	9.9	29.0	45.0	45.0	45.0	25.7	13.2
Manufacturer	6.0	9.5	31.5	46.5	46.0	42.5	25.0	17.0
381H x 787W x 3000L								
Predicted	1.7	5.0	14.5	42.5	34.8	18.6	10.4	7.2
Manufacturer	1.5	4.0	17.0	41.0	35.0	16.5	12.0	8.0
584H x 889W x 3000L								
Predicted	1.2	3.6	10.7	31.2	18.0	10.7	7.3	5.9
Manufacturer	0.5	3.5	14.0	37.0	20.5	10.5	8.0	6.0
100mm LININGS (PERFORATED FACED)								
50H x - W x 900L								
Predicted	7.1	14.1	27.8	45.0	45.0	45.0	45.0	39.0
Manufacturer	8,7	16,16	27,27	45,41	55,47	55,43	55,37	50,31
150H x - W x 900L								
Predicted	3.1	6.0	11.9	23.5	28.3	23.1	18.4	12.9
Manufacturer	3,5,5	6,9,8	11,18,17	20,26,26	25,35,33	25,27,27	15,19,19	8,16,15
200H x - W x 900L								
Predicted	2.4	4.7	9.3	18.3	22.0	16.2	13.6	8.7
Manufacturer	1,3	4,7	8,14	15,22	19,25	19,19	11,13	3,10
200H x - W x 2400L								
Predicted	6.4	12.6	24.8	45.0	45.0	40.8	28.0	15.0
Manufacturer	6,6	11,16	22,32	39,49	49,50	49,40	29,20	8,14

Duct Details	63	125	250	500	1000	2000	4000	8000
	Attenuation (dB) at frequency (Hz)							
150mm LININGS (PERFORATED FACED)✱								
150H x - W x 900L								
Predicted	5.7	10.0	17.3	25.2	26.4	20.8	16.9	11.4
Manufacturer	8.0	13.0	17.0	22.0	27.0	21.0	16.0	14.0
200H x - W x 900L								
Predicted	4.5	7.9	13.7	19.9	20.9	15.1	12.3	7.7
Manufacturer	7.0	11.0	14.0	25.0	28.0	17.0	15.0	14.0
200H x - W x 2400L								
Predicted	12.1	21.0	36.4	45.0	45.0	37.6	25.9	13.5
Manufacturer	12.0	21.0	37.0	48.0	48.0	37.0	26.0	20.0
300H x - W x 900L								
Predicted	3.2	5.5	9.6	14.0	12.3	9.3	6.4	3.3
Manufacturer	3.0	7.0	12.0	17.0	14.0	12.0	11.0	10.0
300H x - W x 2400L								
Predicted	8.5	14.8	25.7	37.5	32.8	23.9	16.2	8.0
Manufacturer	9.0	16.0	29.0	39.0	41.0	25.0	17.0	14.0
25mm LININGS (TISSUE FACED)								
204H x 255W x 3700L								
Predicted	2.4	5.9	14.5	35.9	45.0	45.0	44.5	21.8
Manufacturer	3,0,2	6,5,4	14,13,11	32,30,25	49,47,48	51,50,52	32,32,31	18,19,1
458H x 560W x 3700L								
Predicted	1.1	2.7	6.6	16.4	39.2	21.0	11.8	7.9
Manufacturer	1,1,0	4,4,2	7,7,6	17,17,16	39,41,49	22,20,20	14,15,15	7,8,8
356H x 763W x 3700L								
Predicted	1.1	2.8	6.9	17.0	42.3	27.4	14.6	9.1
Manufacturer	3,0,0	3,2,2	7,7,6	19,19,16	43,42,47	22,24,21	14,14,13	9,9,8
50mm LININGS (TISSUE FACED)								
408H x 510W x 3700L								
Predicted	1.3	4.0	12.7	39.7	45.0	30.0	17.0	9.2
Manufacturer	1.0	5.0	13.0	43.0	54.0	24.0	18.0	10.0
306H X 713W X 3700L								
Predicted	1.4	4.3	13.3	41.8	45.0	39.4	22.0	12.8
Manufacturer	0.0	3.0	10.0	41.0	54.0	30.0	21.0	14.0

✱ Predictions for 150mm linings are likely to be conservative since test results were for tapered linings.

e Manufacturers' data may include results for both fibreglass and rockwool.

4. ATTENUATIONS DUE TO LINED AND UNLINED BENDS

The attenuation data for bends are based primarily on ASHRAE (3) and Beranek (4). The results for a large variety of bend types have been assembled in Figures 5, 6, 7 where the attenuations have been normalized against the parameter (fD/c) or (D/λ). The data shown in Figure 5 for mitre bends are intended to conform very closely with ASHRAE (3). The curve for "LINED BEND ONLY" has been inferred from the results of tests by two insulation manufacturers on radiussed bends, and is fairly representative for linings with perforated facings. The curve may result in quite conservatively low predictions at high (D/λ) for linings with fibreglass tissue facing in lieu of a perforated facing.

It should be noted that the relevant dimension "D" is the dimension of the duct in the plane of the bend, not the smallest duct dimension. To achieve the full degree of attenuation, the duct sections lined ahead and after the bend should extend at least two to four times (depending on frequency) the relevant bend dimension from the entry and exit faces of the bend. For lesser lengths of lining, attenuations should be reduced. A reasonable approximation would be to either adopt the "LINED BEND ONLY" values or some intermediate values between the "LINED BEND ONLY" result and the result for the appropriate bend type with lining extending two to four times the relevant bend dimension. In practice, a high degree of sophistication is not justified since the prediction schemes have not been authenticated as being particularly precise. In fact, successive editions of the ASHRAE HANDBOOK have shown remarkably different bend attenuation figures in recent years, with apparently new data presented in one edition being notably absent in the following edition.

An important point to realize is that bend attenuation has traditionally been defined as the attenuation achieved over and above that which would have been achieved for a straight duct of the same centreline length with the same internal treatments applied. As such, it is legitimate to account for the effects of the duct lining and the bend separately and add the two effects, provided of course that the bend attenuations have been presented in this way. Unfortunately, the bend attenuation data generally available from two of Australia's insulation manufacturers have not been measured and presented so as to separately identify the individual effects of the duct lining and the bend. Thus, if these data are to be used directly, then the measurement of duct lengths for calculating the attenuation due to the duct lining should exclude the bend entirely, and the overall attenuation figures for the bend should be used to represent the total effect of the bend. When the above test data are considered more carefully, it is found that a significant portion of the bend attenuation indicated by the tests is accounted for by the length of the internal lining, and the effect of the bend is to provide some additional attenuation of the higher frequencies by converting the incoming plane waves to higher order modes as they negotiate the bend.

Lined mitre bends with short chord turning vanes exhibit less attenuation at certain frequencies than would have been achieved with a straight piece of duct having the same centre line length. Consequently, in order to arrive at a generalized prediction scheme which separately produces the attenuation components due to lined duct segments and lined bends, it has been found necessary to define duct lengths as minimum values rather than centreline values (refer Figure 8). This approach is recommended for assessing all bend shapes and configurations so as to produce conservative predictions of total attenuations. The curves in Figure 7 were determined as follows:-

(a) For "BENDS LINED AHEAD AND AFTER", as well as for "BENDS LINED AFTER", the curves shown are an average between the relevant mitre bend without turning vanes and the unlined radiussed bend.

(b) The "LINED BEND ONLY" and "UNLINED BEND" results are based on one insulation manufacturer's test results for three lined bends with vanes.

(c) For "BENDS LINED AHEAD", the curve shown for D/λ greater than two is roughly the average of the result for the relevant mitre bend without vanes and the unlined radiussed bend. For lesser values of D/λ, a smooth transition has been assumed on the basis that no enhancement of performance at low frequencies (i.e. low D/λ) will occur in the absence of any lining after the bend.

(d) For thin linings (eg. 25mm as opposed to 50mm and greater) an upper limit on performance has been deduced, based on a similar curve determined from test results for radiussed bends.

Test data on short radius internally lined radiussed bends (e.g. mean centre line radius approximately equal to the duct dimension) are similar to those traditionally attributed to internally lined mitre bends without turning vanes. It is suggested that mitre bend data can be used with some modifications for very short radius bends having a mean radius (R) less than the duct dimension (D). The modifications involve an "UNLINED BEND" curve from ASHRAE which does not exhibit any enhanced attenuation at low values of (D/λ), a curve for "BEND LINED AHEAD" which similarly does not exhibit any enhanced attenuation at low values of (D/λ) and finally a limit on performance for thin linings (eg. 25mm as opposed to 50mm or greater). For long radius bends it is proposed that the bend attenuation be reduced in accordance with $1/(R/D)^2$. Thus, the long radius bend attenuation (L.R.B.ATTN) can be expressed in terms of the short radius bend attenuation (S.R.B.ATTN) as follows:-

$$\text{L.R.B.ATTN} = \text{S.R.B.ATTN}/(R/D)^2 \qquad \text{for } R/D \geqslant 1$$

Conventional bend attenuation data relate to 90 degree bends. For bends turning through a lesser angle (Θ), the attenuation should be reduced proportionally in accordance with:-

Either $\quad \text{BEND ATTEN } (\Theta) = \text{BEND ATTEN } (90^{\circ}) \times (\Theta/90)$

Or $\quad \text{BEND ATTEN } (\Theta) = \text{BEND ATTEN } (90^{\circ}) \sin^2 (\Theta)$

The two equations above produce very similar results.

5. ATTENUATION OF PLENUM CHAMBERS

The attenuation of a plenum chamber can be calculated by a method due to Wells (5). In the first place there will be reflection of sound at the inlet to the plenum back along the inlet ducts. Of the sound energy that enters the plenum, part will be radiated directly to the outlets, part will enter the outlet ducts after repeated reflections from the chamber walls and the rest will be absorbed by any lining in the plenum. The part that is directly transmitted is proportional to the area of the outlet and inversely proportional to the square of the distance from the inlet to the outlet. It is also dependent on the direction between the outlet and inlet, disappearing altogether if the outlet and inlet are in the same wall.

The attenuation of a plenum is given by:

$$A_P = -10 \log_{10} ((S_A \cos\Theta)/(2\pi d^2) + S_A(1-\bar{\alpha})/(S_T\bar{\alpha}))$$

Where :-

- A_P = plenum attenuation (dB)
- S_A = area of outlet duct from plenum under consideration (m^2)
- S_B = total area of other outlet ducts from plenum not presently being considered (m^2)
- S_T = total area of plenum (m^2)
  = $S_1 + S_A + S_B + S_L + S_W$
- S_1 = inlet area to plenum (m^2)
- S_L = lined wall area of plenum (m^2)
- S_W = unlined area of plenum (m^2)
- $\bar{\alpha}$ = mean absorption coefficient for plenum
  = $(S_L\alpha + S_1 + S_A + S_B)/S_T$
- α = absorption coefficient of plenum wall lining material
- d = distance from inlet to outlet (m)
- Θ = angle between line joining inlet and outlet and centre line of outlet duct under consideration

For high frequencies where the wavelength is less than the plenum dimensions, the attenuation equation is accurate within a few dB. At low frequencies the equation will give conservative answers, the actual attenuation exceeding the calculated values by as much as 5 to 10 dB due to sound reflection by the plenum.

Typical absorption coefficients (α) for various plenum linings are given in Table 4. When a plenum has more than one outlet duct, the sound transmitted down each outlet duct should be evaluated separately. The absorption coefficient associated with the areas corresponding to the inlet and outlet duct openings can be taken as unity. It should be noted that this procedure automatically takes into account the split-up of energy from the inlet duct to the various outlet ducts and no further factor should be included. Where the plenum comprises a series of chambers, the total attenuation is the sum of the attenuations for each chamber.

TABLE 4 TYPICAL ABSORPTION COEFFICIENTS (α) FOR VARIOUS PLENUM LININGS

	Octave band centre frequency (Hz)						
Lining	63	125	250	500	1000	2000	4000
25 mm insulation* with perforated metal facing rigidly fixed to plenum wall	0.05	0.10	0.20	0.40	0.78	0.81	0.75
50 mm insulation* with perforated metal facing rigidly fixed to plenum wall	0.09	0.18	0.38	0.73	0.92	0.88	0.80
100 mm insulation* with perforated metal facing rigidly fixed to plenum wall	0.20	0.48	0.75	0.93	0.95	0.92	0.90
Concrete, painted brickwork or unlined metal	0.01	0.01	0.01	0.02	0.02	0.02	0.03

* Insulation corresponds to typically 48 kg/m^3 fibreglass or 80 kg/m^3 rockwool.

6. ATTENUATIONS DUE TO FILTERS AND COILS

High Efficiency Particulate Arrestance (HEPA) filters and cooling coils can provide some useful attenuation over the entire frequency range of interest. Typical values are as follows:

Item	Attenuation (dB) at Octave Band Centre Frequency (Hz)							
HEPA Filters	63	125	250	500	1K	2K	4K	8K
- 300 mm deep LUWA	1	2	2	2	4	14	12	16
- 300 mm deep EMAIL	2	4	3	4	6	10	12	16
Tempmaster Coil	1	2	4	6	7	9	10	11

7. CONCLUSIONS

A procedure for determining attenuations due to internally lined ducts has been developed which correlates well with manufacturers' test results. The procedure involves normalizing against relevant parameters such as frequency and (D/λ). The method is well suited for generalized computer predictions.

Approximate attenuations due to bare unlined duct, lined and unlined bends, plenum chambers, filters and coils have been determined from various sources and summarized in a form suitable for incorporating into a generalized computer prediction scheme.

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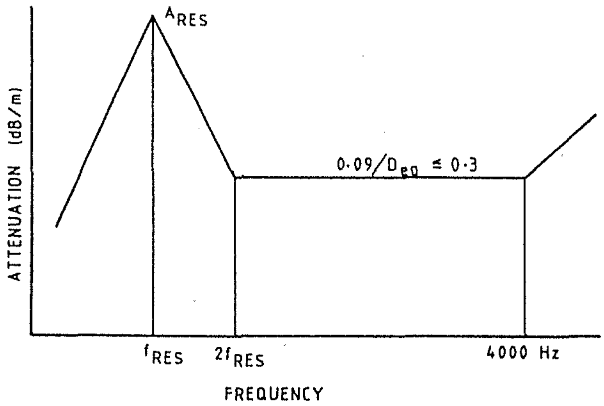

Fig.1. ATTENUATION OF BARE RECTANGULAR STEEL DUCTS.

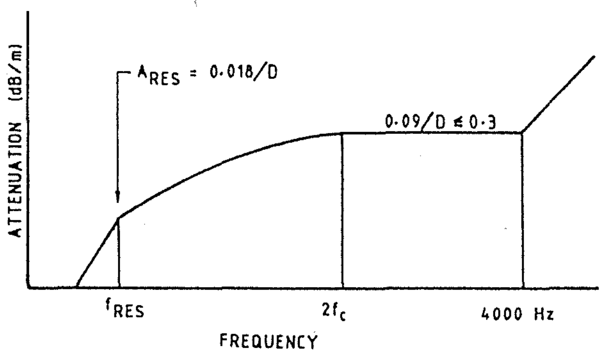

Fig. 2. ATTENUATION OF CIRCULAR STEEL DUCTS OR HEAVY RECTANGULAR DUCTS.

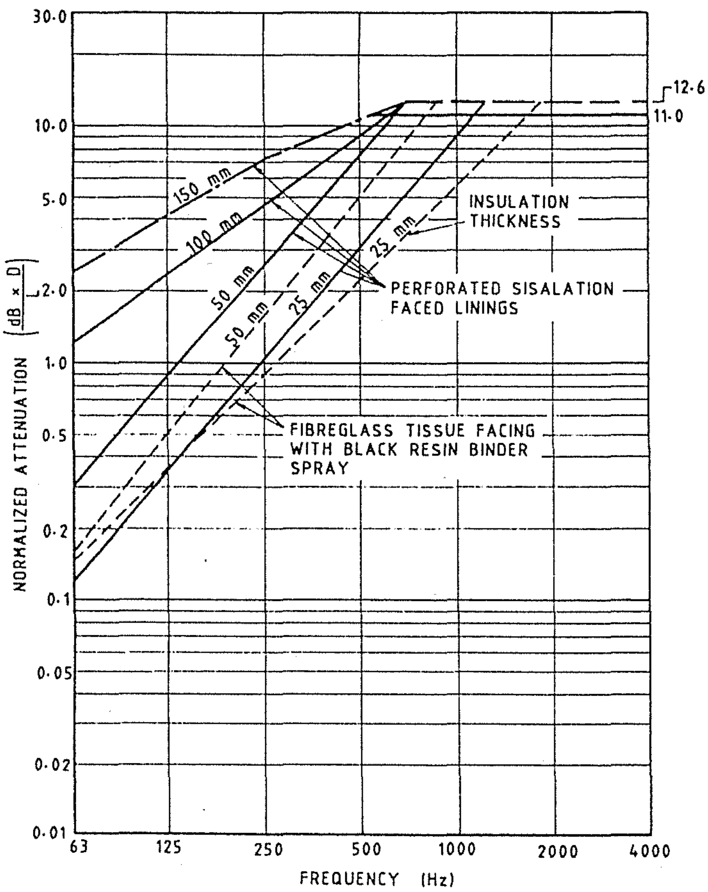

Fig. 3. NORMALIZED ATTENUATIONS AT FREQUENCIES WELL BELOW THE CUTOFF FREQUENCY.

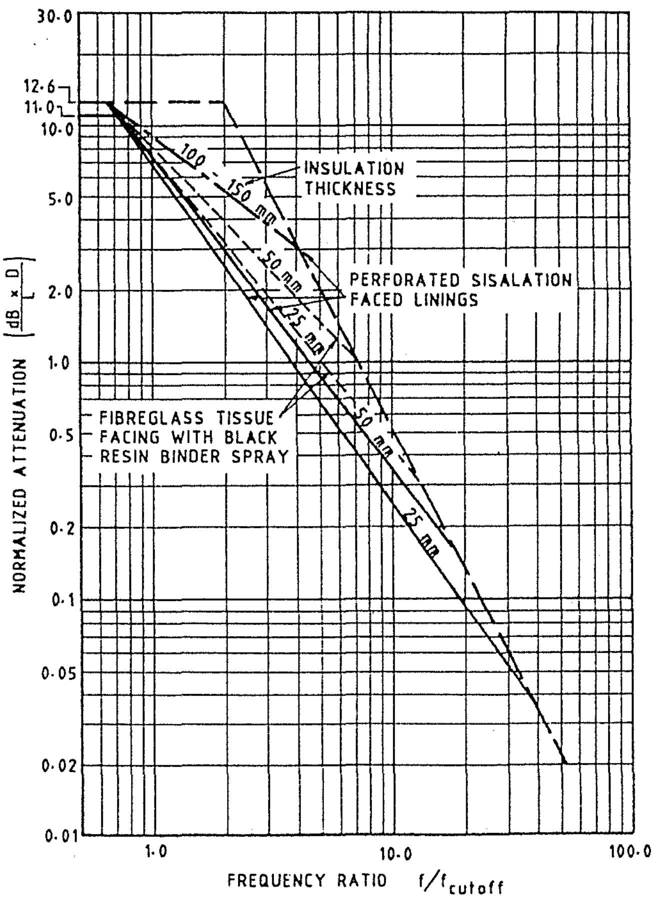

Fig. 4. NORMALIZED ATTENUATIONS AT FREQUENCIES NEAR AND ABOVE THE CUTOFF FREQUENCY.

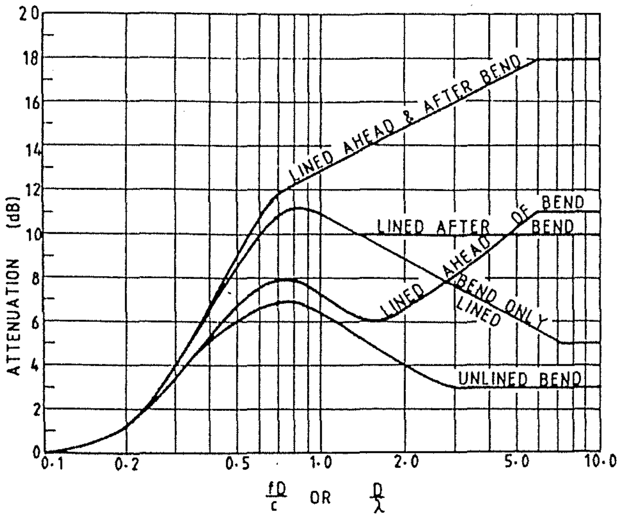

Fig. 5. APPROXIMATE ATTENUATION OF LINED AND UNLINED MITRE BENDS WITHOUT TURNING VANES.

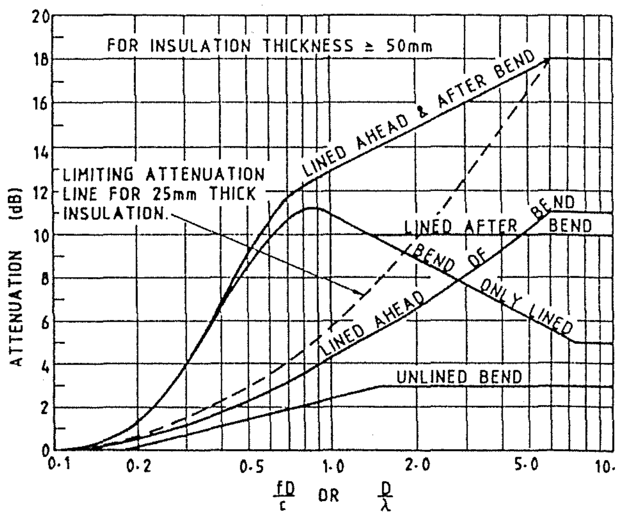

Fig. 6. APPROXIMATE ATTENUATION OF LINED AND UNLINED SHORT RADIUS BENDS (R/D ≤ 1·0).

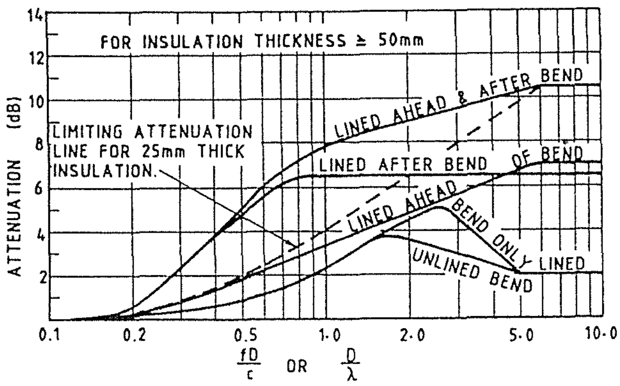

Fig. 7. APPROXIMATE ATTENUATION OF LINED AND UNLINED MITRE BENDS WITH SHORT CHORD TURNING VANES.

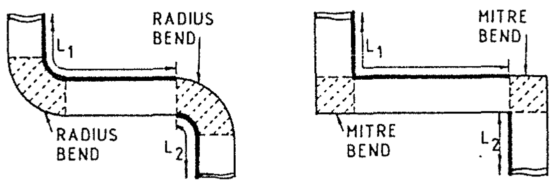

Fig. 8. SCHEMATIC ILLUSTRATION OF HOW TO MEASURE LENGTHS OF LINED DUCT SEGMENTS INCLUDING BENDS. HEAVY LINES INDICATE LENGTHS TO BE INCLUDED IN MEASUREMENT.

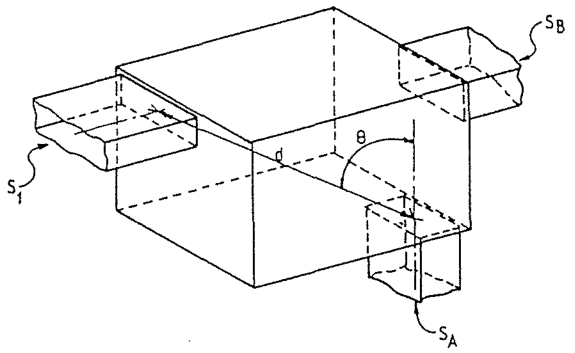

Fig. 9. TYPICAL PLENUM CHAMBER GEOMETRY.

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COMPARISON BETWEEN MOBILE LABORATORY UNIT AND STATIONARY MONITORING TERMINAL FOR MEASURING AIRCRAFT NOISE FLYOVERS

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Summary

Field measurements were conducted to collect data of both road traffic and aircraft noise levels simultaneously in residential areas around the Sydney International Airport. The results of the measurements were intended to characterize the overall environmental noise condition in the surveyed areas. The aim of the particular investigation reported here was to find out the reliability of a field measurement method for collecting the aircraft noise data using a Mobile Laboratory Unit compared with the permanent Noise Monitoring method operated by the Department of Aviation. Data of each individual flyover, including the maximum noise level and the flight information, were matched with data obtained from the Daily Aircraft Movement Report. In order to get more comprehensive results, the investigation was extended to compare the two sets of data with the predicted noise levels provided by the Australian Standard 2021. 109 jet flyovers were analysed using a SPSSX package to obtained statistical inferences of the three sets of data. The linear regression analyses revealed that the maximum noise levels (dB(A)), measured by the Mobile Laboratory Unit, agreed better with the Noise Monitoring Terminal than with the AS 2021. The correlations between the Mobile Laboratory Unit measurements and the permanent monitors showed an acceptable level of accuracy, thus a short sample measurement at any location around the airport may be used for long-term assessment of aircraft noise levels in conjunction with the Department of Aviation data.

INTRODUCTION

The study was conducted as part of a research program investigating the effect of ambient noise levels from road traffic on community response to aircraft noise. The main surveyed areas of the study were selected within the 25 ANEF contour lines resulting from aircraft operation at the Sydney International Airport. Two major survey types were conducted in the study, the first was the Noise Survey, covering both road traffic and aircraft and the second was a Social Survey to obtain data on the magnitude of disturbance due to both noise sources.

The noise measurements, using a Mobile Laboratory Unit, were made within three different residential areas, i.e. areas around the extended line of runway 25 (North East, NE), runway 16 (North West, NW) and runway 07 (South West, SW).

In order to describe the noise condition resulting from the road traffic and aircraft, it was decided to use a simultaneous recording technique where a two-channel recorder and two sets of equipment were set up at each selected site. This technique was intended to determine the noise levels of both sources at the same time.

In order to assess the long-term noise annoyance, ideally, a long-term data collection of the noise exposure of people in the area where they live is also needed. This is to ensure that the results of the objective measurement will be representative of the day-to-day noise environment experienced by the population; therefore, a better correlation between subjective responses and objective levels should be achieved. However, the time-consuming and expensive aspects of a long-term noise measurement often restrict the use of this method (1, 2). Consequently, many studies in community noise have tried to apply various temporal sampling methods in order to reduce time and the cost of the survey. However, the problems of choosing an appropriate sampling technique vary for different sites, time periods and types of noise source (3).

In relation to aircraft noise measurements, the accuracy of a temporal sampling technique for estimating a long-term noise level at a certain site is influenced by some important factors, such as, types of airport (one-direction or multi-direction runways), types of flyover-data structures (stationary or non stationary), types of aircraft usually operating and weather conditions. It is obvious that an attempt to define an accurate temporal sampling technique for measuring aircraft noise is not easy. Some previous studies in this respect (4,5) could not provide a definite conclusion about an adequate sampling method to be applied at sites around an airport. This uncertainty has led some researchers on aircraft noise to utilize data from long-term noise monitoring terminals when they are available (5,7,6).

The present study intended to apply a similar technique, that is utilizing the aircraft noise data from the fixed noise monitoring terminals around the Sydney airport. However, in many cases, the monitoring terminals may be located at some distance from the surveyed areas, thus a field measurement may still be required to describe the levels of aircraft noise at a particular site.

This investigation was conducted to find out whether the method used for the field measurement is statistically comparable with the Noise monitoring system so that the sampled field data can be extended by using additional data from the fixed terminals if necessary.

MEASUREMENTS AND RESULTS

The Noise Monitoring System at the Sydney airport currently operates 11 fixed terminals. Two of them are located within the airport area and the others are mainly positioned on the extended line of each runway (25,16,07 and 34). These monitoring terminals are controlled by the main computer of the Central Processing Unit (CPU). Fig. 1. shows a diagram of equipment and communication lines between the CPU and the terminals. The noise level of a flyover is detected by the microphone and then converted to a frequency in the range of 625 Hz to 3125 Hz. The signal is then transmitted over a normal speech quality telephone line to the CPU. A remote command from the CPU to the terminals, on the other hand, is transmitted over the same line using frequency range of 400 - 500 Hz. This remote command may include microphone calibration, terminal activation or threshold level adjustment. If a terminal detects a flyover which produce a noise level above the threshold adjustment level, the information about the maximum level (dB(A)), the terminal number and the time of the day to the nearest second, are then produced and transferred to the 'on-line' computer. Complete information about the flyover provided by the airport tower is then fed into the 'off-line' computer and matched with the flyover information from the terminal. This process provides a complete report which includes maximum noise levels, noise monitoring terminal number, types of aircraft, takeoff or landing, airline company and the time of event.

On the other hand, the Mobile Laboratory Unit uses an analog signal recording technique for further analysis in the laboratory. Fig. 2.a. shows an arrangement of the recording equipment utilised during the measurements. The time of event and the flyover information (which was identified visually), were recorded through the Cue Microphone. This insured matched information between the recorded signal and the aircraft data, such as, the aircraft type, the number of engines, the time of event, takeoff or landing, the airline company. This may be not as accurate as the previous method , especially during the night-time measurements, however, by taking into account the time of event, the field observations could be matched accurately with the CPU data. The recorded signals were then analysed in the laboratory to produce various noise measures, e.g. PNL, SEL and Maximum Levels. Fig. 2.b. represents the arrangement of equipment used for analysing the signals. The analog signals were converted into digital data by a real-time analyser, GR 1921. Each flyover signal was sampled for every 0.5 sec. to allow either PNL or SEL calculations to be made. The combination of the GR 1921 and a computer provided maximum and minimum levels as well as the sampled data. A Rion SA-25 real-time analyser was used to check the maximum noise levels provided by the computer and to calculate automatically the SEL of each flyover. However, only the maximum levels are compared here with the data from the Noise Monitoring system and the Australian Standard 2021.

STATISTICAL ANALYSIS

Due to the fact that not all of the sites were located reasonably close to the monitoring terminals, a simple extrapolation technique was applied to predict the noise levels derived from the monitoring system levels at the measurement sites.

The flight information and noise levels data from the Noise Monitoring Terminal, the Mobile Laboratory Unit and AS 2021 (7) were matched to create a SPSSX system file which contained the complete data list. Several statistical analyses were then performed to summarize the relationship between measured (MAXM) and monitored (MAXA) levels, between measured and predicted levels using AS 2021 (MAXAS) or between monitored and predicted values.

A linear regression analysis was performed for these relationships. Fig. 3.a. shows a linear correlation between MAXM and MAXA for all aircraft types during landing or takeoff operations. The correlation coefficient revealed by this analysis was 0.921 where $p<0.001$ and the regression line of this relationship follows a linear equation, i.e.,

$$\text{MAXM} = 8.83 + 0.904\ \text{MAXA (dBA)}$$

The linearity of this relationship was also tested by plotting the predicted and residual values. A random distribution pattern was observed along the zero line of the residual values (see Fig. 3.b.). This indicates linearity and homogeneity of variance.

In investigating the relationships between MAXM and MAXAS or MAXA and MAXAS, on the other hand, slightly lower correlations were indicated ($r_{MAXM\ MAXAS}$= 0.703; p<0.001 and $r_{MXA\ MAXAS}$= 0.690; p<0.001). Fig. 4.a. and Fig. 4.b. show the regression lines of the two comparisons. It is interesting to note that a predicted level of an individual flyover at a particular site using the AS 2021 may vary over a range of 20 to 25 dB(A) about the actual (measured or monitored) level. However, these level variations were mainly caused by the level variation of 2-engined aircraft where various types of aircraft may be included, such as B737, DC9 or F28. A much smaller level variation range, of about 7 dB(A) to 11 dB(A), was obtained for the 3 or 4-engined aircraft.

The relationship between the three sets of data for landing or takeoff operation were quite similar. A slightly higher correlation coefficient was discovered for takeoff ($r_{takeoff}$ = 0.939) compared with landing ($r_{landing}$= 0.891) operations between MAXM and MAXA, whereas lower correlations for takeoff were produced when comparisons were made between MAXAS and MAXM ($r_{takeoff}$= o.665; p<0.001, $r_{landing}$= 0.707; p<0.001) or MAXAS and MAXA ($r_{takeoff}$= 0.635; p<0.001, $r_{landing}$= 0.0722; p<0.001).

CONCLUSION

Table 1. summarizes the comparisons of the three sets of data in different surveyed areas. No significant difference can be identified between MAXM and MAXA levels within the three different areas. Therefore, it can be concluded that the measurement method using the Mobile Laboratory Unit is statistically comparable with the method using the Noise Monitoring system. In other words two important findings can be derived from the results, these are,

- an extension of data from field measurements can be obtained from the data from the Noise Monitoring Terminals. The data from the closest terminal is preferable.

- the Mobile Laboratory Unit can be used as a mobile monitoring system for on spot aircraft noise measurement.

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Table 1.
Correlation Coefficients from Linear Regression
of MAXA, MAXM and MAXAS data sets
within the NW, the SW and the NE sites.

Sites	MAXA vs MAXM	MAXA vs MAXAS	MAXM vs MAXAS
NW	r = 0.844 p<0.001	r = 0.730 p<0.001	r = 0.736 p<0.001
SW	r = 0.917 p<0.001	r = 0.569 p<0.001	r = 0.528 p<0.001
NE	r = 0.952 p<0.001	r = 0.830 p<0.001	r = 0.839 p<0.001

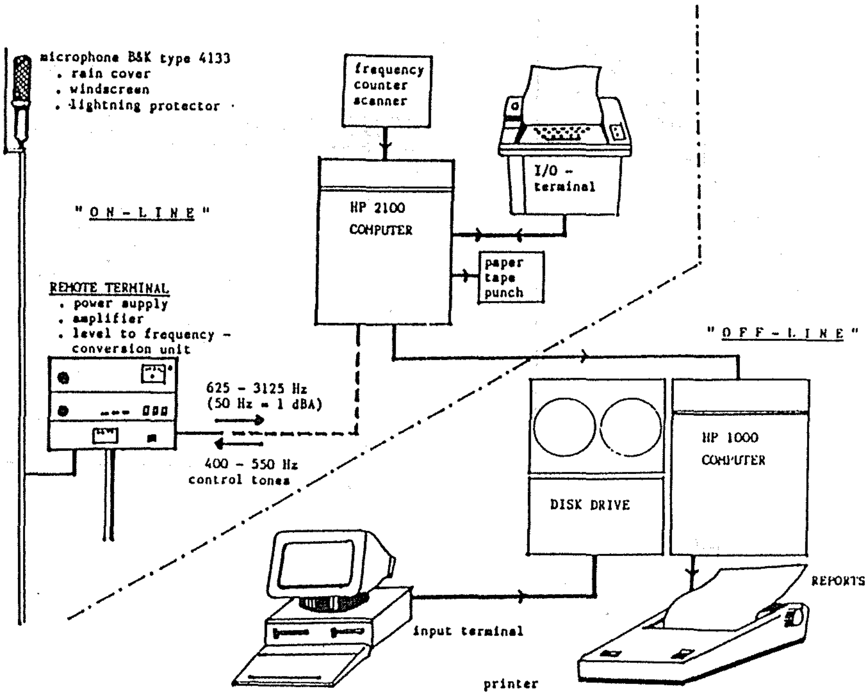


Figure 1.

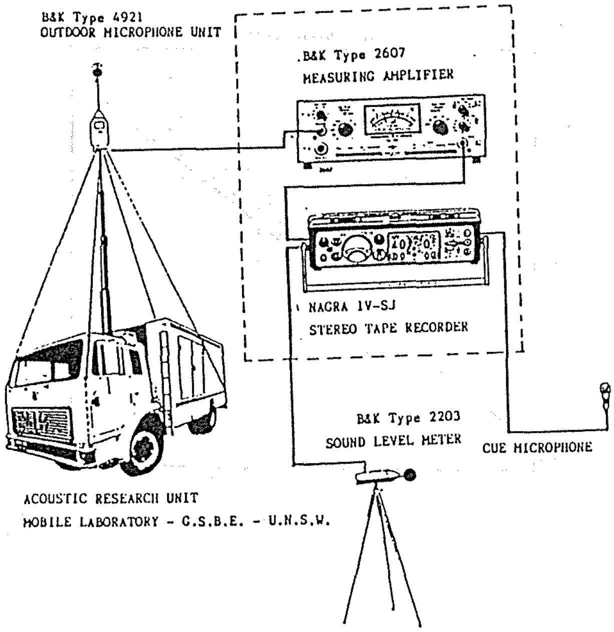


Figure 2.a.

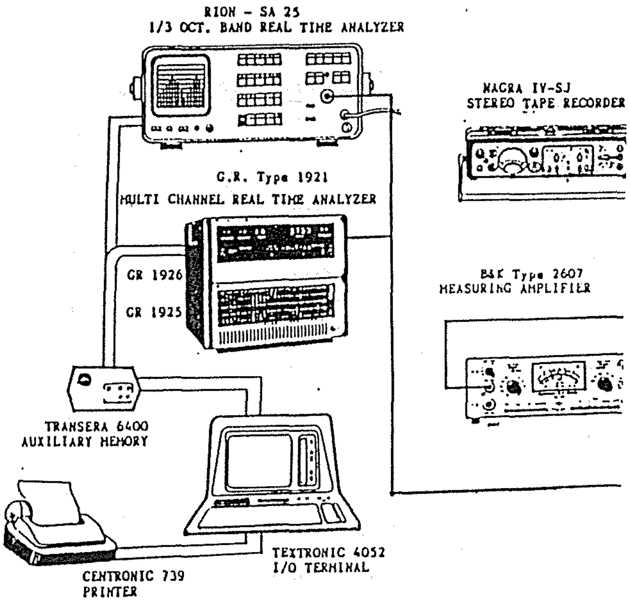
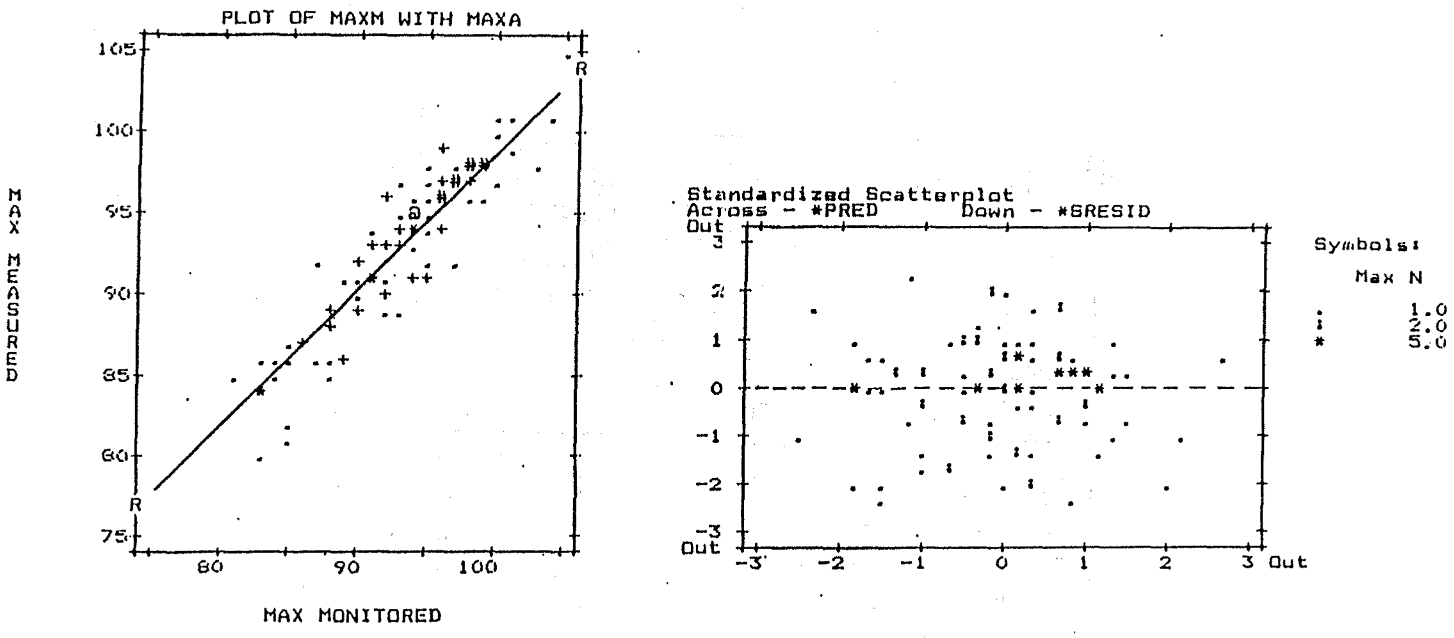


Figure 2.b.

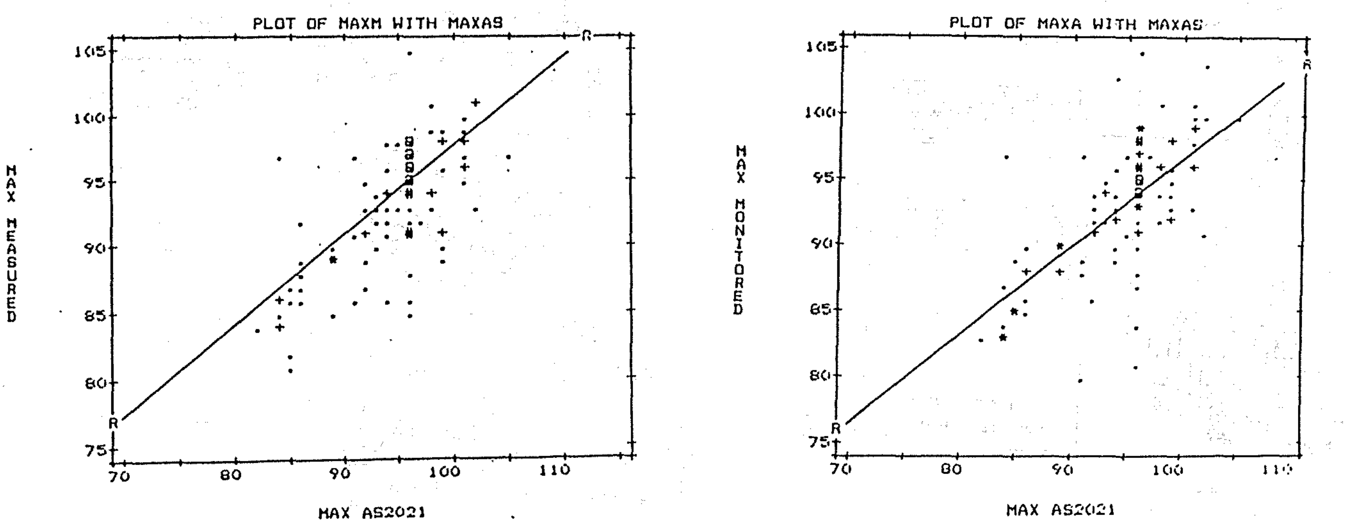


(a) (b)

Figure 3.

Frequencies and symbols used

1 - .
2 - +
3 - *
4 - #
5 - @
6 - @
7 - @



(a) (b)

Figure 4.

Subjective Assessment of Reverberation Time Using Standard Decaying Signals

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Abstract

The work presented in this paper has been undertaken in an attempt to develop an alternative method of measuring reverberation times in rooms. On the basis of the earlier work by Seraphim, it was postulated that it is possible to accurately determine reverberation times by aurally comparing the decay of sound in a room with a standard decaying signal from a tape recorder or other electronic device.

Results of paired comparison tests are presented for different reverberation times and different frequencies. The subjectively equivalent reverberation times and difference limens obtained suggest that most people would be able to use the comparison method to obtain reverberation time measurements which were sufficiently accurate for the determination of transmission loss and absorption coefficient values.

1. Introduction

The just noticable difference in reverberation time has been studied by means of electronically produced decaying signals. Seraphim's results[1] show that in the region of greatest sensitivity, reverberation times between 0.6s and 4.0s, the relative difference limen ($\Delta T/T$) is between 3% and 4%. These results were based on an octave band of noise from 800Hz to 1600Hz and level differences from 20 dB to 40dB. The difference limen was defined as the value which 75% of the test subjects could correctly identify. The dependence of the reverberation time limen upon frequency has been studied by Plenge[2], who showed that the limen (for decays with the first 200ms suppressed) decreases for frequencies below 1000Hz, and at these lower frequencies, approximately concides with the values obtained by Seraphim for octave-band noise between 800Hz and 1600Hz. From the work of Seraphim and Plenge, it can be seen that there is no need to measure differences of less than a tenth of a second for reverberation times between 1.5s. and 2.5s. These results can also lead to an application of reverberation time perception.

The current standard method for the measurement of reverberation time in enclosures, the level recorder method, can hardly reach an accuracy of 0.1s. If human ears are more sensitive than this, to differences in reverberation times, they can be made use of in reverberation time measurement. Tests have been carried out in an anechoic room and in other quiet rooms. The experimenter, who is also one of the test subjects, listened to pairs of sound decays consisting of a room decay recording and an electronically produced sound decay and then made series of comparison judgements (e.g. "longer"). The subjectively equivalent reverberation time can be calculated from the percentages of judgements and taken as estimated reverberation time of the room.

2. Psychometric Methods of Measuring Difference Limen and Subjective Equality - Constant Methods

2.1. Definition of threshold

The ideal, abstract definition of threshold is the minimal amount of energy required for the accomplishment of a perceptual task. The operational and practical definition is the amount of energy necessary for task accomplishment at some arbitrary probability criterion, usually 0.50.

2.2. Constant methods

The constant method of stimulus selection and the forced choice of indicator response are chosen as the most approporiate method of this difference limen (discrimination threshold) measurement.

The experimenter selects the stimuli to be presented, so that they are likely to cluster around the standard stimulus. Ordinarily, a relatively small number (4 to 6) of stimuli is used, typically 5, and each is paired at random with the standard stimulus and present to the subject a predetermined number of times (50 to 200), typically 50, with half the time in one order and half time in the reversed order of the paired stimuli. The pairs of comparison stimuli may be randomly intermixed or presented in blocks. It has been found that grouping the pairs of comparison stimuli in blocks of ten - that is presenting the same pair of comparison stimuli in the same order on ten successive trials - yields more stable results than that does random presentation.

After each pair of comparison stimuli is presented to the subject, he or she is to report whether the one stimulus is apparently "longer" than another stimulus. Experience shows, that for most purposes two-category (longer or shorter) data are better than three-catagory data which include "uncertain" answers. When the subject is uncertain, the experimenter lets the subject report "doubtful" and repeats such trials until the subject can make a "longer" or "shorter" judgement in order to eliminate the "uncertain" or "doubtful" from the data.

Considering the application of measuring discrimination threshold, an unconventional test method was also used in order to get a reference results for the experiment. That is, instead of presenting one stimuli pair to each subject more than 50 times, the experimenter presented each stimuli pair twice in two different orders, to more than 25 subjects. The sequence of the stimuli pairs was randomized. The test subjects made judgements after listening to each pair in the same order twice, so 50 judgements for each comparison pair can be obtained from 25 subjects. This unconventional method generally gives a greater threshold and a greater error than the conventional method does, due to the unfamilarity and uncertainty (which could cause guessing) of the subjects after listening to each pair only twice. Also the concentration of the subjects and the confidence which is built up from perceiving the previous stimuli play very important role in the test.

In presenting the comparison pairs of stimuli, the time interval between the two stimuli was kept as small as possible to make comparisons easier.

2.3. Determination of difference limen and subjective equality

The difference limen (discrimination threshold) is determined from the raw data which consist of a set of percentages, i.e. proportion of "longer" or "shorter" judgements, which associated with each of the comparison stimuli. Since one set of proportion is a complement of the other, only one distribution is left for analysing.

The psychometric function is the probability of the discrimination of "longer" judgements versus the intensity of stimuli:

$$p = \frac{1}{1+e^{-(x-x_o)/C}} \qquad (2.3.1)$$

Where p = the probability of the discrimination of "longer" judgements
x = reverberation time in sec.
x_o = the subjective equality of reverberation time in sec.
C = constant

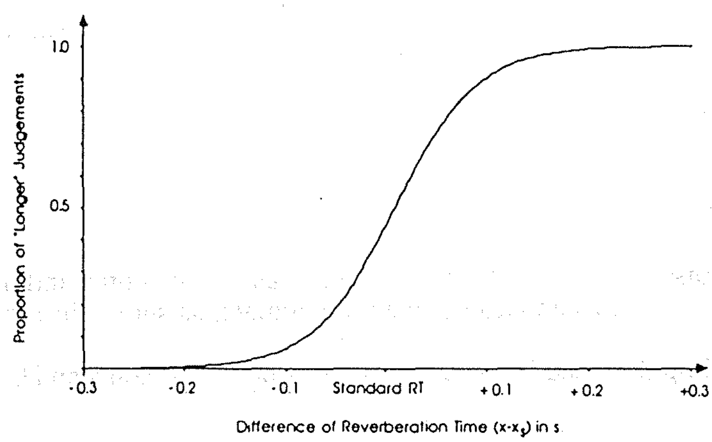

Fig. 2.3.1 The psychometric function shows the probability value (p) of discrimination (e.g. "Longer" judgements) versus difference of reverberation time ($x - x_s$), x_s is the standard reverberation time. The slope of the curve varies according to a constant C

The first step in data analysis is to correct the obtained percentages of discriminations for the chance factor. In this case, with two alternative responses, the probability of being correct by chance is 0.50. This formula is used:

$$p = \frac{p_o - p_c}{1 - p_c} \qquad (2.3.2)$$

Where p = the corrected percentage
p_o = the obtained raw percentage
p_c = the percentage correct expected by chance

In order for p to be 50% of correct discrimination, it is necessery for p_o = 75%. So 75% correct discrimination is taken as the threshold point.

So the size of the difference limen equals to the stimulus distance between 50% to 75% points on the psychometric function, which is the best fitting ogive of the data.

In order to obtain the best fitting ogive, i.e. to determine the value of C and x_o in formula 2.3.1, it is possible to transfer equation 2.3.1 to a linear function as below:

$$\ln\left(\frac{1-p}{p}\right) = -\frac{x}{C} + \frac{x_o}{C} \qquad (2.3.3)$$

Then C and x_o can be determined by linear regression of this function.

The subjective equality x_o is the mean of the distribution, which corresponds to the p = 50%. So the value of the difference limen (Δx) equals to the stimulus distance (x) between p = 50% and p = 75% from the equivalent equation

$$x = -C\ln\left(\frac{1-p}{p}\right)+x_o \qquad (2.3.4)$$

Whereas

$$\Delta x = C\ln 3 \qquad (2.3.5)$$

3. Experiment Design

3.1 The sound source

Impulsed filtered white noise (octave band filter) was used as sound source and the electronic decay of this sound source was used as the standard comparison signals, as shown in Fig. 3.1.1.

The length of the signal impulse used in the experiment is between 0.04 s and 0.18 s.

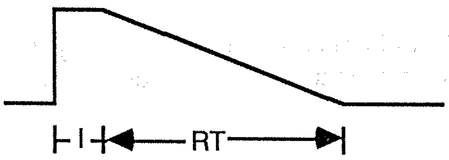

Fig 3.1.1 RT is reverberation time and l is the length of the impulse.

3.2 Instrumentation

For reverberation signal recording, the instrumentation is shown as below.

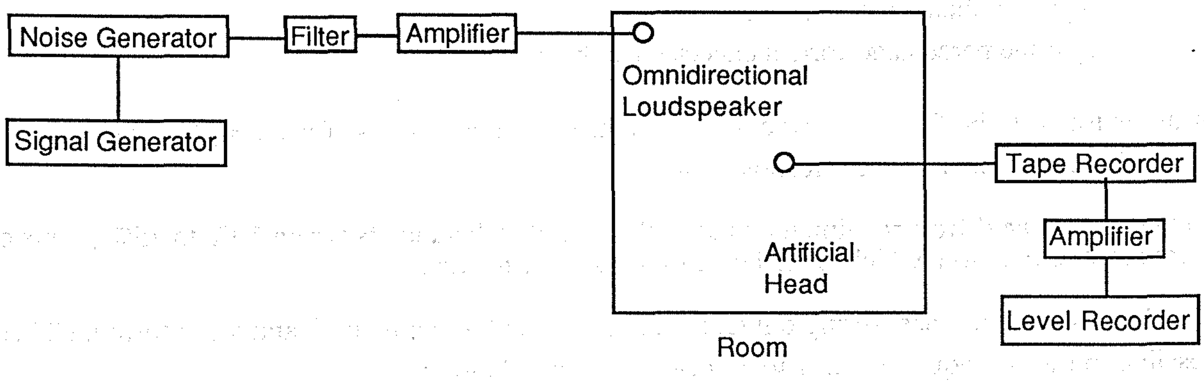

Fig. 3.2.1 Instrumentation for reverberation recording and measurement

For generating a standard sound decay signal the reverberation processor is used. The instrument installation is as shown in Fig. 3.2.2.

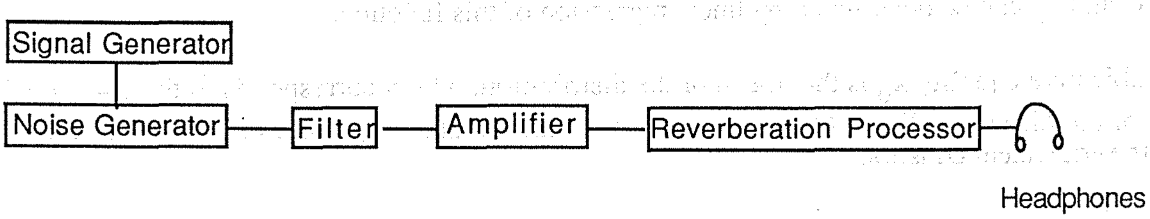

Fig. 3.2.2 Instrumentation for generating electronic sound decay

The test was carried out in a quiet rooms, including a quiet classroom, a quiet lab and an anechoic chamber. The test subjects listened to the sounds through headphones.

For the comparison test, the beginning level and the level difference between the beginning level and the end of sound decay, which were measured in dBA, have been adjusted to be identical. This was done by amplifying one of the sound signal and providing a constant white noise as background masking noise to both of the sound decays.

3.3. Experiment method of estimating reverberation time by comparison method

The sound decay in a room at each octave band centre frequency, from 125 Hz to 4000 Hz, are used as standard sound decays in the comparison test. In the preliminary test, at each octave band centre frequency, five or four electronically produced decays of sound, of which the decay times most likely cluster around that of the room reverberation sound, were chosen as the comparison sounds and paired with the room reverberation sound.

The reverberation time of the electronically produced decays of sounds can be varied from 0.3 s up, in steps of 0.1 s.

The test subjects make judgements, for each comparison pair, about whether the electronically produced decay sounds longer or shorter than the room decay. Then among the electronically produced decays of sound, two groups of sounds were derived, i.e. one group has been identified as "longer" and the other group has been identified as "shorter" than the room decay.

Paired comparisons are then made within each group of sounds (normally of two) to identify the sound decay which is closest to the room decay, i.e. to identify the shortest decay in the "longer" group and the longest one in the "shorter" group.

The subjective equality of reverberation times then gives the "measured" value of reverberation time.

4. Results of Difference Limens of Reverberation Time

4.1 Results of the unconventional method

The difference limens of reverberation times were measured, comparing the electronically produced sound decays.

The unconventional method of presenting stimuli was used in this test. The test subjects made 2 judgements for each stimuli pair after listening to them twice in the same order. 35 subjects were tested, most of them were young students.

Both of the difference limens and the subjectively equivalent reverberation times were calculated. Table 4.1.1. shows the difference limen of reverberation time between 0. 5 s to 2.0 s.

The dynamic range is 50 dBA and the impulse length is 0.11 s.

Table 4.1.1. The difference limen of reverberation time at 1 KHz octave band centre frequency. 35 subjects were tested, each subject made two judgements for each stimuli pair. The dynamic range is 50 dBA

Reverberation Time (s)	0.5	0.7	0.8	1.0	2.0
Difference Limen (s)	0.1	0.1	0.1	0.1	0.2
Subjective Equality (s)	0.5	0.7	0.8	1.0	2.0

Since only two judgements for one stimuli pair were made by each subject, it is impossible to obtain the psychometric function of each subject and then it is impossible to calculate the standard deviation of the difference limen.

4.2. Results of the conventional method

Fig. 4.2.2 shows the difference limen of reverberation time of 0.8 s versus frequency (filtered octave band centre frequency). These results were obtained by the conventional method of presenting stimuli. Three test subjects were tested and each test subject made 10 to 50 judgements for each pair of stimuli. Generally for a comparatively larger difference of the comparison pair, e.g. 0.2 s and 0.3 s, the subjects are very confident of the judgements after listening to this pair of sounds in the same order for five times. So the number of times of judgements are made for these kinds of stimuli pair can be reduced, with 5 judgements for each order of the two stimuli.

The dynamic ranges of the electronically produced sound decays at each filtered octave band centre frequency are as shown in Table 4.2.1 below.

Table 4.2.1 The dynamic ranges of the electronically produced sound decays versus octave band centre frequency

Frequency in Hz	125	250	500	1000	2000	4000
Dynamic Range in dBA	46	50	56	61	65	66

Both the difference limens and the subjectively equivalent reverberation times were calculated from the psychometric functions. Fig 4.2.2 shows measured data and psychometric function obtained by regression (average of three subjects) for a standard reverberation time of 0.8 s at 2 KHz.

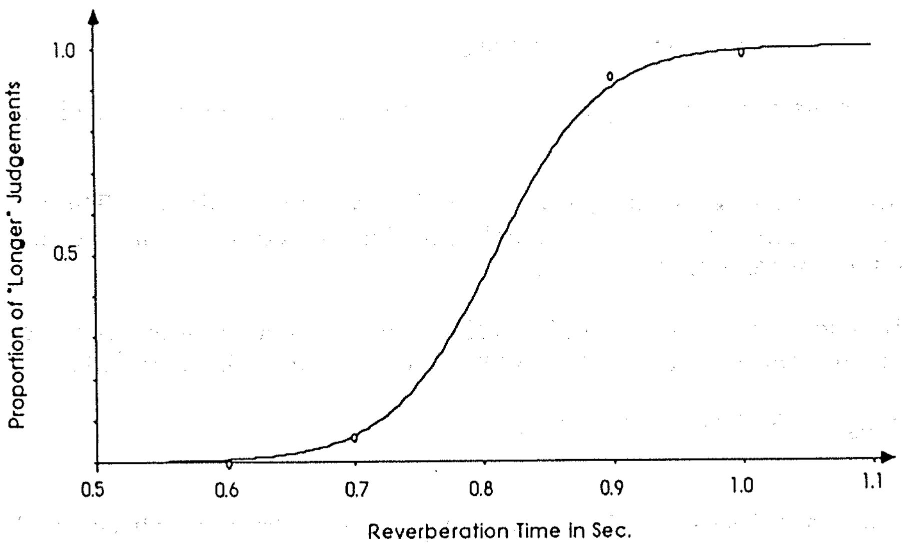

Fig. 4.2.1 The measured data and psychometric function obtained by regression (average of three subjects) for a standard reverberation time of 0.8 s at 2 KHz

The results agree with those of Seraphim's to a certain extent. One reason why there are differences is that different sound sources were used.

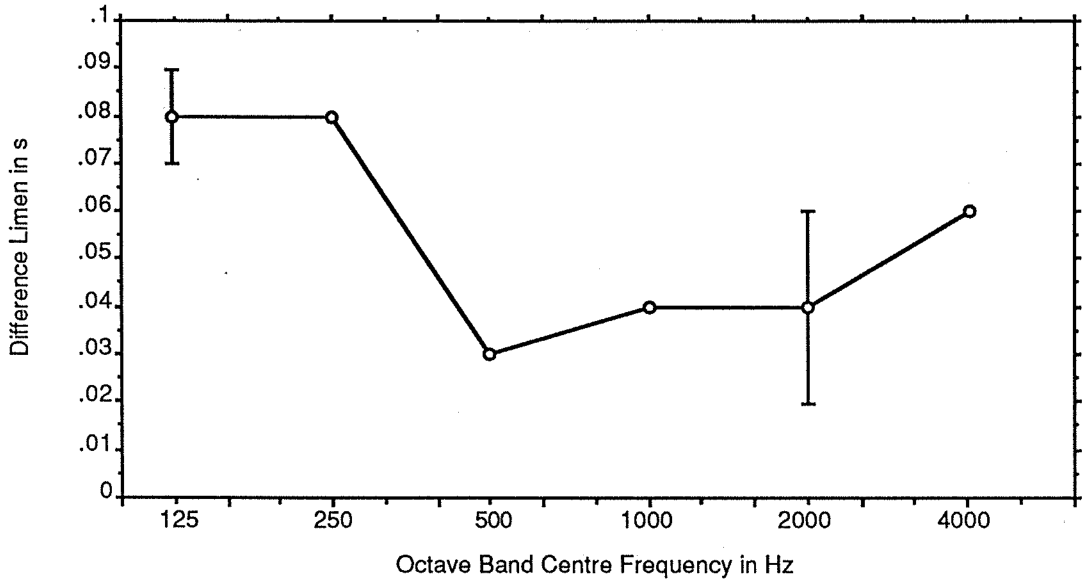

Fig. 4.2.2 The difference limen for a standard reverberation time of 0.8 s versus frequency (octave band center frequency). 3 subjects were tested, each subject made 10 to 50 judgements for each stimuli pair

5. Results of Estimating Reverberation Time by the Comparison Method

The reverberation times of an office at each octave band centre frequency was estimated using the comparison method. Two subjects were tested and each subject made 50 judgements for each pair of stimuli.

The subjectively equivalent reverberation time is taken as the measured result. Fig.5.1 shows the measurement results of both level recorder method and the comparison method.

Table 5.1. The dynamic range of the sound decays for the comparison test

Frequency in Hz	125	250	500	1000	2000	4000
Dynamic Range in dBA	25	35	35	35	35	35

The errors of reverberation times measured by level recorder method are ±0.1 s.

It can be seen that the differences of reverberation times which were measured by the standard level recorder method and estimated by the subjective comparison method are within 0.1 s.

In order to obtain general conclusion, more experiments need to be carried out.

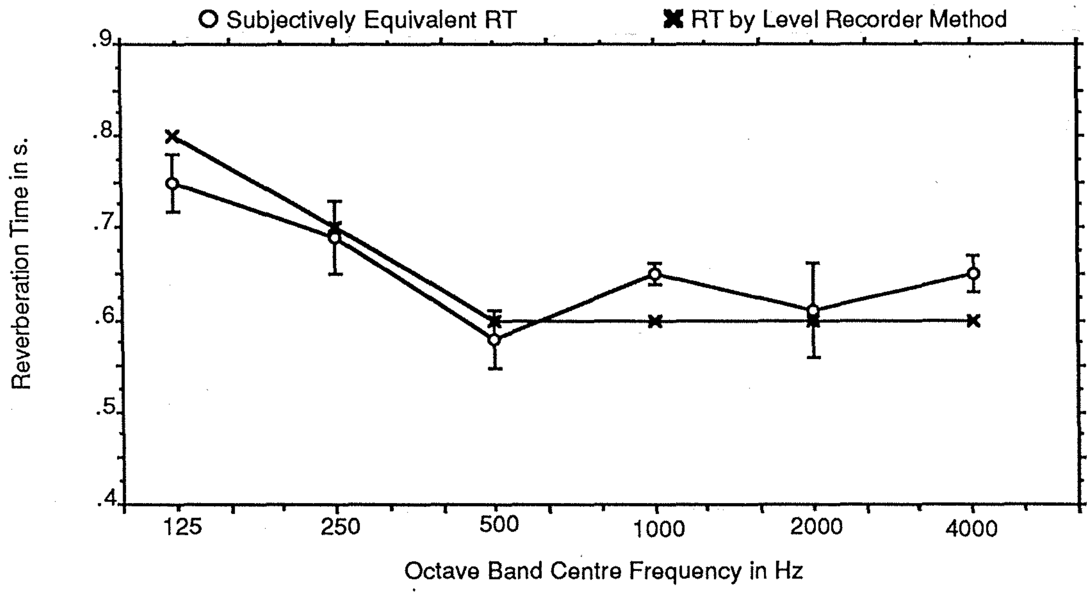

Fig. 5.1 The results of reverberation times measured by the standard level recorder method and the subjective comparison method. In the comparison method two experienced subjects were used and each subject made 50 judgements for each stimuli pair

6. Conclusion

The work presented indicates that accurate assessments of reverberation times can be made aurally. Such estimates are sufficiently accurate for most purposes. In auditoria there is little point in making reverberation time measurements to greater accuracy than people can perceive. Subjective assessments of reverberation times using the comparison method can also be used to measure reverberation times for sound transmission loss and sound absorption purposes.

Further work is being undertaken on the effects of signal dynamic range and "live" comparisons to check the viability of the method in practice.

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