ACOUSTICS
and
SOCIETY

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ACOUSTICS AND SOCIETY

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PREFACE.

These proceedings contain the papers to be given at the Annual Conference of the Australian Acoustical Society at Cowes, Phillip Island, Victoria on 18th and 19th September, 1981. The papers have been reproduced from typescripts provided by the authors. This form of publication was chosen to ensure that papers were available for delegates at the time of registration.

For more than a decade the Australian Acoustical Society has arranged annual conferences. On many of these occasions the conferences have allowed members of the Society and interested persons to examine some specific aspect of acoustics in depth. On this occasion as the title for the conference indicates, the convenors have left the way open for papers on a very wide range of topics.

Everyone knows that modern society uses acoustics in many ways and is heavily dependent on the technology associated with it. David Attenborough in his book "Life on Earth" describes man as the "compulsive communicator" and there is little doubt that man's talents in this direction are largely responsible for his "success" as a species which now overrides all others. Not only is acoustics a fundamental part of communication but it is also used extensively in medicine,chemistry, engineering, entertainment and a host of other things for the benefit of society.

It is significant that although the many beneficial uses of acoustics are an essential part of our everyday life, they receive very little attention in this conference, apart from three papers on hearing and the perception of sound. The remaining papers are devoted to the negative side of acoustics which includes noise emission and control and the effects of noise on individuals and the community. The papers are drawn from members of the Australian Acoustical Society and they therefore reflect the interests of Australian acousticians today.

Although "technological progress" is often blamed for the increasing levels of noise and the resulting predicament in which we find ourselves, society itself must take at least part of the blame. Society has allowed the existing noise problem to develop and now the increasing use of motor vehicles, machinery and various forms of entertainment and styles of living will inevitably generate more noise.

Clearly society in general likes the things technology has given it which increase the ease and enjoyment of day to day living. While complaints about noise are increasing it is also clear that only limited attempts are being made to reduce the problem. It seems that society generally is prepared to accept additional noise, preferring to spend money on further purchases of the technology which will make the future more comfortable, rather than less noisy.

It will be some time before the effects of increases in environmental noise on society are well known. Dr. Henning von Gierke in his closing comments at the Third International Congress on "Noise as a Public Health Problem" highlighted the priority that should be given to the study of these effects, by saying: "The long-term general health effects of living in noisy environments must be identified, clarified, quantified and documented in authoritative studies ---." If the results show noise to be deleterious to general health, perhaps society will be more vocal and active in its approach to reducing the problem.

Members of the Society are continuing to contribute to the knowledge of the interaction of noise and the community. We give them many thanks for their efforts in presenting the papers contained in these Proceedings.

R. A. PIESSE.

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(Papers are individually page-numbered)

LIST OF AUTHORS

PAPER TITLE: SOCIETAL NUISANCE CAUSED BY LORRIES: THE RELEVANCE OF ACOUSTICAL FACTORS

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ABSTRACT

This paper discusses the relevance of acoustical factors to lorry nuisance, and in particular, focusses on the subject of low-frequency sound generated by such vehicles. Indications as to why these vehicles appear to be making a disproportionate contribution to the noise annoyance generated by road traffic, are given. It is concluded that Heavy Vehicle Parameters, which currently figure prominently in general traffic noise annoyance prediction equations, are not acting as surrogates for low-frequency noise scales. The implications of this for the efficacy of recent Australian and U.K. annoyance prediction equations are discussed.

This paper is the second of three papers which are based on work carried out with the assistance of the Greater London Council (Scientific Branch), and which deal with the prediction of low-frequency traffic noise scales and their relevance to vibration disturbance, lorry nuisance and building vibration.

INTRODUCTION

Background

1. A good deal of attention is beginning to be focussed on the subject of the detrimental environmental effects of heavy road vehicles. The recent Armitage Enquiry Proceedings in the U.K. attest to the fact that the public has a multifarious aversion to these particular vehicles. However much research work remains to be done to differentiate the relative contributions which different aspects are making to overall community dissatisfaction, and to identify any complicating synergistic effects. Some undesirable characteristics of lorries (which differentiate them from smaller vehicles) considered to date have included - associations of danger, heavy axle loads, generation of large amounts of low-frequency as well as audio noise, and generation of visually or olfactorily unpleasant fumes and particulate pollution.

ACKNOWLEDGEMENT: The author would like to acknowledge the assistance given him during the study by Messrs Crompton, Gilbert and Hanson of Imperial College and Messrs Blackaller and Vulkan of the Greater London Council. Any opinions expressed are those of the author.

2. While it is not within the scope of this paper to discuss these areas in detail, it should be noted that there is a growing amount of evidence from surveys (e.g. see Jeanes 1980) to support the hypothesis that acoustical factors play an important part in determining public attitudes to lorries. This is not hard to appreciate when evidence from less recent surveys (see Brown 1978 and Morton-Williams et al 1978) has shown that road traffic noise is that which is heard* and felt to be bothersome, most commonly in the home (over that of other noise sources) and further, that when the heterogeneous aspects of traffic noise are considered, the noise of heavy vehicles is reported as being "noticed" most often (Brown 1978), and as being "bothersome" most often (after motor cycle and brake-tyre-squeal noise : Morton-Williams et al 1978). Lorry noise can thus be seen to be an important source of environmental nuisance to society, and an important determinant of community attitudes to lorries.

3. While the higher frequency acoustical output of lorries is no doubt an important causal factor leading to the above survey results, the insidiousness of less audible low-frequency noise, its output at high levels by the lorry population, and its ability to generate forced and free vibrations within building structures and homes (vide Martin 1978, Martin 1979, and Martin et al 1978), has recently focussed attention on the importance of this type of sound.

4. Furthermore, the current difficulties associated with the use of recent traffic-noise-annoyance prediction equations (vide Langdon 1976 a and b, and similar work by Rylander et al, Yeowart et al and Brown) have encouraged this author to investigate, as part of a wider study on prediction of low-frequency traffic noise, whether heavy vehicle flow parameters in these equations might not be acting as proxies for low-frequency noise scales. This subject is developed as a central theme within the rest of this paper.

General Approach to the Wider Study

5. The general approach was to measure low-frequency noise and traffic parameters at a number of sites, specially chosen (on acoustical grounds) to minimise inter-site variations in measured levels, which might result from differences in site propagation, absorption, diffraction and reflection characteristics. Thereafter, low-frequency noise scales would be related to traffic variables alone.

6. From the data gathered in the wider study, a number of interesting inter-noise-scale correlations were able to be made. The results of this work form the basis of this paper.

PLANNING THE SURVEYS

The Problem of Which Variables to Survey

7. The first problem to be overcome in this area was that of the choice of a suitable noise scale or index which correlates conveniently

* For this, Brown found it applied not only overall, but at each site as well - though this result is influenced by choice of site.

with people's response to the various vibratory disturbances induced by low-frequency sound. An extensive review of the literature on low-frequency sound in Hollingworth 1980 encouraged the adoption of Leq (40-125 Hz) as the most probable all-round correlator of general vibration annoyance (though predictions for other scales were made in the study).

8. The second problem regarding variables was that of the choice of traffic parameters for the study. After consideration of a number of variables (such as total flow, composition, mean speed, level of service, rate of arrival, location of exhausts on lorries etc.) and resource constraints, it was decided to measure only flow, composition and % of commercial vehicles not in top gear.

9. A review of the literature on composition variables, together with a re-analysis of data presented in MoT 1970, encouraged the adoption of a tentative classification of vehicles similar to, but slightly more extensive than, that of Nelson and Piner 1977.

10. In relation to site variables, a set of only 7 geometric layout parameters needed to be measured and recorded for each site, after the site was found to satisfy a check-list of 10 acoustic-related criteria.

11. Finally, in relation to environmental variables, none were varied in the study and none therefore required recording (other than in the form of survey comments). Thus all measurements were made under the conditions of dry road, low wind (force 0 or 1 on the Beaufort scale), and clear or patchy skies.

The Problem of Where to Survey

12. The problems of choosing a large number of reasonably uniform measurement sites are discussed fully in Hollingworth 1980.* Suffice to say in this paper, that the accuracy of the final equations developed attested to the wise choice of sites.

13. In all, 32 general site areas were chosen (5 of which were on grade), which covered quite adequately the ranges of various traffic compositions of interest.

14. Final microphone positions at each site area were determined after consideration of a number of conflicting requirements, but were generally taken 1.2 metres above road crown level, and 1 m from the nearside building facade.

The Problem of Choice of Sound Measurement, Recording and Analysis Equipment

15. The equipment chosen for field noise measurement and recording consisted of B and K Windsock, One inch B and K microphone Type 4145, B and K Preamplifier Type ZC0007 and connecting cable, a tripod, rubber bands (for suspension of preamp to tripod, to prevent interference from ground borne vibration), B and K Sound Level Meter Type 2209, a specially fabricated signal splitting box (to divide the SLM signal into two, for recording on two separate frequency sensitive tracks of the tape recorder), a Nagra

* The reader is referred to this work for details of all work henceforth summarised in this paper.

IV-SJ tape recorder (with attendant cables, commentary mic. etc.), and a B and K Calibrator Type 4230 (used for calibration purposes).

16. The equipment selected for laboratory analysis of tape recordings was B and K Sine Random Generator Type 1027, General Radio Real Time Analyser (1926 Multi/Channel RMS Detector), B and K Measuring Amplifiers Type 2606, and two B and K Noise Level Analysers Type 4426 driving two B and K Alphanumeric Printers Type 2312.

17. Equipment-noise tests have shown that the above system represents a high quality noise measurement, recording and analysis system.

SURVEY AND ANALYSIS PROCEDURES

18. Full details of all survey and analysis procedures are documented in Hollingworth 1980.

SOME PRELIMINARY RESULTS AND CONCLUSIONS

19. The actual noise level results have been shown to be accurate, both for the higher frequency scales and the low-frequency scales. For the former scales, (Predicted Minus Measured)* Error Populations for $L_{10}A$ have been shown to compare favourably with those reported by Delany et al 1976, and those by Saunders and Jameson 1978. Moreover, the results for $Leq,_A$ have been shown to be no different statistically from those quoted by Saunders and Jameson 1978.

20. In support of the accuracy of the low-frequency scales measured, Table 1 shows the type of errors involved when the prediction method developed as part of this study is applied to low-frequency noise scale data published by other sources.

21. The form of the equations used to predict values in Table 1, is taken up in more detail in another paper. Suffice to say here that the equation is based on a bivariate regression and because of its form, is able to show that medium commercial vehicles (including buses) make 11 times the contribution of "cars and light vans" to low frequency noise levels, while heavy commercial vehicles make 41 times the same contribution!

The insidiousness of the type of noise investigated (in terms of its potential to disturb pedestrians, as well as its ease of penetration into dwellings with resultant bodily, auditory and general vibrating effects), is thus postulated to be at least a contributory cause of the general public dislike of heavy vehicle flows.

LOW FREQUENCY NOISE SCALES AND dB(A) LEVELS SOME FURTHER RESULTS AND CONCLUSIONS

22. An hypothesis was then tested. This was that $L10_A$ scales, because they are derived from levels of sound where low frequency content has been greatly attenuated, are not an accurate measure of the amount of low frequency noise at a site, and as such, need to be supplemented by proxy measures of heavy vehicle flow to enable reasonable predictive accuracy of community annoyance to traffic noise. (See the Introduction of this paper for background to this hypothesis.)

* Predictions were based on the DoE (1975) Method.

23. It was decided to regress the broad-band vibration annoyance scale Leq (40-125 Hz) on $L10_A$ to assess the degree of correlation, if any. The result was:

$$\text{Leq (40-125 Hz)} = 16.6 + 0.89 \times L10_A$$

with R = 0.94, R^2 = 88.3%. S.E.E. = 1.4 dB

and the regression was found significant at the 1% level.

This high degree of shared variance (reference the Coefficiency of Determination) between the two scales was a most interesting result. Figure 1 shows the data plotted. It was found that not only were this low-frequency scale and $L10_A$ highly correlated, but also that there was no consistent pattern of high-gradient and/or "low-gearing" sites having higher low-frequency levels than expected from their $L10_A$ measurements.

24. It was therefore concluded that the current inclusion of heavy vehicle flow parameters in traffic-noise annoyance prediction equations, was not because such were acting as proxies for varying levels of low-frequency noise.

25. Brown 1978 has postulated that, at least for his data, his heavy vehicle flow parameter was acting as a surrogate for noisy vehicles and events. This was based on the comments received during his survey of households. It is considered that this observation is probably correct since detailed spectral analysis observations show that many short-duration noisy events (such as gear changing, engine revving, brake squeal, man-holes bumping, air brakes hissing etc.) quickly become submerged into overall octave band levels and are thus insensitive to identification by usual energy-equivalent or percentile scales. This commonly leads to equivalent dB levels in one-third octave 32 second RMS integrations whether a small number of very noisy events are included or not. Thus it is quite probable that heavy vehicle flows act as a better proxy for noisy events than do low frequency noise scales. It is thus concluded that heavy vehicle flow parameters will probably remain a consistent and unchanging part of traffic-noise annoyance prediction for some time to come. Prediction of annoyance under noise barrier, housing-insulation or other ameliorated conditions can probably not be predicted accurately by recourse to conventional noise scales alone, but will need to be determined by survey of response of residents under such conditions.

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TABLE 1

Verification of Study's Leq (40-125 Hz) Prediction Method (which includes Gradient Corrections)

Measurement Reference	Source of Data*	Compliance with Site Constraints	Measured* Leq (40-125)	Predicted Leq (40-125)	Error (P-Meas.) dB
(1) Ludlow (10% grade)	after Martin 1978	Yes - except height of mic	88.3	89.7	+ 1.4
(2) Lewes (4% grade)	" " "	ditto	85.0	88.3	+ 3.3
(3) Guildford	" " "	No - but an adjustment made therefore	87.0	88.8	+ 1.8
(4) Slough Road Uxbridge	after Martin, Nelson and Hill 1978	Yes	86.3	84.1	- 2.17
(5) Ditto	Ditto	Ditto	85.7	84.1	- 1.6
(6) Warsaw	After Miazga and Janicka 1979	Corrections made	74.8	77.5	+ 2.7

Error Population mean = + 0.9

Error Population s = 2.27

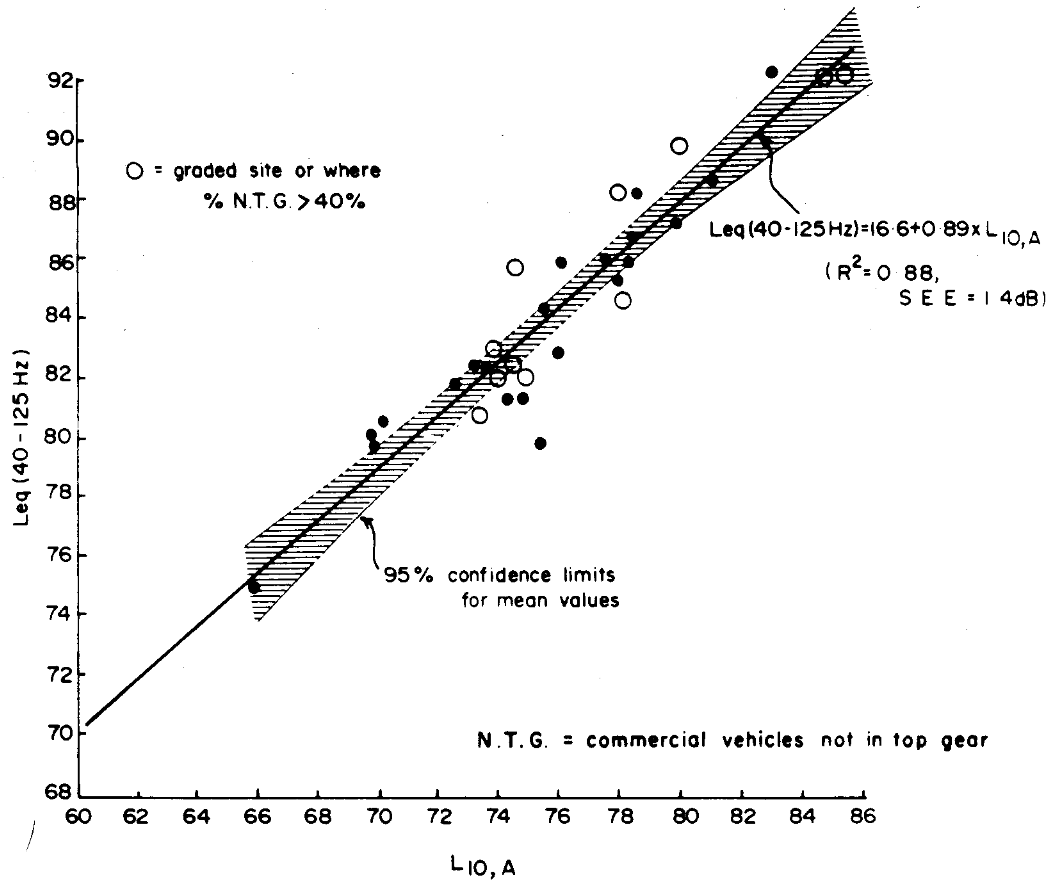

FIGURE 1.

NOISE EMISSION OF ENGINE/EXHAUST BRAKES ON HEAVY VEHICLES

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Abstract

Noise from engine and exhaust brakes fitted to heavy vehicles is a source of annoyance in the community, particularly in urban areas. This paper describes the results of a limited study carried out to document the noise emission due to the use of the brakes. A special noise test, designated as Reverse 28A was devised.

Overall, the differences in measured peak A-weighted noise level between brake and no-brake conditions suggested that there should be little or no subjectively perceptible change in loudness. It was hypothesized that the annoyance due to the brakes is a result of a change in the spectral characteristic of the noise emission rather than due to an increase in the overall peak noise level.

INTRODUCTION

One of the major sources of annoyance due to medium and heavy truck noise is the exhaust. This often dominates the other possible source contributions (e.g. the engine, transmission, cooling fan and air intake).[1] The effect of fitting an exhaust/engine brake (EEB) to a truck is to increase the pressure pulsations in the exhaust at the fundamental firing frequency[2]. It is not surprising, therefore, that the use of EEB's in urban areas is a major source of community annoyance.

This paper describes the results of a limited study carried out to document some of the significant aspects of the noise emission associated with the use of EEB's on heavy vehicles.

VEHICLES TESTED

A representative mix of EEB's on the most common engine configurations was identified and the ten combinations listed in Table 1 were tested. The vehicles fitted with transmission retarders were also tested to allow additional comparisons to be made.

TEST PROCEDURE

In order to document the noise due to engine and exhaust brakes, a special test, designated Reverse 28A, was devised. This test method was based on the present vehicle noise test requirements, Australian Design Rule 28A[3], and is described below.

The vehicle was driven past two precision Sound Level Meters (SLM) and the maximum sound level in dBA was recorded on each side of the vehicle. The microphone of the SLM was 1.2 metres from the ground and 7.5 metres from the centre line of the vehicle. The vehicle was driven at maximum engine speed in a gear which resulted in a vehicle speed between 49 kph and 55 kph.

As the vehicle drew level with the SLM the driver removed his foot from the accelerator and engaged the EEB so that the brake was activated as it drew level with the SLM. The entire procedure was repeated to ensure consistency of results. The difference in sound levels was then taken to equal the effect due to the EEB. The test site and microphone layout were as prescribed by Australian Design Rule 28A.

RESULTS

Peak noise levels were obtained on both the left hand side (LHS) and right hand side (RHS) of each vehicle. Table 2 addresses the difference in EEB versus no brake conditions. The noise level differences were calculated by arithmetically averaging both LHS and RHS noise levels for all runs for a given condition and by then obtaining the differences. Calculations show that some of the differences between EEB versus no brake conditions reach a 95% level of statistical significance.

In addition, even though there is a tendency for the Jacobs brake to be slight noisier than the other types, there is no significant variation in the peak noise level difference (PNLD) between the types of brake. Table 3, which shows the average PNLD between brake on/off conditions in dBA for different brake types also shows this.

Thus, the results would indicate that exhaust brakes are no quieter than engine brakes. They also further suggest that there should be little or no subjectively perceptible change in loudness between the brake-on and brake-off condition.

The peak spectra obtained over the measurement period for each of the vehicles tested were plotted and the data presented in $^1/_3$ octave and octave-band form. In general, the peak octave-band shape did not vary significantly between the vehicles. On the other hand, in terms of $^1/_3$ octave bands, some significant differences occurred. It is therefore suggested that the frequency content of the noise emission is an important factor in determining the annoyance resp

DISCUSSION

The results obtained in Table 2 were somewhat surprising as it was commonly expected that the use of EEB's would lead to a significant increase in A-weighted peak noise level during passby and Reverse 28A tests. This expectation was based on the wide experience of people living near main roads and intersections who complain of a "barking" effect due to EEB's. In conjunction with some evidence from peak spectral analysis, the conclusion is however formed that the annoyance due to EEB's is a result of a change in the spectral characteristic of the noise emission rather than the increase in overall A-weighted peak level of the noise emission.

Another result of Tables 2 and 3 is that exhaust brakes are not significantly quieter than engine brakes in the sense that the PNLD is similar for both types of brake. This result is not in accordance with the subjective opinion of some members of the trucking industry who have indicated that engine brakes are noisier.

CONCLUSIONS

1. For some vehicles, a statistically significant A-weighted peak noise level increase occurred between engine/exhaust brake versus no brake condition.
2. No significant difference in the peak noise level difference occurred between types of brake.
3. The A-weighted peak noise level is not an adequate descriptor for assessin the changes in annoyance due to the engaging of an engine/exhaust brake. The evidence indicates that the perceived annoyance is, in fact, due to a change in the spectral characteristics of the noise emission. Thus, the relationship between noise level, frequency and time should be more fully explored so as to allow the derivation of a useful predictive metric for engine/exhaust brake annoyance.

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3. AUSTRALIAN DESIGN RULE 28A - Motor Vehicle Noise, Department of Transport, Australia.

Engine/Exhaust Brake Type	Vehicle
Butterfly - Exhaust	Fiat
"	Mercedes
"	UD-Nissan
"	M.A.N.
"	Volvo
Cone-in-seat - Exhaust	Volvo
Jacobs - Engine	Atkinson
"	International
Dynatard - Engine	Mack (Laden)
"	Mack (Unladen)
Telma - Electro-magnetic	Bedford
Renk - Hydraulic Retarder	M.A.N.

TABLE 1 : List of Heavy Vehicles Tested

Vehicle	Type of Brake	Reverse ADR 28A 55 - 49 kph
Mercedes	Exhaust	1.3
M.A.N.	Exhaust	2.2
U.D.	Exhaust	1.7
Fiat	Exhaust	-1.1
Volvo Bus	Exhaust	0.4
Volvo	Exhaust	0.3
Atkinson	Engine	3.2
I.H.	Engine	2.6
Mack (Laden)	Engine	2.0
Mack (Unladen)	Engine	1.3
Bedford	Telma Electro Magnetic	1.1
M.A.N.	Hydraulic	1.3

TABLE 2 : Peak Noise Level Difference Between Engine/Exhaust Brake Engaged to Not Engaged in dBA

Brake Type	Reverse ADR 28A 55 - 49 kph
Exhaust	1.5
Engine	2.3
Retarder	1.2

TABLE 3 : Average Peak Noise Level Difference Between Brake On/ Brake Off Conditions in dBA for Different Brake Types

VICTORIAN EPA RECREATIONAL VEHICLE NOISE STUDY

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Abstract

The Victorian EPA has completed a study of noise from mini bike racing tracks at the request of the Melbourne and Metropolitan Board of Works (MMBW). This study included a program of field measurements and an annoyance survey and led to a 350 metre buffer zone being recommended for future mini bike tracks. This distance corresponds to a maximum noise level of approximately 60 dB(A) L_{eq} and can be compared with a 500 metre buffer zone introduced in Belgium for Formula 1 racing circuits which corresponds to a maximum noise level of approximately 70 dB(A) L_{eq}.

1. Introduction

The purpose of the EPA study was to develop guidelines that can be used by the MMBW when assessing future permit applications for mini bike tracks to ensure that noise-compatible land use planning results. The MMBW originally requested that the study include go-karts, racing cars, speedway bikes etc., but, because of the limited amount of time available, the study had to be restricted to one kind of racing vehicle only. Mini bikes were selected because they constitute the main problem.

2. Field Measurement Program

Thirteen recordings, each of at least 30 minutes duration, were made at 6 mini bike tracks in the Melbourne Metropolitan area. A stereo Nagra was used so that spurious noises such as light aircraft overflights could be edited out during subsequent analysis. The descriptors L_1, L_{10} and L_{eq} were used.

Figure 1 shows L_{eq} plotted against the distance from the measurement point to the nearest part of the track. The spread in noise levels can be attributed mainly to the range of "source strengths" of different tracks rather than to propagation or shielding effects. All measurement locations had a line of sight view of the track and were located down wind. Nearly all measurements were taken when the wind speed was less than 6 metres per second. However, at one site the wind ranged from 6 - 8 m/s and at another from 5 - 10 m/s.

These results can be compared with those of Cops and Myncke which are for motor cycle motor-cross races in Belgium (see Figure 2). As would be expected, the maximum levels for the motor-cross races are higher than those for mini bikes as the former motor cycles have larger capacity engines and more are allowed on the track per race. Cops and Myncke also report the results of an extensive measurement program of racing car noise levels.

3. Determination of the Acceptability Criterion

Cops and Myncke (1977a) indicate that the Belgian Government has specified the following buffer zones between racing circuits and residential areas (and parks) : 350 metres for tracks holding one event per year and 500 metres for permanent circuits. Unfortunately the authors do not state explicitly how these distances were determined. Their data shows, however, that maximum levels of 65 dB(A) L_{eq} (15 minute) and 70 dB(A) L_{eq} (15 minute) can be expected 500 metres from motor-cross and Formula 1 car racing circuits respectively. It is possible to derive from their data that the L_{95} levels measured during intervals between races ranged from 35 to 40 dB(A). Hence the 500 metre buffer zone specified for permanent circuits permits a surprisingly large protrusion above ambient. (Perhaps the acceptable level was chosen from an analysis similar to that by Schultz (1978) whose regression line fitted to the results of eleven surveys of annoyance caused by transportation noises shows that an L_{DN} of 70 dB(A) will cause 24% of the population to be highly annoyed.)

Because the level used by Cops and Myncke suggests that residents may be prepared to tolerate higher levels of racing vehicle noise than a standard such as AS 1055 (1978) would permit, it was decided early in the EPA study that the acceptable noise level would be based on the results of a survey of residents' reactions to mini bike noise. Fortunately EPA was able to have access to the raw data from such a survey planned and carried out by a staff member and students of the Burwood State College (Victoria). In this survey residents located at distances of from 220 to 1430 metres from a track at Waverley were asked twelve questions concerning the noise from the track. The last question was: How would you rate your annoyance?

NONE
ONLY SLIGHTLY
MILDLY
FAIRLY ANNOYED
VERY ANNOYED

(Although it is more common to use a semantic differential scale when measuring annoyance , semantically labelled scales such as this have advantages when it comes to choosing an acceptable noise level).

The raw data was analyzed in two ways: by plotting individual annoyance scores against actual distance to the track and by grouping the data and plotting mean score against mean distance for each of the groups of houses. For the grouped data the semantically labelled score was converted to a numerical score by assuming "very annoyed" = 5, etc. (See Brown and Law (1976)) As the Annoyance Survey was not supported by noise levels except at two points a log distance scale was used as substitute for a (reversed) noise level scale. Regression lines were fitted to both graphs the slopes agreeing to the fourth decimal place and the intercept to the second. Figure 3 shows the locations of the houses relative to the track and Figure 4 shows the grouped annoyance data. "Mildly annoyed" was selected as the acceptable level of annoyance and the required buffer zone of 350 metres read from the regression line in Figure 4. (It should be noted that the closest group of houses include some at 350 metres from the nearest part of the track. This was the reason for choosing 350 metres rather than the mean distance of 300 metres). An EPA field measurement at 350 metres from the Waverley track indicated that L_{eq} (30 minute) levels as high as 62 dB(A) can be expected due to the mini bikes. This is certainly higher than a standard such as AS 1055 (1978) or EPA Policy N-1 would suggest for an area where the background, measured as an L_{eq}, would be around 40 dB(A). Nevertheless, the answers to the other eleven questions on the survey sheet support the conclusion that residents find levels as high as 62 dB(A) L_{eq} (30 minutes) only mildly annoying.

4. Discussion

It is purely coincidental that the same distance, 350 metres, has been selected as a buffer zone for motor-cross and Formula 1 tracks in Belgium holding only one event per year and by the Victorian EPA for mini bike tracks holding events year round. The Cops and Myncke data shows that maximum levels of approximately 68 dB(A) and 74 dB(A) L_{eq} can be expected 350 metres from motor-cross and Formula 1 race tracks respectively. This compares with maximum levels of approximately 60 dB(A) L_{eq} at the same distance from mini bike tracks.

For mini bikes, the area occupied by the race track itself is typically 150 by 250 metres. The pits area, warm up track, car park and spectator area lie outside this. As the buffer zone of 350 metres is to be measured from the nearest part of the track, parcels of land approximately 850 x 950 metres will be needed. Effectively, this means that new mini bike tracks will have to be located in the outer suburbs of Melbourne.

Acknowledgements

The Author wishes to acknowledge the assistance given by R.Monteith and C.Senese in the field work program and in the preparation of the diagrams for this paper. The Author would also like to thank Mr. A. Keeble of Burwood State College for making available all the raw data for the Waverley Annoyance Survey.

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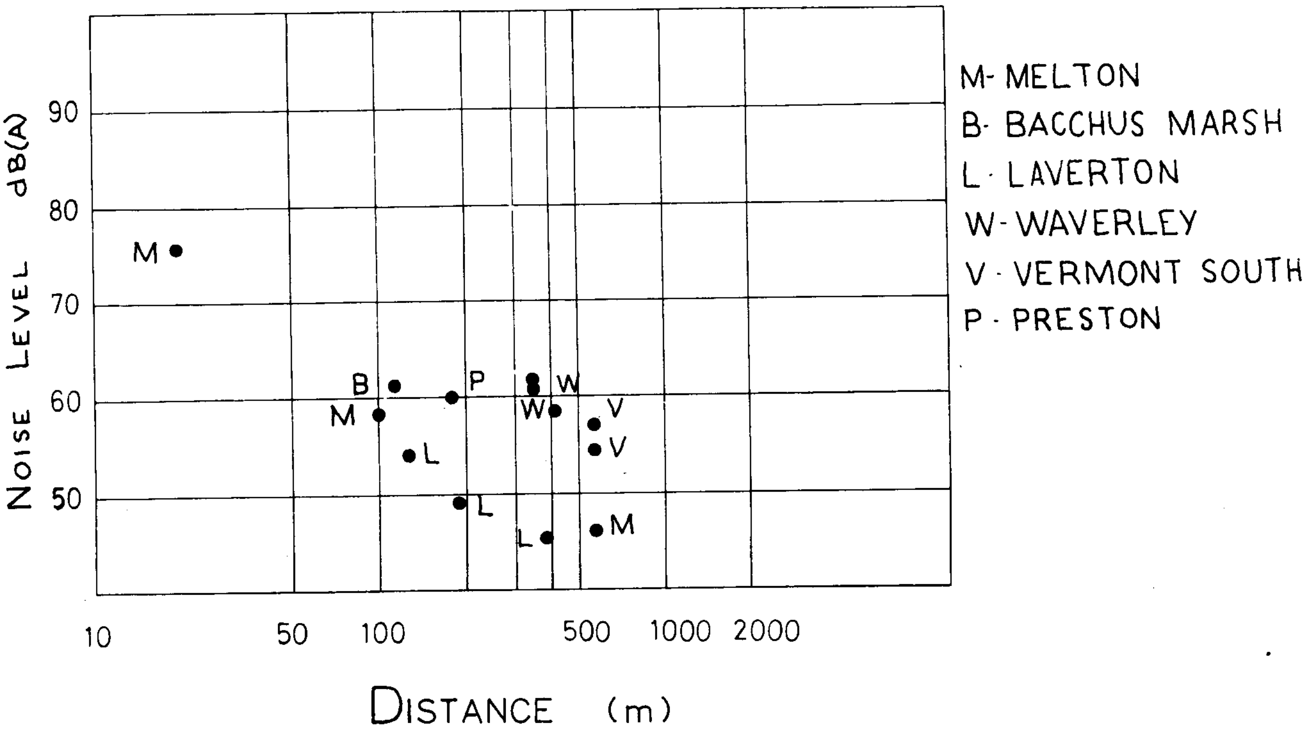
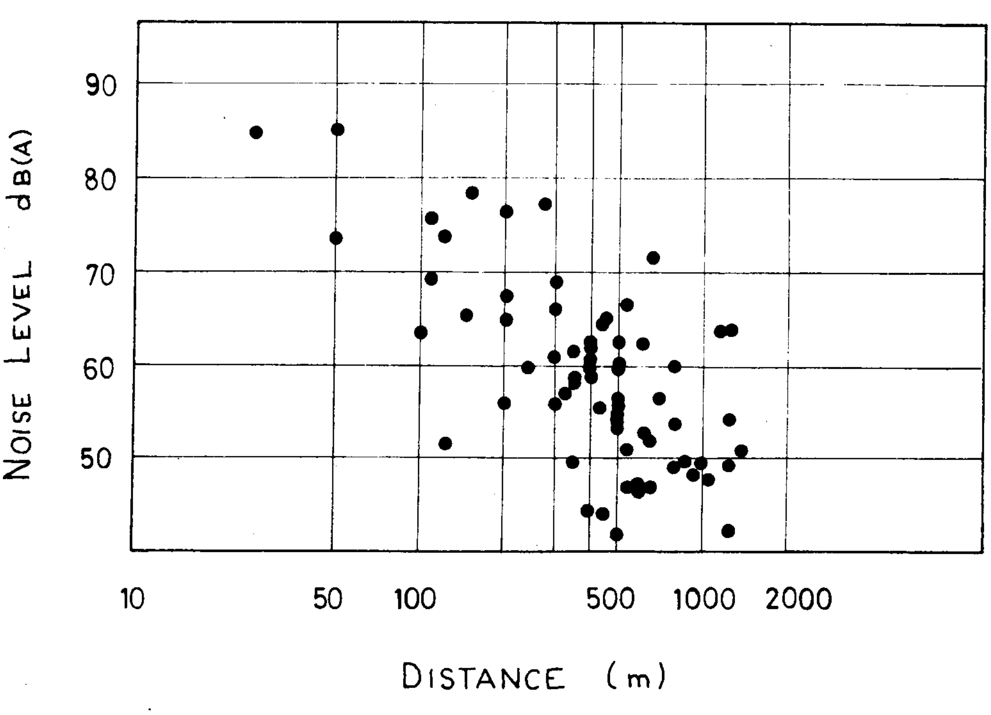
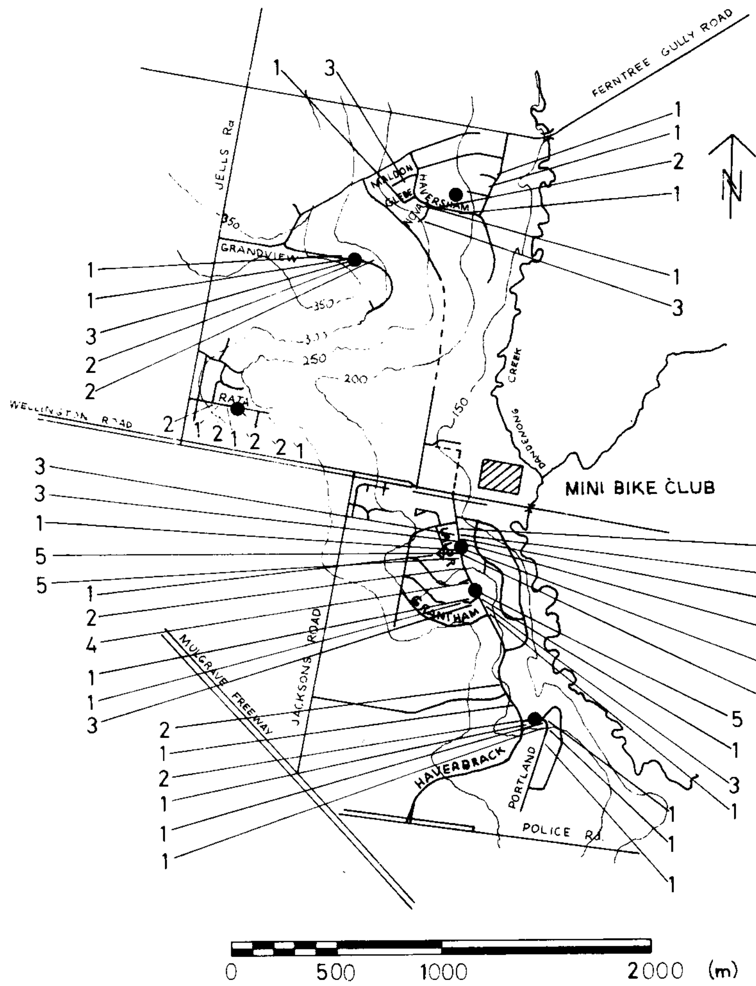
RESULTS: E.P.A MINI BIKE NOISE SURVEY Leq FIGURE 1



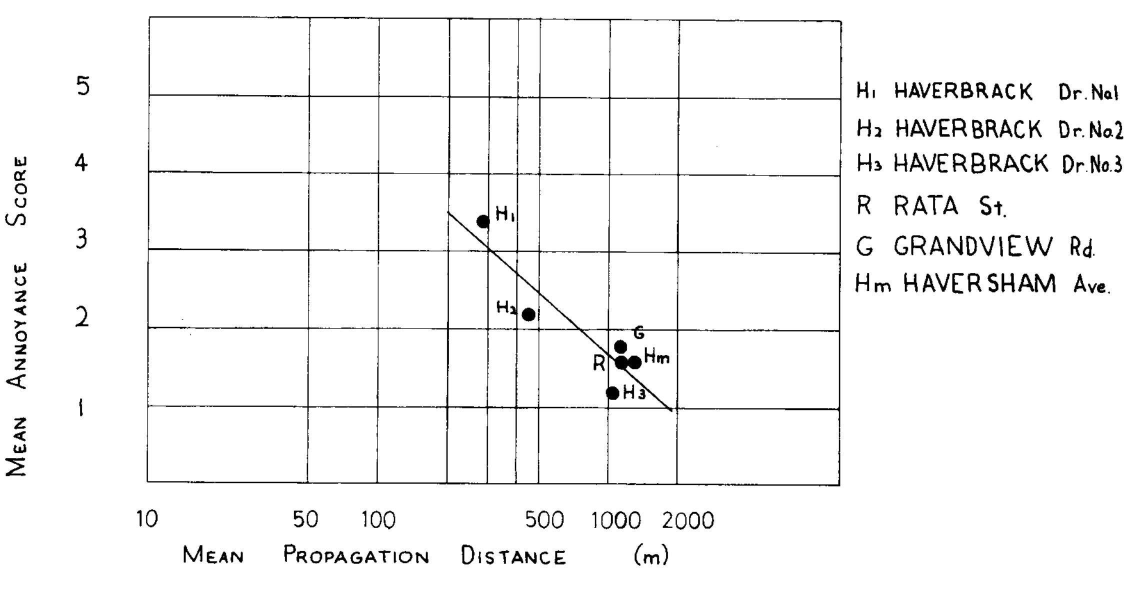
RESULTS: COPS & MYNCKE (BELGIUM) Leq FIGURE 2
MOTORCYCLE CROSS COUNTRY RACES



Noise Annoyance Survey - Waverley Club   Figure 3



Noise Annoyance Survey Waverley Club   Figure 4



SOME COMMUNITY NOISE PROBLEMS AND SOLUTIONS, REVIEWED RELATIVE TO E.P.A.CRITERIA

Dr. N. Broner
Vipac & Partners Pty. Ltd.

ABSTRACT

Many factors play a role in determining the annoyance and complaint response of an individual to given noise sources. Attention to only the absolute physical acoustic factor (the overall A-weighted level) is not always the most effective approach to eliminate that annoyance. For example, the removal of tones in conjunction with a reduction in absolute noise level and with good public relations can result in acceptable noise levels even though the E.P.A. noise limits have not been met. The paper briefly details three case histories demonstrating this points.

INTRODUCTION

It has been well established that community annoyance due to various noise sources such as aircraft and traffic is well correlated with an overall measure of the noise exposure experienced such as the A-weighted level. Thus, one is able to predict the annoyance response of a group of individuals to a given noise source in terms of such gross noise measures; however, with respect to each individual alone, evidence from social surveys suggests that there are many factors which influence the perceived annoyance. Time-of-day, the activity interfered with, the socio-economic status of the individual, beliefs held regarding the necessity of the exposure and previous exposure to such noise may all affect the response of the individual. Social surveys have shown that the gross measure of the exposure can explain only up to 20-30% of the variance in the individual annoyance experienced but that the inclusion of other significant factors can raise the explained variance in response to about 70-80%.

THE VICTORIAN E.P.A. POLICY N1

On May 4th, 1981, the Victorian E.P.A. Policy for the "Control of Noise from Industrial, Commercial or Trade Premises" came into force. This Policy has as its goal the elimination of complaints resulting from noise nuisance. The procedure that is followed accounts for the physical acoustic factor and results in either a pass/fail assessment of the justification of the complaint. It should be realised, though, that the elimination of a complaint action does not necessarily result in the elimination of annoyance. Thus, while according to the Policy a Premises may be found not to be emitting excessive noise, it is still feasible that significant annoyance could be experienced (which may be explained by non-acoustic factors). Vice versa, the annoyance and thus the complaint action, may be eliminated in some circumstances even where the Policy limits are not met due to the influence of some acoustic as well as other factors. For example, often just the fact that the individual has "won" against a large company, so causing the company to introduce some form of noise control (resulting in a change in the acoustic exposure) is sufficient to eliminate the annoyance.

In other cases it has been found that the real cause of the complaint was not due to the acoustic exposure but rather that the impacted individual was choosing the particular avenue of a noise complaint to draw attention to some other grievance.

CASE HISTORY 1

The problem of community complaints due to noise emissions from large industrial fans and cyclones is well known. In this case, a Cyclone Mill Exhaust Fan in a paper coating and conversion plant was found to be the major source of annoyance in that is caused significant tones above the remaining background at the fundamental blade passing frequency of 167.5 Hz as well as at up to five harmonics (see Figure 1). With appropriate adjustments, the Policy Effective Noise Level was found to exceed the Permissible Noise Level by 19 dB(A). Due to stringent space constraints and to the perceived requirement to eliminate the tonality from the environmental emission, a tuned muffler with two additional absorptive faces was inserted between the paddle-wheel fan and the cyclone. The success in reducing the tonality of the noise emission is clearly seen in Figure 1. The reduction in Effective Noise Level was approximately 13 dB(A).

Following introduction of the noise control treatments, no further complaints or indications of annoyance were received. Thus, although the Permissible Noise Level was still exceeded by 6 dB(A), the broadband nature of the resulting noise emission in conjunction with a reduced absolute level and with the fact that noise control was seen (or rather heard) to be done, resulted in an acceptable noise emission.

CASE HISTORY 2

The second case deals with complaints due to the noise emission from a large chemical complex. One of the main contributors to the overall environmental noise emission was flare within the complex. In addition, significant tones due to such items as rootes blowers, hydraulic power packs, vacuum pumps, fans and cyclones, compressors and superheaters were identified. With appropriate adjustments, the Policy Effective Noise Level was found to exceed the Permissible Noise Level by up to 13 dB(A). The major reason for the variation in excess was that the flare noise dominated the emission on many occasions with its sound pressure level depending on the steam setting.

Due to the number and complexity of contributing noise sources, a number of differing administrative and engineering noise controls were instituted. The flare is required for safety, but tests indicated that with minimum steam at night, the resulting flare noise was acceptable to nearby residents. Similarly, certain operating conditions on the off-gas compressors were required to be avoided. Standard-type noise control treatments were introduced on the pumps, some fans and power packs. In addition, the Company made available to the neighbouring residents a "hotline" so that they could register noise complaints at any time. Following the treatments, residents expressed satisfaction with the resultant noise emission with some even expressing the opinion that the resulting noise, which sounded "like waves on a beach", was soothing. The reduction in Effective Noise Level was approximately 8 dB(A), thus, the final noise emission still exceeded the Permissible Noise Level by up to 5 dB(A).

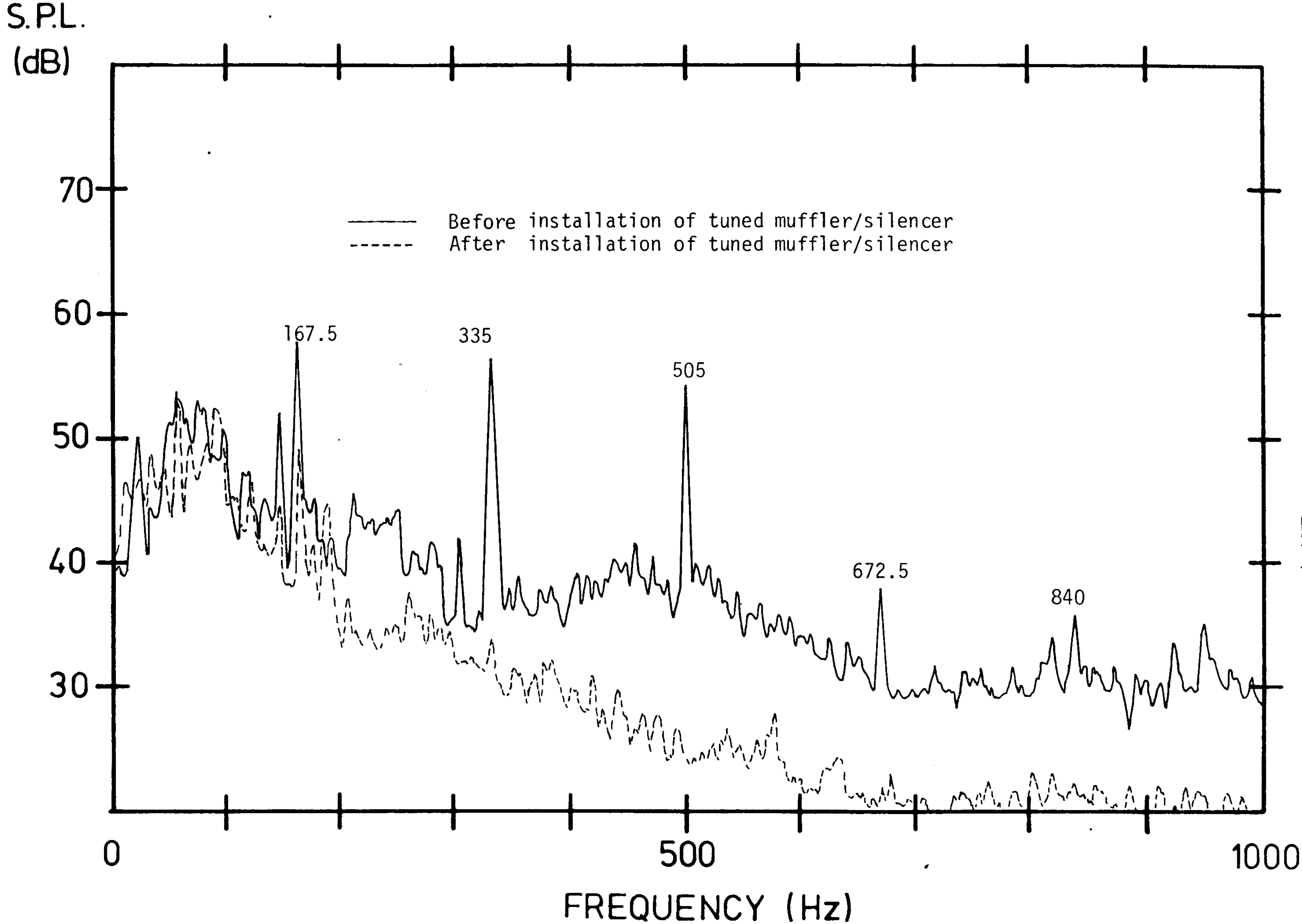
CASE HISTORY 3

The final case history deals with complaints by neighbours regarding the noise emission from an adjacent suburban supermarket. A survey established that the significant noise sources were the cooling towers, compressors and the turbo-chiller within the plant room. During start-up, an impulsive noise was recorded which settled during normal operation. It was assessed that the removal of the tonal and impulsive noise from the emission would result in a broad-band noise which would be regarded as acceptable. The treatment was reasonably successful in removing the tonality and impulsiveness but did not achieve a very significant reduction in absolute noise level. The resulting Policy Effective Noise Level of 56 db(A) still exceeded the Permissible Noise Level by 12 dB(A).

As the annoyance was still present, a further treatment of the cooling tower noise emission was required. The most cost-effective treatment was suggested as a partial barrier. At the time of writing, this barrier is being installed so it is not yet known whether the annoyance has been fully eliminated.

CONCLUSION

The above case histories and considerations indicate that the removal of tonality from a noise emission, in conjunction with a reduction in absolute noise level, can result in the elimination of annoyance even where Policy N1 level criteria are not met. Thus, each complaint case should be assessed on its own merits with a view to achieving the elimination of annoyance. The Policy N1 criteria should be viewed as a guideline only.



S.P.L.
(dB)
70
60
50
40
30
Before installation of tuned muffler/silencer
After installation of tuned muffler/silencer
167.5
335
505
672.5
840
0
500
1000
FREQUENCY (Hz)

1A4-4

Title: CAUSE OF NOISE INDUCED DEAFNESS

Author: A/Prof. B.M.Johnstone
in collaboration with
Dr. D.Robertson & Mr. A.Cody.

Abstract:

Recent experiments suggest that at low sound levels, noise deafness (PTS) is metabolic in origin but at higher levels, a direct mechanical injury also occurs. It is the recovery after noise exposure which appears to be more important for PTS then the initial level of threshold shift.

Text:

There are three major problems in any discussion of noise deafness. The first is just what is the mechanism of noise deafness, particularly that deficit resulting from low level (90-100 dB) long term exposure. The second is the relationship of temporary threshold shift (TTS) or temporary deafness to permanent threshold shift (PTS) or noise deafness. The third is the equal energy hypothesis, or the relationship between duration of exposure and intensity of sound which creates a definable amount of deafness.

It is obvious that TTS precedes PTS, i.e. a short exposure at say 100 dB SPL causes a recoverable deafness but if the exposure is increased, then the TTS becomes permanent. It is no wonder that many workers have sought to use TTS as a predictor for PTS.

What happens during a noise exposure? Because of a desire to use highly controlled conditions, we use pure tones and rather short exposures. By recording from a single nerve fiber, we can see exactly what happens in the cochlea. After all, our percept of hearing is a result of what happens in a collection of single nerve fibers.

The normal single nerve fiber threshold audiogram (fiber theshold curve or FTC) shows a very sharp frequency selectivity. During exposure to an appropriate loud tone, the shape of the FTC changes. It loses its sensitivity and becomes less selective. After 30 minutes at 100 dB SPL the sensitivity has decreased by 50 dB and the frequency selectivity is largely lost. It is important to notice that only the sensitive part of the FTC is lost. This is probably why after a TTS, sounds are distorted and speech intelligibility falls. If the exposure is continued, then the losses become permanent and some damage to the hair cells becomes evident. The FTC appearance is now permanently like the last stages of TTS, i.e. a large loss of threshold and a gross lack of frequency selectivity. The loss of selectivity and change of shape of the FTC can be so dramatic that a tone at what was its most sensitive frequency is now unable to excite the fiber at all. That is, the fiber is now mis-tuned. This completely destroys the place principle of hearing and this is one reason why partially deaf people have so much trouble in understanding speech.

From these and similar results it may appear that TTS is a good predictor of PTS, i.e. PTS is just an extension of TTS and as will be shown later, there are indeed many further similarities. However, efforts to predict an individual PTS from TTS measurements have failed. We tried to check on this point by giving a short controlled sound to some animals, measuring TTS and then giving a much longer tone to the same animals to give a PTS. There is some variation in TTS amount between guinea pigs, but the amount of TTS did not correlate with the PTS in the same animal.

Whilst we usually use pure tones, most normal sound fields are more complex. As a first step in studying this, we gave two tones, e.g. a 16 kHz tone alone and then compared this with a 16 kHz and a 2 kHz presented simultaneously. To our surprise, there was a dramatic difference. Despite the wide separation of frequencies, the two tones together reduced the TTS markedly over that of each tone alone!

This brings into question the concept of "equal energy", or "time-intensity trade off". Here we have a case where obviously the total energy input for two tones is much greater than each along, but the TTS is now much less. The same paradigm is effective for PTS. If the same intensities are continued for longer times, PTS occurs, but it is very much less than for each tone alone.

This is only one of the many reasons that we query any simple time-intensity trade-off function.

Another case arises if we look at a constant time of exposure (one hour) and varying intensity. The amount of damage to the outer hair cells is an approximate direct function of sound pressure level. However, the damage to the inner hair cells (which are more critical for large PTS) is a very sharp non-linear function of sound level. There is no damage for SPLs below 112 dB, but above 115dB, the amount of damage increases precipitously.

We believe that different mechanisms are responsible for PTS at various intensities. Below about 115 dB SPL, the PTS is mainly caused by physiological or metabolic factors. Above this level an additional mechanism comes into play, possibly direct mechanical damage to the hair cells.

The above data, together with the two tone studies suggest that time-intensity trade-off relationships have no firm basis and can be expected to hold only over a limited range.

The results of the two-tone experiments, i.e. less PTS than when when each tone is presented separately are probably due to a greater accelerated recovery after TTS. Normally recovery from a single tone TTS (110 dB, one minute) takes hours, but with two tones, recovery takes only ten minutes.

It is possible that monitoring recovery from TTS may be a useful index of susceptibility to PTS.

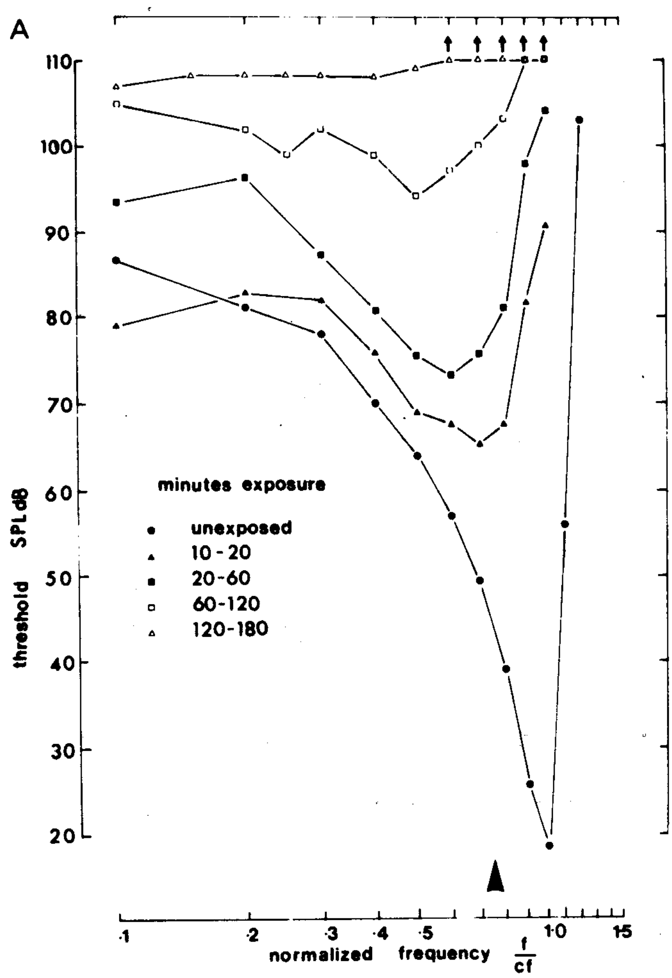


Fig. 1. Single auditory nerve fibre threshold tuning curves (FTC). The ordinate is in dB SPL. The absissa is in normalized frequency. The most sensitive frequency is taken as 1 and other frequencies are a proportion of this i.e. the scale is effectively in octaves, 1.5 = 1 octave higher and .5 = 1 octave lower in frequency. The filled circles show the normal FTC. Loud sound (110dB) is given continuously, except for intervals to measure the FTC. The curves are averages of 6 fibres grouped according to the indicated times of recording. Note that between 20-60 mins, the higher frequencies have lost up to 50dB, but the lower frequencies are unaffected.

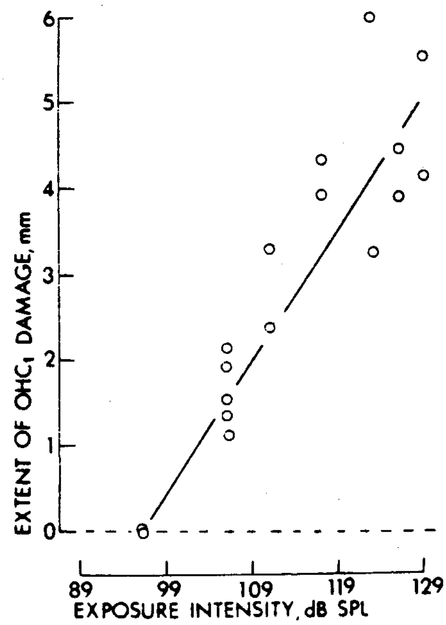
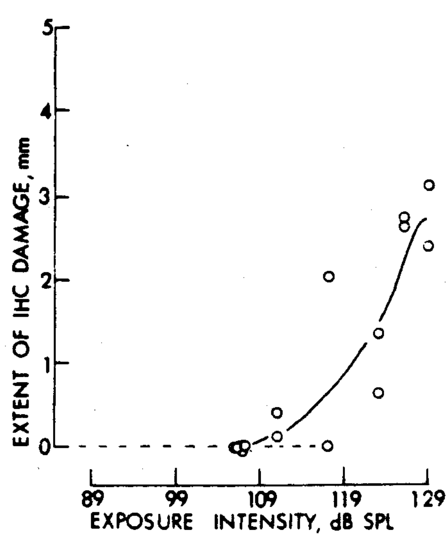


Fig. 2A. Extent of damage to first row of outer hair cells as a function of intensity of a pure tone. Exposure time is 1 hour.



2B. Extent of damage to inner hair cells.

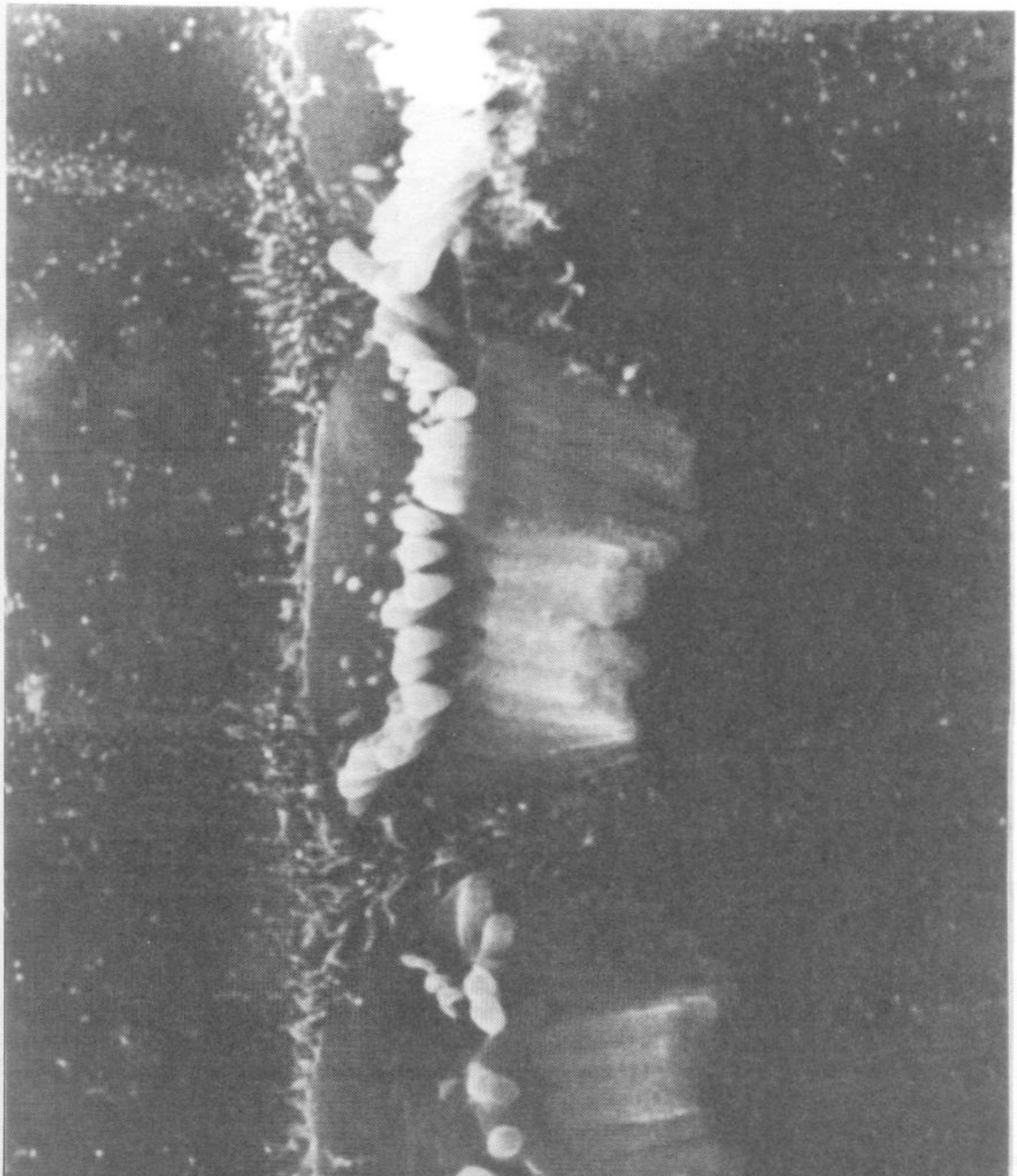


Fig. 3. Outer hair cell damage immediately after an exposure of 110dB for 1 hour.

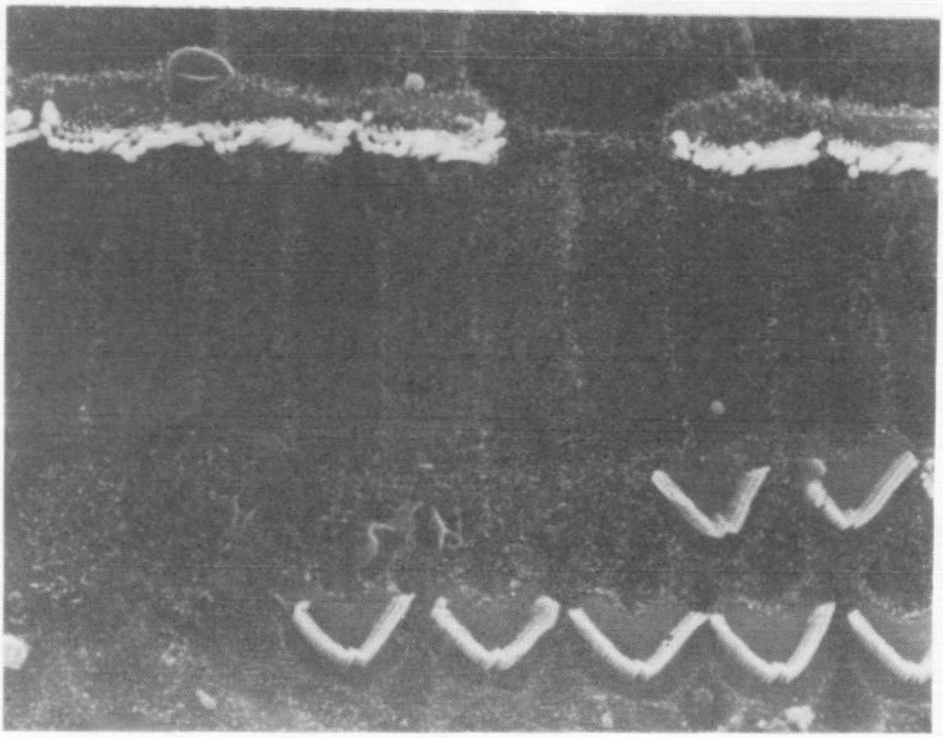


Fig. 4. Surface view of organ of Corti 4 weeks after pure tone exposure of 108dB for 1 hour. The injured cells have disappeared. The remaining cells appear normal. This is not always the case as sometimes cells persist for a long time in the condition shown in Fig. 3.

A STUDY OF RIFLE RANGE NOISE AND ITS EFFECTS ON A SUBURBAN COMMUNITY

Andrew J. Hede, National Acoustic Laboratories

ABSTRACT A socio-acoustic investigation was undertaken in order to assess the impact on the residential community of noise from Hornsby Rifle Range near Sydney. Personal interviews with 201 residents provided data on subjective reaction. Noise recordings made at ten sites around the range were analysed to obtain estimates of noise exposure. The study indicated that the most useful exposure indices for rifle noise are A-weighted Sound Exposure Level and unweighted Peak Sound Pressure Level. The dose-response analysis from this study is seen as having implications for lang-use planning around small-arms ranges.

INTRODUCTION The rifle range studied is located in a bushland setting in the northern Sydney suburb of Hornsby. The range is used almost exclusively on weekends. In the year 1979-80 shooting occurred on 48 Saturdays and 41 Sundays, mostly in the afternoons, with a total of 150,000 shots being fired. The weapons used were mainly 7.62mm calibre rifles and the range of firing varied from 100 yds to 800m.

NOISE STUDY To estimate noise exposure around the range, recordings were made at 10 selected sites on each of 7 days spread over a three month period (see Figure 1). At least 10 rifle shots were recorded each time using a B&K type 4165 microphone and a Nagra IV-SJ tape recorder. Preliminary measurements showed that this equipment was satisfactory.

The noise analysis did not entail simply averaging the levels of the recorded shots. Rather, the various factors responsible for variations in the recorded sound levels were evaluated using multiple regression analysis. These factors were: the distance and angle between the measurement site and the line of firing, the vector wind speed, the effect of air absorption, and the effect of shielding by buildings or topography. Figure 2 illustrates how some of these factors influence the sound of a rifle shot. Note that the levels were higher for site 1 which has a smaller angle from the line of fire but is further away than site 9. Also, the shapes of the frequency spectra and the effects of sound absorption by the air differ at the various sites (cf. Fig 2).

Eight different exposure indices were calculated for each survey dwelling using the predictive equations derived in the noise analysis:

i) Mean ASEL: A-weighted Sound Exposure Level
ii) Mean LSEL: Unweighted Sound Exposure Level
iii) Mean APEAK: A-weighted Peak Sound Pressure Level
iv) Mean LPEAK: Unweighted Peak Sound Pressure Level
v) L 50:10)
vi) L 50:30)
vii) L500:10) L x:y ASEL level exceeded by x shots per day on y days
viii) L500:30) per year.

Note that the four 'mean' indices estimate the energy averages of all rifle shots fired in a year, whereas the four 'level' indices are based on only the loudest of the shots fired. These two types of index make different assumptions about the nature of human reaction to noise.

SOCIAL SURVEY The survey area was defined by a rough semi-circle of 1km radius around the range. The sample set comprised every second dwelling in this area. Only one interview was sought at each dwelling and the respondent was selected randomly. Out of the 245 dwellings in the sample there were 201 successful interviews. The response rate of 82% is considered satisfactory. The refusal rate was 12.7%. None of the respondents knew in advance that the survey concerned the rifle range.

The questionnaire was designed so that the early questions inquired about general satisfaction with the neighbourhood, and gradually led to questions about the rifle range. There were various questions on reaction to the noise. Overall subjective reaction was assessed by a measure called GR (General Reaction) which is a composite of the scores the respondent obtained on several psychological scales. These include ratings of how much AFFECTED and DISSATISFIED the respondent felt, as well as four different ratings of annoyance due to the rifle noise. In addition, GR includes scales based on activity disturbances caused by the noise and the number of complaint actions the respondent felt like taking. GR scores ranged from 0-10. It is argued that GR provides a reliable and accurate estimate of the total extent to which the individual is affected by the noise.

Respondents with the GR scores of 8 or more were classified as "seriously affected" and those with GR $\geqslant$ 4 as "moderately affected". These criterion scores were chosen on the basis of an examination of how the GR scale was related to responses on a variety of independent questions (see Figure 3).

DOSE-RESPONSE ANALYSIS The eight exposure indices proved to be comparable as predictors of subjective reaction - the correlations with individual GR scores ranged from r= 0.27 to 0.29. The 'level' indices were no better than the 'mean' indices. Since the level indices are considerably more difficult to calculate, there appears to be no justification for using them to estimate rifle noise exposure. The analysis focussed on the two indices judged to be the most practical and theorectically defensible:

1) Mean ASEL which measures the total noise energy in the rifle shots adjusted according to the A-weighting frequency response.
2) Mean LPEAK which measures the maximum instantaneous sound pressure of the shots.

(Note that both indices are obtained by averaging on an energy basis over a large and representative number of shots).

The percentage of respondents moderately and seriously affected are shown in Figure 4 as a function of exposure in dB LPEAK. Similar curves were obtained for ASEL although the functions appeared to be more linear in the former case. High correlations were obtained for group data (percentage seriously affected in each exposure zone), namely r= 0.81 for ASEL and r= 0.95 for LPEAK.

The only other published socio-acoustic study on rifle noise is one in Sweden by Sorensen and Magnusson (2). While this study covered four rifle ranges it used less reliable methodology for assessing both subjective reactio and noise exposure. Despite marked differences between the two studies the present dose-response relationship is quite similar to that reported for the Swedish study (see Figure 5).

MODIFIERS OF REACTION Human reaction to noise is known to depend not only on the physical properties of the noise itself, but also on a number of psychological factors. For example, people's sensitivity to noise in general and their attitudes about the particular noise serve to modify the effect that a given amount of noise will have on them. The present questionnaire included psychological scales designed to measure these 'modifiers'.

Scores on a scale of a Negative Attitude were found to be highly correlated with reaction scores (GR): r= 0.78. That is, about 60% of the variation in individual reaction can be explained by attitudes towards the rifle range, the shooters, the authorities etc. This contrasts with noise exposure which accounts for less than 10% of the variance in reaction. Noise Sensitivity scores were also significantly correlated with individual GR but this factor explains only a small proportion of the variance (4.4%).

The effect of these modifiers is clearly illustrated in Figures 6 & 7. These plot the dose-response regression lines for sub-groups of respondents with positive, neutral and negative attitudes or high, medium and low noise sensitivity. People with positive attitudes were almost completely unaffected irrespective of the amount of noise. By contrast, those with negative attitudes were appreciably affected even at moderate exposure levels (cf. Fig 6). Modifying variables such as attitude and sensitivity explain why one person may be seriously affected while a neighbour with identical noise exposure, may be totally oblivious to the noise.

LAND USE IMPLICATIONS Caution must be exercised in generalising from a single survey. However, given the similarity in the dose-response functions found in the present study and in the Swedish study (2), it does seem appropriate to consider the implications for land-use planning and noise regulation. As can be seen from Figure 5 there is relatively minor community disturbance below about 80dB (LPEAK) in both studies. But a significant proportion of the community is seriously affected by noise levels above 90dB. It would appear, then, that a mean unweighted peak sound pressure level around 85dB would be a reasonable criterion for land-use planning. At this level approximately 10% of a residential population would be expected to be seriously affected. Although the LPEAK index is attractive because of its simplicity, it is argued that an energy-based measure (eg, A-weighted sound exposure level) provides a more generally applicable exposure index for impulsive noise (1). With this index the corresponding criterion level would be 55dB (ASEL).

Further research is needed to determine whether and by how much the criterion level should be varied to take account of variation in the number of shots. Results from a laboratory study (3) suggest that subjective reaction to impulsive noise follows the energy principle, that is, that changing the number of impulses by a factor of two is equivalent to a 3dB change in noise level. This contrasts with Sorensen and Magnusson's (2) finding that the dose-response functions were essentially the same for two shooting ranges which differed greatly in the number of shots fired. In deciding suitable criterion levels, consideration also needs to be given to possible weighting for night vs. day and weekend vs. weekday firing.

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2. SORENSEN, S & MAGNUSSON, J. Annoyance caused by noise from shooting ranges. Journal of Sound and Vibration, 1979, 62, 437-442.

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URE 1 Noise measurement sites around Hornsby Rifle Range.

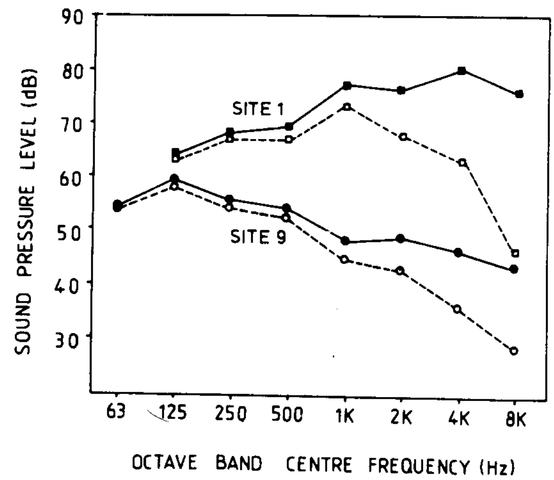
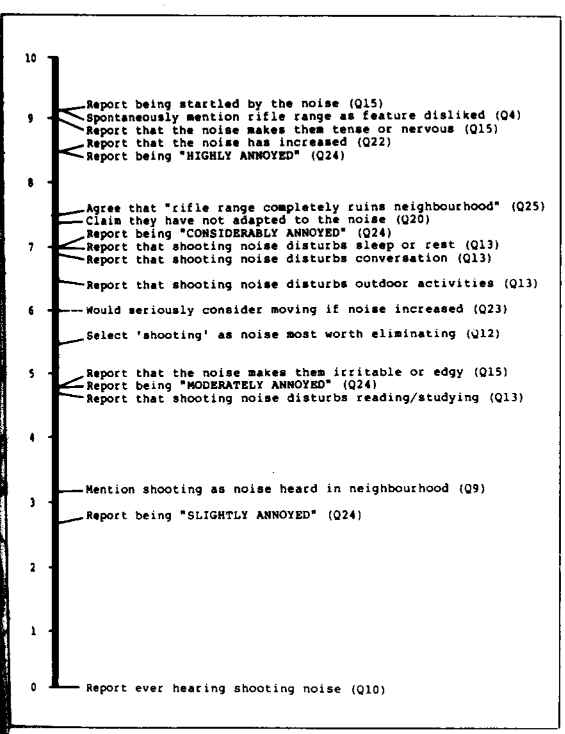


FIGURE 2 Spectra of typical rifle shots recorded at two sites. (------ spectra as measured; ——— spectra corrected for air absorption.)



URE 3 Points on the GR scale at which 50% of respondents...

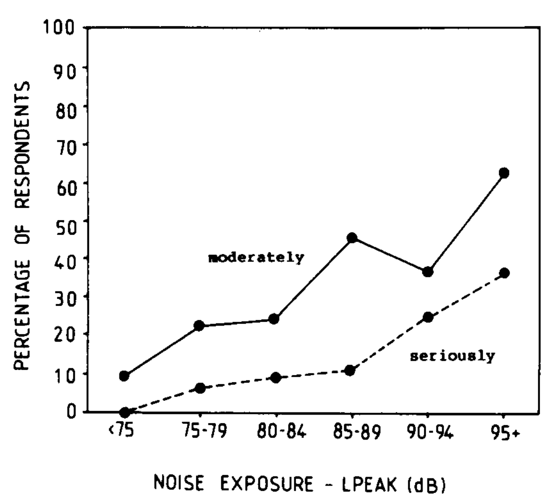


FIGURE 4 Percentage of respondents seriously affected (●---●) and moderately affected (●——●) as a function of noise exposure in LPEAK.

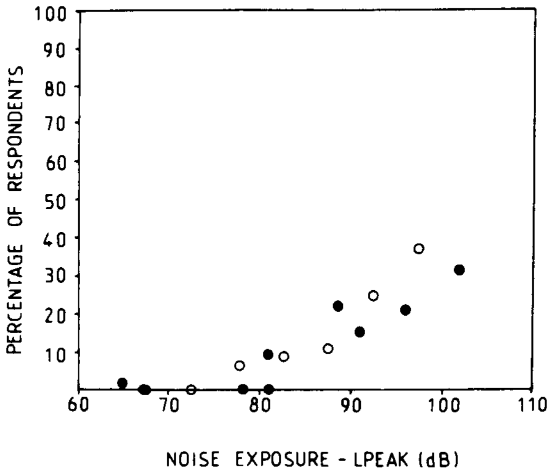


FIGURE 5 Comparison of present results (O % seriously affected) with the Swedish results (● % very annoyed) for exposure measured in LPEAK.

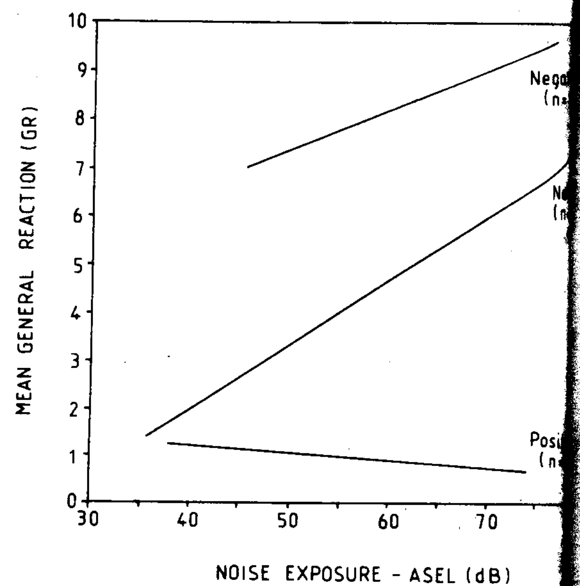


FIGURE 6 Dose-response regression lines for respon
with positive, neutral versus negative at

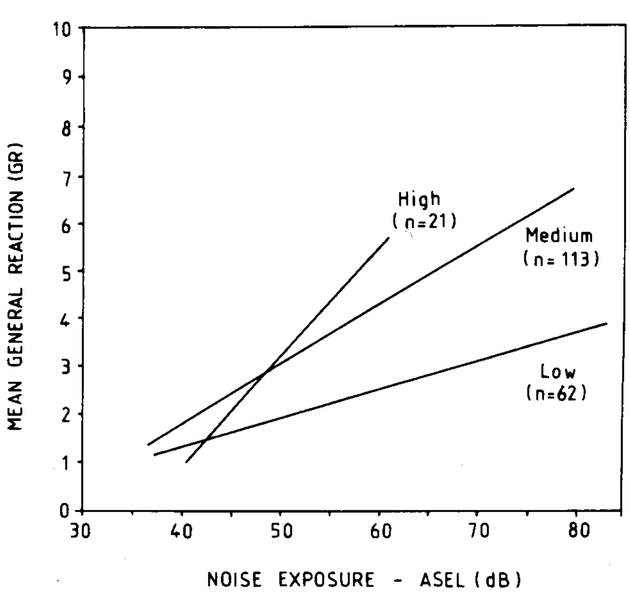


FIGURE 7 Dose-response regression lines for respondents with high, medium versus low noise sensitivity.

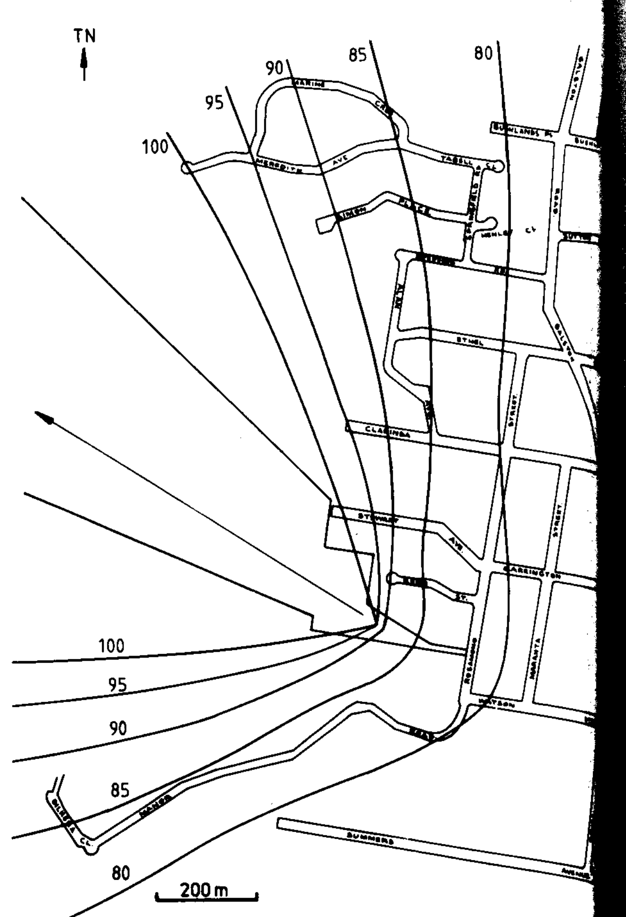


FIGURE 8 Noise contours around Hornsby Rifle Rang
(Unweighted peak sound pressure level -

NOISE AND VIBRATION ASPECTS OF THE SPACE SHUTTLE ORBITER VEHICLE

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Abstract

The various phases undertaken in the development and validation of an analytical model for use in prediction of the noise levels inside the payload bay of the Space Shuttle Orbiter Vehicle, are summarised. Application of the modelling procedure to calculations of the vibrational response of the Orbiter Vehicle structure is also discussed. The methodology is suggested as being appropriate for wide application to vehicle noise and vibration problems.

Background

During the lift-off phase of launch the Space Shuttle Orbiter Vehicle is subjected to intense exterior acoustic noise produced by the interaction of high velocity exhaust gases with the atmosphere and launch pad. This jet noise propagates from the orbiter aft section vertically over the Orbiter Vehicle fuselage and is transmitted into the payload bay to develop high noise levels inside. The payload bay is located closer to the rocket engine exhausts than in previous launch vehicles so that the potential for fatigue of lightweight, fragile payload components is considerable. Further the payload bay construction is markedly different than the cylindrical payload shrouds of previous vehicles (having flatter, more flexible surfaces) and the levels of low-frequency acoustic energy during launch were predicted to be greater than for early launch vehicles. Consequently there was a strong need to develop an accurate procedure for estimation of the payload bay noise levels during launch, for both the payload bay empty and with various proposed payloads present. Bolt Beranek & Newman [References 1 and 2] were funded over a 3 year period by N.A.S.A. and other interested organisations to develop and validate such a predictive methodology. The various stages involved in the development of the analytical model and its validation are summarised below, together with an example of its application to structural vibration assessments and a brief discussion of the general benefits of such analytical models.

Space Shuttle Orbiter Vehicle

The concept of a re-usable space shuttle vehicle, in which the Orbiter Vehicle, mated to its external booster rocket, is launched vertically like previous space vehicles but where the Orbiter Vehicle returns to land horizontally like an aircraft, has forced a new geometry of launch vehicles with space craft paylaods. The configuration successfully used in the recent launch is sketched in Fig. 1 showing Orbiter Vehicle fuselage components of present concern.

The payload bay structure can be readily divided into six structurally distinct, yet potentially important transmission regions: payload bay door, fore and aft sidewalls, fore and aft bottom panels and the aft bulkhead. These surfaces bound the payload bay cavity. When a payload is present, the single cavity formed by the empty bay is broken up into a series of interconnected subvolumes which surround the payload (Fig. 2). Subvolumes are identified by bold numbers and transmitting openings between subvolumes are shown by encircled numbers and zig-zag lines.

Analytical Modelling and Validation Program

The basic aim of the program was to develop a computerised prediction methodology to enable interested users of the Space Shuttle Orbiter Vehicle to calculate with good accuracy as part of the structural design process, the lift-off sound levels in the various subvolumes adjacent to the payloads and to determine to which levels the paylaod should be tested in acoustic fatigue tests prior to flight. Fig. 3 outlines key elements in the program to develop this prediction methodology. Model development was conducted in parallel with experimental verification. Both facets were essential to develop a prediction methodology having user confidence.

The basic approach followed in the development of the analytical model was that of an acoustic power balance. The net power flow into a volume is equated to the power dissipated (absorbed) within that volume. When the payload bay was broken into inter-connecting subvolumes, the acoustic power flow between subvolumes had to be included in the power balance equations. Such equations describe the power flow to the resonant and non-resonant modes of the cavity via the resonant and non-resonant (mass-controlled) modes of the transmitting structures. The relationships encompass the normal mode approach at low frequencies and a form of statistical energy analysis at high frequencies.

The payload bay structure was idealised as a series of equivalent flat or curved orthotropic panels, derived by distributing the bending rigidity of stringers and frames uniformly over associated panel surfaces and then used to find mode shapes and resonance frequencies. At low frequencies, however, modal data from detailed finite-element analysis was used to more closely represent the structural response shapes. Damping loss factors for the structures were obtained from published empirical data for similar structures.

Joint acceptance functions were used to predict the structural responses of the various panel elements. The acoustic power flowing into the volume or subvolume was determined, at low frequencies, by an analogous structural-acoustic coupling factor and at high frequencies where there are many structural and acoustic modes in a given frequency band, this coupling was calculated from reverberant field radiation ratios for the various structures. The coupling between interconnecting subvolumes was modelled by using impedance functions for rectangular or circular pistons.

The analysis is applied to each of the structural regions and volumes to find the total power flow into the volumes. Fig. 4 shows the calculated total power flow into the empty bay, together with contributions due to resonant and non-resonant transmission through the payload bay door. Resonant power flow controls the transmission at most frequencies and the door forms the dominant transmission path.

Validation of the analytical model was carried out for both empty payload bay and the bay with payload. Two tests on the OV101 orbiter vehicle were conducted. The first employed random-noise driven speakers as an acoustic source (Fig. 5) using both diffuse noise excitation and a propagating wave excitation. Space averaged payload bay and exterior surface noise levels were measured from which payload bay noise reductions were calculated as shown in Fig. 6: the noise reduction with diffuse excitation is less than for progressive wave excitation, demonstrating the importance of correct simulation of the correlation characteristics of excitation. Two F-104 jet aircraft were used to provide a progressive wave exterior excitation in the second OV101 test, as a close simulation of the launch noise excitation characteristics. Fig. 7 shows the test layout while the measured exterior-to-interior noise reduction for this test is plotted in Fig. 6 and shows close agreement with the speaker simulation of the launch progressive wavefield. Comparison of analytical predictions for this test for the space-averaged acoustic levels in the payload bay are shown in Figure 8 with the corresponding space-averaged measured data. Close agreement between measured and predicted interior levels is shown, although major refinements in models for damping and door structure in the verification process to achieve such close comparison.

Studies of the acoustic effects of payloads in the orbiter were performed using a ¼ sclae dynamic model of the Orbiter Vehicle. Three rigid ¼ scale payload models were used and the results for Space Lab 2 (Figure 2) are presented here. In these tests speakers were used to produce a broad-band progressive wavefield over the payload bay surfaces. Empty bay baseline noise levels derived from up to 80 microphone measurements, were compared with space-averaged subvolume noise levels. The measured changes in bay levels due to introduction of the payload are shown in Fig. 9 with 99% confidence intervals and compared with predicted changes. In most cases, the predicted change in sound level lay within the 99% confidence limits. Changes in sound level varied rapidly from frequency band to band in both measured and predicted data.

Structural vibration measurements on the payload bay door and bottom structure were made during OV101 jet noise tests. These were used to provide limited validation of the analytical model as regards its capacity to predict structural response for acoustic excitation as part of a Dynamic Verification Assessment, preparatory to the first vehicle flight [3]. Comparison of measurements and predictions are shown in Figs. 10 and 11 for the door and bottom structure respectively in terms of the transfer function relating structural acceleration level to excitation noise level in 1/3 octave bands. The agreement is close for the door response but less satisfactory for the bottom panel response. For the latter two predicted spectra are presented, one for the forward region of the bottom structure and the other for the aft. In both cases the predictions show differences of up to 10 dB relative to the measured data: apparently the model for the bottom structure is not sufficiently refined at mid to high frequencies. The model was then used to make vibrational response predictions for the major payload bay surfaces during lift-off and in-flight, maximum aerodynamic excitation for review of design calculations and test specifications which had been scaled from Saturn/Apollo structures and measured data. Fig. 12 presents results using the current analytical model for the door vibrational response and compares this with the (scaled) test specification. Good similarity occurs generally but the predictions exceed the test criterion by about 3 dB at several frequencies. The comparison can be interpreted as indicating a lack of conservatism in the vibration criteria.

Study Benefits

The payload bay acoustics study was carried out to allow qualification of payload configurations with respect to acoustic fatigue during launch during their design and to so avoid potential failure situations. Obviously few configurations warrant such detailed analysis. However, the study has demonstrated how elaborate vibro-acoustic analysis can be applied to practical engineering, albeit fairly complex, structures within a logical framework termed 'power flow analysis'. The approach has wide application particularly in the area of the structural response of and noise transmission into various vehicle types, since it encompasses both low and high frequency regimes within the one methodology thereby overcomming the limitations of finite element models at high frequencies and statistical energy analysis at low frequencies.

Successful application will generally require, at least, a limited verification program to ensure that major errors in structural input have not been made, for example, in the selection of a structural damping model. The method has proved to give an order of magnitude improvement in the accuracy of noise transmission predictions relative to previously available methods.

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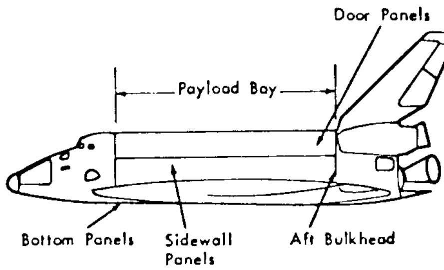

Fig. 1 Space Shuttle orbiter vehicle showing mid-fuselage structure.

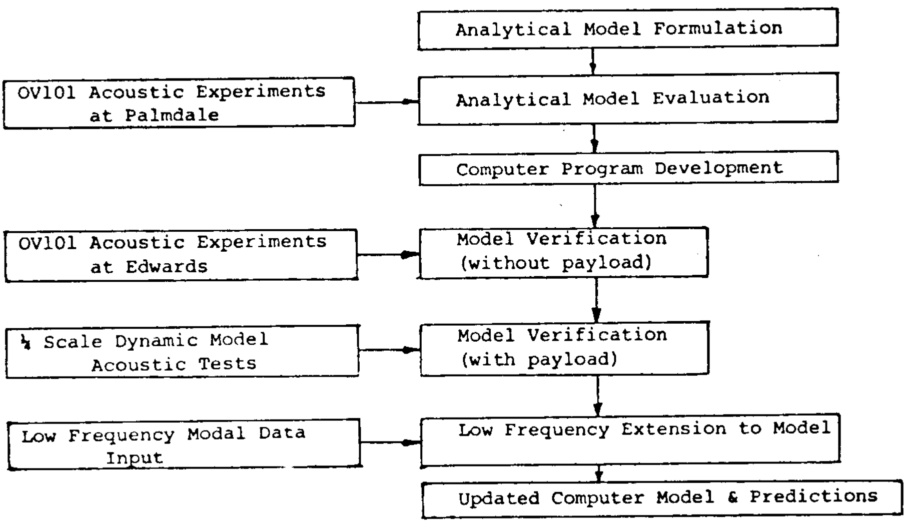

FIGURE 3

Outline of Space Shuttle Payload Bay Acoustic Prediction Study

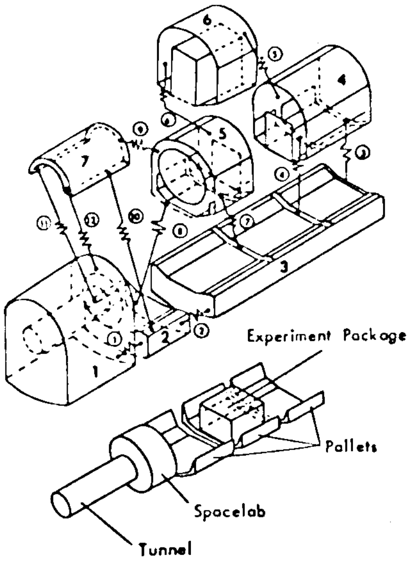

Fig. 2
Spacelab configuration 2 idealization.

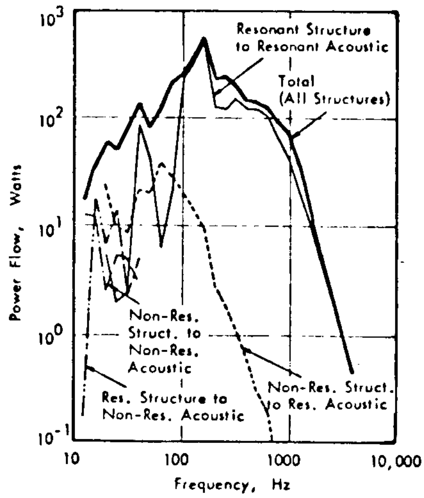

Fig. 4
Calculated power flow through payload bay door.

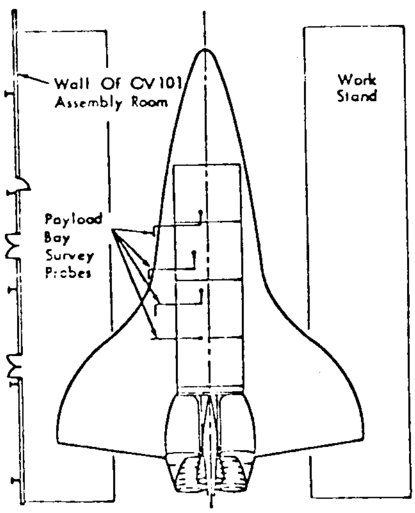

Fig. 5
Survey microphone locations for OV101 speaker tests.

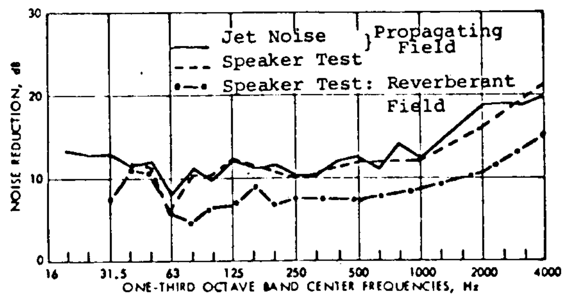

Fig. 6
OV101 payload bay noise reduction for uniform level propagating wave and for reverberant field excitations.

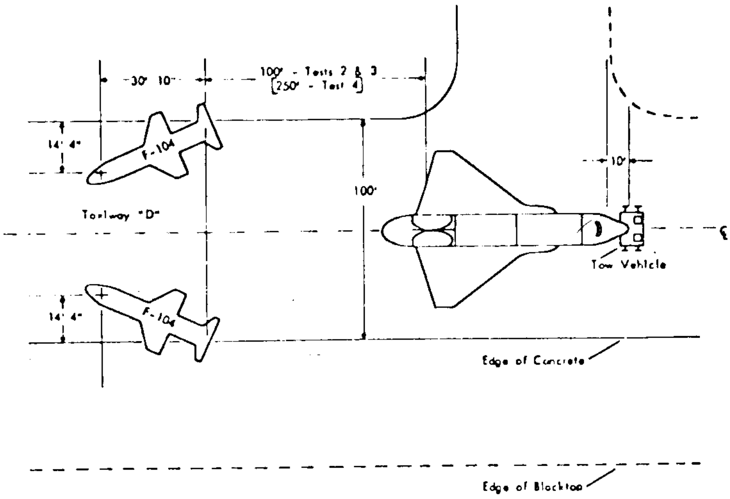

Fig. 7
Position of F-104 aircraft for OV101 jet noise tests.

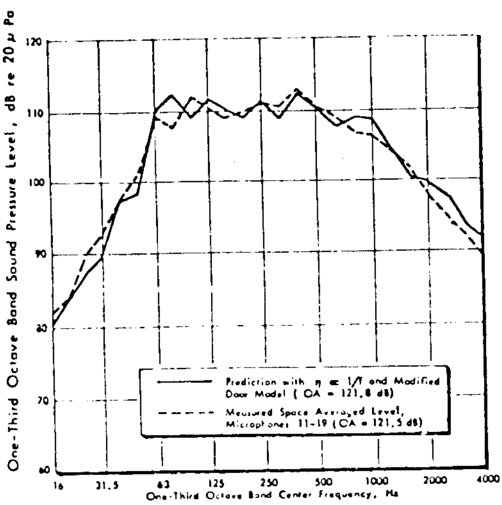

Fig. 8
Measured and predicted payload bay acoustic levels for OV101 tests 2 and 3.

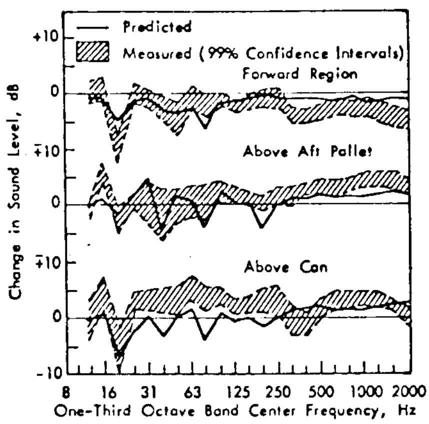

Fig. 9
Comparison of predicted and measured changes in sound levels when Spacelab payload placed in bay (one-quarter scale model).

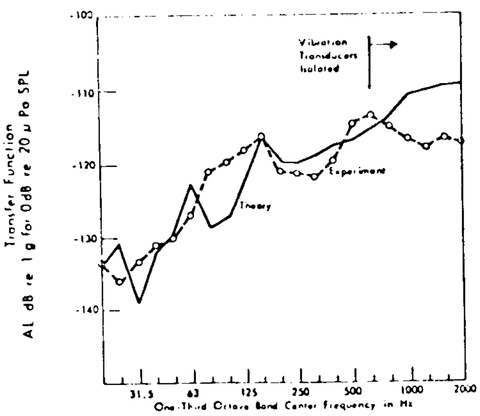

Fig. 10
DOOR RESPONSE: OV101 TEST DATA

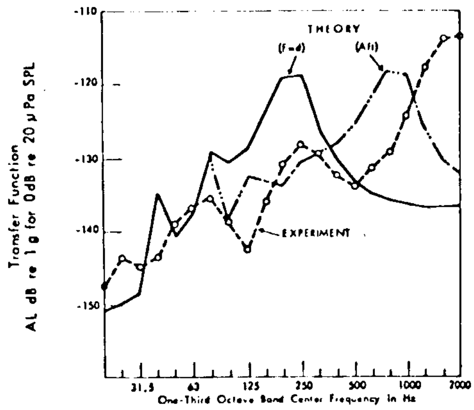

Fig. 11
BOTTOM VIBRATION RESPONSE: OV101 TEST DATA

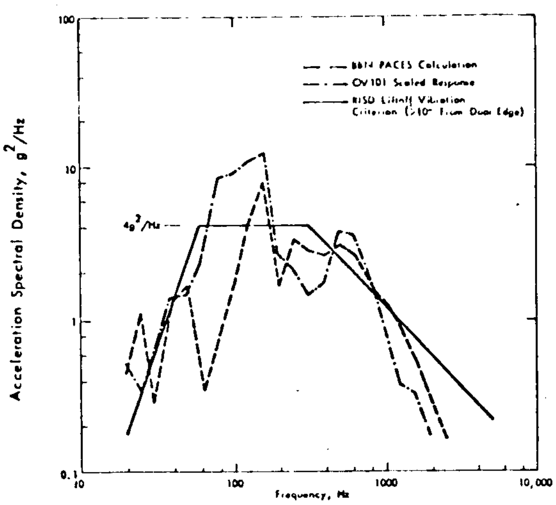

Fig. 12 PAYLOAD BAY DOOR ACCELERATION SPECTRAL DENSITY FOR LIFTOFF EXCITATION

ACOUSTIC IMPACT OF SOME OFF-ROAD VEHICLES

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ABSTRACT

Bushland areas away from roads have lower levels of background sound than found in quiet suburban environments, especially at low frequencies. Sound level measurements of well maintained four wheel drive vehicles indicate that their noise can be audible over large distances. Small areas of roadless country remain in eastern N.S.W. and nothern Victoria, where the intrusion of vehicle noise would be uncharacteristic of these natural environments. Multiple use of bushland would be made acoustically more satisfactory by restricting off-road vehicles to perimeter tracks and by quietening vehicle noise in the 250 Hz octave band.

BACKGROUND SOUND LEVELS IN THE BUSH

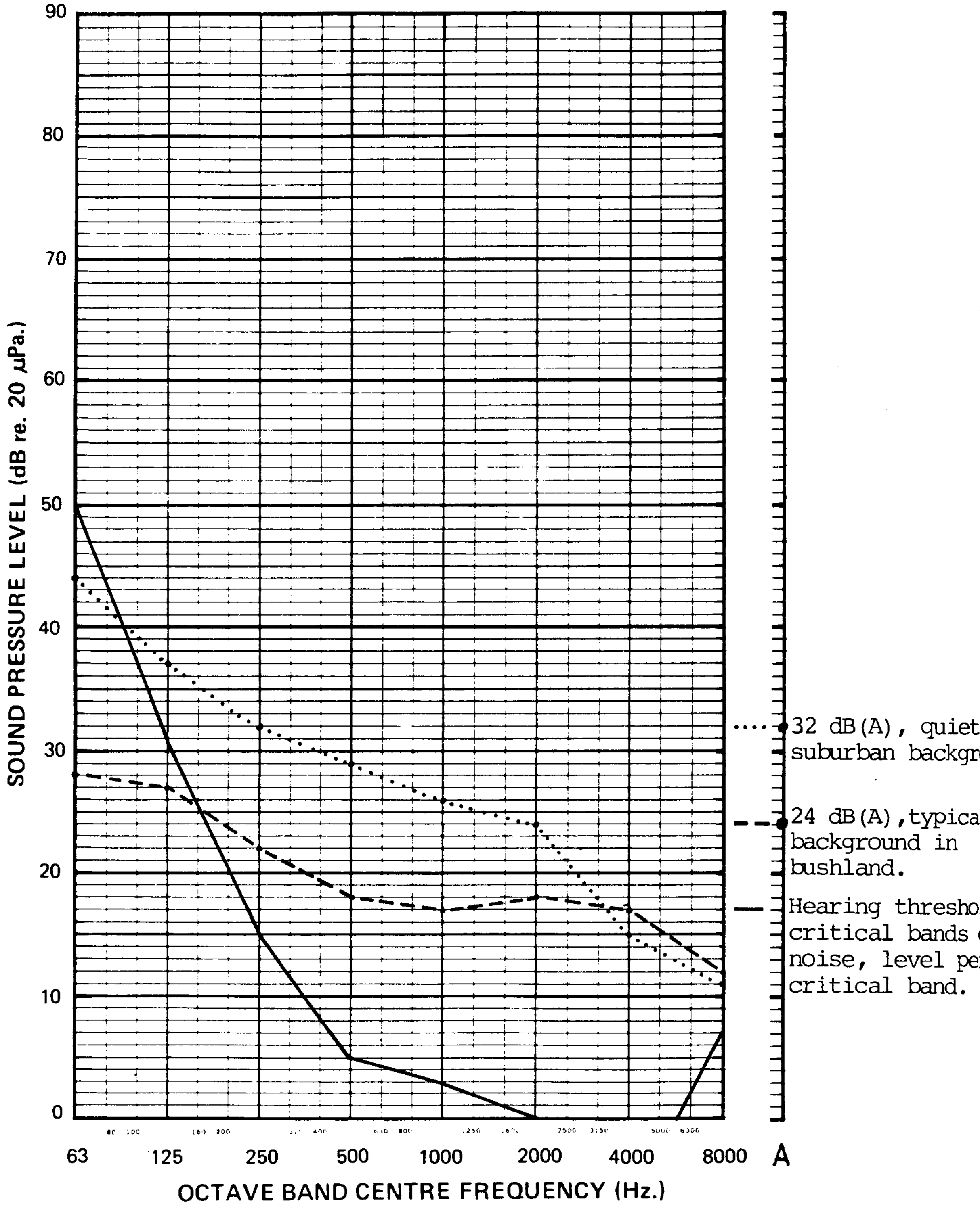
The Australian bush can be a very quiet place, but the impact of aeroplanes and road traffic carries for long distances. Figure 1 shows as a dashed (--) line the measurements of the background or ambient sound level on a hill overlooking the River Nattai. The Warragamba Dam Catchment Area was to the west with 8 km of relatively undeveloped bush to the east before one reaches the first sealed road. Although the area to the east is being developed for houses and so trail-bike riders etc. use the roads into the bush for recreation, none were audible during the periods measurements were taken.

The measured 24 dB(A) is a low background level for daytime. There was an intermittent wind which when blowing the trees nearby raised the level to 34 dB(A). Bird calls were higher at 38 to 40 dB(A). By walking into a nearby gully below a ridge (without a sound level meter because it's heavy) ther were times when it was quieter still, occurring when the trees in the gully were not being blown at all be the breeze. The minimum sound level on the hill was 22 dB(A).

The frequency analysis shown in Figure 1 shows a slight rise in sound level above 1000 Hz due to the breeze, birds and insects. The sound level at low frequencies is greater than at 1000 Hz, but not as great as levels usually found in quiet suburban areas.

The results of background sound level measurements in suburban Sydney are shown on Figure 1 as a dotted (...) line. Although the overall level is louder due to the suburban environment at 32 dB(A) the spectrum shows higher levels of low frequency sound. The difference of 6 dB in the overall A weighted levels is small compared to the differences at low frequencies:16 dB at 63 Hz and generally 10 dB below 1000 Hz.

The greater the difference between the intruding and background sound levels, the more likely it is that the intruding sound will be heard. As the low frequency sound is attenuated less in air, over obstacles and through enclosures, it is more likely that low frequency noise from off-road vehicles will be heard in bushland where there are low levels of low frequency noise.



BACKGROUND SOUND LEVELS, DAYTIME

Figure 1

Figure 1 also shows (as a solid ——— line) the threshold of audibility for critical bands of noise in a free field 1),2). We suggest that there would be times and places in the bush or natural environments when there is an evolutionary or survival advantage to be able to hear sounds at or close to the threshold of audibility and that therefore there must be times and places when the hearing threshold exceeds or is the background sound level in the bush. Examples might be windless days and nights when there is minimum activity from wildlife.

VEHICLE NOISE MEASUREMENTS

Measurements of sound pressure levels at 7.5 metres away from the two well maintained Range Rovers and a Land Rover are shown near the top of Figure 2. The vehicles were stationary in a flat open area away from reflecting surfaces apart from the ground. The A weighted and octave band spectrum sound levels are averages of four measurements taken for each vehicle at the front, rear and both sides.

Engine speed was adjusted to simulate typical driving conditions such as driving up a hill in low gear, negotiating obstacles under power and highway travel at typical road speeds.

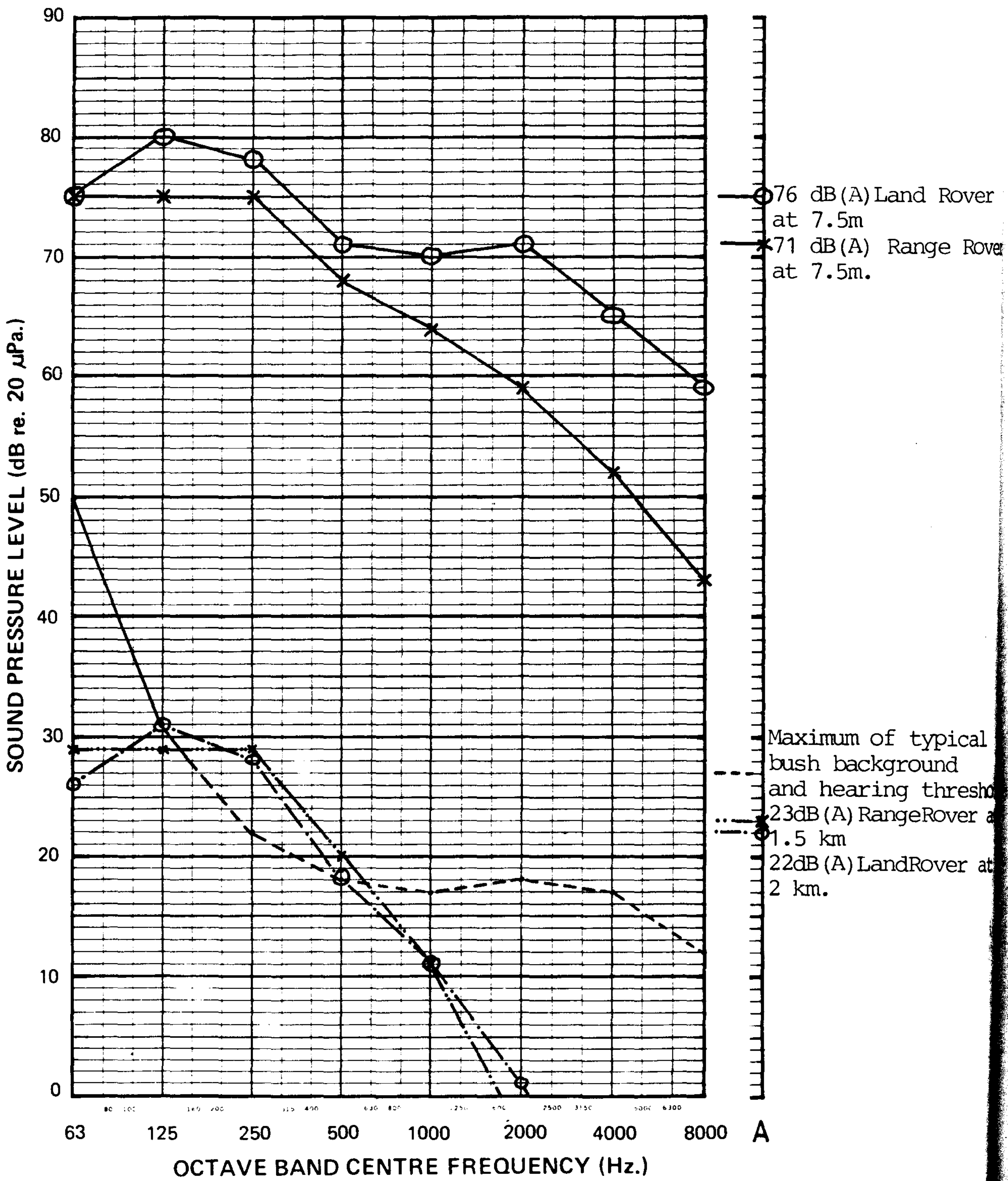
The main noise source was generally from the exhaust, except for the front of the vehicle where the cooling fan was dominant. This is shown by the louder levels at high frequencies especially 2000 Hz for the Land Rover. To test the significance of neglecting gear box and transmission noise, the Land Rover was driven in first and reverse gears, low range, and similar results obtained. This demonstration of the lack of significance of other noise sources is belied by the high sound levels inside a Land Rover which get even worse at highway speed due to tyre and aerodynamic noise. However, as noise external to the vehicles is of interest here, internal sound levels are not referred to further apart from commenting that drivers of less well insulated vehicles that Range Rovers should consider wearing hearing protection devices if they use their vehicles frequently.

External sound levels for a Range Rover are similar to that for a conventional car. Land Rover sound levels are slightly higher.

ATTENUATION DUE TO DISTANCE

The following analysis is based on there being no shielding due to barriers between the vehicle noise source and a person, such as a walker in the bush. Such a situation would occur when the vehicle is driving along a ridge or along the face of a mountain when a person is able to see that area from another ridge or elevated position. This is a common situation as roads are often built along the easiest access, namely ridges and bushwalkers also use these routes for easy access and the views.

Attenuation due to distance of 6 dB each time the separation distance is doubled has been used to arrive at the predicted sound levels at 1.5 and 2 km separation shown near the bottom of figure 2 (as dots & dashes). Attenuation due to molecular absorption in the air is included which greatly reduces the sound levels at high frequency. Field measurements at 250 and



VEHICLE NOISE MEASUREMENTS AT 7.5 METRES AND PREDICTED VEHICLE NOISE AT 1.5 and 2 KILOMETRES

FIGURE 2

600 metres were used to verify these results over medium distances.

ASSESSMENT OF VEHICLE NOISE

The lower part of figure 2 also shows the octave band hearing threshold at 63 and 125 Hz (solid line), with the typical bush background sound level at higher frequencies shown dashed(--).

It can be seen that a Range Rover at 1.5 km distance would generate 23 dB(A) as an average maximum level (or the level exceeded 10% of the time), exceeding the background sound level by 7 dB at 250 Hz. At 2 kilometres a Land Rover generates 22 dB(A) exceeding the background sound level by 6 dB at 250 Hz.

One of the reasons people walk off roads into bushland is to get away from the mechanical noise of urban society. To be sure of getting away from the sound of a Range Rover they would have to travel more than 1.5 km and more than 2 km to get away from a Land Rover.

ACOUSTIC IMPACT ON MULTIPLE USE OF BUSHLAND

There are six small areas of the Victorian Alps where it is possible to be more than 5 km from a road.[3] The furthest distance it is possible to get away from a road in eastern N.S.W. was 6 miles (9.6 km) in 1969 and now may well be less.

To illustrate the impact of vehicle noise on the few remaining roadless areas, consider an area of 250 square kilometres (a circle of 9 km radius). The outside 1.5 km in from the edge affected by Range Rover noise has an area of 77 sq. km., comprising 31% of the total. The area affected by Land Rover noise (2km in from the edge) is 100 sq. km or 40%. Motor bikes with little low frequency exhaust muffling and badly maintained off-road vehicles would affect greater areas. Field days held by four wheel drive clubs involve more than one vehicle, increasing noise levels and the area of affected bush.

Restricting four-wheel drive and off-road vehicle enthusiasts to the perimeter of the limited existing roadless areas reduces the area of bushland interfered with by their vehicle noise emission. Drivers of such vehicles, even at the perimeter admire and enjoy wilderness, using it in their own way.[4]

Further reductions in the conflict between drivers and walkers can be achieved by installing exhaust mufflers with good low frequency performance, especially at 250 Hz.

ACKNOWLEDGEMENT

Land Rover supplied by G. Eden, driven by P. Eden.
Range Rovers supplied by P.Eden and J. Eden, driven by G. Eden and B.Eden.

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NOISE ASPECTS OF THE ENVIRONMENT EFFECTS STATEMENT FOR THE PROPOSED DRIFFIELD POWER PROJECT

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ABSTRACT

An Environment Effects Statement for the proposed S.E.C.V. Driffield power station development has recently been prepared. The noise assessment portions of this work indicate that changes in the noise character of the area would occur.

This paper outlines the methods of sound level prediction and sound level assessment that were used and gives some results of the analysis. It indicates some of the difficulties encountered in estimation of the relevant (future) background sound and in allowing for other factors which modify community response. Consideration of additional or changed criteria assessment methods is proposed.

INTRODUCTION

The proposed Driffield power project is the next planned extension to the S.E.C.V. baseload electricity generating facilities. It is projected to have a capacity of 4000 MW available progressively from approximately 1993. The plans encompass a new open cut south of the Princes Highway, west of the present Morwell cut. Eight 500 MW boilers and their turbo generators would be located south west of the Driffield open cut. The main station buildings would be 6 or 7 km south of the highway and approximately 10 km west of Churchill. A major diversion of the Morwell River would become necessary during the life of the open cut. Open cut overburden would be carried by belt conveyor into the existing Yallourn open cut, north of the highway. Figure 1 locates the project with respect to the S.E.C.V. Latrobe Valley coalfields.

The present Driffield Valley is predominantly rural with small farming/service townships. The valley flats are used for dairying. Small hobby-farms are becoming more numerous. The lower hills to the western edge of the valley are used for APM pine forest. Extensive changes in land use are planned within the S.E.C.V. designated boundary proclaimed by an order of the Governor in Council. It is expected that the sound levels which characterise the existing environment would be significantly increased in areas close to the power stations and to the open cut. The worst case noise levels are compared in the E.E.S. with relevant future background sound using the methods of AS 1055 and S.E.P.P. N1.

THE DRIFFIELD ENVIRONMENT EFFECTS STATEMENT

A comprehensive statement was prepared during late 1980 and the first half of 1981 by the S.E.C. and Kinhill Pty. Ltd. Background noise surveys and project engineer's design forecasts were provided by the S.E.C. Sound level projections and effect assessment was made by Vipac while ameliorative measures were jointly approached. The E.E.S. was issued for public comment in August, 1981.

BACKGROUND SOUND

The existing sound levels in the area within 15 km of the site were surveyed by the S.E.C.V. in February 1981. Preliminary indications from the three night and two daytime surveys have enabled the characterisation of ambient sound levels and ambient sound frequency spectra for the Morwell River Valley (Driffield Valley) and adjacent areas. Sound levels for much of the area are representative of those found in rural communities. Urban, traffic and industrial (power plant) noise influences are significant to the north and east of Driffield. Lowest ambient sound levels (those exceeded for 90% of the quietest measurement periods) were found to be between 15 and 20 dBA near Yinnar and Boolarra.

Higher minimum ambient values of around 25 and 30 dBA were characteristic of the areas to the north and east of the project area. The E.P.A. S.E.P.P. N1 approach uses an average of hourly L_{90} levels over an extended time period whereas the AS 1055 may require consideration of a level typical of a much shorter time period. Minimum measured sound levels as indicated in Figure 3 were taken as a conservative estimate of present day background sound levels. These were used as a basis for prediction of relevant future background sound levels. A map showing contours of minimum measured sound levels is presented in Figure 3.

Project sound level emission and of sound propagation was made using computerised mathematical models, considering the sources and the propagation of sound. A simplified block diagram of program layout is shown in Figure 2. Noise source data is based on the preliminary layout and size and capacity information available for Driffield, tempered by information relating to Loy Yang and at Yallourn W, Unit 1. From each individual equipment item and its location an overall noise emission model was prepared. It was checked against a simple (mathematical) scaling up of the Yallourn W1 station noise and emission and found to be in reasonable agreement.

The propagation of sound through the atmosphere is a complex phenomenon. The variability of the atmosphere leads to continuing changes in the transmission of sound. Atmospheric inversion conditions are expected to occur in the Driffield Valley for approximately one quarter of all night times. This will tend to force the return to the ground of some sound that would otherwise have radiated only towards the sky. An estimation of weather conditions conducive to accentuated propagation has been made. These occur for around 10% of all night times in the Driffield Valley. Increased transmission of sound is predicted (alternatively) to the east or to the west of the valley.

For a 20 km grid about Driffield predicted sound level data were obtained for each grid point by octave band frequencies as well as overall A-weighted sound level (dBA).

An iterative noise emission and control prediction approach was adopted. Initial calculation output showed that the most significant noise sources influencing the resultant sound level at Yinnar would be the induced draft fans (breakout and station stack efflux). McDonald's Track would be influenced mostly by the conveyor system (transfer towers) and the open cut machinery (overburden dredge). No significant reduction in noise level at these locations could be achieved without engineering treatment of the sources themselves (noise control) or of the sound propagation path into the community (barriers). Final calculations were made on the understanding that the S.E.C.V. would incorporate practical and cost effective noise controls using proven technology. Agreed treatments have been incorporated into the E.E.S. Figure 4 shows the noise levels predicted for the final engineered noise control option. Results show that the air absorption and barrier effects lead to the finally received noise have a significant low frequency bias. The worst case (loudest) Driffield noise levels are predicted to raise the quietest ambient sound levels by up to 15 dBA. The highest noise levels from the development influencing a residential area would be around 45 dBA at the north western project boundary, (around McDonald's Track). The prediction is made for full station operation (eight units) and worst case atmospheric propagation conditions.

At present there is no legally prescribed method for assessing the effect of noise emissions from a major industrial development in an essentially rural environment such as the proposed Driffield projects. As a guide the E.E.S. indicates assessment of the predicted noise levels against the Victoria E.P.A. Noise Policy N1 for metropolitan Melbourne and the Australian Standard AS 1055. Results of these are given in Table 10.29 of the E.E.S., reproduced below.

Table 10.29 NOISE ASSESSMENT COMPARISON WITH AUSTRALIAN CRITERIA[a]

Residence Location	AS 1055: Noise Level Excess (dBA)	AS 1055: Likelihood of noise annoyance	S.E.P.P. N1: Noise Level Excess (dBA)	S.E.P.P. N1: Would policy criterion be met (if policy applied)
Boolarra (town centre)	0	None	- 11	Yes
Yinnar (town centre)	+ 11	Positive	+ 2	No
Churchill (northern boundary)	+ 4	Marginal	- 3	Yes
North-west project boundary	approx. + 15	Positive	approx. + 8	No
Morwell (south west)	+ 2	Marginal	0	Yes

(a) Based on assessed future background levels, eight Driffield units operating on 10 per cent of nights with worst weather conditions prevailing.

The numerical indicators of noise level excess above may alone not be sufficient to consider noise effect, a notion accepted within the AS 1055 itself. A number of assessment issues have therefore been faced and incorporated into the E.E.S. These concern the relevant background sound, the community acceptance of certain noise and sound spectrum content.

The present background sound level is not considered appropriate as an assessment base due to changes that will be evident in the area during the construction phase. For example, population changes would significantly alter the area; major rezoning from rural to power production use is involved; long term (15 year +) construction and startup phases would be experienced. The Australian standard does recognise that communities may become habituated to and therefore accept some noise environments. Also, it accepts that low frequency tonality may require special analysis of the sound but does not indicate what form this is to take. Some evidence to assist assessment of the former of these issues is available to the Latrobe Valley. In this regard, none of the 250 people or parties questioned in the social assessment interview process raised noise as an issue. Similarly, indications from the local government authorities are that no complaints with respect to power plant or coal winning plant noise have been received in the four year period of their records. This is in spite of some fairly high residential noise levels.

This raises the possibility that currently accepted Victorian methodologies, the E.P.A. Melbourne Noise policy and the A.S. 1055, are not sufficient indicators of noise acceptance in the Latrobe Valley and similar rural/industrial situations. Consideration of possible methods involving social surveys coupled to objective measurement may be required.

It is submitted here that if there are to be social-survey linked measures of environmental noise assessment then those measures should possess a number of qualities. These should encompass the physical nature of the sound as well as situational, demographic and sociological factors in the community. Any such work carried out in Australia should be linked for logical objectivity to accepted indicators.

Two such indicators proposed for consideration here are the CNR (community noise rating) and the LDN (day night sound level) methods. The latter, for example, has been quoted in U.S. E.P.A. documents as "one of the best methods of community noise assessment available today". The main advantage afforded by such measures is the ability to indicate effect on the community (i.e. the extensity and intensity of annoyance or disturbance) rather than merely a statement of "noise excess".

Neither of these two methods, nor indeed any similar indicators, can gain widespread Australian acceptance without their objective linkage with local social surveys. No such work will be undertaken without the Australian Environment Council or the individual State Authorities taking a keen interest in assessment of overall social effect and in providing appropriate research funding. Perhaps then we can look forward to a rationalisation of the diversity of environmental noise criteria and methods of assessment around Australia.

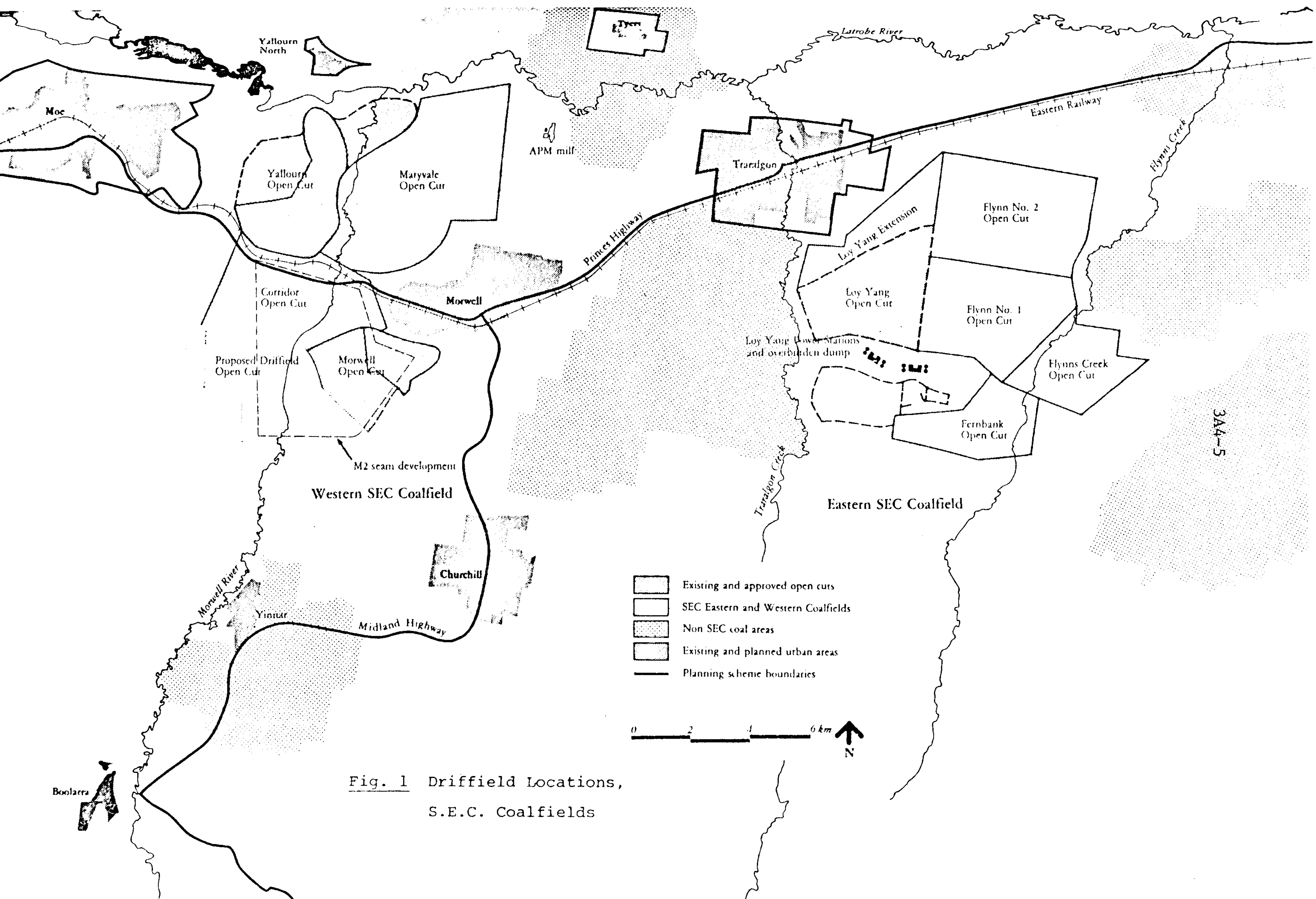


Fig. 1 Driffield Locations,
S.E.C. Coalfields

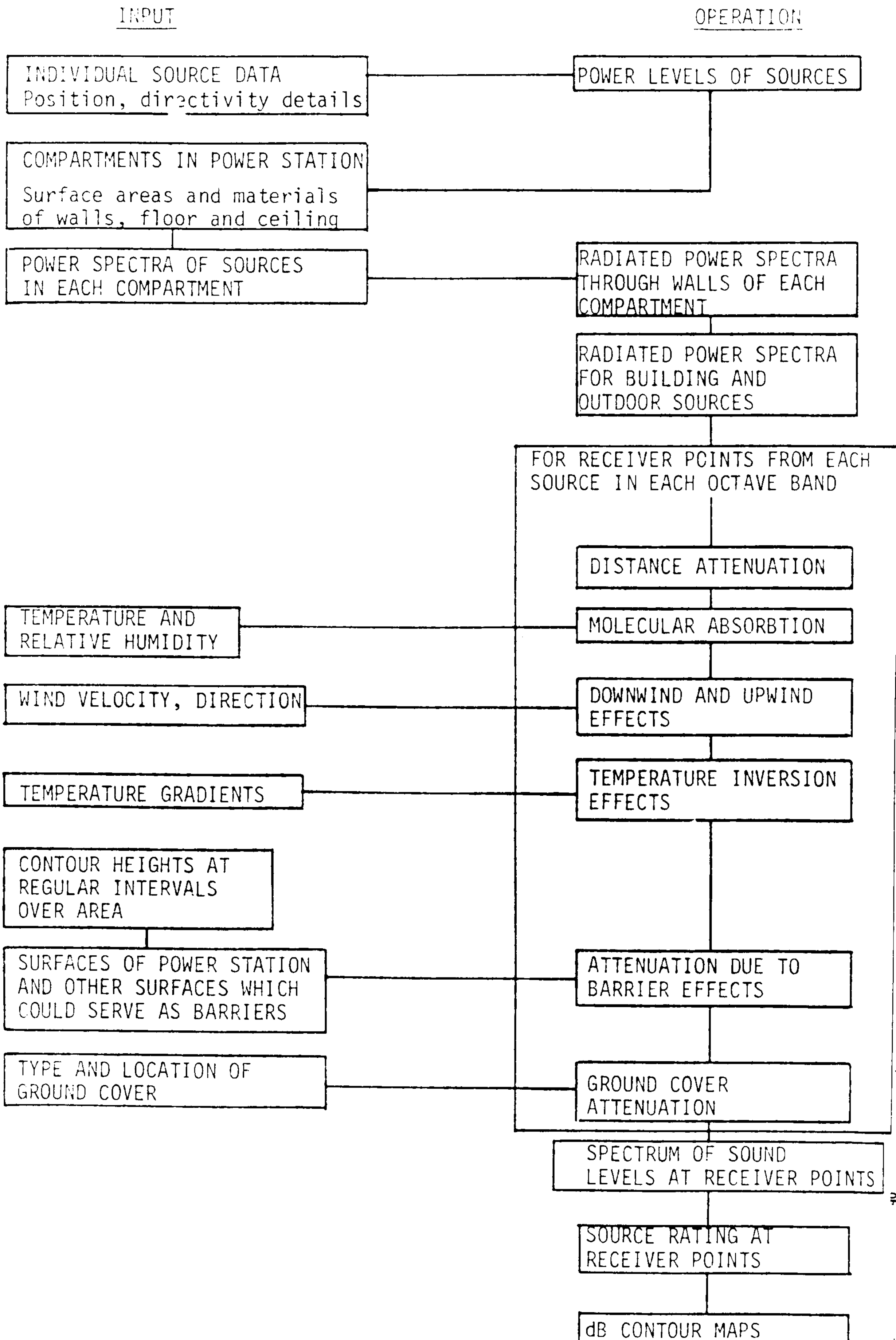


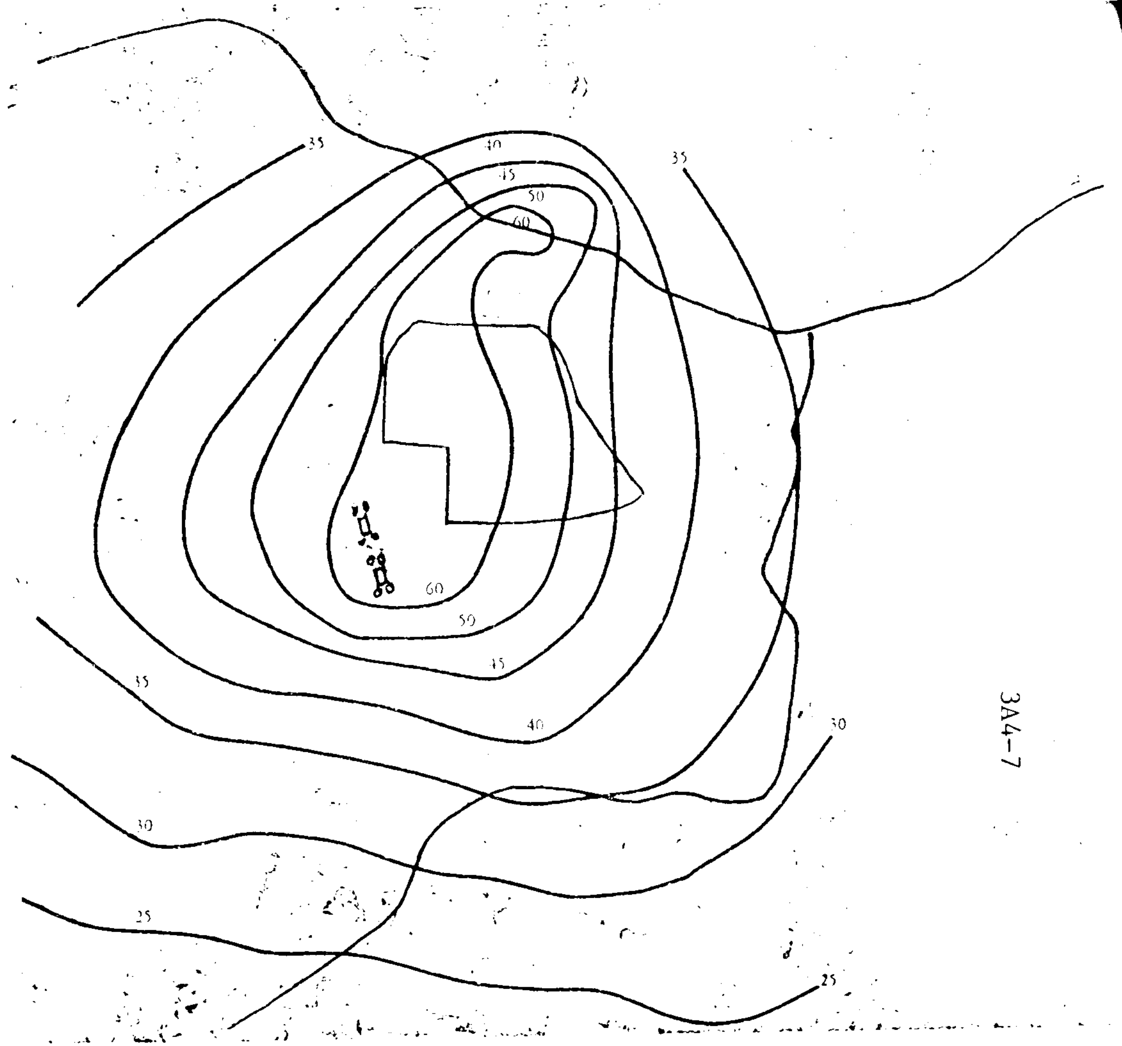
FIGURE 2

Flow Chart of Computer Program for Predicting Sound Propagation into Surrounding Communities

0 1 2 3 4 5 km
Approximate scale
N

FIGURE 3

Estimation of minimum existing L_{90} sound level (dBA)



0 1 2 3 4 5 km
Approximate scale
N

FIGURE 4

Predicted worst case sound emission dBA (L_{10})

THE ACOUSTICAL RATINGS FOR CONSUMER AIR CONDITIONING COMPONENTS

By Louis A Challis
Louis A Challis & Associates Pty Ltd

Once upon a time the standard definition of noise was unwanted sound. Today this definition has become "any sound that the perceiver believes he can have stopped". Of those possible sources of noise one of the easiest to have stopped may well prove to be the noise of a neighbour's air conditioner. The reduction of the noise of one's own air conditioner tends to be a little bit more difficult but there are a number of documented cases where even this has been rectified under the guise of various Consumer Protection acts. The noise of other air handling equipment including kitchen range hoods and kitchen and bathroom extraction fans would hardly seem the basis for a scenario of a classical "Who Done It", but a number of other consultants and we feel that we could well tag our respective experiences in such a way.

Given this strange and anecdotal introduction, acoustical engineers and manufacturers find themselves in the interesting situation where new guidelines are now being set prior to introducing noise labelling for such products in Australia, as well as many other countries. The problem is that the parameters that need to be assessed are not just noise related, as air flow, electrical efficiency and thermal performance may be of equal, if not greater importance.

2. Noise Labelling

The first draft document relating to noise labelling of consumer products in both the U.S.A. and Australia have been released to enable manufacturers to evaluate their position in the market place and to determine what steps they will take to provide the data necessary for the noise labelling programme. Whilst most people think in terms of dishwashers, vacuum cleaners and refrigerators, inside the house, the more important and significant products tend to be those which affect the internal and external environment alike. These are obviously air conditioners, both window type and split systems, heat pumps, pool pumps, heaters and spas, as well as range hoods and exhaust fans.

Most acousticians are familiar with the draft document Australian Environment Council, "Technical Basis for the Noise Labelling of New Air Conditioners in Australia, [1]". It is unlikely that they would be aware of the work being fostered by the U.S. E.P.A. [2] or associated work being proposed in Japan by the Japanese Standards Association [3]. Both of these documents present their assessment criteria in terms of a single figure rating measured on either a prescribed surface, or at 1m from the face of the piece of equipment being evaluated, specified in terms of dB(A). As you will realise such a criteria ignores problems created by various shapes and sizes of equipment, impact of source directivities and ignores tonality which in the case of so many air condensing units creates the major complaint of loss of amenity.

3. Noise Criteria

Over the last fifty years a multitude of criteria have been proposed for evaluating noise. They have included Phons, dB(A), Sones, NC criteria, PNC criteria, NR criteria, octave band levels, one third octave band levels and even PNdB although not specifically for air handling equipment. A further complication and compounding of the criteria problems has been created by various authorities who have nominated that sound pressure level be determined either on a prescribed surface or at prescribed distances on the one hand or as total sound power, typically on an octave band basis, on the other hand. Whilst from the acousticians point of view, any one of these criteria is acceptable, from the layman's point of view one could generalise by a statement that none of them is acceptable, in that the more comprehensive and complex the criteria, then the less comprehensible it is to the general community.

The A-weighted sound level criteria is obviously the easiest for the layman to understand and so for pragmatic reasons, it has tended to become the preferred criteria. This is supported and supplemented by its ease of determination by law enforcement, council, environmental officers, and last but not least by acousticians who are able to evaluate a simple "go", or "no-go" criteria. With this background of pragmatism you may well ask why are there so many antagonists lobbying for sound power ratings and other more complex criteria which may bear little relevance, at first sight to the preferred field measurement which Mr. Average would like to see documented on that noise labelling slip.

4. Laboratory Studies

Over the last ten years we have carried out numerous rating tests for manufacturers, importers, environmental agencies, councils, consumer groups and even individual owners of ventilation and air conditioning equipment, to determine how they perform acoustically, aerodynamically and thermally under controlled laboratory conditions. The majority of these tests have been carried out in accordance with one or more national or international standards and many of them have involved multiple tests to compare various national standards, particularly for European, American and Japanese equipment.

One significant series of tests performed for the Australian Environmental Council [4] highlighted the ease with which variations in testing procedures could be inter-related, provided one was prepared to accept the variability in precision that accrues from differences in testing procedure. This was subsequently confirmed in another series of tests conducted for the Australian Electronic Industries Importers Association [5] which provided additional correlation with the Japanese Standard (reference 3).

The laboratory tests were performed utilising a 340 cubic metre reverberation chamber, a 25 cubic metre clear volume anechoic chamber and freefield measurements utilising a demountable prefabricated wall, in which a series of window type air conditioners were installed. The primary series of measurements were designed to evaluate a prescribed microphone array technique for evaluating air conditioners in the field. These measurements were then correlated with measurements performed both in an anechoic room and in the reverberation chamber.

The results of these investigations showed clearly that the reverberation chamber is unquestionably the best location for evaluating window type air conditioners. Reasonable precision is also achieved in the anechoic chamber and in outdoor freefield environments. By contrast, split system air conditioning units only appear to be adequately evaluated in either a reverberation chamber, a very large anechoic room or in a specially selected outdoor environment where hemispherical or prescribed surface measurements can be performed.

The reverberation chamber provides other attributes not necessarily achievable in other environments including the ability to control the thermal environment, the ability to achieve the noise levels close to the threshold of hearing and simplified measurements in which sound power may be determined on either an octave band or one third octave band basis. It is for this reason that the Australian Environmental Council have proposed that the nominated testing environment be a large reverberant chamber.

For evaluation of less complex equipment such as kitchen range hoods and general exhaust fans, only the reverberation chamber provides a fully satisfactory test environment where air flow, electrical performance and sound power may be simultaneously evaluated, and controlled.

5. Evaluation of Air Flow and Electrical Parameters

Evaluations of air flow currently being practiced range from the strict objective testing procedures nominated by national standards and recommended by the National Association of Testing Authorities, on the one hand, to the subjective evaluations typically performed by the testing personnel of the Australian Consumers Association, on the other hand. Obviously, there are different approaches being practiced in what one would normally think is an area where objective testing is the only method for rating a products performance. The classical case with which you may be aware is the testing of range hoods reported by the Australian Consumers Association in the latest issue of "Choice". It is our belief that such testing should only be performed on an objective basis, but A.C.A. view this matter from a different standpoint. Our research and discussions with the senior lecturing staff and N.A.T.A. assessors from the Univerisity of Sydney led us to believe that the accurate determination of velocity profiles coupled with the determination of total air flow, extracted by a range hood, would provide the best descriptor of the aerodynamic performance. Using this criteria we performed detailed evaluations of air velocity over a prescribed surface delineated by the external extremities of each range hood. By contrast A.C.A. personnel, at different times, placed a pot of steaming water under each of the range hoods that they were evaluating and subjectively assessed the amount of steam escaping into the test room. Based on this visual and highly subjective assessment they assigned a priority and ranking of performance for each unit.

Whilst we do not recommend subjective evaluations in such a situation it is nonetheless necessary to point out that there are factors which may not be elucidated by objective testing, where that testing has not been specifically planned to provide the required information. Obviously the classical situation is the ability of a range hood to extract cooking odours and this particular test still worries A.C.A. and a lot of other people as well.

6. The Development of Quieter Equipment

Our laboratory investigations and our studies overseas have shown conclusively that quieter air conditioners, fans and other consumer oriented equipment is a reality and not a pipe dream. As if to underscore this point, our research for the Australian Environmental Council showed that real attenuations of between 3 and 7dB are readily attainable in the ubiquitous window type air conditioner.

The Japanese manufacturers have obviously never read our reports, for their latest batch of "quiet air conditioners" have achieved lower sound levels than we initially forecast. This has been achieved by developing advanced rotary compressors and new forms of fan impellors with noise generation characteristics far superior to those which we investigated. Not content with that, the latest Japanes split systems have achieved noise levels that are bordering on "state of the art" and their research and development programmes to develop new equipment are by no means yet complete. As a case in point the latest Sanyo 1.5 h.p. units achieve figures which meet the Woollahra Council criteria with a unit located 2m from a boundary. Next years models are reputed to be even quieter than the current model and as such may well solve many of those difficult problems that many of us have faced in the past.

Obviously AS 1866 is a good start and the Australian Environmental Council's technical basis for noise labelling of new air conditioners is a further step in the right direction. Both of these need to be expanded to take into account the evaluation of other performance parameters to avoid the obvious pitfall, where in trying to quantify one difficult parameter, they lose sight of the importance of the other parameters which are often equally, if not more, important.

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RESPONSES TO A REDUCTION IN TRAFFIC NOISE EXPOSURE

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Current criteria for road traffic noise have been based on studies of community response (annoyance) measured under relatively steady-state conditions. The paper examines various mechanisms which may operate under such conditions to alter the annoyance of a group of people and which impinge on the validity of annoyance measures. A study of response to a reduction in noise level indicates a much larger reduction in annoyance than would be anticipated from responses under steady-state conditions.

INTRODUCTION: Field surveys of community response to environmental noise, particularly noise from transport sources, have appeared in the literature since the 1960's. The primary instrument used for measurement of community response to noise has generally been some scale of annoyance (or dissatisfaction) often in association with measures of the effect of noise on behaviour. Respondents' annoyance scores, either as individual scores or some aggregated measure of group response, have then been correlated with physical measures of noise exposure. Except where the effect of specific physical changes were to be monitored most surveys have been carried out under what could be described as "steady state" conditions, viz., no obvious change in the noise environment apart from that attributable to the annual growth in traffic, and a stable residential population with no unusual inward and outward migration rates. The existence of such steady-state conditions has not been reported explicitly in the various studies, but it seems reasonable to infer that study sites were selected with such conditions in mind.

Griffiths *et al* (1980) have commented that while it is possible in such studies to demonstrate statistical relationships between measures of exposure and measurements of such subjective reactions as annoyance and dissatisfaction, the fundamental requirements for the validity and reliability of the subjective measurement techniques have not been demonstrated. It is the purpose of this paper to pose some questions regarding the validity of the subjective measurement technique when it is applied under steady state conditions as defined above. In other words, do annoyance scores obtained from an exposed group in a steady-state situation really measure the effect of noise on that group? It is convenient here to postulate various mechanisms in a longitudinal model of the effect of noise on a group of people (say, all residents of the one roadway) which could influence subjective reaction to noise as measured at one slice in time.

CONCEPTUAL MODEL: The model postulated here makes no claim to comprehensiveness. It considers only three mechanisms: differential susceptibilities to noise annoyance between immigrant and emigrant populations, adaptation to noise, and repression of noise annoyance. Other mechanisms, such as change in community awareness, or expectations regarding the acoustic environment, could perhaps be included. However, it should be regarded as an initial attempt at providing a dynamic framework for existing conceptual models of subjective reaction to noise (e.g., Hede *et al* 1979) which are essentially static.

The first mechanism involves the normal turnover in population experienced in most residential areas. It is possible that outward migrants may include those people most affected by the noise and that inward migrants may include those who will be little affected. The latter requires an intending immigrant to be aware of both the noise exposure at the site and the potential effect of that exposure. While this mechanism could clearly influence the annoyance scores measured from a group of respondents, it cannot be said to affect the validity of the measurements as the change in measured annoyance truly reflects a change in the effect of noise on this group. The results are applicable to the surveyed population at one point in time. However, such a mechanism, if active, would invalidate the application of these survey results to the assessment of the impact of an increased noise exposure on a different population.

A second mechanism which may influence subjective reaction to noise is *adaptation*. This term, which is preferred here to the more usual term *habituation* in order to clearly differentiate it from the next mechanism, refers to the postulated ability of a person to "get used to the noise". It implies that a person's subjective assessment of a new noise environment would become less adverse with continuing exposure. We have chosen to clearly define *adaptation* as an *accommodation to the noise without cost to the individual*. The existence of an adaptation mechanism is part of the folklore of noise. However, several field studies (e.g., Vallet *et al* 1978) found no evidence that habituation to noise had taken place, though these limited studies could not be regarded as conclusive. Taylor and Hall (1976) reported a positive correlation between length of residence and level of noise disturbance; support for the absence of adaptation and/or the presence of a migration mechanism. As in the case of migration, the presence of adaptation does not invalidate assessment of subjective reaction. Again, change in the measured annoyance reflects real change in the effect of noise.

The third mechanism we have termed *repression*. It is suggested that a person, subject to an adverse noise exposure which is relatively beyond his or her control, may actively exclude annoyance from conscious awareness, or at least attenuate the magnitude of subjective reactions to the noise. There is adequate evidence that many people exposed to high road traffic noise levels perceive futility in trying to achieve any mitigation of the noise (Brown, 1978), and Langdon in Alexandre *et al* (1975) comments, "... its victims experience difficulty in identifying individual offenders or locating responsible agencies, nor are they easily able to formulate schemes capable of giving immediate relief". For many people, relocation or expensive house insulation may not be real options. Whereas *adaptation* was defined as accommodation to the noise without cost, *repression* is defined as accommodation to the noise in the absence of any other course of action, and presumably cannot be accomplished without cost. It is suggested that this mechanism may produce a change in measured subjective reaction to noise, without any change having occurred in the real effect of the noise. In part, the following study was designed to explore this repression mechanism.

THE STUDY: In 1979 a residential street in Brisbane which had carried heavy traffic volumes for many years was relieved of through traffic by the opening of a freeway extension. Traffic volumes reduced from 20,000 veh./day to 3,000 vehicles/day. Residents living along the roadway experienced a major improvement in environmental quality including a large drop in noise levels. It was believed that this would be accompanied by a change in the residents' subjective reaction to the roadway noise, from high annoyance to lower annoyance.

Residents along this roadway will be referred to as the experimental group. Two control groups were also included in the study. These were residents along two roadways which were respectively matched to the before (high traffic volume) and after (low traffic volume) conditions on the experimental roadway. Matching was based on traffic volumes, noise levels and physical characteristics such as roadway cross-section and housing type (see Table 1).

Surveys of residents on all three roadways were conducted between November 1980 and April 1981; a minimum of 15 months after the change had occurred on the experimental roadway. Interview surveys were completed with 60, 52 and 40 respondents on the experimental, before control and after control roadways respectively. Response rates were high on all roadways (85-90%) with the smaller after control sample resulting from a large number of non-contactables (three calls back were required). The questionnaire, developed specifically for the study, sought information on all effects associated with living in proximity to a roadway. Only one item is reported in this paper, viz., responses to the question, "To what extent does traffic noise annoy you here?". The scale presented to respondents was the seven-point semantically labelled scale used in Brown (1978) (see Fig. 1).

In addition to their assessment of the existing (low volume) conditions, the experimental group were asked, "To what extent did traffic noise annoy you *before* the freeway extension was completed?" The same seven-point scale was used in this retrospective assessment.

RESULTS AND DISCUSSION: The distribution of annoyance scores obtained from each group of respondents are shown in Figure 2.

Control Groups: Annoyance scores reported from the two control roadways were distributed over the whole scale, but with the expected result of the after control group (low volume) generally reporting low noise annoyance and the before control group high annoyance. The literature has tended to report the annoyance of a group either by median scores or the percentage of respondents scoring higher than some arbitrary break-point on the annoyance scale (e.g., respondents scoring at the 6th or 7th point on the scale defined as ("highly annoyed"), and these statistics are shown for all groups in Fig. 2. In addition to the results from the control roadways, Fig. 2 shows noise annoyance distributions which were obtained in an earlier study (Brown, 1978) from sites somewhat similar to the control roadways. It can be seen that the distributions are generally supportive of those measured from the control groups, though on the low volume roadway, a higher proportion of respondents reported high annoyance.

Experimental Group: The noise annoyance distributions from the experimental group were markedly different. In the retrospective assessment of annoyance for conditions before the change, the experimental group reported very high annoyance, with more than 80% recording the highest point on the scale. No respondents recalled their annoyance as rating less than the mid-point on the scale. After the change, most of the group reported low annoyance, with only one respondent reporting annoyance higher than the mid-point of the scale.

If the noise annoyance distributions measured from the control roadway under steady-state conditions are regarded as typical - and there is little reason to presume otherwise, then members of the experimental groups, reported atypically high annoyance with conditions which existed before the change and atypically low annoyance with noise level conditions after the change. In other words, the people who had experienced a large reduction in noise levels outside their dwellings have reported a much larger improvement in acoustic amenity (i.e., reduction in noise annoyance) than would be anticipated from a simple comparison of annoyance distributions obtained under steady-state conditions. This difference is highlighted if conventional measures of group annoyance are used. The median score for the group experiencing the change dropped from 6.9 to 1.9 (5 points of the 7 point annoyance scale) while the steady-state drop was only 4.6 to 2.9 (1.7 points). Similarly, the percentage of respondents in the experimental group who were highly annoyed dropped from 87% to 2%, compared to the steady-state change of 40% to 4%. Obviously some explanation of these differences is required.

The first and most obvious explanation is that respondents in the experimental group were not able to correctly recall how much they were annoyed before the change. However, two factors should be borne in mind when considering the magnitude of any memory distortion. The first is that memory is dependent, among other things, on the learning period and that some 80% of the experimental group had lived on the roadway for more than five years. The second is that memory distortion may be in the direction of dampened response (Campbell and Stanley, 1963).

An alternative explanation is that this difference may be evidence of the repression mechanism postulated above. If it is argued that people subject to "unavoidable" high levels of noise repress their annoyance, then the experimental group, relieved of the excessive noise exposure, were no longer in need of a defense mechanism, and were able to articulate the effect that the previous noise levels had had on them. By contrast, the before control group, surveyed under steady-state conditions, with no expectations of remission from high levels of noise may be more affected than is indicated by their self-reported annoyance.

If this is so, it has important ramifications for any study of the dose-response relationship for noise. Measures of group annoyance, particularly at higher levels of exposure, may be an attenuated measure of the true effect of noise on the group. A repression mechanism could contribute to the low correlations and the low gradients of the dose-response curve, generally reported from such studies.

CONCLUSIONS: This paper has suggested various mechanisms which may alter the annoyance of a group of people who are exposed to noise. The existence of a repression mechanism would cast doubt on the validity of measurements of subjective reaction to noise under steady-state conditions with important ramifications for the interpretation of past studies of dose-response relationships. The finding that people who have experienced an improvement in their acoustic environment report a much larger reduction in annoyance than expected, suggests such a mechanism. The finding should, however, be treated with some caution until the effect of memory distortion on this result is explored.

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TABLE I

NOISE LEVEL AND TRAFFIC DATA FOR THE EXPERIMENTAL, CONTROL AND SUPPLEMENTARY SITES (All roadways were two-lane, two-way roadways.)

	Experimental Site		Control Sites		Supplementary Sites	
	High Volume	Low Volume	High Volume	Low Volume	High Volume	Low Volume
Traffic Volumes						
24 hour	19864 (est. from 12hr)	2903	19776	2673	20460	3810
12 hour	14898	2177	14832	2005	16480	2820
% Heavy Vehicle Content	6.9	2	6.8	3.8	11	3
Distance from Centreline of Roadway to Dwelling Facade (m)	17	17	16	14	14	20
Noise Levels (dB(A))						
L_{10} (12h) (0700-1900)	74.3[1]	64.5	75.1	65.2	78.0	64.3
L_{10} (18h)	n.a.	61.7	73.8	62.4	75.5	63.0

1. Noise levels for all groups were measured except those for the experimental roadway before conditions. These were predicted using the U.K. Department of the Environment procedure using available 12-hour traffic data. All measurements, site selection, the questionnaire and social profile of the study groups are available in Kyle-Little (1981).

not at all	*1*
very little	*2*
a small amount	*3*
a fair amount	*4*
quite a bit	*5*
a lot	*6*
a great deal	*7*

(Numercial values are not part of the scale, but are used for convenience to present the results of Figure 2.)

FIGURE I : The Annoyance Scale

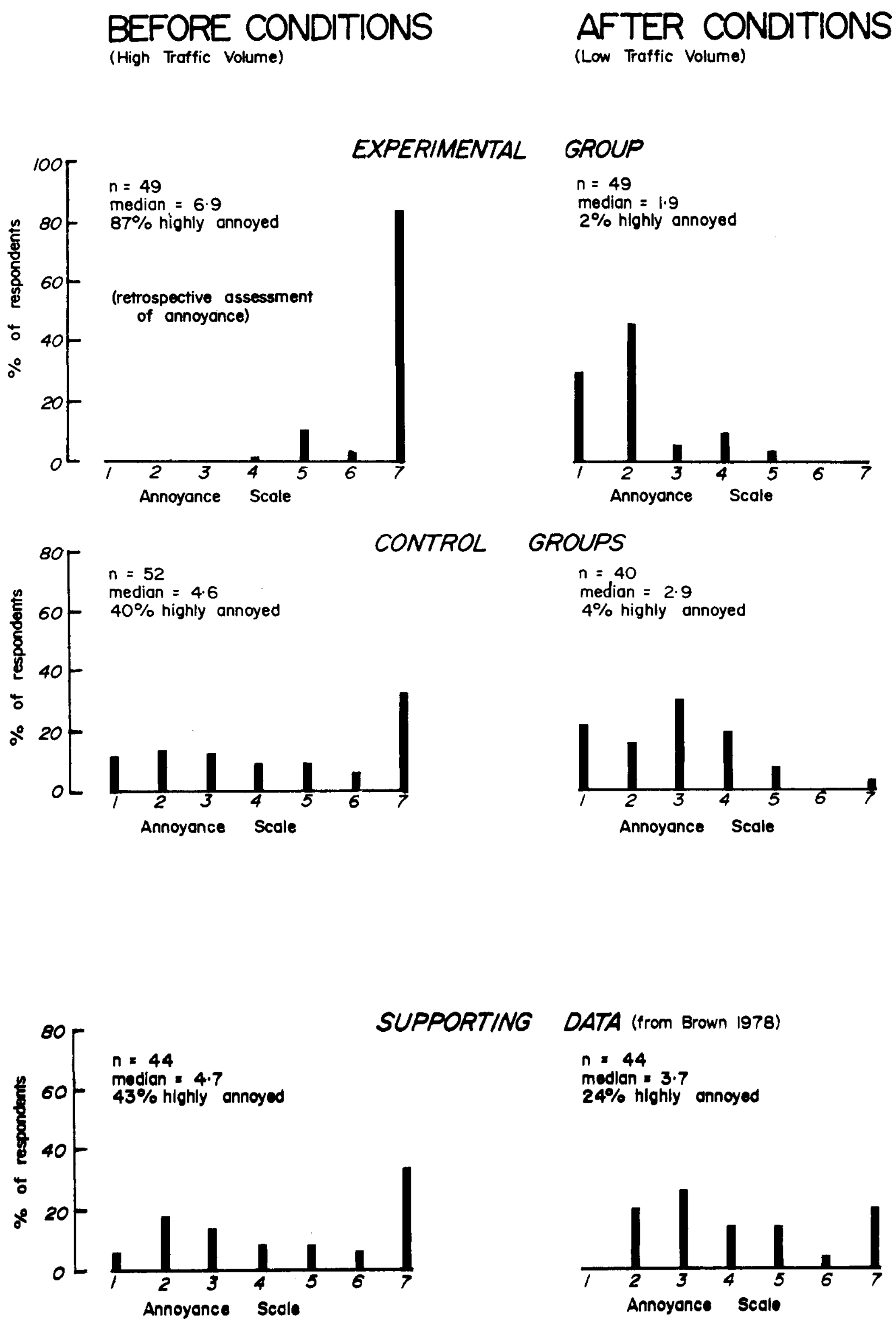

Figure 2: Distribution of noise annoyance scores for each group of respondents. Of the 60 people interviewed on the experimental roadway, annoyance scores are shown only for the 49 who had been in residence before the change.

NOISE PROBLEMS ASSOCIATED WITH ENTERTAINMENT PREMISES

T.J. STUBBS

S.A. DEPT. OF ENVIRONMENT AND PLANNING *

Abstract:

Noise associated with the operation of entertainment premises, licenced and otherwise, is a major source of annoyance and complaint within the community. The paper examines the range of problems - internal, avoidable external, and 'unavoidable' external noise - and of control options - preventative and complaint - responsive. The perspective is that of the regulatory authorities and the theory is made as tangible as possible by concentrating, in particular, on the S.A. experience and on the area of licenced premises. Whilst in the final analysis, the nature of people and of government may mitigate against the solving of particular problems, an understanding of their complexity, of the evolution systems involved and of the points where interference by government and/or acousticians can fruitfully occur will hopefully prove beneficial in minimising community trauma.

* The views expressed in this paper are those of the author and not necessarily of the Department.

This paper purports only to introduce the subject and some of its many aspects in the relatively short space available. What follows is discursive and forms a background to what I will say on the 19th Sept. It is not erudite (even to the extent of omitting formal references) and it is certainly not technical. However I do consider it important, as those of us fortunate enough to live in 'quiet' suburban back-blocks cannot even begin to appreciate the disruption and trauma that entertainment noise brings to a great many people in the community. We all know that noise of 100 dB(A) can be a health hazard. We ought to start learning that a level of 50 dB(A) can be equally hazardous, albeit in a different way.

PROBLEM SCOPE

There are nearly always two distinct components of the annoyance as far as nearby residents are concerned - noise arising from activities within the premises (internal), and noise from patrons in the street outside the premises (external).

Internal noise

Music of the recorded disco variety and from live bands is a problem for a number of reasons:

(i) The audience and the disc jockey or musicians find sheer volume an essential ingredient of enjoyment.

(ii) Much of the music has a pronounced beat and an imbalance towards the low frequencies. This is particularly unfortunate as low frequency noise, because of its long wavelengths, is relatively difficult to attenuate.

(iii) Many of the hotels, halls and restaurants are unsuitable for the containment of loud music. They are literally sound sieves, often because they are being used for a purpose for which they were not designed.

(iv) Large crowds often attend the venues and this, coupled with poor ventilation, leads, especially in summer, to doors and windows being opened.

(v) The entertainment tends to reach its peak around or well after midnight, at a time when ambient noise levels are low and taffic noise in particular is dropping right away.

External noise

Whilst as many people enter as leave these premises, the noise associated with the arrival tends to be masked by traffic noise or by the activities which the surrounding residents are engaged in at the time. The real trouble occurs when people depart after their night's entertainment. The sources of annoyance are numerous:

(i) There are bursts of internal noise as doors are opened while people move onto the street.

(ii) Many people have consumed large quantities of liquor so voices are raised and inhibitions are lowered. People often behave antisocially using offensive language, trespassing, urinating etc.

(iii) Car doors are slammed, engines started and revved, horns blown and cars driven in a reckless and noisy manner. Often they have been parked illegally throughout the evening.

(iv) Noise in the previous three subcategories could be classed as 'avoidable' or 'unnecessary' but most importantly, even people who leave in an orderly fashion cannot avoid creating noise levels high enough to wake, or keep awake, surrounding residents.

At major venues, where the entertainment lasts till well after midnight, this process may continue intermittently for several hours.

CONTROL OPTIONS CURRENTLY ADOPTED IN S.A.

At present in S.A. control of noise from entertainment is mostly shared by the Noise Control Section of the Environment Department and the S.A. Police. They deal respectively with the problems of internal and external noise. The Licensing Act, 1967-1977 provides amongst possible grounds for objection to the grant or renewal of any licence (except a packet licence or vigneron's licence) that:-

(a) the quiet of the locality in which the premises, are situated will be disturbed

or

(b) the owners or occupiers of premises in the locality will be affected to an unreasonable extent

Accordingly, the Licensed Premises Division of the Department of Public and Consumer Affairs has also had some involvement with the particular issue of noise. In one case, Tramps, the Licensing Court imposed severe conditions upon the restaurant licence by restricting hours of trading. Although imposed for reasons other than grounds of noise disturbance, the condition effectively contained noise problems associated with the premises resulting in cessation of the discotheque type entertainment and subsequently a change of licensee.

Internal noise

This is covered by the Noise Control Act, 1976-1977 and the Industrial Noise Control Regulations, 1978, since most of the premises involved are classed as non-domestic for the purpose of the Act. The assessment procedure essentially revolves around four schedules from the Regulations:

The noise level from premises alleged to be emitting excessive noise is measured from the premises of the nearest complainant, and with the addition of appropriate loadings (Schedules 2, Appendix 1) is compared to the appropriate maximum permissible noise level (Schedule 3) or the adjusted traffic noise level (Schedules 4 &5), whichever is the greater. The measurement is made at a time representative of the annoyance, usually nominated by the complainant.

Several points are worth noting at this stage. Firstly these premises are treated in exactly the same way as all industrial and other non-domestic premises. Victoria, on the other hand, have chosen to exempt this type of noise from their policy until a more satisfactory assessment is developed. Secondly the penalty for tonal components is limited to 5 dB(A). Dr. Broner, a contributor to this Conference is one of a number of a people who have done useful work in attempting to find a suitable assessment criteria where the noise has a distinct imbalance towards the low frequencies. In essence though, the simple fact is that the A-weighted sound pressure level and a small penalty may be a grossly inadequate indicator of annoyance potential. Thirdly the extent to which the music is masked by traffic noise is open to question since the lulls in the traffic are filled by the unwanted noise. Finally, and perhaps most importantly, the criterion used is not inaudibility. Thus it is highly likely that the noise will be audible even when not excessive as defined in the Act.

External noise

This problem is currently handled by the Police using the provisions of the Police Offences Act. The major problem is, of course, that there are not enough officers available to adequately police the various venues. Nor can the police be expected to apprehend people who leave in an orderly fashion and yet still cause a disturbance sufficient to interrupt or prevent sleep.

AVAILABLE RANGE OF CONTROL OPTIONS

An adequate control strategy must address all three components of the annoyance:

- internal noise - noise escaping from within the premises
- avoidable external noise - noise caused by unruliness or resulting from inconsiderate behaviour outside the premises
- 'unavoidable' external noise - noise resulting in particular from people leaving the premises in an orderly fashion but which is annoying because of its character and time of occurrence

Options are really of two kinds:

- those which can prevent the problem from arising - screening mechanisms
- those which essentially are in response to complaint where screening mechanisms prove ineffective.

Before ...

(a) Land Use Planning

This is an area beyond the scope of this forum but suffice to say that to prevent problems arising from new premises, it is necessary to gazette land use zoning regulations embodying clear exclusions of certain types of land use, and to base those exclusions on the detrimental effects that the uses could have on the amenity and quiet enjoyment of land within the specified zones. It is vital for planners to understand and espouse the concept of buffer zones.

An area which needs immediate consideration is that of changing land use, whereby quiet pubs are transformed into much more lucrative youth catchment centres and the struggling restaurant becomes an investor's dream - a popular discotheque.

(b) Acoustic Planning

It hardly needs repeating, but costs, at the initial building stage, are of the order of 20% of what would be incurred later in acoustic modification. Things to be considered include: proximity to noise sensitive areas; mass, limpness and transmission characteristics of building materials; sealing; internal absorption; externally, the provision and placement of car parking facilities.

... and after the horse has bolted

(a) Enforcing planning constraints

In respect of problem premises the following aspects should be checked and policed:

- whether the use is permitted by zoning regulations
- whether, for a permitted use, development standards are in fact being complied with
- whether the operation, in consent use cases, is in accordance with all provisions of the granted consent
- whether, for 'existing use' cases, the use is identical with the use existing on the day the zoning regulations were gazetted: if the use has intensified since that day, it is a change of use which requires the Council's written consent.

(b) Volume reduction

This may involve the costly business of building modification or be a case of control at the source . To quote from the British Handbook of Noise & Vibration Control :

"This is the new field. Where machinery noise is being considered volume reduction usually means a reduction of speed. With an electronic group, all it means is turning down a volume control - what could be simpler? The truthful answer is - almost anything. We have seen in the introduction that the present level is the result of a power escalation and we have already stated that musicians are egotists. They just will not turn down. Nothing, and that it not too strong, nothing will make them turn down and stay at the low level. They give various excuses. "We can't feel the music". "It doesn't sound right". "People like it loud". Whatever thinking people may feel about this is really irrelevant; if the musician will not turn down the volume himself it must be forced upon him. "

So we now have limiters, visual devices such as the Electronic Orange and Peace Pulsa or carefully designed and placed arrays of multiple speakers. Yet these devices can be subject to tampering or by-passing and, if the level is to be subjectively satisfying to the 'believers', a certain minimum building shell performance is necessary anyway. A particular problem is often the entrance where people come and go all night so that neighbours can experience an unattenuated performance but intermittently. If the entrance cannot be shrewdly placed it must be converted to a virtual duct silencer with bends and multiple doors.

(c) Hearing Conservation

This is perhaps a regulatory approach to (b).
In recent years there has been considerable discussion as to whether people attending discotheques and the like risk permanent damage to their hearing. An American review of research in the area (Rintelmann, Maico Audiological Library Series, 1970) typifies much of the literature in concluding :

"... insufficient evidence exists concerning the ultimate effects of rock and roll music upon the hearing of young people. Undoubtedly, there are some effects, and avoiding prolonged exposure to intense music would reduce whatever risk may exist. This is especially important for persons who are highly susceptible to auditory damage resulting from noise exposure. Unfortunately however, efforts to date to develop tests for isolating the noise-susceptible individual have not been very successful. Thus, individuals who are exposed daily to rock and roll (e.g. employees in discotheques) should wear ear protective devices (plugs or muffs), since for this group of people rock and roll represents an occupational rather than a recreational (part-time) exposure. On the other hand, most young people could probably tolerate one or two exposure periods per week for a few years with negligible risk of an educationally or socially significant hearing impairment.

It is important that the public be properly informed on this topic. Parents of teenagers should know that there may be a minor risk to auditory damage, but that their youngsters are not 'doomed to defness' as the popular press often emphatically states.

A conclusive answer regarding the effects of rock and roll music on hearing probably must await exposure data (to rock and roll music) that is correlates with long-term threshold studies on humans. But, as a safeguard for anyone frequently exposed to high levels of either occupational or recreational noise, periodic (at least annual) pure-tone threshold tests are advisable as a means of monitoring possible significant changes in hearing that might otherwise not be detected."

Irrespective of whether a severe damage risk exists, the hearing conservation provisions of the Noise Control Act provide a possible avenue for the reduction of internal noise levels.

(d) Abatement directions

Some states have fairly powerful short-term powers for use, with respect to internal noise,when all else fails. They typically take the form:

> "An inspector or member of the Police Force may direct the occupier or the person apparently in control of the premises or part of the premises as the case may be and such other persons therein as appear to him to be responsible for causing, suffering or permitting the noise, to abate the excessive noise forthwith <u>and</u>
>
> A person to whom such a direction is given, shall comply with the direction forthwith; and shall, for a period of 12 hours from the time of direction given, refrain from the emission of or contributing to the emission of excessive noise from the premises to which the direction relates. "

(e) Licensing laws

Section 19 of the South Australian Licensing Act provides, inter alia, that the holder of a full publican's licence is authorized to sell and dispose of liquor in any quantity in the house or premises therein specified -

(c) upon any day at any time for consumption, in such parts of the licensed premises as are fixed by the court, with or ancillary to bona fide meals.

Section 31 of the South Australian Licensing Act provides that every restaurant licence shall authorize the licensee to sell or supply liquor of any kind in the premises specified in the licence for consumption at any time on any day with or ancillary to bona fide meals. As well as 'bona fide meal' and 'ancillary to' not being defined under

Section 4 of the Licensing Act there appear in the Licensing Act various degrees of meals to be provided by licensees, all undefined. It is felt that such variation and lack of particularity has led to the present sham of some licensees providing 'token' meals which are often deposited in available receptacles by patrons who wish to remain on licensed premises after 12 midnight Monday - Saturday, or on Sundays, to drink and/or for the entertainment.

I believe that the only solution to many of the problems which we face is the imposition of conditions (notably hours restrictions) on licensed premises where internal and/or external noise problems persist. Unfortunately some Courts have not seen it as their duty or prerogative to assign responsibility to the licensee for external noise problems. Judge Campton in respect of Her Majesty's Hotel, Sth. Yarra stated:

> "If patrons misbehave after they have left the licensed premises, through no fault of the licensee, the remedying of that situation lies in the hands of those whose duty it is to prevent such behaviour in public streets or on private property."

His stance has been supported by S.A. judges and the answer appears to lie in the type of licensing legislation adopted by New South Wales.

S. 57 B (A) of effect from 30th June, 1980, Act 32/80 refers - upon a complaint being made to the Licensing Court by the City Municipality onShire Council in whose area premises with respect to which permits under s. 57 B are in force or by 20 or more persons who reside in the vicinity of any such premises that the premises are being conducted in a manner that habitually or frequently disturbs unduly the quiet or good order of the neighbourhood, or that persons after resorting to the premises have habitually or frequently disturbed unduly the quiet or good order of the neighbourhood, the Licensing Court shall summons the holder of the permit to appear before the Court and show cause why the permit should not be cancelled.

Having said all this, I must confess that I still find myself drawn to the conclusion, pessimistic though it may be, reached by the British Parliamentary 'Committee On The Problem of Noise" (July 1963):

> "Technical achievements, such as the transistor, may introduce new means of amusing some people and annoying others, but we doubt whether technical advances will contribute much to the solution of the problem. A contribution may come from town and country planning and from the better insulation of buildings from external noise, but fundamentally the problem is one of human behaviour. The primary cause of annoyance from entertainment and advertising noise is that the people making the noise lack consideration for the interests of others. On the other hand some of the complaints may reflect a lack of reasonable tolerance. The problem cannot, therefore, be solved by laws, although they may help to reduce it."

APPENDIX 1

INDUSTRIAL NOISE CONTROL REGULATIONS, 1978

SCHEDULE 2

Adjustments to the Measured Noise Level in dB(A)

Characteristics of the alleged excessive noise			Adjustment dB(A)
Total components -			
Perceptible tonal component		...	+5
Frequency and/or amplitude modulation		...	+5
Impulse component - Impulsive noise		...	+5
Intermittency -	Between -		
Duration of alleged excessive noise expressed as a percentage of the relevant time period	100 and 56	...	0
	56 and 18	...	- 5
	18 and 6	...	-10
	6 and 1.8	...	-15
	1.8 and below	...	-20

SCHEDULE 3

Maximum permssible noise levels for areas

Description of area in which the noise source is situated	Maximum permissible noise levels dB(a)	
	7a.m. - 10p.m.	10p.m. - 7a.m.
Rural or predominantly rural ...	47	40
Urban residential ...	52	45
Urban residential with some commerce, or with a school, hospital or the like	55	45
Urban residential with some manufacturing industry, or with some place of public entertainment or place of public assembly or a licensed premise	58	50
Predominantly commercial ...	65	60
Predominantly industrial ...	70	70

SCHEDULE 4

Noise levels for premises adjoining roadways carrying more than 100 vehicles/hour

Vehicles per hour (Two-way count)	Noise level dB(A)
100	54
200	58
300	60
400	62
500	63
700	65
1000	67
1500	69
2000	71
3000	73
4000	75
5000	76

NOTE - Interme diate values as may be determined by linear interpolation between the tabulated values.

SCHEDULE 5

Adjustment to noise level to allow for the distance (m) between the noise source and the centreline of the nearside carriageway

Distance (metres)	...	15	30	60	120
Adjustment, dB(A)	...	-4	-8	-12	-16

SO-YOU HAVE TAKEN A SOUND LEVEL READING

.... OR HAVE YOU TAKEN A SOUND LEVEL READING?

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Steven M. Tasker
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Abstract

Random pink noise in third octave steps between 100 and 4000 Hz was measured at various sites. Some dramatic variations in measured sound spectra were observed.

While the authors accept that precise measurement and analysis may yield quantitative explanations, they present this paper merely as evidence of possible variability and inaccuracy in field measurements, and note, in particular, the significance of these variations in legally judging noise in neighbourhood confrontations.

....

After twentyfive years of feeling confident about taking sound level readings, we have data which may explain some of the many conflicts we find between the anticipated reaction to a noise level reading and the real reaction to the noise.

This paper will show only a few of the variations in spectral analysis of a 'standard' noise source that can be expected. There is no claim to originality and no attempt to establish experimentally the cause of the variations in the data. We merely call attention to the difficulty of obtaining meaningful data in neighbourhood studies.

From a single cassette recording of a Bruel & Kjaer third octave random pink noise (from Bruel & Kjaer Record Type QR2011) in the frequency range between 100 and 4000Hz, played through a Solid State 100 w amplifier into two or four Plessey 8MX speakers (Diagram 1) sound level recordings were made with a system consisting of a Rion Type NA61 Sound Level Meter, a Rion Type NX02 Third Octave Filter, and a Rion Type LR04 Level Recorder.

Data was taken at different microphone heights, different speaker heights, different speaker angles, and over different surfaces (grass, concrete, asphalt cement) for spectral comparison. The graph of the data from the speakers in a moderately reverberent condition (a 4 m x 7 m wooden shed) is offered for comparison purposes (Figure 1). Although a reasonable attempt was made to maintain the same level into the speakers, we offer no guarantee of the level; we are offering the data for spectral comparison.

Diagramatic representations of the six experimental configurations along with corresponding sound spectra are given in Figures 2 - 4.

We offer that some of the differences recorded are caused, in some cases, by:-

(1) the phase cancellation due to the difference in distance of the microphones from the various speakers in our array.

(2) Phase cancellation due to the phase relationship and the distance travelled between the direct line from the speakers and the line reflected from the experiment base (grass, concrete, asphalt)

(3) absorption of sound, incident on the ground.

(4) the difference in energy from the speaker due to polar considerations.

We see no reason to believe that a factory noise source could not duplicate some or all of these phenomena.

We make the point that all of these considerations may be applicable to a neighbourhood noise measurement. We believe that the data establishes that even a noise with no pure tones can have at least 10 decibel more (or less) of the annoying character of the particular noise entering a house than a perfectly honest boundary survey may show.

In view of the variability of measurements that this study has highlighted, it may be misleading to quote a reading accuracy of ± 1dB (± 25%) for environmental studies.

Conclusions

We believe that the data supports our long-standing philosophy that the only sure test of whether a neighbourhood noise is too loud is whether it is annoying someone. We believe that, regardless of what the sound level meter shows, every complaint should receive an honest effort on the part of those contributing the noise source directed towards reducing the noise.

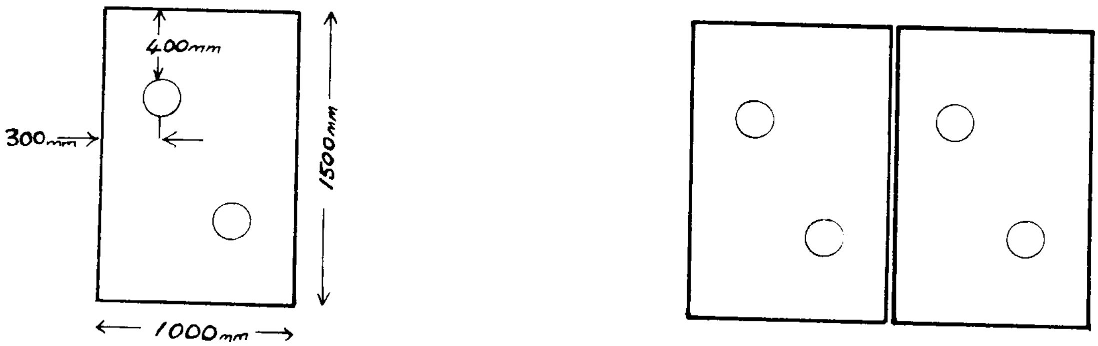

Diagram 1   2 and 4 Speaker Configurations, showing positions of speaker cones

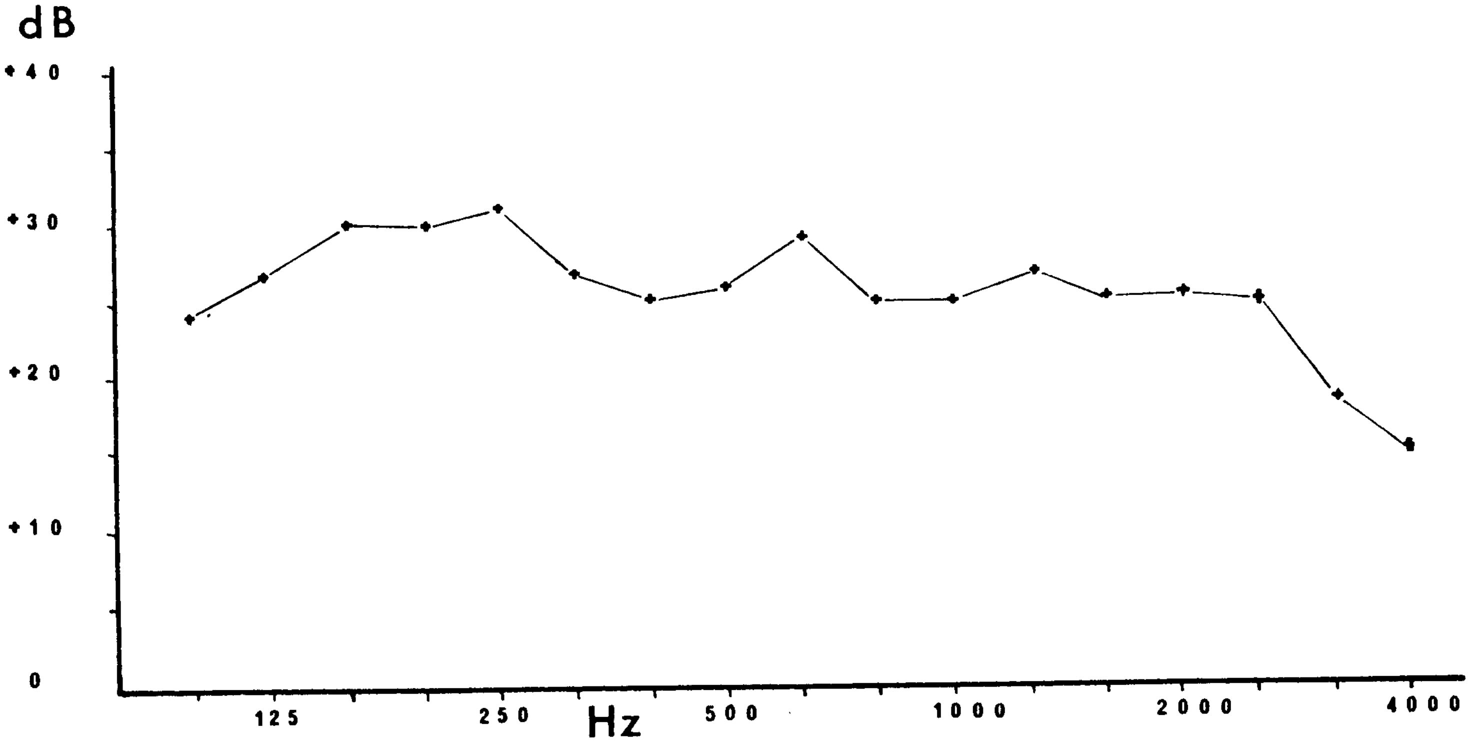

FIG. 1 4 Speaker Source in Reasonably Reverberant Room

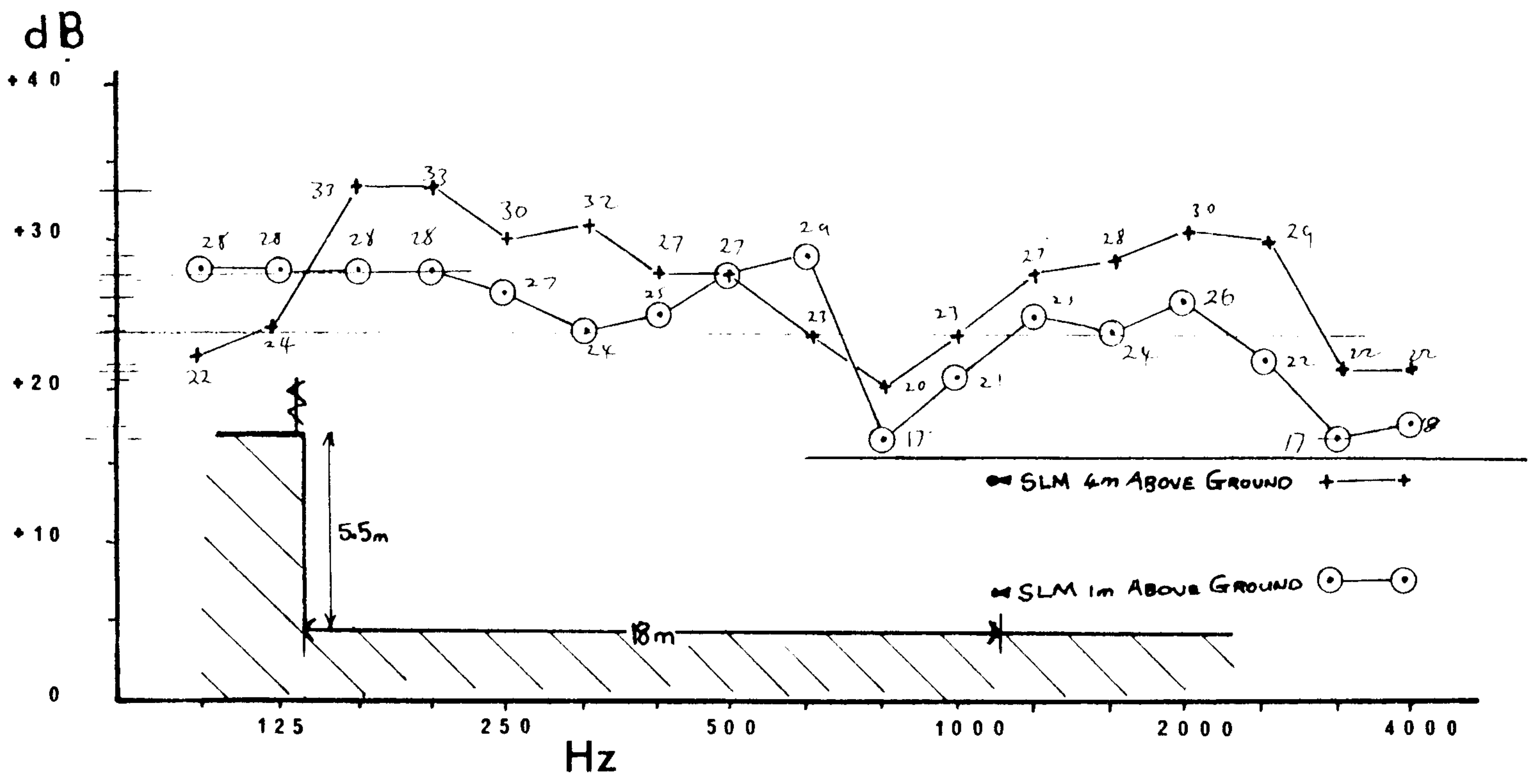

FIG. 2 Elevated 2 Speaker Source Transmitting over 18 metres of Grass

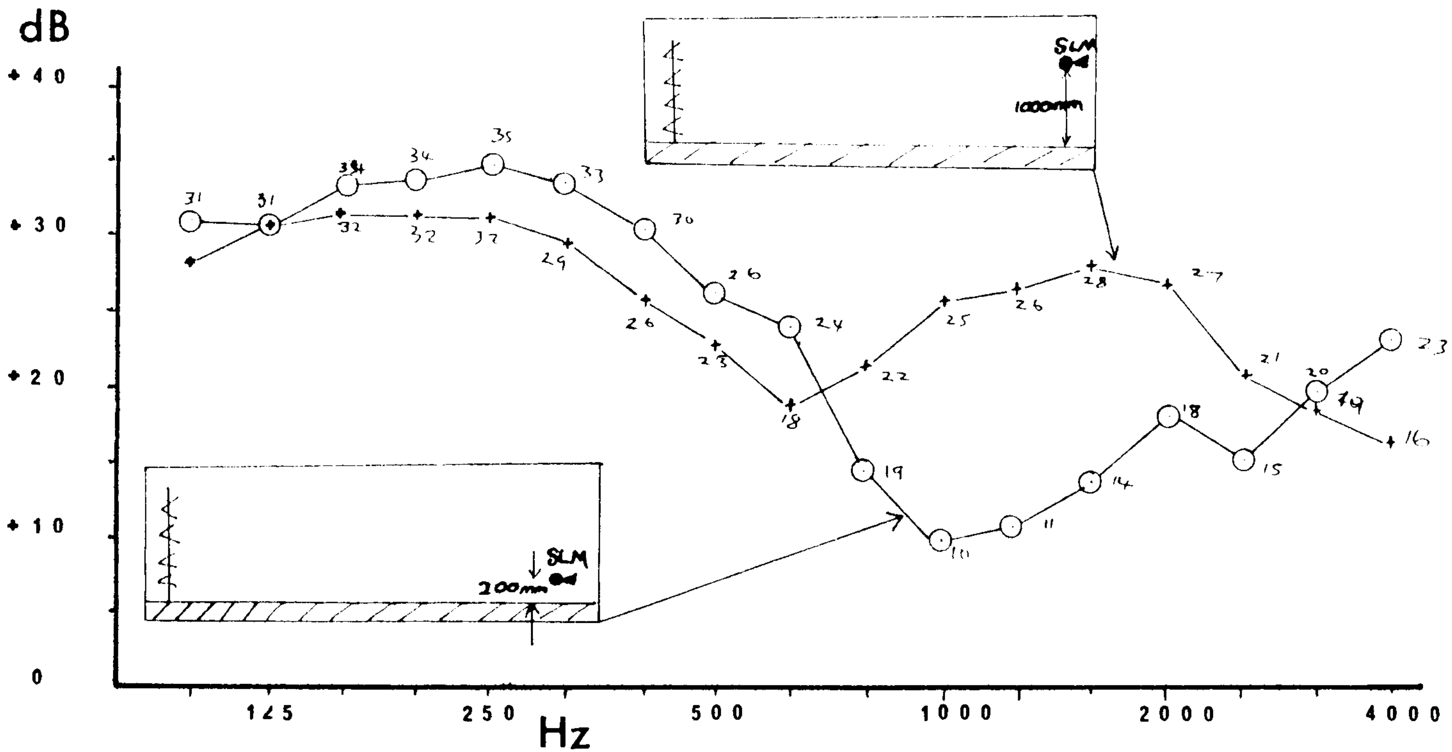

FIG. 3 Four Speaker Source, Transmitting over 5 metres of Grass

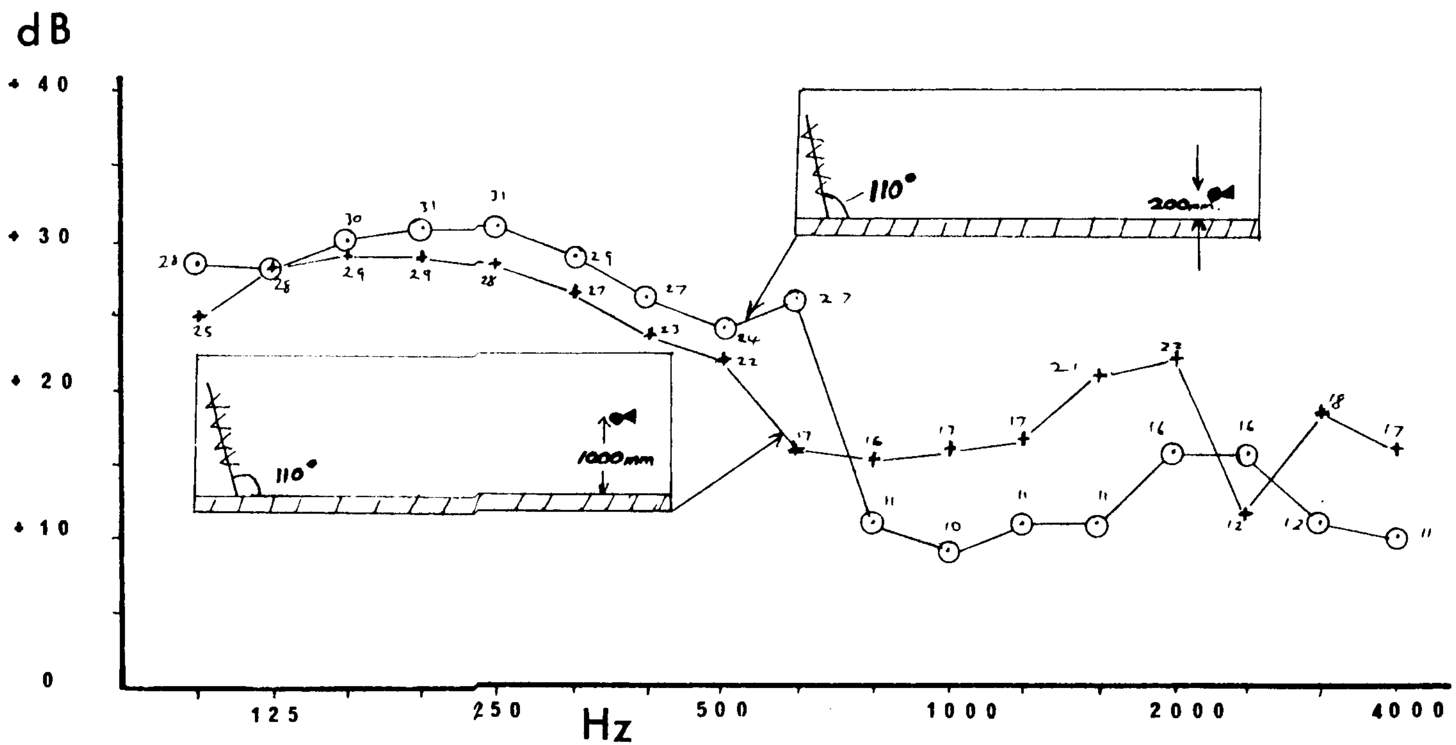

FIG. 4 Tilted Four Speaker Source, Transmitting over 5 metres of Grass

SHOULD ROAD TRAFFIC NOISE BE MEASURED BY THE NUMBER OF NOISE EVENTS?

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The paper proposes, without formal definition, the concept of number of noise events as a measure of road traffic noise. Various studies which relate traffic noise annoyance to traffic stream composition and others which examine the frequency of reports of a specific vehicle nuisances are examined, and from these it is suggested that the noise event measure could be the determinant of community annoyance. The relationship between noise events and current noise scales is examined. It is suggested that the effectiveness of most traffic noise control strategies in reducing annoyance could be significantly underestimated by the use of scales such as L_{10} and L_{eq}.

Introduction

The noise scales of L_{10} and L_{eq} are almost universally accepted as measures of road traffic noise best suited to the prediction of human response and hence to be used in assessment and design of acoustic conditions near roadways. There has been considerable support for these scales, particularly L_{eq}, from a variety of laboratory and field studies examining the relationship between community response and noise exposure.

However, a succession of studies over the past five years have also reported the rather surprising result that some traffic measure, usually the number of heavy vehicles on the roadway (NHV) or its derivatives (log NHV, %HV, log%HV) have correlations with response to noise which are as high as, or higher than, the acoustical scales. For example, Langdon (1976) found that under free-flow traffic conditions, both total traffic flow and the number of heavy vehicles correlated with dissatisfaction with noise as well as did L_{10} or L_{eq}. For non-freely-flowing traffic, most of the noise scales were not significantly correlated with dissatisfaction, whereas various measures of the number of heavy vehicles in the traffic stream were. Yeowart et al (1977), from their study of free-flow, congested and motorway conditions, suggested a combination scale (L_{10} + 0.18NHV, where NHV was measured between midnight and 0600) as the best predictor of group dissatisfaction. Rylander, Sorensen and Kajland (1976) reported that the number of heavy vehicles in the traffic stream had almost as high a correlation with annoyance as L_{eq} and L_{10}, and made passing reference to a combined scale L_{01} + 10 (NHV). Finally, Brown (1978) reported from an Australian study that the correlation of group annoyance with various measures of number of heavy vehicles was significantly higher than its correlation with any of the conventional noise measures.

Why should a "non-acoustic" variable such as the number of heavy vehicles on the roadway be a predictor of annoyance with traffic noise? Langdon (1976) has pointed out that to a resident indoors the passage of a heavy vehicle must be indicated by the noise it makes. It is not unreasonable then to suggest that the vehicle passage is identified by the excess of the vehicle's peak noise level above other noises. This means that the non-acoustic variable "number of heavy vehicles" is probably a surrogate for the acoustic variable "number of peaks from heavy vehicles".

Further, Bodsworth and Lawrence (1978) found from a selection of Sydney roadways that while, on average, 85% of "noticeable peaks" were generated by medium or heavy trucks, the remainder were generated by motorcycles or poorly muffled cars. It seems reasonable to assume that a person inside a dwelling may regard these peaks from other vehicles in the same way as the peaks from heavy vehicles and it is suggested that the acoustic variable which might correlate best with annoyance may be a more general measure such as the number of noise events in the traffic stream.

Defining a noise event

The suggestion is put forward here that the effects of traffic noise on people may be determined more by the number of interruptions - defined in the broadest sense - which they experience, than by some time-averaged level of the noise. Interruptions are likely from discrete noise events which are discerned by an observer above prevailing noise levels. Near a continuous stream of traffic a noise event would result from the passage of an exceptionally noisy vehicle. However, the definition could also be extended to low volume roadways where a noise event may result from the passage of almost any vehicle. This implies that the "threshhold" above which a noise event is discernible varies according to the situation of the listerner. Annoyance with such interruptions could occur at any noise level, subject to a lower limit of noticeability of noise events and an upper limit of intolerable background level. Any definition should include minimum and maximum durations for a sound to be recognized as an event.

To an acoustician,a noise event measure should be more conceptually satisfying as a predictor of traffic noise annoyance than are the simple measures of traffic composition reported in the above studies. The idea should not appear too alien to anyone familiar with the assessment of aircraft noise.

Nuisance caused by noise events

Apart from the performance of number of heavy vehicles as a predictor of annoyance, there is some evidence to support the concept that annoyance with traffic noise might be determined by noise events, particularly those generated by the noisier vehicles in the traffic stream.

Most surveys of the effects of traffic noise annoyance have sought, in addition to some global measure of annoyance, respondent's comments on the noticeability and nuisance of noises from different components of the traffic stream. Table 1(a) shows results from a national U.K. survey of road traffic nuisance (Morton-Williams et al 1978). Only respondents who reported that they heard a specific noise source were asked if they were bothered by it.

Despite the fact that heavy vehicles and motorcycles constitute a minority of vehicles on the road, the table shows the importance attributed to these sources and to the noise of vehicle operations - all high level noise events. Not only did they bother more people than did general traffic, but any person hearing such a noise was more likely to be bothered by it. Langdon's (1976) data, shown in Table 1(b), is different in that a higher proportion reported bother with traffic, and few with motorcycles, though in this case, the sample were all located along urban roadways carrying moderate to high traffic volumes rather than being drawn from the whole U.K. population. However, trucks and buses still bothered a high proportion of respondents, and, more importantly, were rarely heard without the concomitant experience of nuisance. Similar results were obtained in a survey of 818 respondents along 19 urban roadways in Australia (Brown, 1978). Unprompted responses were sought to the question regarding what noise respondents noticed most in their area. Over half reported noise from a specific vehicle source including 23% who gave a first mention to heavy vehicles and 12% a first mention to motorcycles. In this study, the proportion of heavy vehicles at sites ranged only from 1% to 12%, and motorcycles never exceeded 2%.

This disproportionate attention given by respondents to such vehicles, far exceeding what would be expected from their actual numbers in the traffic stream, and in particular, respondent's imputation of nuisance to them, suggests that when people are asked to assess their global annoyance with traffic noise they may give particular weight to the presence of noisier vehicles in the traffic stream.

Relationship of the number of noise events to current noise scales

An important question is, even if noise events such as motorcycle and heavy vehicle passages do contribute so much to annoyance, is it not likely that noise scales in current use adequately take this into account? For example, one would expect that the number of noise events would be closely related to L_{eq} and L_{10}. If so, there would be little to gain from the use of the noise events concept.

One of the few studies which provide data to test this relationship is that by Bodsworth and Lawrence (1978) who measured noise peaks, L_{10} and L_{eq} on a sample of roadways in Sydney - mainly on multi-lane roadways carrying 1000 to 3500 vehicles per hour under a mix of free-flow and stop/start conditions. The measured number of "noticeable noise peaks" in a 10 minute(?) period is plotted against L_{10} in Figure 1. It is clear from this sample that if annoyance *was* determined by the number of noise events, then the L_{10} scale would have little power to predict annoyance. A similar plot could have been produced for L_{eq}. In the light of this finding, it is interesting to note that Langdon (1976) reported no correlation between dissatisfaction and the noise scales for congested traffic conditions. (However, his result was not replicated by Yeowart et al (1977)).

There is no equivalent data to examine this relationship under free-flow traffic conditions. However, in both the samples of free-flow roadways reported in Langdon (1976) and Brown (1978) it can be shown that each of L_{10} and L_{eq} have reasonably high correlation ($r \approx 0.8$) with the number of heavy vehicles on the roadway. Assuming number of noise events and number of heavy vehicles to be similar under free-flow conditions, this correlation implies that both the noise scales and the number of noise events would have similar relationships to annoyance.

The link between noise events and currently used noise scales needs further investigation. Traffic volume, composition and flow regime, as well as distance from the source and even different definitions of what constitutes a noise event, would all influence the relationship.

Using the noise event concept

If annoyance *were* found to be determined by noise events, the most important ramification would be in the manner in which strategies for control of road traffic noise are evaluated. At present, the quieting of vehicles, the use of distance, shielding and insulation, and the implementation of various traffic control techniques are all evaluated by their effectiveness in reducing the magnitude of L_{10} or L_{eq}. However, with a noise event concept, the aim would be either to eliminate the source of the event, or to reduce its magnitude below the threshhold. The latter entails reducing the peak levels from individual sources.

Each of the control strategies listed above can be shown to produce a far larger reduction in peak noise levels than in the time averaged noise scales. Consequently, the effectiveness of such strategies in reducing annoyance may be significantly underestimated at present. For example, pursuing reduction in emission levels from heavy vehicles would dramatically reduce the number of noise events, yet Nelson and Fanstone (1974) demonstrated that quieting heavy vehicles alone would produce minimal changes in L_{10}. Again, the peak level of a noise event is reduced far more than is L_{10} or L_{eq} by distance (spherical rather than cylindrical divergence) and by shielding (point source rather than a line source). It is interesting to speculate that, in situations where barriers constructed near roadways have been found to reduce residents' annoyance despite minimal reductions in the L_{10}, the reduction in annoyance may have resulted from a significant attenuation of the peak noise levels. To date, such reduction in annoyance has been acribed to "psychological" factors. (eg. Dunn 1976).

A noise event concept could also prove useful in "environmental capacity" studies in residential street networks (eg. Holdworth and Singleton 1980). L_{10} and L_{eq}, particularly when averaged over long periods, lack ready interpretation at the low traffic volumes experienced in such studies. Similarly, these scales appear insensitive in assessing the likely impact of an increase in the number of heavy vehicles utilizing a roadway which previously carried only automobiles. For example, 30 trucks per hour from a new quarry traversing a route normally carrying 300 cars per hour would increase L_{10} and L_{eq} by 3dB(A) or less. It is suggested that the impact on residents of an additional noise event every 2 minutes may be more than this small increment suggests.

Summary

The concept of measuring noise events for road traffic noise instead of conventional noise scales was developed from the high correlations found in field studies between annoyance and number of heavy vehicles on the roadway. The concept receives some support from the high incidence of reports of nuisance from heavy vehicles and motorcycles.

The number of noise events generated by a traffic stream may be a unifying measure for predicting annoyance under all traffic situations, *viz* free-flow and congested conditions as well as conditions of low traffic volume. If annoyance is caused by noise events, then the effectiveness of most traffic noise control strategies is currently underestimated by evaluating them only on the basis of reductions achieved in L_{10} or L_{eq}.

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TABLE 1

	% hearing	% bothered	% of those hearing who are bothered
(a)			
n = 5686			
General traffic	73	18	25
Start/gears	72	21	29
Motorcycles	67	26	39
Lorries	66	22	33
Cardoors	64	20	31
Car Horns	57	13	23
Brakes/tyres	46	22	48
(b)			
n = 2933			
Road traffic (unspecified)	70	48	68
Private cars	14	5	36
Motorcycles	6	3	45
Trucks and buses	25	21	83

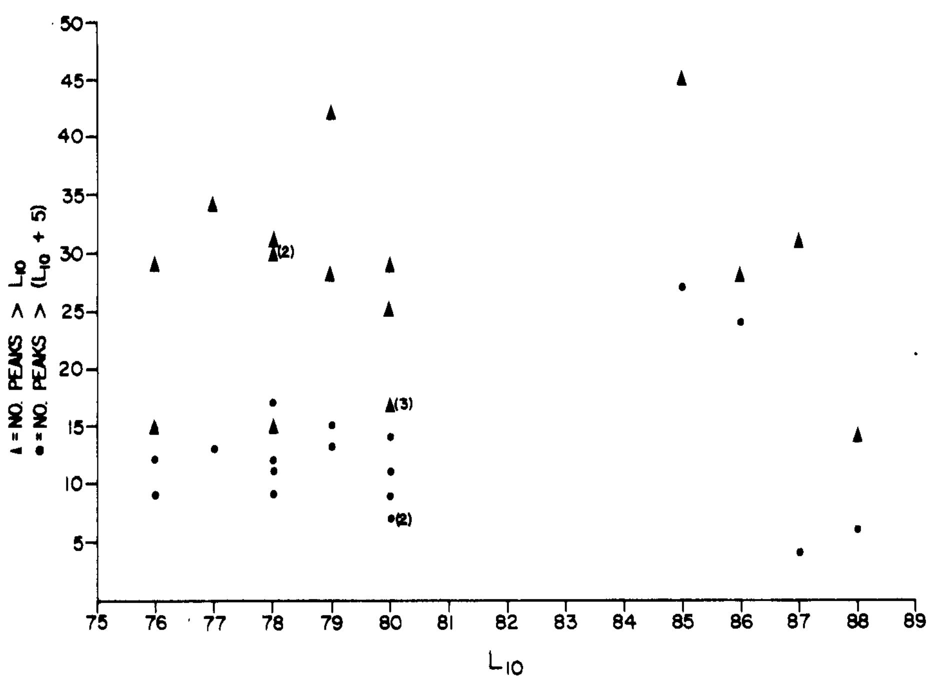

FIGURE 1. The relationship between L_{10} and the number of noticeable noise peaks in a 10 minute(?) period. Noticeable peaks were defined in two ways: > L_{10} and > L_{10} + 5dB(A). The results are from roadways in Sydney and have been plotted from data in Bodsworth and Lawrence (1978).

APPLICATION OF FINITE ELEMENT MODELLING TO THE PREDICTION OF INTERIOR NOISE LEVELS OF AN ARMOURED PERSONNEL CARRIER

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Abstract

A formalism is presented for incorporating modal output data from finite element structural models into the more conventional framework of space and frequency averaged vibrational response and interior noise predictions. The methodology is applied to calculation of interior noise-to-force transfer functions for an armoured personnel carrier with good results. Considerable computational cost savings and more convenient and improved data interpretation result from the use of the outlined method.

Introduction

Calculation of the flow of vibrational energy resulting from mechanical excitation, into a structure composed of beams and plates, is quite complex for all but the simplest structural configurations. Resort is therefore often made to discreetized structural modelling methods such as finite elment analysis. Such techniques allow the generation of point-to-point, narrow-band transfer functions which tend to be strongly frequency dependent. Valid interpretation requires extensive additional non-standard computer-based data reduction and this is rarely conducted.

In achieving engineering solutions to such problems, one seeks analytical frameworks which simplify data interpretation. Presently it has been found useful to develop frequency and space averaged formulations to bridge between the output data of accurate (finite element) structural models and the conventional formalism of vibro-acoustics.

This paper presents a summary of the analytical method used to predict the development of the vibrational response and interior sound levels of vehicle structures to mechanical excitation. Normal mode analysis is extended to derive, in terms of the output parameters of finite element analysis, relationships for the space-averaged vibrational response of and acoustic power radiation from the vehicle surfaces, averaged over say, one-third octave, frequency bands. Application of the formulation to the prediction of the interior noise levels of an armoured personnel carrier, is presented.

Vibrational Response of a Complex Structure to Mechanical Excitation

The general situation considered consists of a complex structure which is driven by an excitation force of amplitude $F(\omega)$ at frequency ω located at $\bar{x} = \bar{x}_F$ and loaded by interior and exterior pressure fields, p_i and p_2 respectively, as shown in Figure 1. The structure is represented by its set of independent normal modes (uncoupled and orthogonal), so that the total displacement of the structure $w(\bar{x},\omega)$ at a point $\bar{x} = (x,y)$ on the surface is given by

$$w(\bar{x},\omega) = \sum_r \bar{\xi}_r \Psi_r(\bar{x}) \qquad (1)$$

where $\bar{\xi}_r = \bar{\xi}_r(\omega)$ is the displacement of the r^{th} normal mode. The response to the forces and pressure fields acting is found by integrating the product of the structure's Greens function and the forces acting on the surface, all over the vibrating surface. Neglecting the forces developed by the adjacent pressure fields which will be negligible for heavy structures surrounded by air, the total response of the structure is derived to be

$$w(\bar{x},\omega) = F(\omega) \sum_r \frac{\psi_r(\bar{x})\psi_r(\bar{x}_F)}{M_r Y_r} \qquad (2)$$

Here ψ_r, M_r and Y_r are the mode shape, generalized mass and admittance function for the r^{th} structural mode. η_r is the sum of the structural and external radiation loss factors.

The time average, mean square displacement at frequency ω

$$\begin{aligned}\overline{w^2(\bar{x},\omega)} &= \tfrac{1}{2}\,\mathrm{Re}\,[w.w^*]\\ &= \frac{F^2(\omega)}{2}\left\{\sum_r \frac{\psi_r^2(\bar{x})\,\psi_r^2(\bar{x}_F)}{|Y_r|^2 M_r^2} + \mathrm{Re}\sum_r\sum_{\substack{s\\ r\neq s}} \frac{\psi_r(\bar{x})\,\psi_s(\bar{x})\,\psi_r(\bar{x}_F)\,\psi_s(\bar{x}_F)}{Y_r Y_s^* M_r M_s}\right\} \quad (3)\end{aligned}$$

Thus the structural response depends on the spectrum of the excitation force, the mode shapes at both the point of application of the force and the observation point, and on the modal masses and admittance function.

As frequency increases above the fundamental structural modes, the number of modes accumulates rapidly with frequency. Certain structural elements may tend to respond more strongly (independently) than connected elements, and may tend to contribute most to the structural modal mass and hence kinetic energy at that resonance frequency. An estimate for the space-averaged, mean-square displacement of each of the major structural elements to a band-limited force excitation of mean square force in $\Delta\omega$ of F_Δ^2 is of considerable interest and use in the evaluation of the effects of structural modifications on both structural vibration response and the associated interior acoustic radiation.

The space-average, mean-square displacement, for an element A of the structure, averaged over a narrow frequency band containing several resonant modes $(r\varepsilon\Delta\omega)$, $\overline{w^2_{\Delta,A}}$, is found, on integrating equation (3) over the elemental areas A and over the frequency band $\Delta\omega$, as

$$\begin{aligned}\overline{w^2_{\Delta,A}} &= F_\Delta^2 \sum_{r\varepsilon\Delta\omega} \frac{\left[\frac{1}{A}\int_A \psi_r^2(\bar{x})\,d\bar{x}\right]\,\psi_r^2(\bar{x}_F)}{M_r^2} \cdot \frac{1}{\Delta\omega}\int_{\Delta\omega} |Y_r|^{-2}\,d\omega \;+\; \text{2nd Order Terms}\\ &\simeq \frac{\pi}{2\omega^3}\cdot\frac{F_\Delta^2}{\Delta\omega}\sum_{r\varepsilon\Delta\omega}\frac{\left[\frac{1}{A}\int_A \psi_r^2(\bar{x})\,d\bar{x}\right]\;\psi_r^2(\bar{x}_F)}{M_r^2\eta_r} \quad (4)\end{aligned}$$

Here the modal cross-coupling terms $(r \neq s)$ do not contribute to the response and the response of mass and stiffness controlled modes is assumed to be of second-order importance relative to that of modes resonant in $\Delta\omega$.

The quality in square braces [], is inversely proportional to the degree of restraint provided by the remainder of the structure on element A. It can be calculated for each structural element using, for example, finite element analysis, as

$$\frac{1}{A}\int_A \psi_r^2\,(\bar{x})\,d\bar{x} = \frac{1}{A}\sum_i\sum_{j\varepsilon A} \psi_r^2\,(i,j)\;A(j) \simeq \frac{1}{A}\sum_{j\varepsilon A}\psi_r^2\,(3,j)\;A(j) \quad (5)$$

where A(j) is the area associated with the j^{th} node and where i and j represent the nodal coordinate directions and node number respectively. It can be assumed that only displacements normal to the surface are significant.

The contribution of structural elements to the total modal mass can be calculated as

$$M_{r,A} = \int_A m(\bar{x})\psi_r^2(\bar{x})\,d\bar{x} = \sum_i \sum_{j\varepsilon A} m(i,j)\psi_r^2(i,j) \simeq \sum_{j\,\varepsilon A} m(3,j)\psi_r^2(3,j) \tag{6}$$

where normal displacements are most important. Here m(i,j) is the nodal mass of the j^{th} node referred to the i^{th} coordinate direction.

Acoustic Power Radiated to Interior Acoustic Field

The time-averaged acoustic power radiated due to the motion of the r^{th} structural mode is given in terms of the modal radiation resistance as

$$W^r_{rad}(\omega) = R^r_{rad}(\omega) < v_r^2 > \tag{7}$$

where $< v_r^2 > = \omega^2 < w_r^2 >$, is the space-time average, mean-square velocity of the r^{th} mode.

At those frequencies were several acoustic modes occur in the analysis band, the band-averaged radiation resistance will provide a good estimate for the structural-acoustic coupling for each structural element. Then, if there are N modes in a frequency band $\Delta\omega$ and assuming equipartition of energy between these modes applies, the total time-average acoustic power radiated W_Δ is

$$W_\Delta = v_\Delta^2 \cdot \frac{1}{N} \sum_{r\varepsilon\Delta\omega}^{N} R^r_{rad}(\omega) = v_\Delta^2 \cdot \overline{R_{rad}} \tag{8}$$

where v_Δ^2 is the space-average, mean-square velocity of the radiating surface and $\overline{R_{rad}}$ is the band-averaged radiation resistance. Eq. (4) may be used to calculate v_Δ^2, and $\overline{R_{rad}}$ can be calculated using the methods of statistical energy analysis. For an element A of structure, the band-limited radiated acoustic power is

$$W_{\Delta,A} = \omega^2 \overline{w^2_{\Delta,A}} \cdot \overline{R_{rad,A}}$$

$$= \frac{\pi}{2} \frac{F^2_\Delta}{\omega\Delta\omega} \cdot \sum_{r\varepsilon\Delta\omega} \frac{[\frac{1}{A}\int_A \psi_r^2(\bar{x})\,d\bar{x}]\ \psi_r^2(\bar{x}_F)}{M_r^2\eta_r} \cdot \overline{R_{rad,A}} \tag{9}$$

Application to the Hull of an Armoured Personnel Carrier (APC)

A finite element package was used to compute the resonance frequencies, modal masses and mode shapes for the hull structure of an M113A1 APC. 221 nodes were used to represent half of the structure, bilateral symmetry being assumed for modelling economy. Plate and beam elements were used. The nodal arrangement indicated in Figure 2 for half of the structure, was found satisfactory to give adequate shape definition for about 70 modes extending up to 350 Hz for the full structure. The major panel elements are named in Figure 2(a).

Figure 2 presents computed shapes for a range of representative modes. The lowest frequency modes involve breathing, racking and ovaling of the structure. As frequency increases the modal displacements tend to be confined to individual panel elements, mainly the top, bottom and upper side plates.

Values of the parameter describing the contribution of panel elements to the total hull modal masses $M_{r,A}$, normalised by the total modal masses M_r, are presented in Table 1 for several modes. As the normalised mass parameter approaches unity for any element, the modal energy of vibration becomes concentrated in that element. For example, for the anti-symmetric mode resonant at 66 Hz the top plate contributes 83% of the modal generalised mass while, for the symmetric modes at 55 Hz and 296 Hz, the major contributions to M_r are provided by the vehicle sides and sponson (82%) and the bottom (at least 77%) respectively.

The transfer function between surface response and idler force $v^2_{\Delta,A}/F^2_\Delta$ is calculated with equation (4), for example as shown in Figure 3 for the hull top plate for a vertical idler force. In addition to the transfer function due to resonant modes ($r\varepsilon\Delta\omega$), the contributions due to mass-controlled ($r<\Delta\omega$) and stiffness-controlled. ($r>\Delta\omega$) modes are shown. It is clear that, for all bands in which resonant modes exist, the resonant response dominates the response-force transfer function.

The total acoustic power radiated into the hull interior volume from the various structural elements in response to a driving force at the idler attachment position will be absorbed on the hull surfaces. From this power balance, the interior noise-to-idler force transfer function for each structural element can be derived as

$$\frac{\langle p_i^2 \rangle}{F^2_\Delta} = \frac{\rho c^2}{V\eta_i} \; \frac{\overline{R}_{rad,A}}{\omega} \cdot \frac{v^2_{\Delta,A}}{F^2_\Delta} \qquad (m^{-4})$$

where, in frequency band $\Delta\omega$, $\langle p_i^2 \rangle$ is the space-average mean-square interior acoustic pressure, F^2_Δ is the mean-square idler force and $v^2_{\Delta,A}$ is the mean-square velocity of elemental area A. V, ρc and η_i are the volume, characteristic impedance and acoustic loss factor of the hull interior space. Then, from calculated values of $\overline{R}_{rad,A}$ and measured values of η_i and V, the noise-to-force transfer functions for each major radiating element can be calculated, and then summed to give the overall noise-to-force transfer function. A typical example is presented in Figure 4 together with measured noise-to-force transfer function for vertical excitation of the idler attachment. The agreement is generally satisfactory, validating the general approach followed.

Conclusions

A method for incorporating modal output information generated from finite element analysis models for complex structures, into the more familiar (to acousticians) framework of frequency and space averaged response and transfer functions, has been developed. While the method is fully consistent with the conventional approach of using transfer function spectral densities, the averaging procedures presented are computationally much more efficient and provide, at least for vibro-acoustics problems, results more directly useful in understanding the effects of various structural parameters or of changes to basic structure.

Table 1

Normalised Generalised Mass of Structural Elements ($^{M}r,A/^{M}r$)

Resonance Frequency Hz	Top	Upper Side	Lower Side	Structural Element Sponson	Bottom	Rear Plate	* Total
29.5	.89	.030	.014	.014	.024	.013	.990
55.0	0.051	0.265	0.252	0.303	0.069	.055	.995
66.9	.827	.046	.036	.022	.035	.023	.988
105.1	.309	.058	.069	.026	.401	.009	.872
134.9	.948	.039	.002	.005	.003	.003	1.000
176.1	.233	.105	.014	.009	.263	.011	.634
213.3	.388	.389	.053	.062	.024	.090	1.007
265.4	.507	.185	.044	.073	.031	.143	.983
296.6	.049	.022	.015	.010	.765	.009	.870

* In evaluating the vehicle generalised masses, the inclined sections of the vehicle noise have been omitted from present calculations for simplicity. When the 'total' normalised generalised mass is less than 1.0, significant vibrational energy occurs in this area of the vehicle. Such hull vibrations are screened by the engine cover and do not radiate into the interior space.

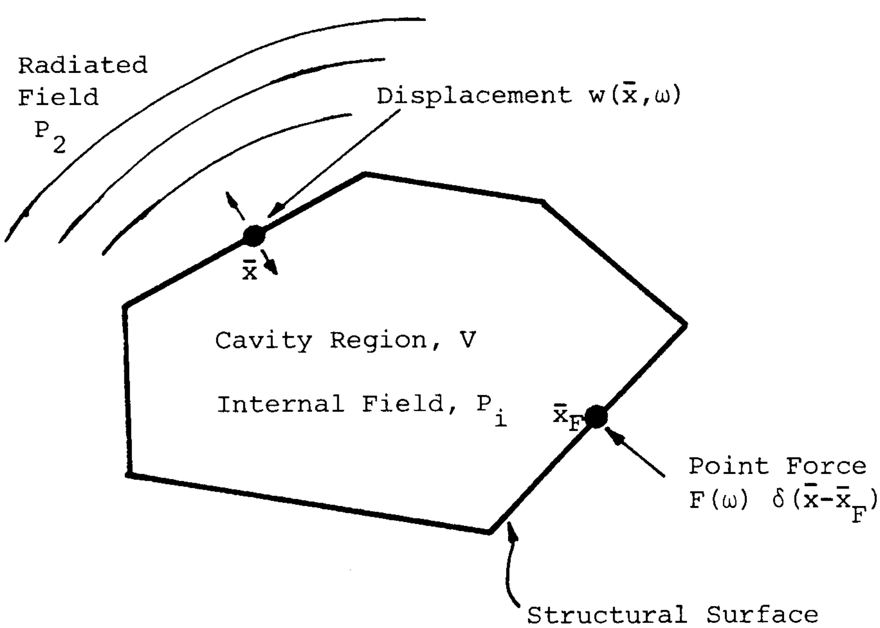
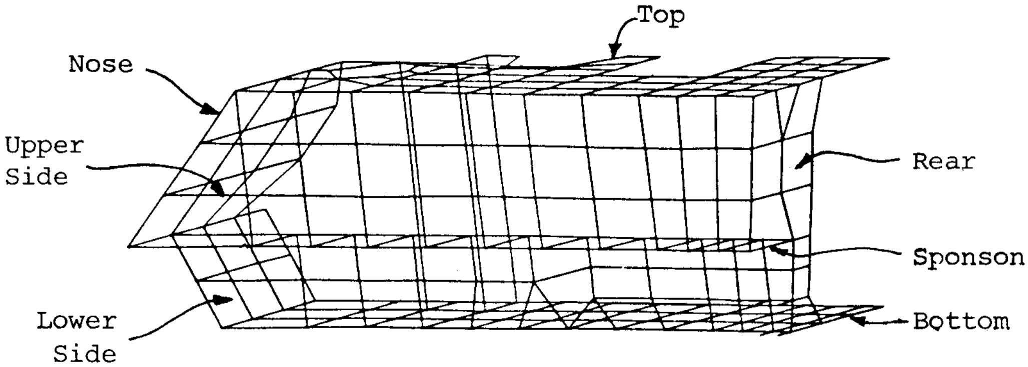
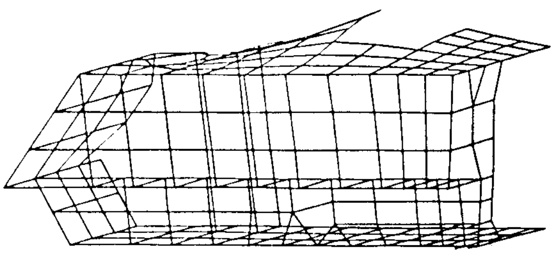


FIGURE 1

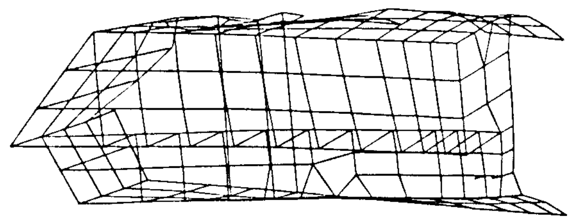
General Structural Configuration



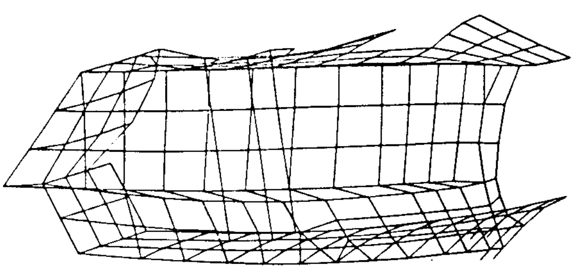
(a) Nodal configuration for M113A1 model.



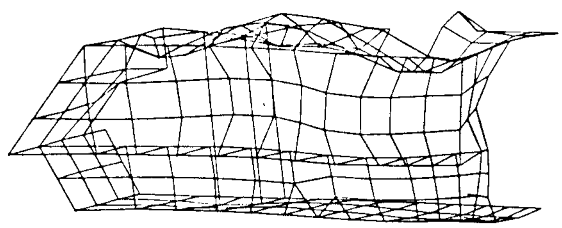
(b) 29.5 Hz symmetric



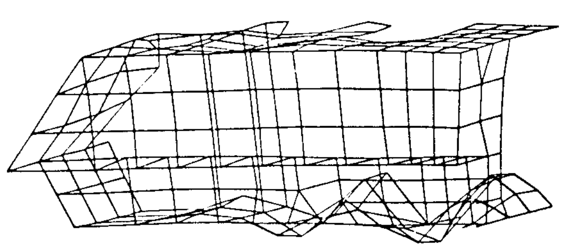
(c) 41.3 Hz anti-symmetric



(d) 55 Hz symmetric



(e) 213.3 Hz anti-symmetric



(f) 296 Hz symmetric

FIGURE 2 : Typical Vibration Mode Shapes

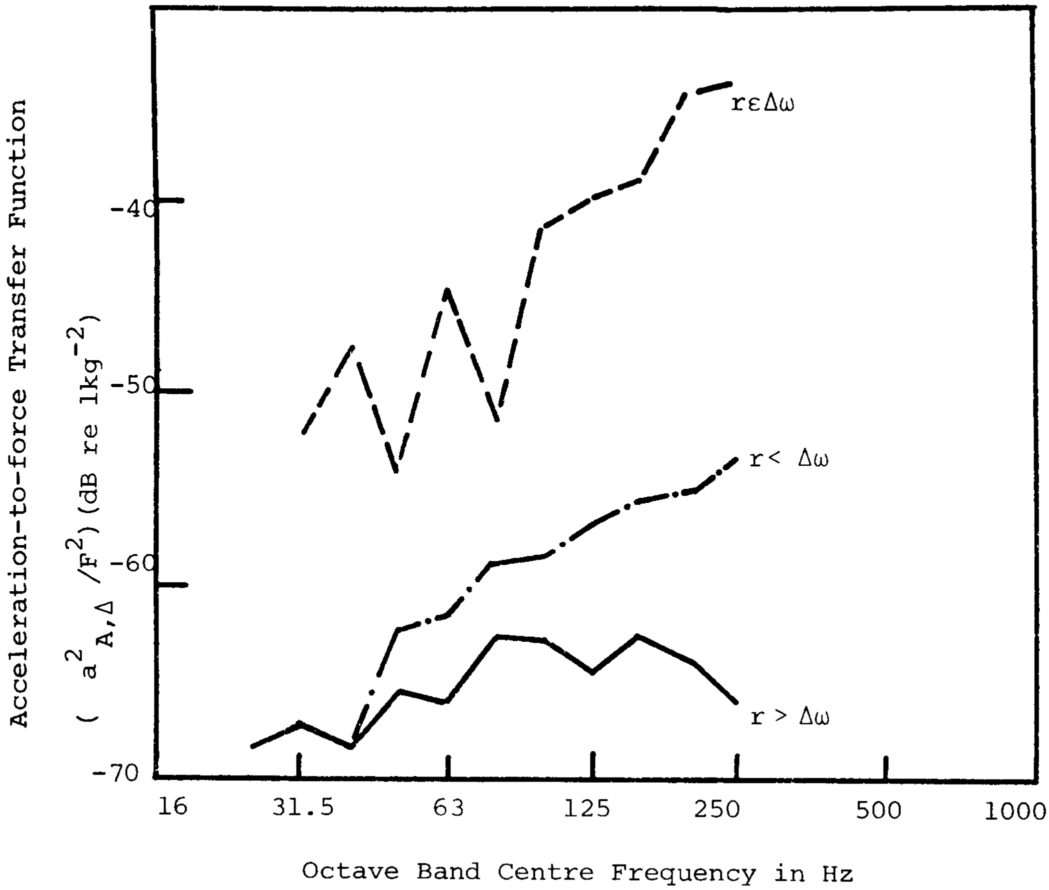


FIGURE 3 Contribution of resonant and non-resonant modes to top plate response for vertical excitation of idler.

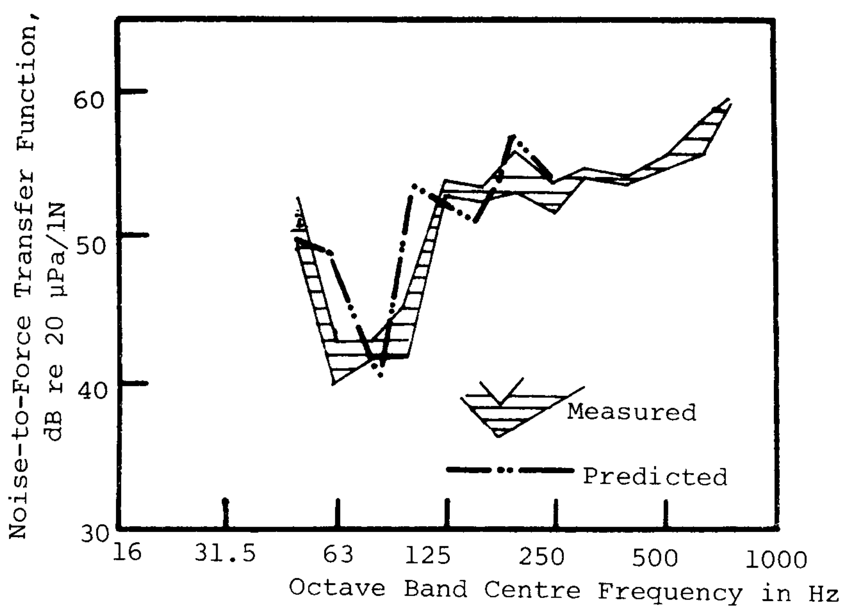


FIGURE 4

Comparison of measured and predicted noise-to-force transfer function for vertical excitation of idler spindle on M113A1.

SIMPLE NOISE CONTROLS IN THE TEXTILE INDUSTRY

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ABSTRACT. There are many noise sources in industry for which simple and cost-effective controls exist. These controls involve engineering modifications at the source, rather than enclosures around the machine. A selection of such controls, drawn mainly from the textile industry, is presented. However, the techniques are relevant to other industries.

There are two basic ways in which noise is generated: by fluctuations in fluid flow and by vibrating surfaces. The general controls for fluid flow noise are to slow and smooth the flow and to prevent resonant feedback or subsequently to insert reactive and/or dissipative mufflers. Controls in two areas of fluid flow noise, namely, steam heating of liquors and air-moving systems, are given. The general controls for vibrating surface noise are to reduce the exciting force, reduce the area radiating and its radiation efficiency, and to increase the damping. Controls for vibrations arising from both impacts and rotating parts are given.

Steam Heating of Liquors

It is common practice in textile finishing to heat liquors to near boiling using steam injected directly into the bath or vat via long perforated pipes. The noise varies with the liquor temperature as shown in Fig. 1. Peak noise levels as high as 106 dBA have been observed during the heating phase. Even though noise levels are high for only part of the cycle, the repeated cycles of several dye vats can easily lead to operator daily noise doses in excess of one.

The noise is due to the violent collapse of the steam bubbles as the steam inside condenses. The noise thus drops off near boiling because the bubbles collapse more slowly as the rate of heat transfer slows. The decreased noise levels at low bath temperatures seem to be due to the steam condensing before the jet has travelled far into the liquor. The pressure pulses from bubble collapse set the bath walls into vibration. These are usually of thin stainless steel, and hence are efficient noise radiators.

A possible, but expensive, control would be to damp the fairly live steel surfaces. An alternative is to replace the perforated pipe by commercially available silent steam injectors such as those by Bestobell [1], Penberthy [2] or Horne [3]. The first two use the steam to entrain a liquor flow which then mixes with the steam inside a thick-walled tube or duct. The typical noise reduction for the first two silencers is 10 to 12 dBA and the third has given 5 to 9 dBA under a limited range of test conditions. A disadvantage of all three types is that a single unit replaces a long length of perforated pipe and steam distribution is not as even.

A novel and simple method of noise control that does not suffer from the same disadvantage is to add air to the steam before discharge. The air inside the steam bubbles then acts to slow and cushion the bubble collapse and drastically reduces the pressure fluctuations. The amount of air required is about 1/10th by volume of the amount of steam. The method has achieved reductions of 20 dBA depending on the initial noise level and amount of air used.

An elegant means for introducing the air is to use the steam flow itself to draw in atmospheric air using a commercially available steam ejector or steam siphon (Fig. 3). Since a large volume of air is not required the ejector nozzle can be bored out to increase the steam flow (by 60% to 70%). A control valve or orifice is required on the air inlet because too much air leads to excessive bubbling.

The one disadvantage of adding air is that it will cause foaming when surfactants are present. This is not usually a problem when dyeing but can be when scouring.

<u>Air Moving Systems</u>

The most common noise source of air moving systems is the centrifugal fan. The methods of noise reduction at the source have been reviewed by Neise [4]. The noise is either pure tone, at the blade-pass frequency and harmonics, or broadband due to turbulence. No satisfactory controls that can give a significant reduction in broadband noise appear to be available at present. However, a number of possible options are available for the reduction of the pure tone noise. The cut-off clearance and the radius of the curvature of the cut-off can be increased. In general, this appears to have negligible effect on fan performance. Alternatively, the blades or cut-off can be angled. This will give an effective reduction in pure tone noise provided the cut-off spans at least two blades. The authors have tried this option but have achieved reductions of only 3 to 4 dB on fans with a small number of blades.

If the fan cannot be modified, then the noise can be muffled before it escapes. The most common controls are the plenum or acoustically lined duct. However, if the noise consists of pure tones then an often overlooked alternative is the tuned side-branch. The quarter-wavelength side-branch produces a reflected wave that is 180^{o} out-of-phase with the sound wave travelling down the duct. The net result is that the transmission loss of the duct at this wavelength is greatly increased [5]. The main disadvantage is that only the fundamental and every second harmonic are attenuated. In principle, the transmission loss of a side-branch is zero for the 1st, 3rd, 5th,.. harmonics, but, in real systems the side-branch does have an effect on the impedance seen by the source at the odd harmonics. Experiments have shown that these harmonics can actually be increased slightly if the end of the side-branch is n /2 from the source. However, the odd-numbered harmonics can be removed with a second side-branch half the length of the first. It is best to experimentally determine the length of each side-branch, but /4 - 0.2 d (d = duct diameter) is usually a reasonable approximation.

It has been recommended that a side-branch should be placed at a pressure maximum of the outlet duct [6]. This is not mandatory but can increase the impedance seen by the source and so lead to less noise being generated.

Side-branches have been successfully applied by the authors to various centrifugal fans [7] including those of a bakery where the pure tones were causing neighbourhood annoyance. A great advantage of the side-branch is that it can be applied in situations where acoustically absorbing material would clog up, e.g. from flour. It can also be applied to fluid-borne sound and has been used successfully on the output line of a hydraulic pump.

Impact Noise

Impact generated noise is surprisingly common in the textile industry. On machines such as looms and gill boxes it is difficult to cushion these impacts by a material that will wear satisfactorily. However, on other machines many of the impacts can be treated. A good example is the yarn winder in which metal yarn guides oscillate to build up a uniform yarn package. The yarn guides impact their track guides at each reversal of direction. Replacement guides of plastic have enabled reductions of from 4.5 to 10 dBA on various winders [7]. The reduction arises because the plastic part is lighter, better damped and the impacts are somewhat cushioned.

Another machine, called a comb, has a reciprocating motion as it repetitively combs a beard of fibres. The reciprocating motion leads to numerous impacts. In the case of the "shovel plate", a sheet metal component of the comb, each reversal of direction causes an unwanted rotation which results in impacts with neighbouring parts. The simple control of altering its pivot point (Fig. 3) greatly reduced its tendency to rotate. On one type of comb an additional noise source was a light gauge aluminium cover plate which bounced on other metal components. This source was effectively eliminated by attaching a sheet of Soundfoil constrained-layer damping (available from Nylex). Together, these modifications reduced the noise of the comb by 3 to 4 dB without interfering with machine operation.

Rotating Machinery

Spinning frames, which twist fibres to produce yarn, have a multiplicity of rotating parts which, together with the air-moving system, produce most of the noise. A typical frame is shown in cross-section in Fig. 4. The spindles which carry the yarn package are rotated at speeds of up to 14,000 rpm via a tape (not shown) whose position is controlled by the idler pulleys. The yarn passes onto the package via a metal loop, called a traveller, which slides around a metal ring at typical speeds of 30 m/sec. The vibrations that give rise to the noise are due to the rotating parts but most of the noise is radiated by nearby large surfaces to which the parts are structurally connected.

Wide experience has shown that in many mills the level of most of these spinning frame noise sources can be easily reduced. The noise level of such multiple machine installations can be brought to below 90 dBA at a total cost of less than 5% of the capital cost of the machines. Moreover, the different sources have different frequency signatures and so their relative importance can often be gauged from a simple frequency analysis (Fig. 5).

The general solution to the noise of the rotating parts is to vibration isolate the sources from nearby large surfaces. For example, on one machine the small idler pulleys were the major noise source but most of the noise was radiated by their metal support plates. A pair of soft neoprene rubber washers isolated the vibrations which were further reduced by slowing the speed of idler rotation by enlarging the diameter of the pulleys. The combined controls reduced this noise source by 5 dB.

Noise from the spindle vibrations can be reduced using commercially available elastomeric washers to isolate the spindles from the much larger spindle rail. Ring/traveller noise can be reduced by elastomeric mounts that isolate the ring from the ring rail and the structurally connected separator plates (Fig. 6). Further details of these controls can be found in reference 7.

The purpose of this paper has been to illustrate a range of simple noise controls of a type that should be applicable to many industries and none of which has involved an enclosure.

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Figure 1

Steam injection noise level vs. temperature

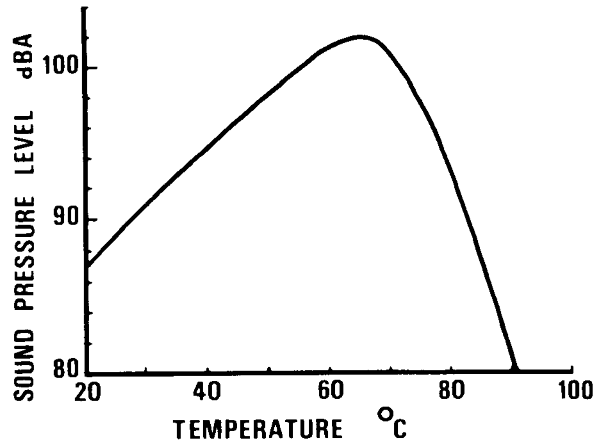


Figure 2

Schematic diagram of steam injection showing steam siphon

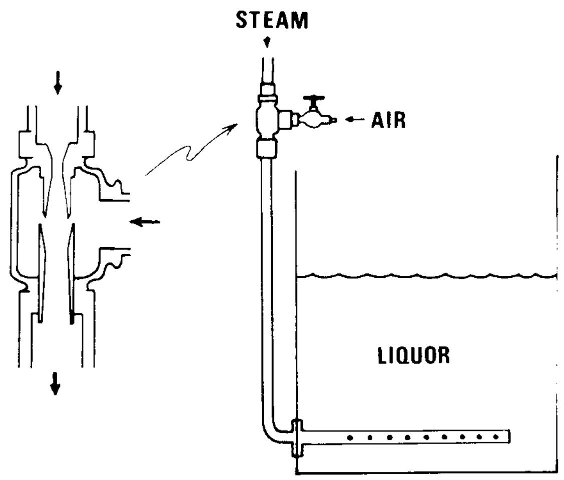


Figure 3

Schematic diagram of original and modified shovel plate linkage

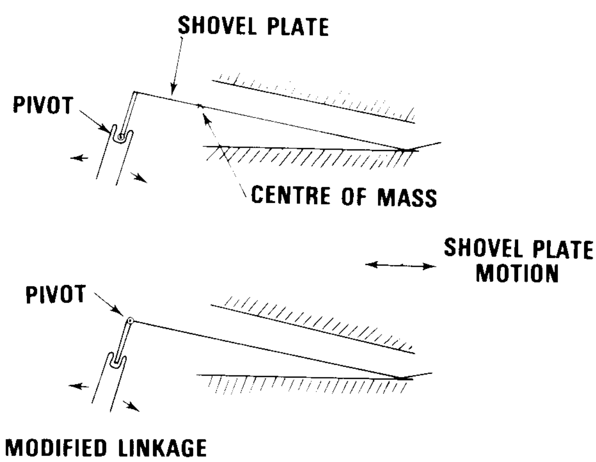


Figure 4

Schematic diagram of spindle-drive system of a spinning frame

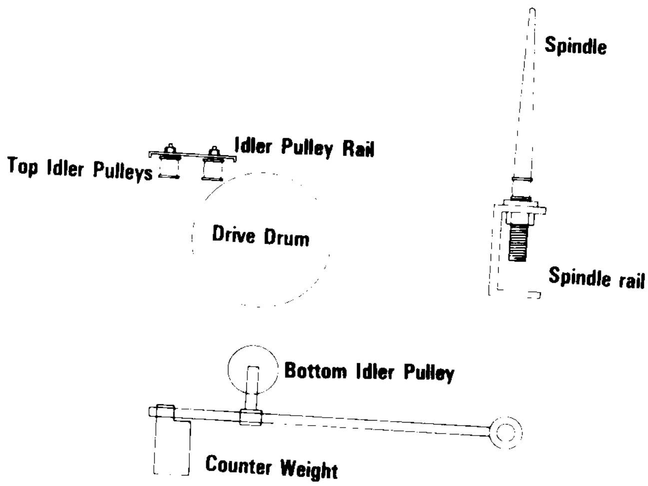


Figure 5

Major frequency bands of noise sources on spinning frames

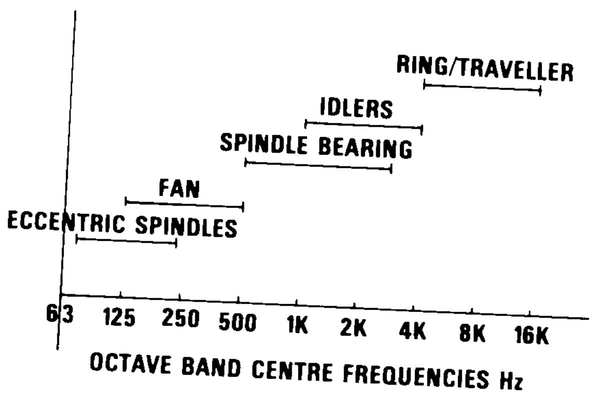
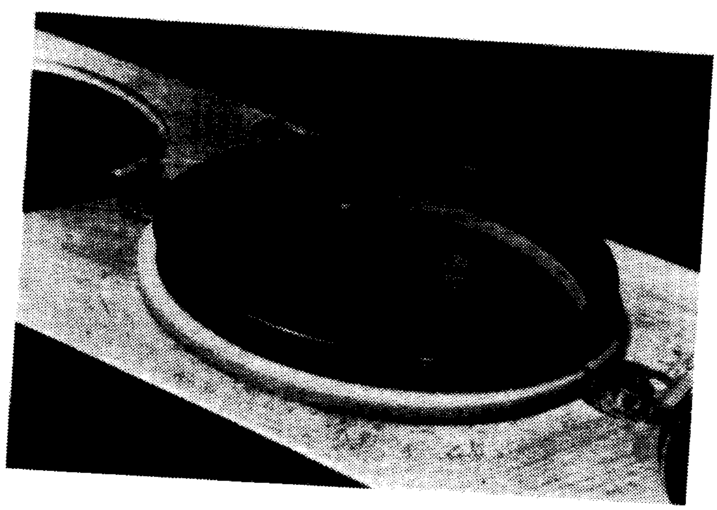


Figure 6

Ring and elastomeric ring-mount in position on ring rail



MECHANISMS OF NOISE GENERATION IN PUNCH PRESSES AND MEANS BY WHICH THIS NOISE CAN BE REDUCED

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ABSTRACT

In recent years the mechanisms of noise emission from presses have been analysed, and it is now possible to identify broadly what action should be taken in designing a press to ensure that the noise output would be lower than that generated by presently available machines. In this paper the causes of press noise emission are outlined, and the broad features required of a low noise press are illustrated by a design proposal.

INTRODUCTION

This paper outlines the theoretical principles that may be employed to explain the magnitude and spectrum of noise emitted by a press, and on the basis of this indicates the action needed to reduce this noise. An 'ideal' design to replace the C-frame press is put forward. This incorporates a variety of essential features required of a low noise press. Largely such features are no more than adaptions of techniques used in other types of press e.g. drawing presses, and no radical departures from established design approaches are involved.

MECHANISM OF NOISE GENERATION

At its simplest the energy radiated by an impulsive machine during a single stroke can be expressed as

$$E = \int_{-\tau}^{\tau} S\rho c\hat{v}^2 \sigma_r dt$$

where ρc = characteristic impedance of air

S = total surface area

$\hat{v}$ = mean surface velocity

σ_r = mean radiation efficiency.

Such an approach does, however, reveal little regarding the mechanism of noise generation and it is necessary to transform the analysis into the frequency domain to make it possible to see fully how the force generated by the tool translates into radiated sound. Over the years techniques for this have been refined mainly at Southampton University and have recently been updated by Herbert and Halliwell [1]. The preferred form of the transformation into the frequency domain is along the following lines.

$$E = 2S\rho c \int_0^\infty (V(\omega))^2 \cdot \Sigma_r(\omega) . d\omega$$

$$= 2S\rho c \int_0^\infty (F(\omega))^2 . \left(\frac{V(\omega)}{F(\omega)}\right)^2 . \Sigma_r(\omega) d\omega$$

$$= 2S\rho c \int_0^\infty (F(\omega))^2 . (H(\omega))^2 . \Sigma_r(\omega) d\omega$$

where ω = frequency in Hz

$F(\omega)$ = spectrum of force applied to the machine by the tool

$V(\omega)$ = spectrum of mean surface velocity of the machine

$H(\omega)$ = $V(\omega)/F(\omega)$ = mean structural admittance spectrum

and $\Sigma(\omega)$ = mean radiation efficiency spectrum.

The perceived L_{eq} of the noise depends on the time, T, between impulses; on the hemispheric surface area $R = 2\pi r^2$ representative of the distance, r, from the source to the observer, and also on the A-weighting effect of the ear thus

$$L_{eq}(A) = 10 \log[\frac{1}{w_o} 2 \frac{1}{T} \frac{S}{R} \rho c \int_0^\infty (F(\omega))^2 . (H(\omega))^2 \Sigma_r(\omega) . A(\omega) d\omega.$$

In this form it is easy to see how the force reacts with the structure resulting in a surface velocity, the effects of which are selectively radiated and then selectively perceived by the ear.

Halliwell and Herbert point out that the general form of each component can be broadly categorised as shown in Fig. 1. The force spectrum starts at a level

$$F_1 \, \alpha \int_{-\tau}^{\tau} F \, dt$$

and has cornerfrequencies at $\frac{1}{T_1}$ and $\frac{1}{T_2}$ where $T_1 = F_1/F_{max}$ and $T_2 = F_{max}/\frac{dF}{dt}_{max.}$, thus dropping finally at 40 db/decade. The admittance curve rises sharply at the first resonant frequency. Coupling between the surface velocity and the air depends on the coincidence frequency - frequencies below this being ever more poorly radiated. The perceived radiation is further modified by the A-weighting curve and the net effect of adding all the logarithmic components is as indicated. Most of the radiated noise can thus be expected at the fundamental natural frequency and for a decade or so above this.

Fig. 2 shows the actual spectrum measured on a press under test in the laboratory. The coincidence frequency is about 1 kHz and the fundamental natural frequency is a low 200 Hz. In this case the A-weighting effect attenuates noise at the fundamental frequency and somewhat higher modes are the main contributors to the noise.

METHODS OF NOISE REDUCTION

It is apparent that noise reduction depends mainly on achieving a separation of the force spectrum from the structural admittance spectrum. A lowering of the force spectrum can be achieved by reducing the area cut on the item being blanked, but this of course says nothing useful in practice. Movement of the break frequencies to the left can however be achieved for a given item by applying shear to the tool since both F_{max} and $\frac{dF}{dt}_{max}$ will then be reduced. Reducing the speed of the press also has a desirable effect, since again the rate of change of the forces on the press are reduced.

In principle the admittance spectrum of the structure can be lowered in amplitude and moved to the right by improving the damping of the structure and increasing the natural frequency. But both approaches within the conventional layout offer little practical opportunity for improvement. Experiments [2] performed on the cast foot assembly of our press show that even when substantial efforts are made to increase the damping by effectively coupling the vibration to a sand filling, a reduction of about 6 db in L_{eq} is all that can in practice be achieved. Further the technique of sand damping could not realistically be applied to the entire structure. Stiffening the structure to increase the fundamental natural frequency has the problem not only of increasing the weight of the structure but also raises practical difficulties in that excessive structural stiffness increases the likelihood of crank shaft breakages and is unacceptable in general usage.

The consequence of all this is that while a variety of options appear to exist for noise reduction most are discounted for one reason or another - certainly within the constraints of a press which already exists. Work at Adelaide University is proceeding to examine methods by which the noise from an existing press can be reduced by altering the displacement/velocity characteristics of the press stroke : this is being done by inserting a hydraulic actuator at the ball joint between the connecting rod and the ram to achieve a modified displacement/time characteristic.
The question which arises, however, is how could the design of the traditional C-frame press be fundamentally rearranged so that its basic operating characteristics would remain but its noise would be substantially reduced ? The elements of such a new design are discussed below.

THE BASICS OF A LOW NOISE PRESS DESIGN

The principle of noise emission outlined earlier indicates that the following features are necessary in a low noise press, outside of tool modification such as shear :

(a) A reduction in cutting velocity while still retaining the stroke rate of the press.
(b) A major reduction in the strained surface area of the press so reducing S in the earlier equations.
(c) Minimisation of the strain in these parts.
(d) The most effective damping possible of these strained parts.
(e) The limitation of these areas so that a partial enclosure can be put round them without interfering with the use of the press.

Fig. 3 shows a schematic arrangement for a press which could incorporate these features. A departure is made from the conventional near simple-harmonic motion of the ram which is generally used in mechanical presses. Such a motion is incompatible with the need to provide a low rate of change of force on the press during cutting and penetration. The problem is that the velocity of the ram within the last few percent of the stroke length is too high. The toggle mechanism illustrated provides the appropriate dwell characteristic to achieve low velocities at the bottom of the stroke : other more complex mechanisms can be used if desired to produce even lower velocities if this is considered necessary. This approach has already produced considerable success in specific cases[3].

A particular feature of the configuration shown is that the stessed area of the press is greatly reduced compared with that of a conventional C-frame press since the main frame acts in a purely geometric role to support the slide for the ram driven by the moving C piece. It is true of course that the load still has to be transferred through the moving C, but, since this is a discrete item, it is possible to conceive of making it particularly stiff; of engineering extensive damping into the component; and also perhaps of putting the whole assembly behind an enclosure mounted on the unstressed frame. The fact that the noise radiation is basically confined to a specific component makes it possible to direct a number of noise reduction techniques at that item: techniques which it would be impracticable to apply diffusely over the entire structure of a conventional C-frame press.

The features incorporated in this design indicate the direction in which impulsive machine design must move if substantial noise reduction is to be achieved.

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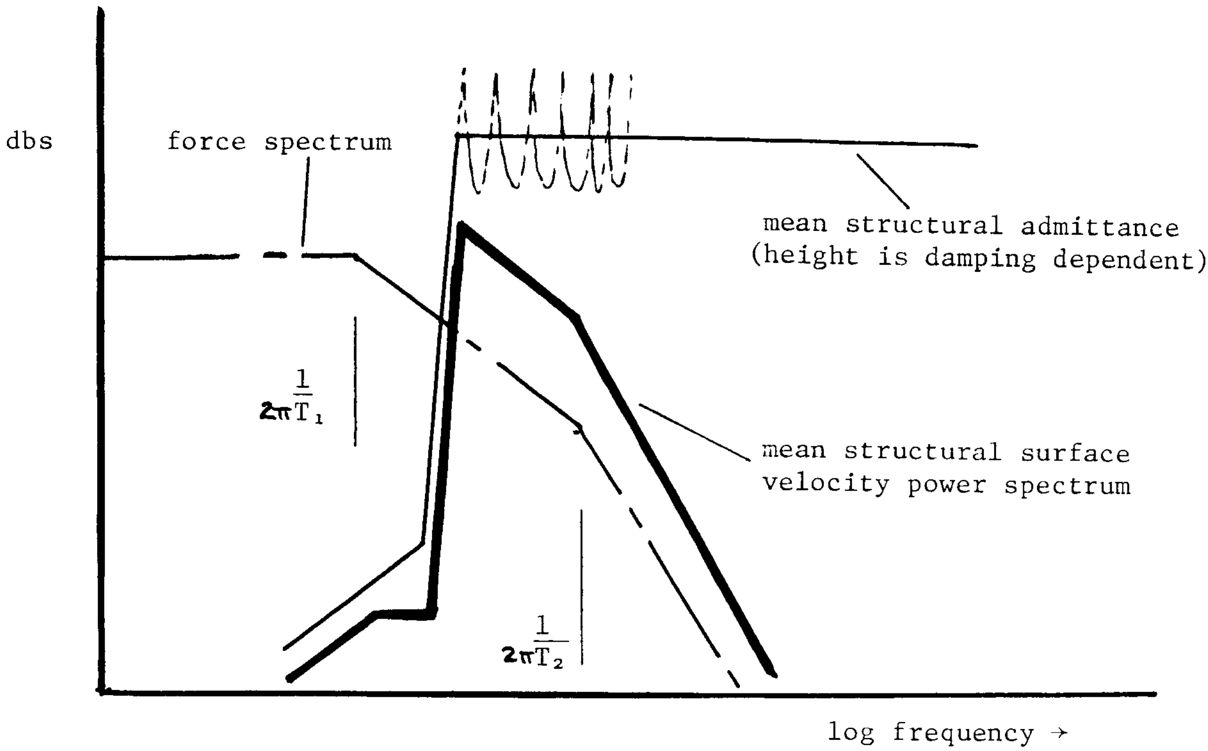

Fig. 1a. Form of velocity power spectrum derived from force and admittance spectra.

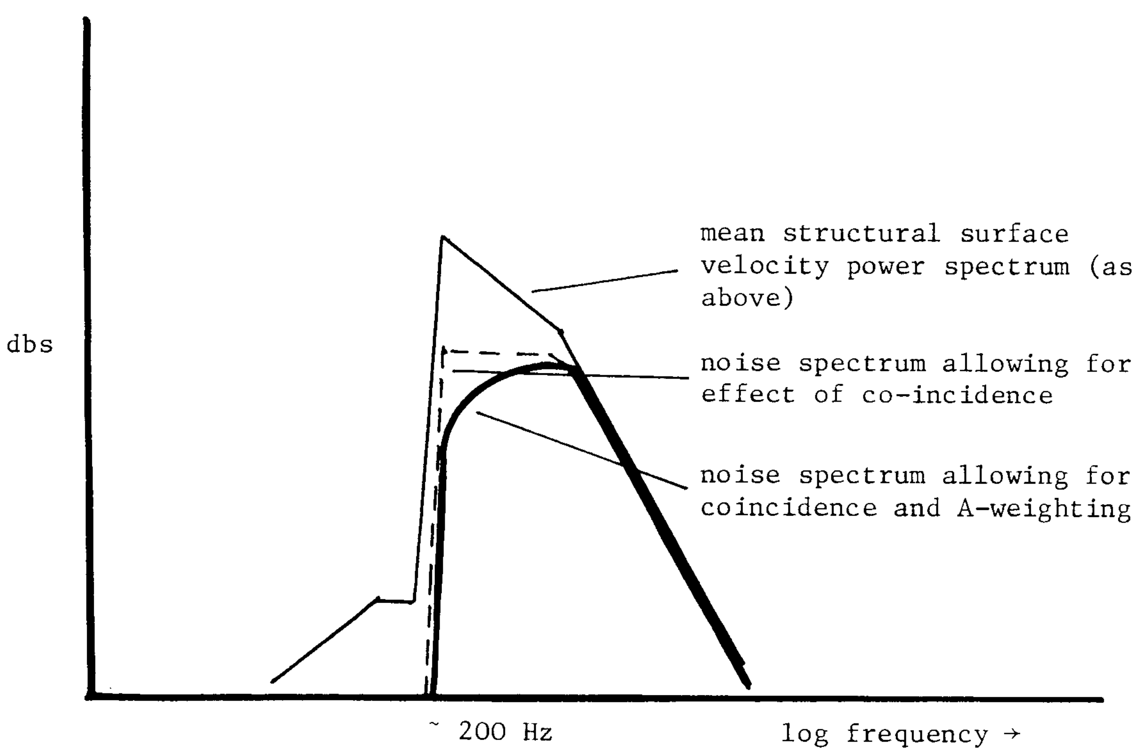

Fig. 1b. Form of noise power spectrum obtained by modification of velocity power spectrum.

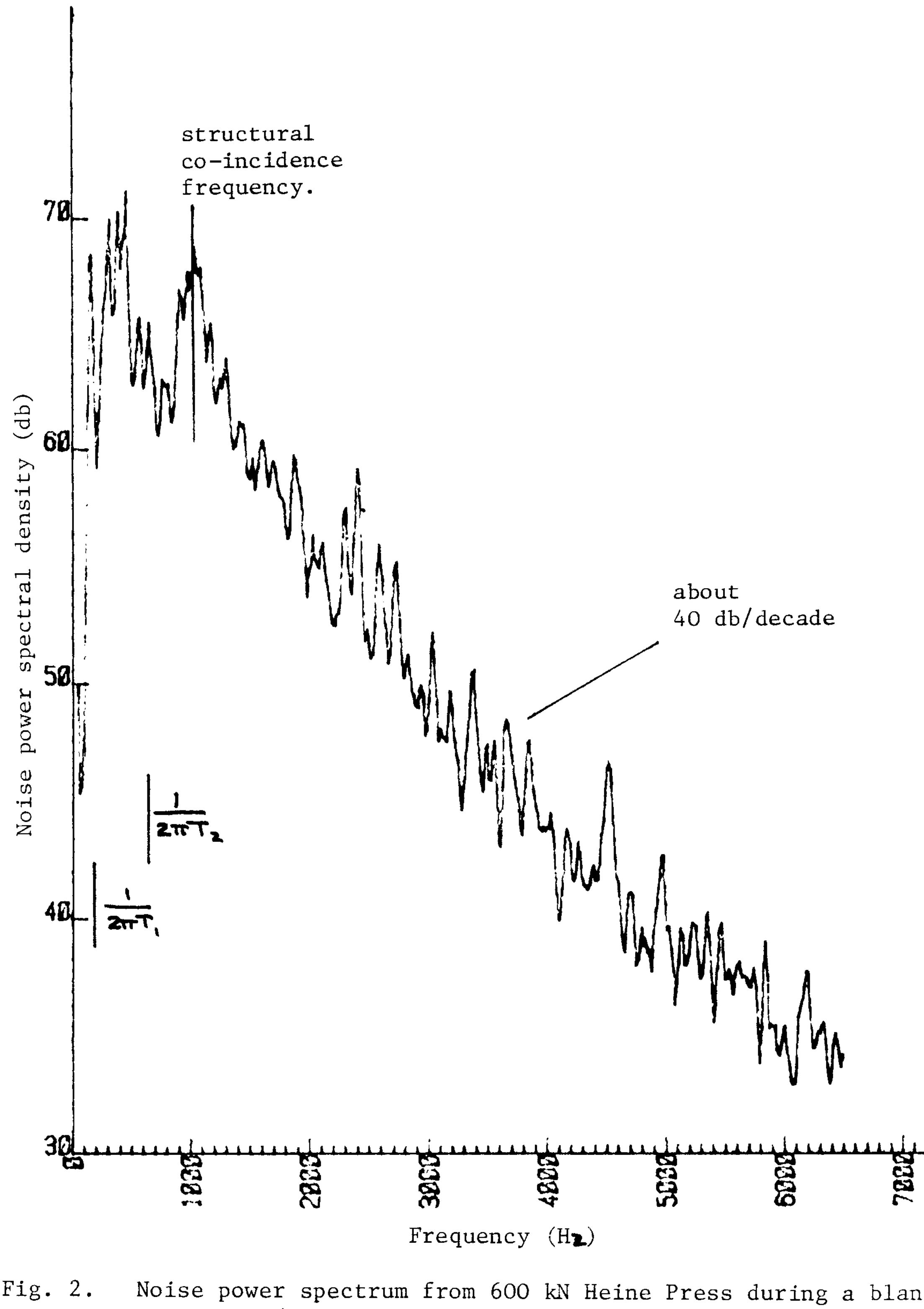

Fig. 2. Noise power spectrum from 600 kN Heine Press during a blanking operation (average of 8 impacts).

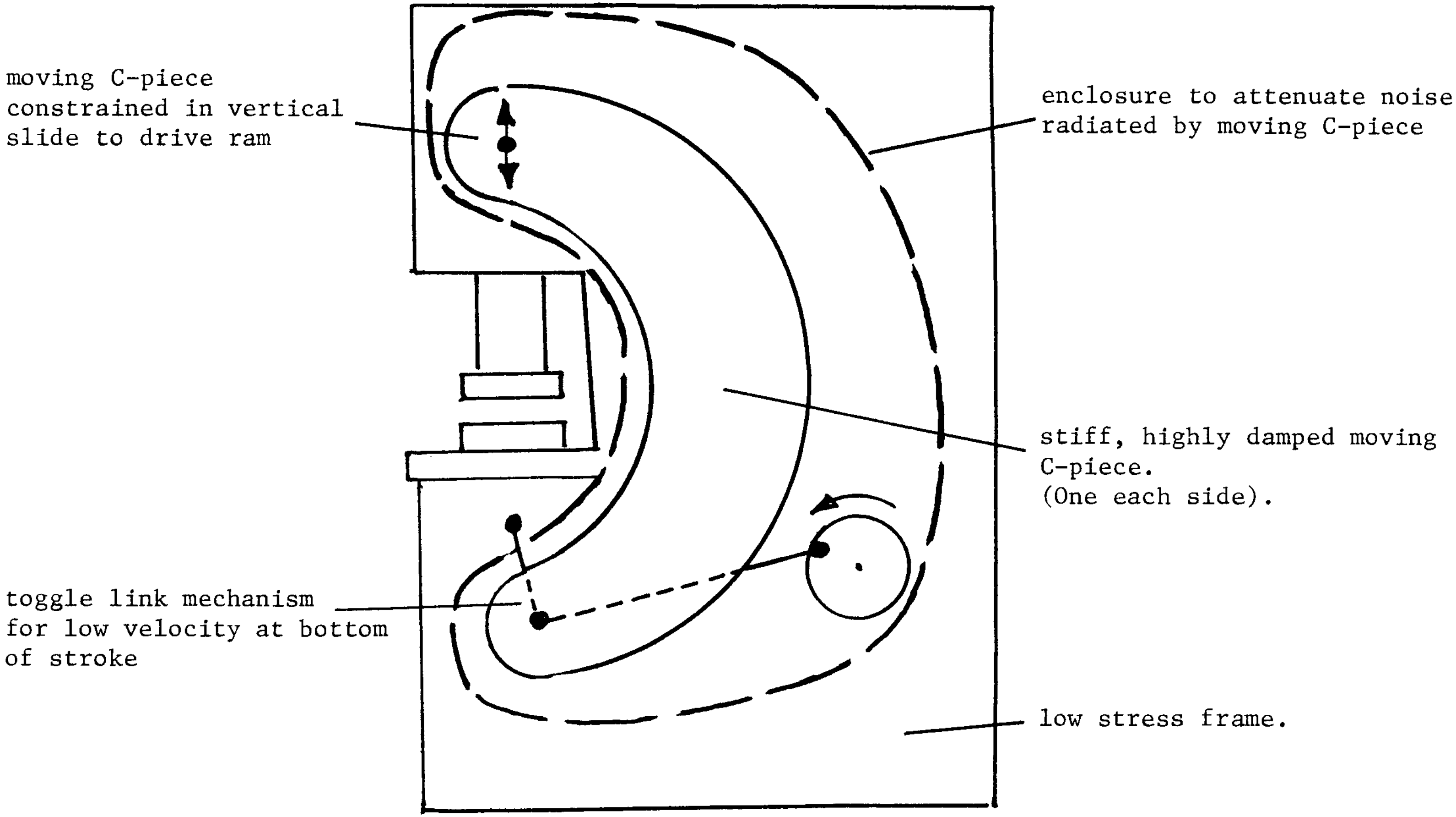

Fig. 3. Configuration of press for low noise emission from structure.

PERCEPTION AND MEASUREMENT OF MUSICAL TIMBRE

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Abstract

The perception of musical timbre involves many inter-related factors that have long defied description or measurement. Some of the factors involved will be discussed and a system of measurement described that attempts to take into account known properties of the hearing process. An important objective of such an acoustic measurement procedure is to extract characteristic features from the signal that are relevant to the functions of the auditory system. A new system of feature extraction will be described which may be applied to the analysis of musical sounds.

1. NATURE OF MUSICAL SOUNDS

Ever since the times of the early Greeks, there has been considerable curiosity concerning the nature of musical sounds and the identification of the factors that make them different from other sounds. Before real progress could be made, two major concepts were needed: frequency and spectrum. Arising out of his experiments on vibrating strings, Galileo (1564-1642) suggested that there was a connection between the perceived pitch of a musical note and the (present) concept of frequency. The first experimental measurement of frequency is generally attributed to Mersenne (1588-1648). The concept of frequency spectrum arose from the theoretical work of Fourier (1768-1830) and the development of spectrum analysis using a set of resonators by Helmholtz (1821-1894).

The perception of musical sounds is usually discussed in terms of four psychophysical characteristics: pitch, loudness, duration and timbre. The corresponding physical quantities are frequency, intensity, time and spectrum. However, as pointed out by Seashore (1938), these quantitites can apply to all sounds and do not enable musical sounds to be distinguished from a variety of other sounds. Musical sounds are combinations of these elements together with a microstructure consisting of deviations from regularity in all the elements. (Seashore discusses musical sounds in terms of four complex 'sensory capacities': tone quality, harmony, volume, rhythm). Thus, a musical sound may be regarded as a complex sound having a definable pitch and consisting of a set of related partial tones (not necessarily harmonic) which are not constant with time but may vary both in amplitude and frequency. The tone itself is usually accompanied by instrumental noises.

2. TIMBRE

From a psychophysical viewpoint, pitch, loudness and duration are one-dimensional quantities whereas timbre is a multidimensional entity, that is, a number of individual factors or dimensions are involved in the perception of timbre. In the past, numerous attempts have been made to systematise verbal descriptions of timbre. For instance, Helmholtz (1885) recognised that timbre was a measure independent of pitch and loudness and attempted to systematise descriptive attributes such as brightness, richness, sweetness, pleasantness, fullness and roughness. Bismarck (1974) studied a set of 30 verbal scales in relation to 35 sounds, consisting of spectrally shaped harmonic complex tones and noises. Two groups of subjects, some with and some without musical training, rated the sounds in terms of scales such as weak-strong, dull-sharp, smooth-rough, clean-dirty full-empty, pleasant-unpleasant, etc. The most significant attribute (accounting for 44% of the variance) was found to be sharpness (dull-sharp scale) which was identified as a measure of the centroid of the loudness spectrum of the sound. Other scales, representing the remaining portion of timbre not accounted for by sharpness, could not be clearly identified psychophysically.

Grey (1975, 1977) applied multidimensional analysis techniques to a set of 16 instrument tones equalised to the same pitch, loudness and duration (0.5 s). Perceptual similarities were determined for all pairs of stimuli formed from the tones. He found that three dimensions were sufficient to give a satisfactory representation of the data: (1) a dimension related to the spectral energy distribution, tentatively identified as the location of the centroid of the loudness distribution, that is, sharpness; (2) the presence of low-amplitude, high-frequency energy in the initial attack stage; (3) synchronism in the higher partials, that is, whether the higher partials rise in level together at the beginning of a note or fall together at the end. Grey found that the level of spectral fluctuation in the tone with time was closely related to the third factor.

3. MEASUREMENT PROCEDURE

A system of measurement is required for timbre that duplicates as far as possible the known characteristics of the ear in terms of time and frequency response. A set of filters is therefore required having frequency and masking characteristics matching the ear's critical bands. In the present study, a set of conventional 1/3 octave filters are used in conjunction with standard methods for loudness computation (ISO, 1966).

Recorded musical notes are played back through each filter in turn, the outputs being recorded on a digital waveform recorder with a sampling rate of 50 kHz. The sampled data is transferred to and stored in a micro-computer (Hewlett Packard type 9825A or Apple II). The signal values are now squared and averaged over one or more fundamental periods of the wave. For notes in the range C_4 to G_4 an rms value is found approximately every 3 ms. The rms values are converted into decibel measures and then further smoothed by fitting a high-order polynomial whose coefficients are then stored for use in later computation and graph plotting. This move also results in considerable saving of storage compared with that required for digitised signals. From the polynomial at 5 ms intervals, loudness values (in sones and phons) are computed using an algorithm based on the method of Stevens (1972).

Graphical displays include landscape plots of band loudness-frequency -time growth curves and the derivative of the sound level as a function of time obtained by differentiating the polynomial functions. The latter type of plot is valuable in studying which partial tones are likely to be dominant in gaining the attention of the ear at any particular time. Fig. 1 shows a landscape plot for the partial tones of a Gedackt C_4 organ pipe. The numbers on each curve refer to the partials in each 1/3 octave band used in the measurement. Fig. 2 shows a dominant slope diagram for the Gedackt C_4 organ pipe comprising segments of loudness derivative curves that are dominant at particular times. Superimposed are dominant loudness level curves taken from Fig. 1. It may be observed that the 7th partial is dominant between 0 and 10 ms, the 5th between 10 and 19 ms and the fundamental is dominant after 19 ms. There are also coincidences between the appearance of a dominant slope and the corresponding loudness level. A maximum positive slope value usually occurs about 10 ms before the onset of the corresponding steady state. It is therefore suggested that the ear uses the dominant slope signal as early warning for the appearance of a strong steady state partial tone.

4. SPECIFICATION OF TIMBRE

A simplified method for representing musical timbre has been developed (Pollard, 1980; Pollard and Jansson, 1980) that extracts three coordinates from the loudness data produced by the sampled filter method. The three coordinates are derived from (i) the loudness of the fundamental, (ii) the equivalent loudness of partials 2 to 4 treated as a group, (iii) the equivalent loudness of partials 5 and upwards. Thus, the total loudness, N, of the sound is

$$N = N_1 + N_2^4 + N_5^n \qquad (1)$$

where N_1 is the loudness of the fundamental in sones, N_2^4 is the equivalent loudness of partials 2 to 4, N_5^n is the equivalent loudness of partials 5 to n. Equivalent loudness is computed using Stevens (1972) equation:

$$N_i^n = 0.85\ N_{max} + 0.15 \sum_i^n N_i \qquad (2)$$

where N_i^n is the required equivalent loudness of the partials i to n, N_{max} is the value of the loudest partial in the group and $\sum_i^n N_i$ is the sum of the loudness values of all partials in the group.

Three coordinates may now be formed:

$$\begin{aligned} x &= N_5^n/N \\ y &= N_2^4/N \\ z &= N_1/N \end{aligned} \qquad (3)$$

from which a tristimulus diagram may be drawn using any two coordinates, e.g. x versus y, since x + y + z = 1. Sounds with a predominance of high frequency components will have large values of x, those with predominantly middle frequency components a large value of y, while those with a prominent fundamental will have a large value of z.

5. TRISTIMULUS DIAGRAM

Fig. 3 shows a tristimulus diagram for the following organ pipe sounds: (a) Gedackt 8' C_4, (b) Principal 8' C_4, and (c) Vox Humana 8' C_4. Times measured from the onset of the sound are shown beside each curve. The final circle in each case represents the steady state. It is immediately apparent how the spectral character of the sound changes between onset and steady state. The Gedackt C_4 has strong high frequency components initially and then progresses towards a predominantly fundamental type of tone. The Principal C_4 starts with some strong high partials and progresses towards an evenly balanced sound. The Vox Humana C_4 starts with a balanced distribution of middle and higher partials and then moves towards a more dominant set of higher partials. Fig. 4 shows a tristimulus diagram for the starting transients of notes from a viola, clarinet and trumpet displaying quite different paths. Data for the latter two sounds are taken from Moorer (1977).

6. CONCLUSIONS

The proposed acoustic tristimulus diagram has the following useful properties:

(a) each point in the diagram corresponds with the instantaneous spectrum of the sound. Contours on the diagram are related to spectral variations and reveal the variations in timbre with time.
(b) the diagram is independent of the total loudness of the sound and of the pitch of the note.
(c) spectral variations during the 'steady state' appear as an area on the diagram.
(d) the effect produced by the early appearance of high frequency components is clearly observed.

The tristimulus diagram offers a simple graphical presentation for following timbre changes during the starting transients of musical sounds. The method is able to reveal similarities and dissimilarities between notes from different musical instruments and is useful for making comparative studies of different notes from the same instrument.

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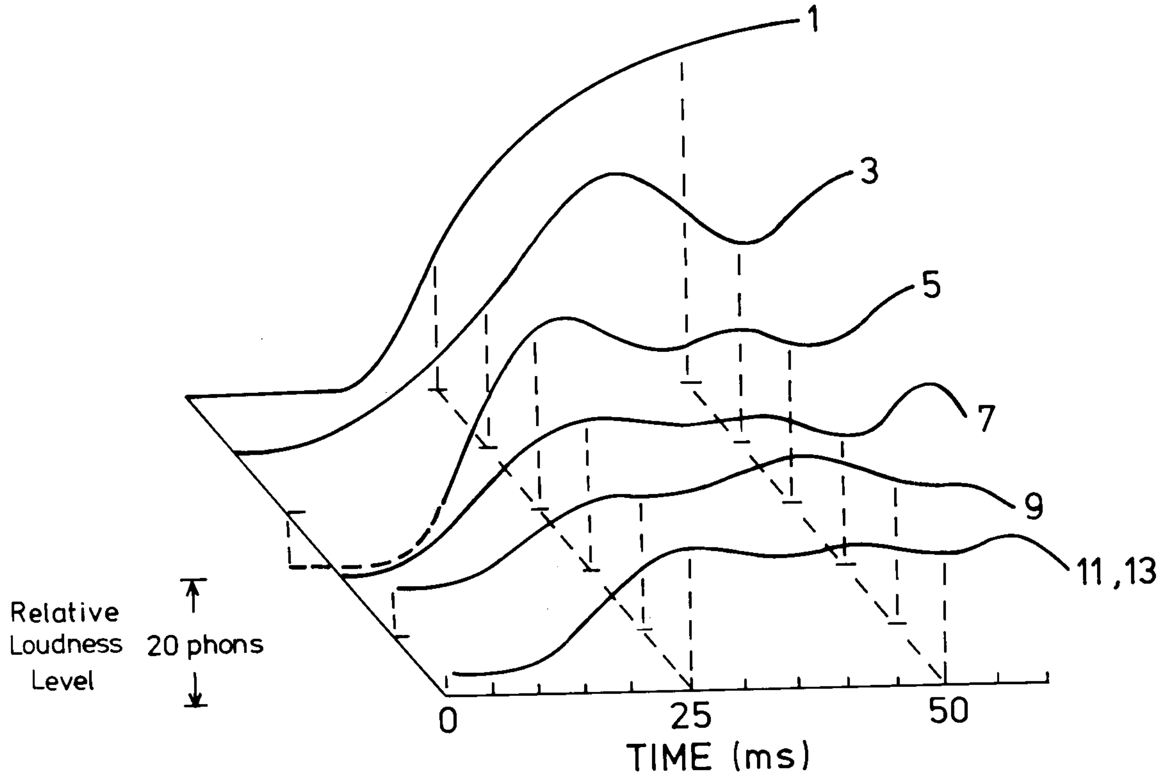

FIG. 1 Frequency - time - loudness level plot for the partial tones of a Gedackt C_4 organ pipe. Numbers on curves indicate partial tones in each filter band.

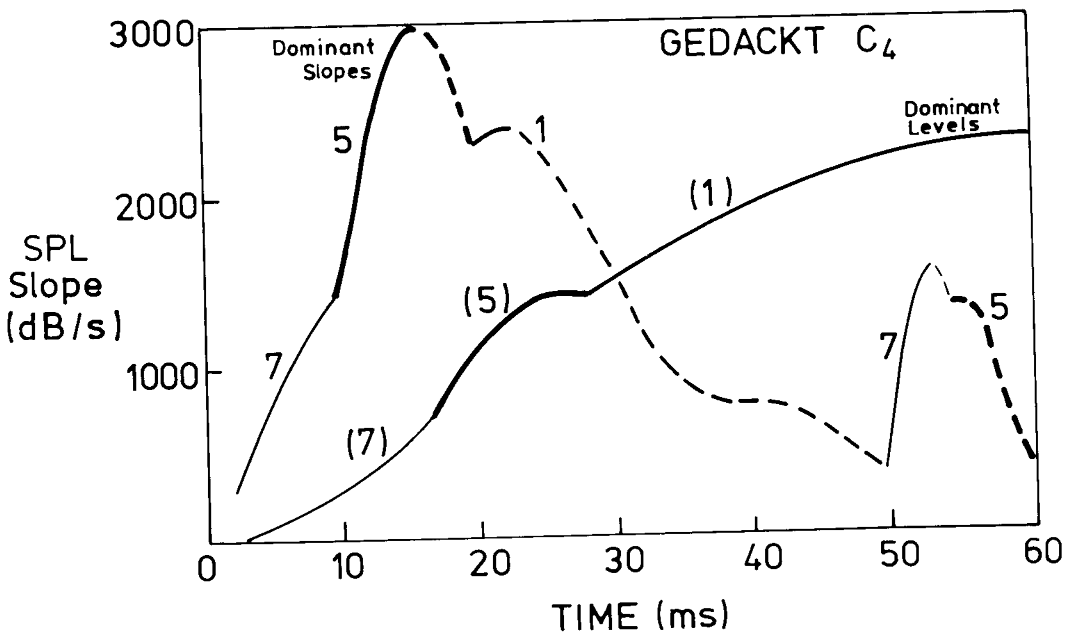

FIG. 2 Dominant slope diagram for a Gedackt C_4 organ pipe with superimposed dominant loudness levels. Numbers indicate partial tones.

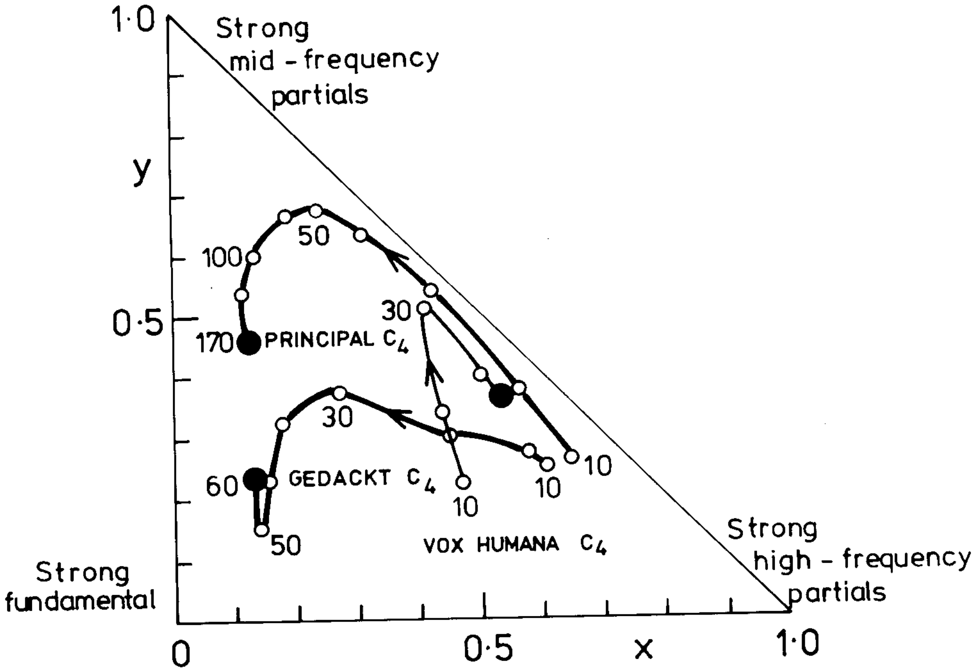

FIG. 3 Acoustic tristimulus diagram for 3 organ pipes. Numbers indicate time in ms after onset of sound. Large circles correspond to steady state.

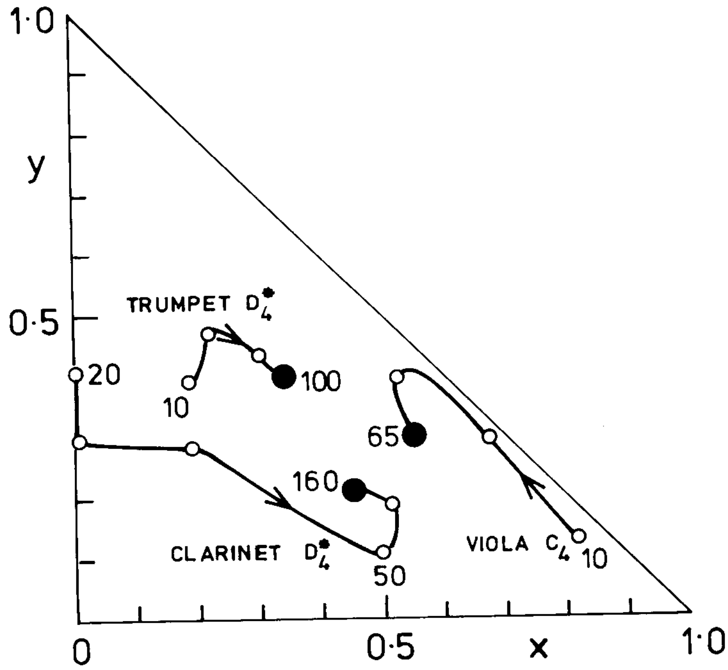

FIG. 4 Acoustic tristimulus diagram for viola, clarinet and trumpet notes.

PERCEPTION OF SOUND IN REVERBERANT FIELDS

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SUMMARY

In industrial buildings the addition of sound absorbing material appears to have a far greater subjective effect than the reduction of sound level would suggest. The magnitude of this effect has been investigated in a workshop and by a series of laboratory experiments. The localization of sound sources in reverberant spaces has also been investigated. It seems that the reduction of loudness due to the addition of sound absorbing material depends on the type of sound and the age of the respondent. Accurate localization of sounds depends on the reverberation time and orientation of the subject.

1. INTRODUCTION

There is an amazing lack of information on the way people perceive sounds in non-anechoic spaces. In this paper the results of some field and laboratory studies on the perception of sound in spaces where free-fields do not exist are reported on.

Information on the loudness of sounds as a function of reverberation time would seem to be as important, if not more important, than the effect of reverberation time on the intensity of sound in rooms. There are some vague statements, in the literature, about the importance of sound absorbing treatment on the loudness of sounds. For instance, Kuttruff(1) when discussing the reduction of noise by acoustical treatment states that " .... a noise level reduction of only a few decibels can increase the acoustical comfort to an amazing degree." Embleton et al (2) suggest that, "in rooms with very good acoustical absorption", the acceptable noise criterion can be raised by 5 phons.

Likewise there appears to be little quantitative or qualitative information on the localization of sound sources in non-anechoic spaces and no information on the perception of the distance of a source. This lack of information is surprising because such information would seem to be of fundamental importance in perception studies as well as of practical importance in indentifying sound sources and sound paths in buildings. Embleton (2) also points out that, "if one is not able to locate directly and subconsciously the position of the sources then one finds them considerably more annoying".

2. LOUDNESS PERCEPTION

2.1 Field Study

The opportunity arose to measure and assess the change in acoustic conditions in a workshop when a new roof, incorporating of sound absorbing ceiling, was put on the building. The measured reverberation times indicated that the sound levels in the reverberant field should be approximately 3 dB less, when the new roof was installed, than they were with the old roof. The subjectively assessed improvement in conditions, as assessed by employees, was much more remarkable (see Fig. 1).

It is suggested that the responses of employees were not unduly influenced by a placebo effect as the employees were very critical of the change in lighting levels in the workshop. (The original roof had skylights while the new roof did not.) The results indicate that the addition of sound absorbing material has an important effect on the perceived noise level in a room. However the reason of this and the reason why the older employees perceive a greater improvement than the younger employees is unknown.

2.2 Laboratory Study

Four different types of sounds were recorded in reverberant and anechoic conditions. The four sounds were music, white noise, road traffic noise and impulse noise. The levels of these four sounds were modified so that the loudness of pairs of sounds with different intensities could be compared by subjects. The pairs of sounds were arranged so that only like sounds were compared, eg. music with music. One of each pair was recorded in an anechoic room and the other in a reverberant room.

The recordings were made using a dummy head. The impulse noise was recorded directly using a tapping machine as a source. The white noise was recorded from a loudspeaker fed by a signal from a white noise generator while the music and traffic noise were prerecorded and played back through the same loudspeaker used for the white noise. The music was from a BBC tape recorded in anechoic conditions.

Two test tapes were prepared. One contained impulse sounds only, each sound lasting for approximately 15 seconds. The levels of the 'anechoic' and 'reverberant' signals were altered so that the levels varied by ±10 dB, in 5 dB steps, about a mean 'impulse' level of approximately 75 dB at the subject's ear. The sequence and levels of each of the paired sounds was then ramdomised.

The second tape consisted of music, white noise and traffic noise. The tape was prepared in the same way as the impulse sound tape except that L_{eqs} were used instead of impulse levels and the levels were varied by ±5 dB instead of ±10 dB.

The results of the laboratory studies are shown in Fig. 2. They show that by adding absorption the average perceived loudness can be reduced by between [illegible] and 8 dB(A). A significant number of subjects perceived the 'reverberant' sound to be 10dB(A) or more louder than the 'anechoic' sound with the same impulse level, for impulse sound. For music, white noise and traffic noise a significant number of respondents perceived the 'reverberant' sound to be 5 dB(A) or more louder than an 'anechoic' sound with the same L_{eq}.

It is interesting to note that the stereo recordings were perceived different to the mono recordings and that the results for music in stereo are not monotonic.

3. DISTANCE PERCEPTION

It is known that in anechoic conditions subjects can accurately determine the direction of a sound source. Data on distance perception does not seem to exist and yet such information would seem to be important in fundamental studies of aural perception and in practical situations where the importance of noise sources have to be subjectively assessed, where a visually handicapped person has to make judgements of distance on aural signals and in concert halls where visual and aural information should not be contradictory if reality is to be maintained.

3.1 Laboratory Study 1

Subjects listening to the 'loudness' tapes described in the previous section were also asked to make judgements about which of each pair of sounds was closer. In each case the sound was recorded 2m from the source so that there was no actual difference in distance.

In the case of the impulse source it seems that most subjects base their judgement of distance on reflected sound although there is an almost constant percentage (approximately 20%) who appear to judge distance by loudness. (If the sound is louder it is judged to be closer.)

With white noise and traffic noise there were a significant number of subjects who perceived the reverberant sound to be closer but for these two sound sources there is a well defined trend to judge the more intense sound as being closer. This trend is least pronounced for the music where the difference between mono and stereo recordings is also largest. The results are presented in Fig. 3.

3.2 Laboratory Study 2

In the second study blindfolded subjects were placed in reverberant and anechoic rooms with two identical loudspeakers placed in line with them but at different distances from them. (The distances of the loudspeakers were known to the subjects.) Pure tones and octave-band filtered white noise were used as the source signals. The signal levels were randomised.

Briefly, the results are as follows for pure tones:

i. In both the anechoic and reverberant rooms the correct source was not accurately determined at the three frequencies used (2, 4 & 8 kHz) but the correct source was identified more consistently at the higher frequencies.

ii. In both the anechoic and reverberant rooms the correct source could be identified better when the subject faced 90° away from the sources. When the subject faced the sources the number of correct identifications could be explained by chance.

iii. For the 8 kHz signal with subjects facing 90° away from the sources the correct source was identified 88% of the time in the reverberant room and 73% of the time in the anechoic room.

These results suggest that the distance of a source could be determined by differences in sound level or phase between the two ears and that reflected sound is also important in determining distance.

4. LOCALIZATION

4.1 Localization Study 1

Experiments were carried out to determine how well subjects could localize sound sources when the sources were located outside the furnished room in which the subjects were seated. Subjects were seated in a normally furnished room. Loudspeakers were located at the trangled positions (1, 2, 3 & 4) indicated in Fig. 4. Loudspeaker 1 was located outside the building, loudspeaker 2 was in a neighbouring furnished room, loudspeaker 3 was in a corridor and loudspeaker 4 was in a reverberant room. The main transmission paths appeared to be at D (window mounted airconditioning unit) for loudspeaker 1, at A (a door) and D for loudspeaker 2, at A for loudspeaker 3 and at C (a vertical crack) for loudspeaker 4.

The experiment was somewhat artificial in that subjects were asked not to move their position and the signals used were pure tones at 1000 Hz and 3000 Hz with a duration of 2 seconds. The signal level at the subject was 60 dB(A) in each case. The background noise level was approximately 40 dB(A).

The results are shown in Fig. 4. They show that subjects can often determine where sound is entering a room when the path is reasonably well defined. (The ringed numbers indicate the perceived direction of the source. The percentage following the ringed number indicates the percentage of subjects who perceive the sound coming from that direction.) However, there is a considerable spread in the results. For instance, in the case of sound source 2 for 3000 Hz signal, a significant number of subjects perceived sound coming from direction A, 2, D and 1. A later objective test indicated that the main transmission path in this case was via D, and the second most important path was via A. It is interesting to note that with the 1000 Hz signal both these paths (and only these paths) were identified. Subjective determination of sound paths was not tried with more diffuse sound paths which it is thought would be more difficult to localize.

4.2 Localization Study 2

In the second localization study subjects were placed in a reverberant room in which the reverberation time could be altered by the addition of absorbing material. The subjects, who were blindfolded, were asked to identify the direction of the source of sound. The source was a loudspeaker and the signals used were tone bursts (1000 Hz and 3000 Hz) with a duration of 2 seconds.

The results indicate that the accuracy of localization decreased as the room reverberation time increased. Ability to localize the 1000 Hz signal was better than the ability to localize the 3000 Hz signal except when the reverberation time was short. Here both frequencies were equally well localized. Accuracy of localization also depended on the subject's orientation relative to the source; subjects can identify the direction of the source better when they face the source. The results are summarised in Fig. 5.

4.3 Localization Study 3

A third localization study was undertaken to investigate precedence effects. A distributed source (9 loudspeakers) was used with a variable delay line incorporated into one of the loudspeakers. Subjects were asked to identify the direction of the delayed loudspeaker.

The results of this study are difficult to present suscintly as many different sound level differences (5, 10, 15 & 20 dB), delay times (10, 15 & 20 ms) and reverberation times (0.5, 1 & 2 s) were used. The signals used were 1/3 octave bands of white noise centred on 1000 Hz and 3150 Hz. The main point to come out of this study was that it is the signal level difference rather than the delay time that is important in assessing the direction of the delay signal. This point is important in the identification of sound transmission paths in buildings.

5. CONCLUSION

Some experiments, concerning the perception of sound in buildings, have been described the results of which indicate that there is more to the subject than meets the sound level meter.

The main conclusions to be drawn from the work described (other work is also being undertaken on the perception of pitch changes in rooms and the effect of the visual field on aural perceprion) are:

i. The addition of absorbing material to a room has a greater effect on the perceived noise level than the reduction in noise level would indicate. This is especially true in the case of impulse sounds where the average young subject perceives an 8 dB(A) reduction in the sound level when reverberant conditions are changed to anechoic, but the impulse level in unchanged. Older subjects perceive an even greater change.

ii. People are able to make judgements on the relative distance of sound sources. The mechanism used for these judgements is not known but it appears that impulse sources judgements are based on the 'reverberation' present whereas for other sources used the judgements seems to be based mainly on 'loudness'. The distance of a source is judged better when the person faces 90° away from the source than when he faces the source.

& iii. Localization of sound sources in non-anechoic environments is not as relative as in anechoic environments. The ability to localize is dependent on reverberation time and the number of sources present. Short reverberation times are more conducive to accurate localization than long reverberation times. When there is more than one source localization depends on the relative level of the sources. Localization ability does not seem to depend on the delay time between sources for delay times up to 20 ms when one source (to be localized) is delayed relative to other sources nearby.

The present work despite its many limitations has shown a number of generally held beliefs about the perception of sound in buildings cannot be justified. In particular the above comment applies to the perception of distance and the effect of absorption on loudness perception. Work in this area is continuing.

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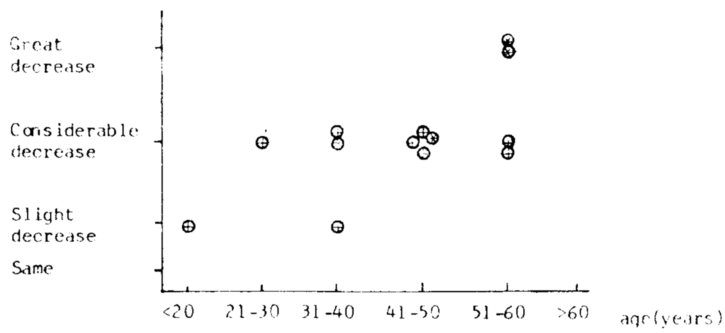

Fig. 1a Noise change with new roof as judged by employees in a workshop

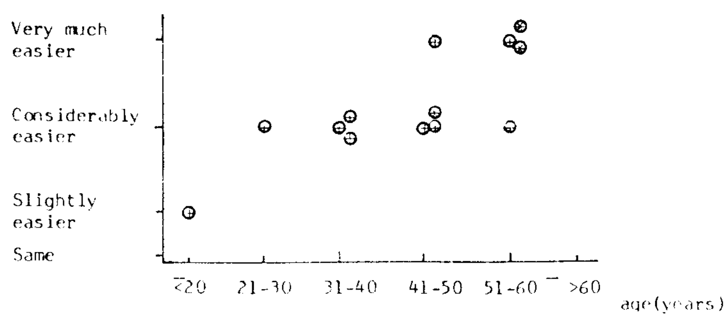

Fig. 1b Change of ease of conversation with new roof as judged by employees in a workshop.

⊖ Not known hearing disability ⊙ Known hearing disability

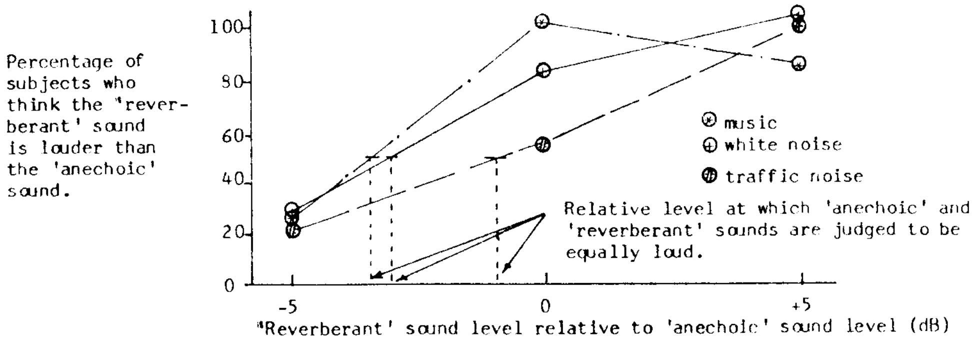

Fig. 2a Perceived loudness of 'reverberant' and 'anechoic' sounds using a dummy head and quasi-steady sources.

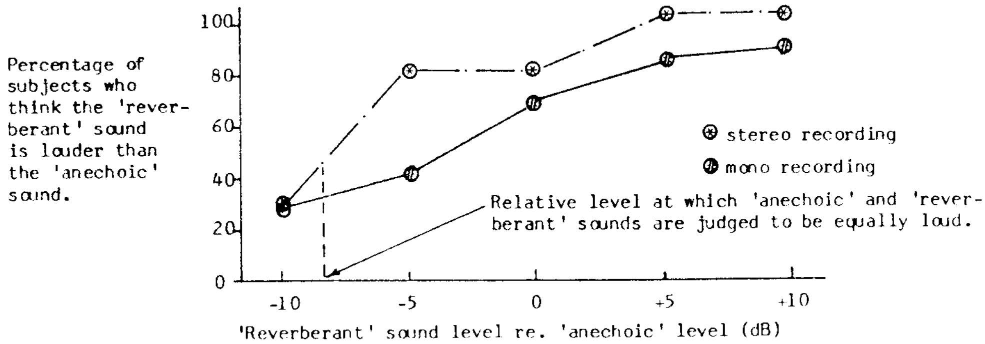

Fig. 2b Perceived loudness of 'reverberant' and 'anechoic' sounds using a dummy head and an impulse source.

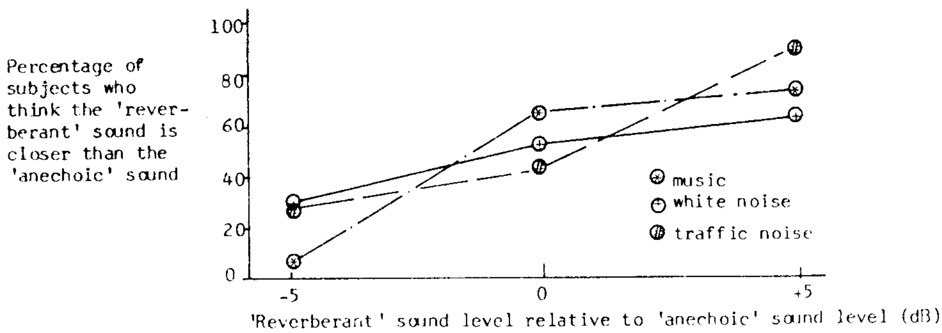

Fig. 3a Perceived distance of 'reverberant' and 'anechoic' sounds using a dummy head and quasi-steady sources.

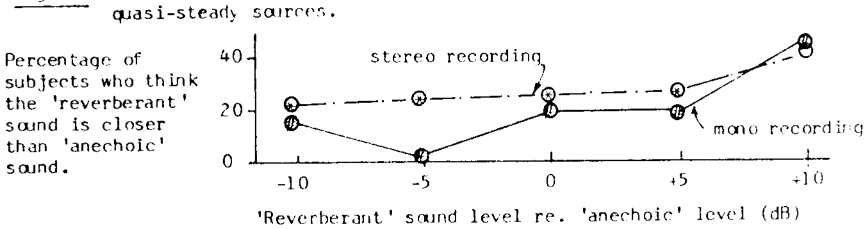

Fig. 3b Perceived distance of 'reverberant' and 'anechoic' sounds using a dummy head and an impulse source.

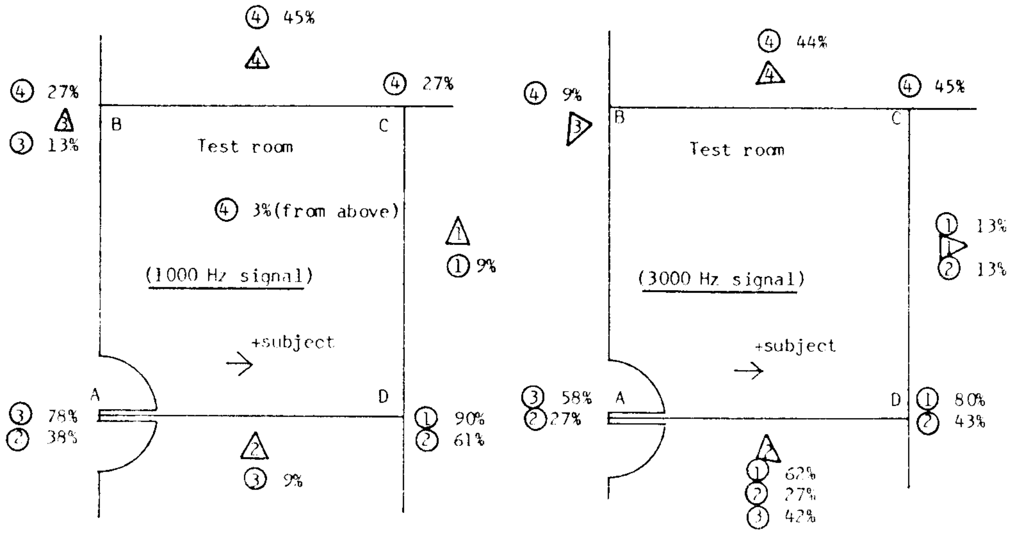

Fig. 4 Localization of sound paths in a room.

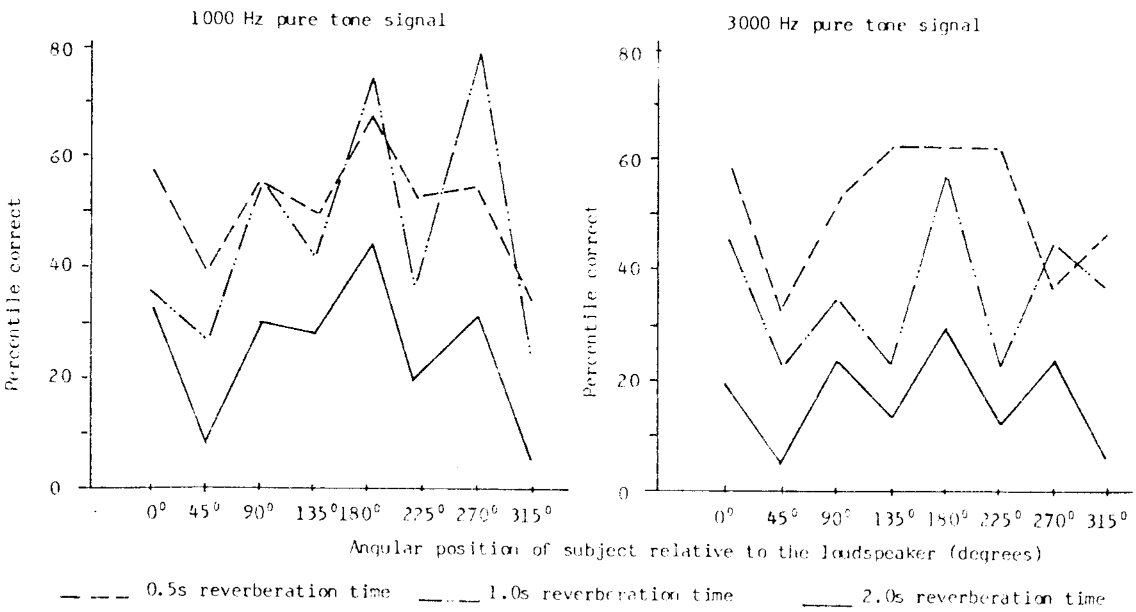

Fig. 5 Comparison of localization results for three different room conditions.

WHAT IS THE PURE TONE AUDIOGRAM REALLY TELLING US?

Dr R.G. Hicks*

Many of us have asked: "what is a pure tone audiogram really telling us?" Some of the assumptions underlying such testing are; 1) the use of pure tones is in some manner associated with real life situations, and ii) the sensory threshold, an all or none concept, is also related to how an individual functions in life. These assumptions are obviously erroneous, but we still use such audiograms as "measurement" devices.

The central concern of hearing loss is the effect of that loss on the individual in his day to day living. This bares little resemblance to listening to pure tones in a silent booth or even listening for speech discrimination in a silent booth. How often does the average individual, in real life, get the chance to listen to pure tones, let alone in silence. Most of us are lucky to find modest silence in which to sleep. Our real life situation is filled with competing sounds that we must separate and analyse. The person with a hearing loss has increasing difficulty performing this discriminatory - analytic task as their hearing deficit increases. In other words, it is the signal-to-noise ratio that becomes increasingly larger with the hearing loss.

What is a pure tone audiogram really telling us? During the early part of 1981 we commenced a study which endeavoured to answer this question by asking another question, "Is it possible to predict an individual's hearing loss on the basis of a paper and pencil test concerning his coping in the world?" To achieve this objective a questionnaire was administered and the information thus obtained was compared with pure tone audiogram data.

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*This paper is the sole responsibility of the author, who would like to thank Dr J.C. McNulty, Commissioner of Public Health, and Dr F. Heyworth, Director of Occupational Health, Clean Air and Noise Abatement for allowing him to present this paper. Thanks also to Barras Brooks for the computer analysis.

The "paper and pencil test" comprised the two most commonly used device available. The questionnaires by Nobel (1978) and Giolas (1970). These measures were not used in their "pure" form although all items from both questionnaires were included. The items were re-organised into forms considered appropriate for a single questionnaire. The ordering was varied to correct for fatigue and other effects. The resultant questionnaire was administered to an initial sample of seventeen (17) men who were also tested for noise induced hearing loss (NIHL). This information was compared with the questionnaire data for the purpose of identifying the questions with the highest degree of association with the pure tone audiogram, the question clusters, and the question groups that are statistically distinct as possible, in other words those would which would weight and align the discriminating variables.

On the basis of this discrimination analysis a revised questionnaire comprising thirty-one (31) questions was devised (Appendix I). This was administered to a second sample of seventy-seven (77) trade lecturers at a Perth technical college. In addition to the questionnaire, data were collected on; i) pure tone audiograms, ii) blood pressure and heart rate as indicators of the autonomic nervous system (Kryter, 1976), iii) eye colour (recently reported as being associated with Noise Induced Hearing Loss) and iv) the usual types of data collected with audiograms; type of work, length of employment, date of birth, sex etc. Two important questions were overlooked on the questionnaire, both in the trial and in the preparation. Firstly, does the subject engage in vigorous exercise at least 3 times a week and secondly, what, (if any) medication is the subject using and if so, in what dosage? On site, it became apparent in testing the first three or four subjects that this data was important and they consequently were collected from all subjects including those tested prior to the realization of the ommission.

Results

The results of the study indicate that heart rate is not related to noise induced hearing loss. On the other hand, diastolic blood pressure has the highest degree of association with NIHL (Pearson's R of 0.38385, significance at 0.0011 level of confidence).

Systolic blood pressure is also associated with NIHL (Pearson's R of 0.20825 significance at 0.05 level of confidence). These figures are related to pure tone hearing losses at 4, 6, 8 kHz and when the clients are not on medication. When all clients' data are pooled, the correlation decreases but still remains significant. For frequencies 1, 2 and 3 kHz, only diastolic blood pressure show a significant (0.0258) correlation (0.23353) with NIHL. Systolic blood pressure become significant when only non medication clients' data are analysed. Age and hearing loss are related at the 0.00001 level of confidence (Pearson's R). Eye colour was related to NIHL at the 0.07 level of confidence.

Discussion

Age and hearing loss are always associated. This effect should not be rationalised as an ageing effect - rather as an exposure effect differing according to the exposure to noise an individual experiences over a lifetime. (Hicks, 1980).

At the commencement of this study I was extremely skeptical of the relationship between eye colour and NIHL. Our data confirm the association of 4,6, 8 kHz but as with all correlations, it is impossible to state the cause of the relationship.

Of course, the resulting questionnaire data present a more complicated picture. On the basis of the discriminatory analysis the questions can be clustered into three major groupings: i) those questions that relate to a general hearing deficit, ii) those questions that relate to a hearing loss at 4, 6 and 8 kHz, and iii) those questions that relate to a hearing loss at 1, 2 and 3 kHz. None of the questions are mutually exclusive but it should be noted that particular questions are more closely associated with different frequency losses.

Questionnaires of this type are not the be-all or end-all and they do have limited application; ie, the client must not only be literate in English but also motivated to complete the questionnaire accurately. Nevertheless questionnaires of this type, with established reliability, and validity, do have a place in a hearing conservation programme as they provide another "number" to an audiologist's repertoire.

The questionnaire compiled on this study provides some broad generalizations. Firstly the OOAA and the NAL have assumed in their formulae for hearing loss that frequencies of 4, 6 and 8 kHz play a non-existance role in the clients everyday life. Our results do not vindicate this assumption. The frequencies of 4, 6 and 8 kHz are not a "no mans land" that can easily be abandoned to the effects of noise. Quite the reverse. The "simplistic" formulae that exist for hearing loss seem to bear little or no resemblance to a real life situation. Such formulae, born from pure pure tone audiometry and beloved by a legal millstream, are zealously maintained inspite of their miscarried birth.

Conclusion

What is the pure tone audiogram really telling us? It may tell us that a client's blood pressure is high or what colour his eyes are. It could also tell us that our client is having difficulty with his human interactions and has changed his life-style to accommodate his deficity. However the pure tone audiogram still remains; 1) a heuristic misfit, 2) an over used, misunderstood expedience, and 3) it fails to interface to the real world.

Finally, I still don't know what the pure tone audiogram is telling us and I suspect that in isolation, it is telling us nothing as the relationships between the pure tone audiogram and real life situation still remain a mystery. The pure tone audiogram does not allow us to prescribe or determine causal factors. In an attempt to specify the physical parameters of the signal and to simplify the testing procedure a test has been created with little or no heuristic value.

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IDENTIFICATION OF FLANKING PATHS IN BUILDINGS

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and
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1.1 INTRODUCTION

The aim of anyone associated with designing or maintaining a building must be to achieve a "comfortable" environment. A significant parameter affecting this "comfort" is the attainment of a certain level of noise privacy. If adequate noise privacy does not exist then the inhabitants may have difficulty communicating, not be able to sleep or suffer increased stress levels which cause increased anxiety and irritation. The attainment of adequate noise privacy is more particularly a problem in multi-dwelling units because of the closer proximity of neighbours.

In cases where satisfactory noise privacy does not occur one would suspect the existence of flanking paths, i.e. sound paths other than the direct path between source and receiver through the separating partition. These flanking paths may be:

(i) airborne paths through cracks and poor seals
(ii) structure-borne paths where sound is transmitted through the structure
(iii) transmission through areas with a lower transmission loss value than the separating partition.

According to a BRE field survey[1] conducted between 1971 and 1976, 55% of the 1270 wall partitions tested failed to meet the requirements of the noise regulations. The worst failures were due to bad detailing or poor workmanship causing cracks in the partitions or producing inadequate sealing. The most common fault was due to the use of a lightweight ceiling material which allowed the sound to easily reach the receiving room via the ceiling space above.

It is the purpose of this paper to review existing methods of identifying flanking paths and to propose a method which is simple to carry out with a minimum of equipment.

2.1 METHOD OF IDENTIFICATION DESCRIBED IN THE STANDARDS

ASTM E336-71[2] describes a series of identification methods.

2.1.1 The Ear

Initially the standard recommends that the room be surveyed by walking around and listening so that a subjective assessment of possible suspect areas may be made. Although very obvious sound leaks would be identified using the ear, Lim[3] has shown that human localisation of sound paths is very unreliable.

2.1.2 The Stethoscope

As an extension of the human ear the standard recommends the use of a stethoscope for locating flanking paths.

The main problems associated with the use of the stethoscope are:

(i) pressure doubling effects occur near edges and corners which increase the sound levels without any flanking paths occurring;
(ii) it would be extremely inconvenient to survey a whole partition with a stethoscope;

(iii) no quantitive information may be obtained. This particular problem may be overcome by replacing the stethoscope with a microphone and sound level meter. The protection grid of the microphone would be placed on the partition and the sound level noted from the sound level meter.

This method, using a microphone and sound level meter, was investigated and increases of at least 12dB were measured when the microphone was placed over a non-obvious crack in the test partition (the crack was a poor joint between a brick partition and a concrete column). Thus, the method may be useful for assessing suspected airborne sound paths but would be unsuitable for locating unsuspected cracks.

2.1.3 The Accelerometer

To locate structure-borne flanking, the use of an accelerometer is recommended. If the average vibration level (usually measured as a velocity level) of the dividing partition is at least 10 dB more than the average vibration level of other surfaces then the contribution of the other surfaces to the sound level in the receiving room is negligible.

Tests were carried out on a timber model. The model simulated a source/ receiver room situation with a 15 mm thick plywood panel used as the separating partition. To minimise mechanical coupling the model was cut through adjacent to the partition and joined with plasticine. Accelerometer measurements were taken on the partition as well as on the surrounding walls and the floor and ceiling of the receiver room. A comparison was made with the model joined with the plasticine and with the two sections of the model rigidly connected with two large clamps. It was found that the velocity level of the surrounding surfaces increased by an average 12.75 dB per $^{1}/_{3}$ octave when the model was rigidly connected.

This method was investigated by Cunliffe[4] under field conditions. He found that there was a dip in the transmission loss of a wall with woodwool shuttering used as the ceiling material compared to the same wall with a plasterboard ceiling above. Accelerometer tests of the two walls showed that in the frequency range where the dip occurred, the woodwool shuttering radiated considerably more sound than the party wall.

2.1.4 Space Insulation

For a comprehensive identification of flanking paths ASTM E336-71 recommends the erection of a temporary room, in effect, within the receiving room. By removing small sections of the temporary room and remeasuring the transmission loss the contribution of the particular uncovered section of the receiving room can be determined. If the transmission loss reduces considerably, then it may be presumed that the flanking path is in the area of the room where the section of the temporary shield has been removed.

As may be readily appreciated this method is cumbersome and time consuming. It requires constructing a complete room within the test room which must, itself, be free from leaks and cracks. To accurately identify the flanking paths many transmission loss tests must be made.

3.1 METHODS DESCRIBED IN OTHER SOURCES

Two other methods of identifying flanking paths are found in the literature and are based on the same principle. The principle is that the important sound paths can be identified by the time taken for the sound to travel from the source to the receiving room microphone. In a simple case there may be two significant paths; through a test partition and through an obvious flanking path. The sound travelling directly through the partition will arrive at the receiving room microphone before the sound travelling via the flanking path.

3.1.1 Impulse Method

This method, described by Raes[5], was intended to measure the transmission loss in the presence of flanking paths but may be adapted to actually identify the flanking paths.

A short duration source such as a pistol shot is used. One microphone is placed close to the source and another in the receiving room. The microphone close to the source is used as a trigger for the receiving microphone. The signal from the receiving microphone is taken to a storage oscilloscope.

If there is a flanking path then the oscilloscope trace should show two obvious amplitude changes. The first should occur when the sound reaches the receiving microphone through the testpartition and the second should occur when the sound via the longer flanking path reaches the receiving microphone. This allows the calculation of the path length difference between the direct and the flanking path and thus the location of the position causing the flanking.

The use of the impulse method was investigated for identifying panels with relatively low transmission loss characteristics. A doorway in brick partition was boarded up with a sheet of 15 mm plywood, and the edges of the plywood were carefully sealed. The amplitude change due to the direct path was not obvious in the trace but the amplitude change due to the path through the plywood was certainly obvious.

3.1.2 Cross Correlation Method

This method is described by Goff[6] and Price et al[7]. It is based on the same principle as the impulse method but a continuous source may be used. Again a microphone is placed near the source as well as in the receiving room. The signal from the source microphone is cross-correlated with the signal reaching the receiving microphone. A peak occurs in the correlogram when the delay corresponds to the direct path and again when it corresponds to the flanking path.

Both the cross-correlation method and the impulse method can locate the flanking path length and thus the position of the flanking path but require a considerable amount of specialised equipment and would be less reliable if there was more than one flanking path. For identification of the flanking path it must be assumed the time taken for the sound to travel through the structure is negligible.

4.1 THE USE OF DIRECTIONAL MICROPHONES

The previously described methods would be difficult for a building inspector to implement because of the amount of specialised equipment required. A simpler approach would be more suitable in most situations.

The directional microphone shown in Figure 1 was studied. It incorporates a sound level meter with a $^1/_3$ octave filter and headlight reflector positioned so that the ¼ inch microphone of the sound level meter is at the focus of the reflector. The directivity characteristics of the microphone are shown in

Figure 2. The response of the directional microphone was investigated at different $^{1}/_{3}$ octave bands. The results at 2kHz, 4kHz and 8kHz for a point in the centre of the receiving room are shown in Figure 3. It may be seen that the amplitude of the peak and the accuracy of the localisation is greater as the frequency increases. In subsequent tests the source and directional microphone were filtered at 8kHz.

Further tests were carried out in a reverberation chamber where the flanking was electronically introduced and thus carefully controlled. Figure 4 shows the results of different levels of flanking transmission. Even where the sound coming through the flanking path is 10 dB less than the sound coming through the direct path the peak is easily recognisable. Since it appears that the directional microphone is very sensitive to flanking paths it would first be necessary to measure the sound transmission loss of the wall to determine whether it is worthwhile reducing the sound transmitted via the flanking path. If the wall transmission loss is the same as that obtained in laboratory tests there is no point in reducing the transmission through the flanking path.

An enclosure with a front panel which opens and a loudspeaker inside was used to investigate the efficiency of the directional microphone for different widths of cracks. The response was uniform where the crack was carefully sealed and, as would be expected, the amplitude of the peak at the position of the crack increased as the crack width increased.

A check was made of the repeatability of the measurements and it was found that for ten tests taken at the same position in the receiving room, at different times, the standard deviation varied from 0.9 dB where the microphone was pointed at the crack to 0.2 dB remote from the crack.

Several tests were carried out to identify flanking paths in existing buildings. The test environment for one is shown in Figure 5 with the polar curve for the receiving microphone position shown as well. Both the flanking path along the corridor and the flanking path through a poor joint where the brickwork reaches the window are clearly visible.

In another case, the directional microphone was used in an attempt to locate panels of relatively low transmission loss, when no airborne flanking path existed, but no obvious peak at the edge or centre of the panel was apparent.

5.1 DISCUSSION AND CONCLUSION

The research indicates that no single method is able to identify the flanking path under all situations. The main conclusions are:

(i) Human localisation is useful as a first subjective approximation but has been shown to be unreliable when the path is not obvious.

(ii) The use of the stethoscope or the microphone with sound level meter does clearly indicate airborne flanking paths but its use would be very inconvenient for locating a non-apparent airborne path. This method may, however, be useful for assessing particular paths which are suspected as being airborne flanking paths.

(iii) The work of Cunliffe, as well as that carried out on the model, indicate that structure-borne radiation may be identified using an accelerometer, but the accelerometer would not be usable if the flanking path were airborne.

(iv) The impulse method appears to be usable for identifying flanking paths through panels of low transmission loss such as through false ceilings. The method is simple to carry out but requires a storage oscilloscope and is not certain how the method would perform with more than one flanking path.

(v) Experimental work on the cross-correlation technique indicates that the method produces less reliable results than the impulse method. It also requires more specialised equipment and is more difficult to carry out.

(vi) The directional microphone has proven to be very reliable for identifying airborne flanking paths. It is simple to implement and requires no specialised equipment other than a sound level meter with a frequency filter. Its results are reliable and repeatable. It cannot, however, predict the importance of a particular flanking path and would not be suitable for identifying structure-borne radiation or radiation from panels of relatively low transmission loss unless the solid-angle of the panel was less than the directivity lobe of the microphone.

The experimental investigation would indicate that human localisation and the stethoscope method could be eliminated as useful methods because the directional microphone provides the same information and is a more practical method. Similarly, the cross-correlation method is less usable than the impulse method and provides the same information in many cases (though it could be more useful in multipath propagation situations).

Thus, a procedure for identifying the main transmission path would be:

1. Carrying out a transmission loss test of the partition to determine whether the partition performs satisfactorily or not. It would be useful to use a simplified method such as that outlined by Lim[3].
2. If the partition was found to be unsatisfactory the directional microphone should be used to locate airborne paths. This method is best carried out first as it is the simplest to implement. The directional microphone need only be placed in the middle of the room and the room may be surveyed. If a path is identified by the directional microphone this does not mean that it is the main transmission path. The main path may be a structure-borne path. Thus, the airborne path should be repaired and the partition should be retested.
3. If the transmission loss of the partition has not improved or if the initial directional microphone test did not show any significant airborne path then the impulse method and/or the accelerometer should be used.

It would be advisable to carry out all the above methods as there may be more than one significant flanking path contributing to a low transmission loss value.

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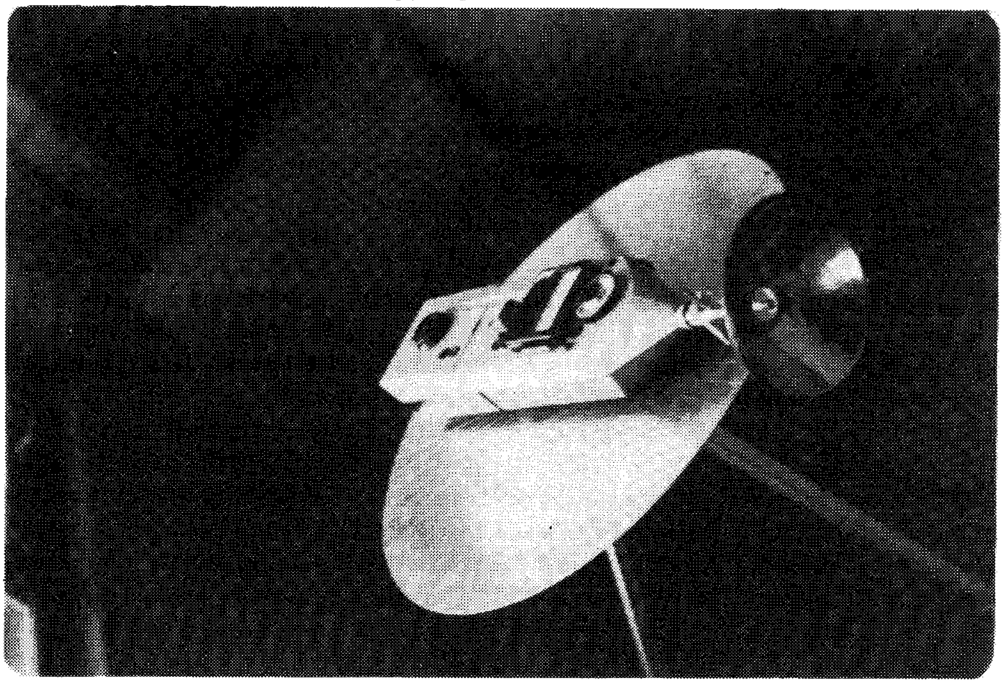


Figure 1: The directional microphone as used in the field.

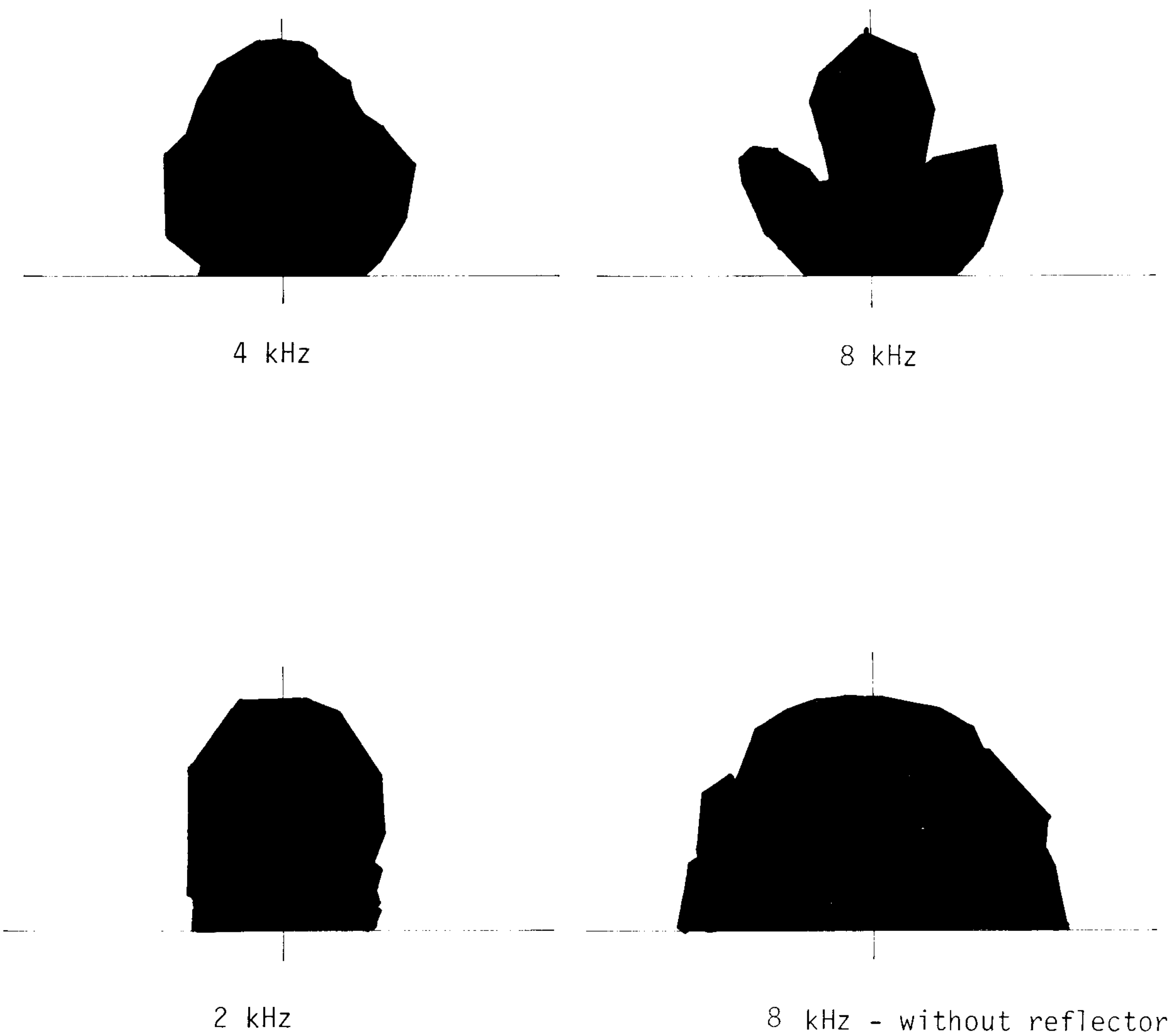
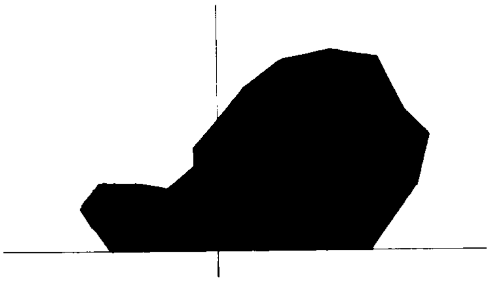
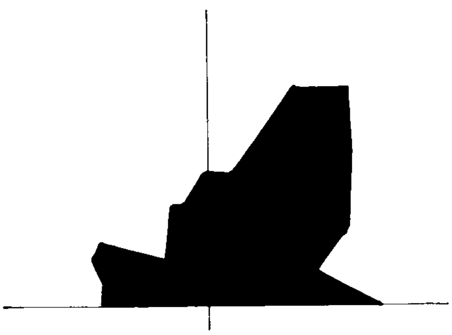


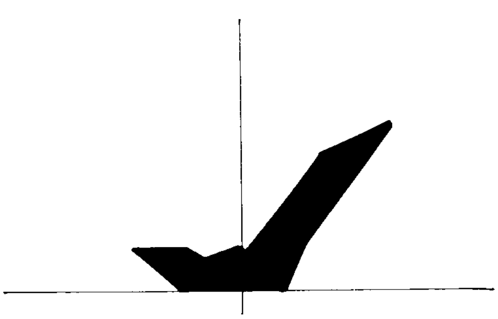
Figure 2: Response of directional microphone in anechoic room.



2 kHz

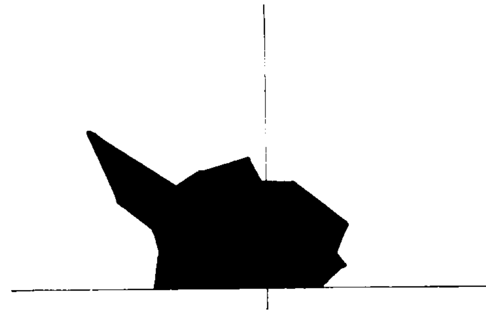


4 kHz

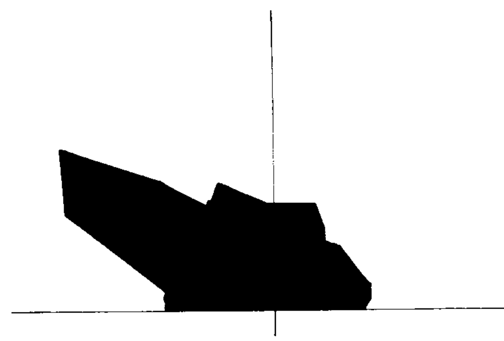


8 kHz

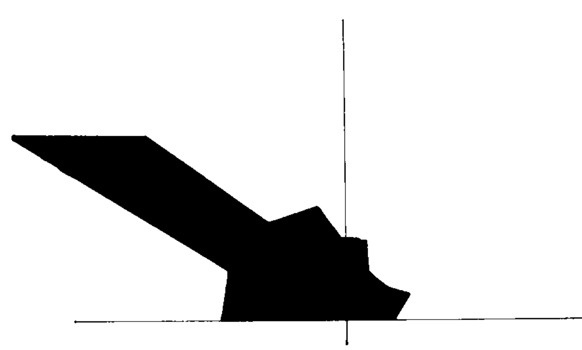
Figure 3: Comparison of Directional Microphone Results When the Signal was Filtered At 2, 4 and 8 KHz.



Flanking 10 dB < Direct



Flanking 4 dB < Direct



Flanking Level = Direct Level

Figure 4: Comparison of Different Relative Levels of Flanking in a Reverberant Chamber (8 kHz signal)

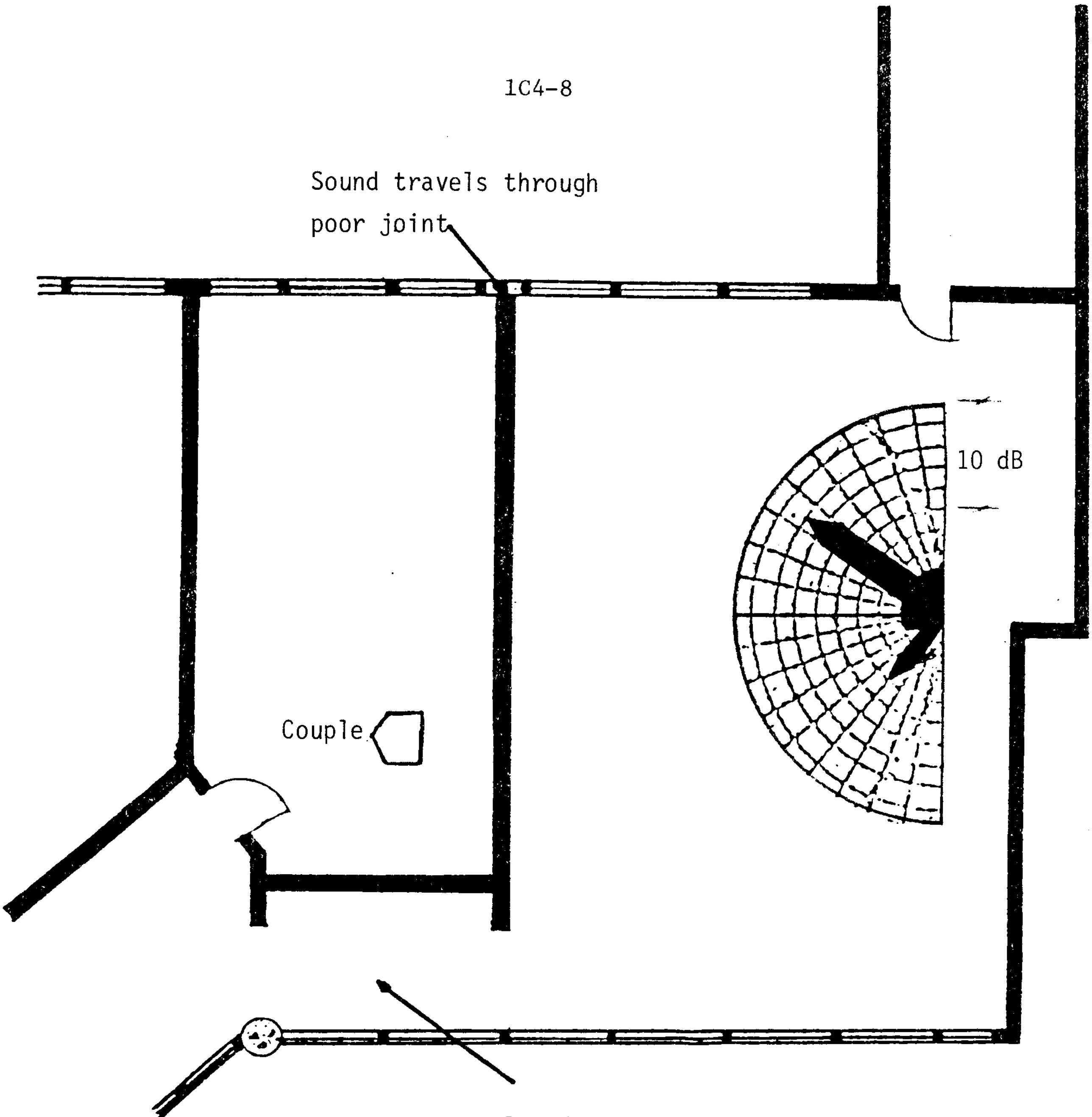


FIGURE 5.

EXAMINATION OF FLANKING IN AN
EXISTING BUILDING

OPTIMISATION OF HERSCHEL-QUINCKE SPLIT DUCT ATTENUATORS WITH AND WITHOUT FLOW

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ABSTRACT

The basic principle of the split duct attenuator is well known. A duct is divided into two branches of differing length which recombine after a short distance. The system acts as an attenuator to a tonal wave travelling down the duct if the branches are of equal cross-sectional area and their lengths vary by half a wavelength. Discussed here is the fact that the attenuating effect of such a system is more complex and more effective than this traditional model implies. Taking account of wave reflections in the branches, a more complete description is used to identify the best relative branch lengths for various conditions of optimality. The effects of finite duct length and flow in the duct are also considered.

INTRODUCTION

The principle of the Herschel-Quincke[1,2] split duct attenuator, as usually explained, is illustrated in Fig. 1. If a plane wave passing down a duct comes to a point where the duct divides into two parts ($S_1=S_2$) then the wave will split into two waves B and C of equal magnitude. At a later time these waves come together again as waves D and E to produce a net wave F which continues along the duct. For the case where A is a pure tone of wavelength λ, total cancellation between D and E will occur when

$$X_2 - X_1 = \frac{1}{2}\lambda \qquad (1)$$

With such a dimensional arrangement wave D will lag wave E by 180°, and the magnitude of wave F will be zero. More generally equation 1 can be written as

$$X_2 - X_1 = \frac{1}{2}(2N+1)\lambda \qquad (1a)$$

where $N = 0, 1, 2$ etc.

An alternative approach is to describe the situation at Q, not as simple wave cancellation, but as the excitation of a cross-mode which decays very rapidly in a narrow duct well below cut-off. This gives a better description of the situation in the immediate vicinity of Q, and is consistent with the simple wave cancellation approach at positions remote from Q.

The above explanation is, however, a misleading oversimplification of the real situation: what happens in a split duct attenuator is more complicated and more effective than is suggested by equation 1a. This was first identified by Stewart[4] who pointed out that cancellation of the downstream wave occurs not only when the conditions of equation 1a are met but also when

$$X_1 + X_2 = M\lambda \qquad (2)$$

where $M = 1, 2, 3$ etc.

These additional criteria arise because the waves D and E at Q do not simply add to produce wave F. Even with continuity of cross-sectional area ($S_1 + S_2 = S_o$), reflections occur at Q back along both branches in the duct. Fig. 2 shows the complete situation. The mechanics of this additional method of cancellation may be visualised by considering a wave travelling along branch 1, being reflected at Q, and then travelling back along branch 2. The total path length is $X_1 + X_2$, but when F = 0 the wave experiences a 180° phase shift at Q since the effect is similar to encountering an open end to a tube. Now in the case where M = 1 and $X_1 + X_2 = \lambda$, the returned wave H will lag the incident wave A by 540° so producing half cancellation at P. A similar delayed wave, D, appears in phase with H having gone round the circuit in the opposite direction. The net effect is total cancellation.

This result was obtained more formally by Stewart applying the conditions for continuity of pressure and volume velocity at P and Q. Defining $E = Ce^{-i\alpha_1}$ and $J = Ge^{-i\alpha_2}$ it is found that

$$L/A = \{[4 \sin (\alpha_2 + \alpha_1)/2][\cos (\alpha_2 - \alpha_1)/2]\} \times$$

$$\{[1 - 2 \cos (\alpha_2 + \alpha_1) + \cos (\alpha_2 - \alpha_1)]^2 + 4 \sin^2 (\alpha_2+\alpha_1)]\}^{1/2}$$

$$\text{if } S_i = S_o = 2S_i = 2S_2.$$

This equation, evaluated for the experimental case studied by Stewart, is shown in Fig. 3, where $\alpha_1 = kX_1$ and $\alpha_2 = kX_2$.

Apart from the large number of notches which can be obtained, what is of particular interest in Fig. 3 - from a practical point of view - is the substantial band width of the attenuation which might be achieved if X_1 and X_2 are appropriately chosen so that a notch from equation 1a is adjacent to one from equation 2. Consideration of this raises a number of questions relevant in the practical use of such an attenuator. What happens when the notch frequencies from equations 1a and 2 are the same? What are the optimal relative duct lengths to achieve wide band attenuation? Are the characteristics seriously affected if the duct is of finite length? Does flow in the duct affect the performance? This paper summarises investigations into these questions performed by solving the various forms of the equations of continuity of pressure and volume velocity in each case.

EFFECT OF SUPERPOSITION OF HERSCHEL-QUINCKE AND STEWART MINIMA

Values of X_1 and X_2 which result in superposition of the minima are obtained when

$$X_2 - X_1 = \frac{1}{2}(2N + 1)\lambda$$

and

$$X_2 + X_1 = M\lambda$$

An array of possible solutions is obtained by varying N and M, and Table 1 shows the values obtained for various N and M values. Although an infinite number of solutions exist, it is unlikely that practical forms would go beyond variations around the solutions in the first column for N = 0. Fig.4 shows the variation of $\sqrt{\ }$ transmission with non-dimensional frequency $W = (X_1 + X_2)/\lambda$ for the case where $X_2/X_1 = 3$. The particular point of note is that superposition of the Herschel-Quincke and Stewart minima produces a rounded minimum instead of a cusp. This could be a highly desirable feature in a practical realisation of the system since one of the usual objections to the use of this type of attenuator, as with quarter-wave stubs etc., is that the notch they provide is very narrow and small frequency changes in the source can greatly reduce their effectiveness. It is now apparent that this objection can be overcome. A further interesting feature of the attenuator in this form is that rounded minima occur also at the odd harmonics: this may be useful in attenuating inputs with a periodic impulsive characteristic where these harmonics are important.

Table 1

		N = 0	N = 1	N = 2
M	1	$\frac{0.75\lambda}{0.25\lambda}$ =3(fig.4)	-	-
	2	$\frac{1.25\lambda}{0.75\lambda} = 1.67$	$\frac{1.75\lambda}{0.25\lambda} = 7$	-
	3	$\frac{1.75\lambda}{1.25\lambda} = 1.4$	$\frac{2.25\lambda}{0.75\lambda} = 3$	$\frac{2.75\lambda}{0.25\lambda} = 11$

Values of X_2/X_1 for various values of N and M for which Herschel-Quincke and Stewart minima are superposed.

WIDER BAND ATTENUATION

Figure 3 shows that by spacing the Herschel-Quincke and Stewart minima appropriately a wide band attenuation effect can be achieved. The problem is to position the minima in an appropriate way so that a suitable compromise between transmission loss and attenuation bandwidth is achieved. It will be seen that as the minima are separated so the intervening hump in the transmission curve will rise. The optimum compromise depends on the requirement, and in fact a variety of optima can be identified depending on the value of W near which the matched minima are to occur. For example, if the wide notch is to occur near the Stewart minimum at W = 1, then a H-Q minimum either to the left or right can produce the desired effect (see fig.4). Table 2 shows the values of X_2/X_1 for such cases which will result in a minimum theoretical insertion loss of either 10 db or 20 db.

Table 2

	Required insertion loss	X_2/X_1	W_{min}	W_{max}
	10 db	2.05	0.93	1.58
or	10 db	5.90	0.62	1.04
	20 db	2.38 (fig.4)	0.96	1.28
or	20 db	4.15 (fig.4)	0.78	1.04

Some Optimal Values for X_2/X_1 to Achieve Specific Insertion Losses Over a Wide Frequency Range with $W \doteq 1$

Figure 5 shows an alternative and particularly effective arrangement which minimises transmission in the frequency range W = 2 to 3. Here X_2/X_1 is required to be 1.5. This case is one previously identified by Bies and Fuller[3] in their optimal 90° duct bend attenuator.

EFFECT OF FINITE DUCT LENGTH

A finite downstream duct length terminating with an open end has the effect of introducing a reflected wave N at Q. From standard theory[5] the impedance at this termination is given by

$$Z_L = \frac{\rho_o c}{S}\left(\frac{k^2 a^2}{2} + j\,\frac{8ka}{3\pi}\right) \qquad (3)$$

and the impedance at Q is given by

$$Z_o = \frac{\rho_o c}{S}\left(\frac{L+N}{L-N}\right) = \frac{\rho_o c}{S}\,\frac{Z_L + j(\rho_o c/S)\tan k\ell}{(\rho_o c/S) + jZ_L \tan k\ell} \qquad (4)$$

Solving the new equations of continuity provides a ratio of L/A as shown in Fig. 6, the exact form of the curve being particularly dependent on the downstream length, ℓ. The mechanics of the situation may be understood if the output of the attenuator is considered as driving a standing wave pattern in the downstream tube: at particular frequencies resonances occur, and L can be large. Although L/A can be > 1, the device does not of course act as an amplifier. Complete analysis requires consideration of power flow, the large downstream flow due to L being almost cancelled by the upstream flow due to N. The essential point to emerge from Fig. 6 is that finite duct length does not affect the broad notch characteristics of the attenuator although it would be advisable to avoid a downstream duct length which would produce a resonance at the input frequency.

EFFECT OF AIR FLOW

Flow complicates the analysis of sound transmission in ducts[6]. By extending the present simple analysis the consequence of low flow rates can, however, be identified from consideration of the altered phase speeds of the waves in the branches of attenuator. The waves numbers upstream and downstream become

$$k_a = k/(1 - M) \quad \text{and} \quad k_d = k/(1+M)$$

and for equal flow rate in each branch equations 1a and 2 change to

$$(1/1+M)X_1 - (1/1+M)X_2 = 1/2(2N + 1)\lambda \qquad (5)$$

$$(1/1-M)X_1 + (1/1+M)X_2 = M\lambda \qquad (6a)$$

and

$$(1/1+M)X_1 + (1/1-M)X_2 = M\lambda \qquad (6b)$$

The latter two equations are required to define the Stewart criteria in equation 2, but now cannot both be satisfied at the same time. The consequence of this is that with flow the depth of the notch, resulting from the Stewart criteria, decreases with increased flow rate[3]. The tuned attenuation frequency from equation 5 also changes.

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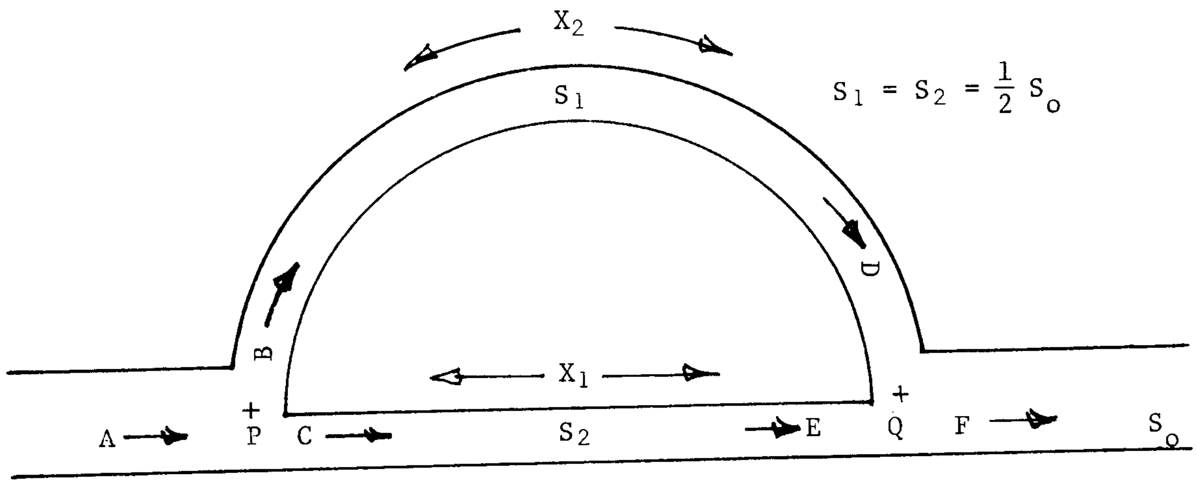

FIG. 1. Conventional plane wave description

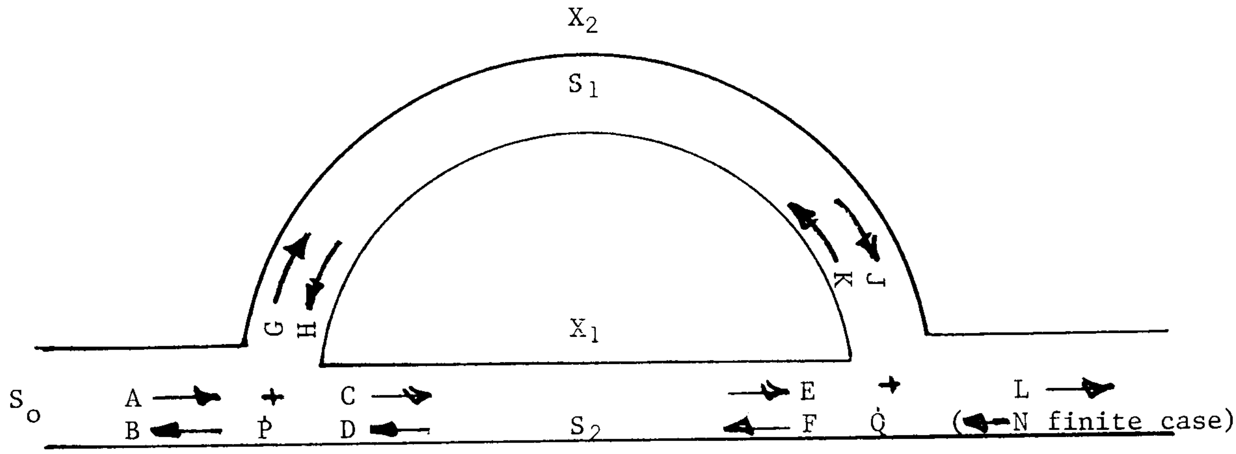

FIG. 2. Complete plane wave description

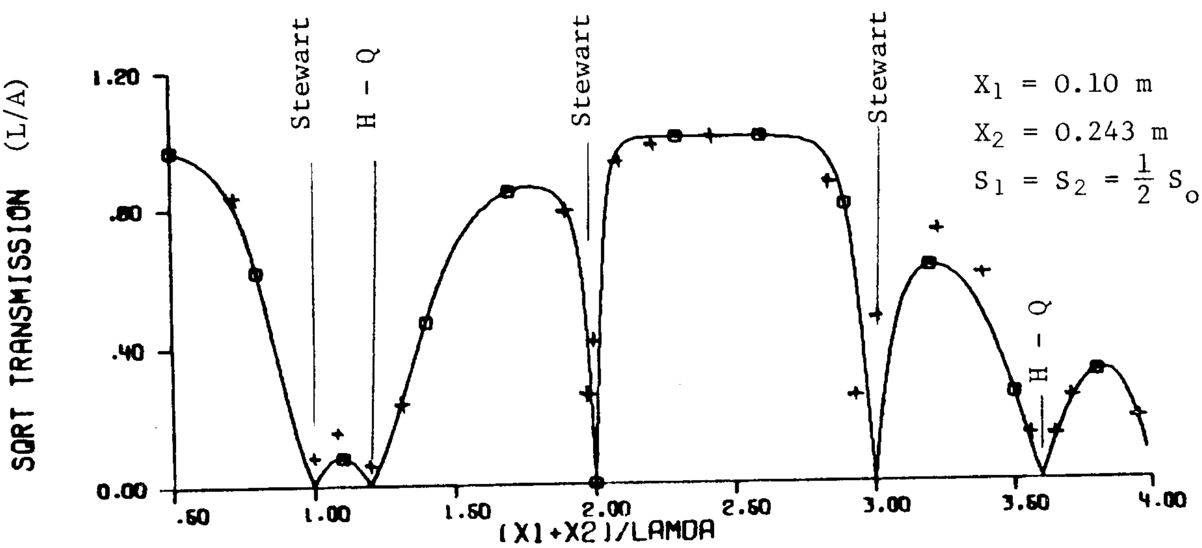

FIG. 3. Stewart's experimental results (+) compared with theory

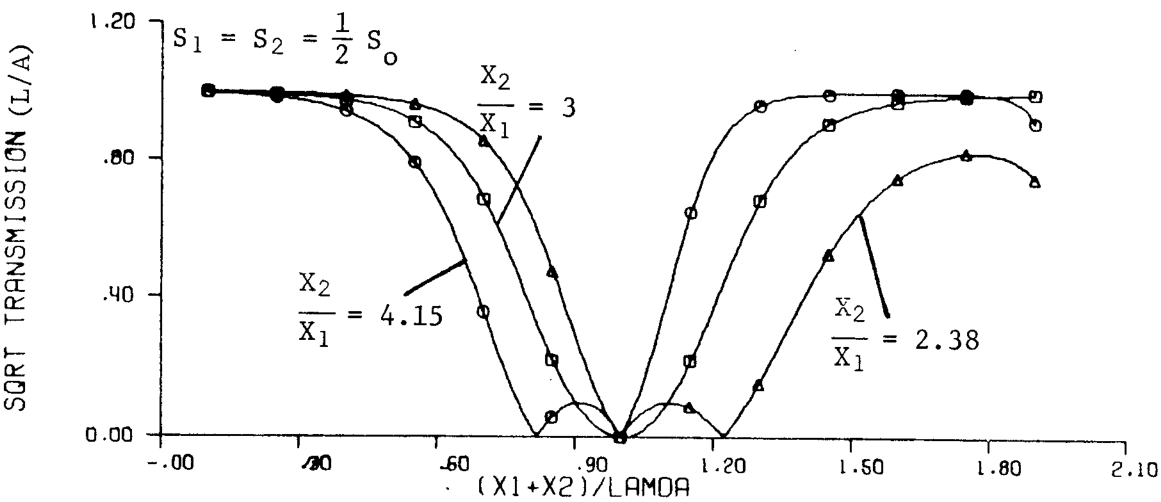

FIG. 4. Some optimal values of X_2/X_1 in the region W = 1.

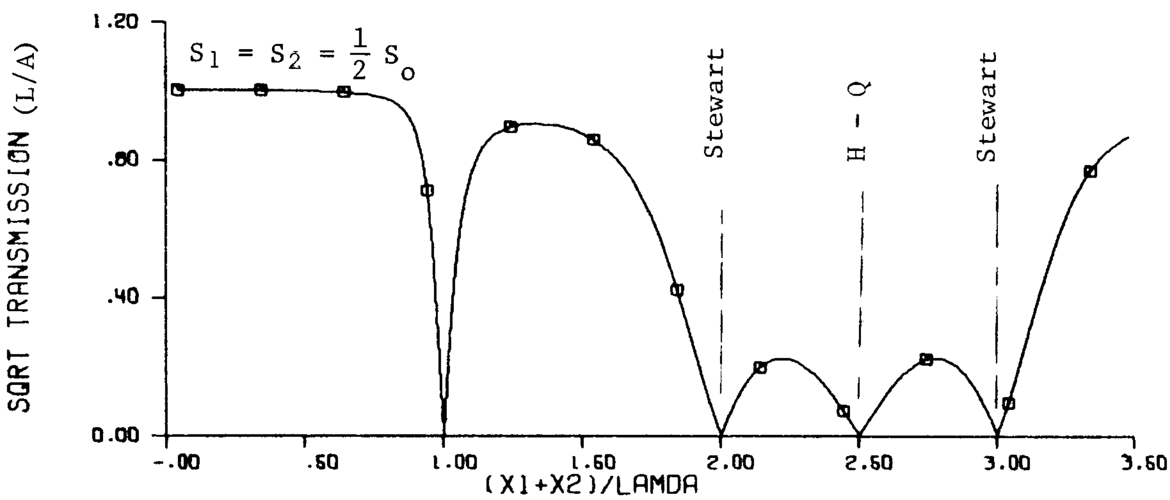

FIG. 5. Wide band attenuation with $X_2/X_1 = 1.5$

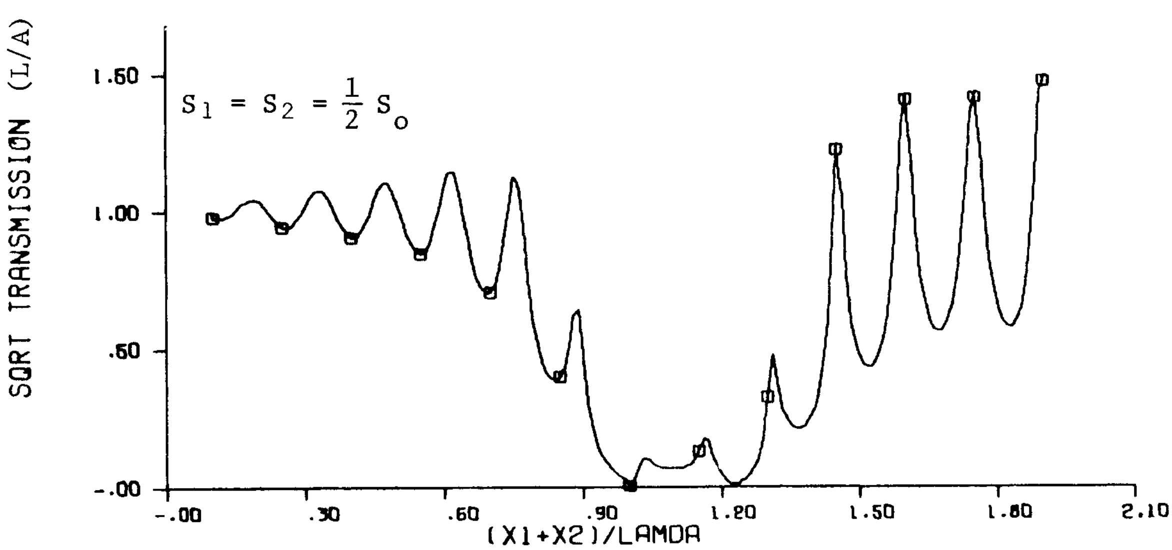

FIG. 6. Effect of finite duct length with $X_2/X_1 = 2.38$
(c.f. Fig. 4)

A MODAL DESCRIPTION OF REVERBERATION DECAY

by

David Alan Bies

ABSTRACT

A modal analysis is used to reconsider the problem of describing reverberation decay in a rectangular room. Growth of a reverberant field and steady state are then considered using the same modal analysis. The calculation of statistical obsorption co-efficients during both reverberent decay and steady state is considered. It is shown that the answers are critically dependant upon the assumed energy distribution among excited modes. It is shown that the usual assumption of equal energy distribution among excited modes is not correct. A possible alternative of equal power flow into all modes is considered.

INTRODUCTION

The classical description of reverberation decay in rooms proceeds upon the assumption of uniform energy distribution over all frequencies within a narrow frequency measurement band and all directions of propogation. On the other hand the modal response of enclosures is well known and the consequence of modal response is that when the source of sound is shut off the acoustic field will always decay at the discrete frequencies of the excited modes no matter how the field was established. The sound energy distribution during reverberation decay will be neither distributed uniformly over frequency nor over all directions of propagation.

The decaying reverberant sound field will be the sum of contributions from all of the decaying modes and their relative contributions will depend upon the model reflection co-efficients, the modal mean free paths, the initial energy distribution among excited modes and the relative phases of the decaying modes. In general, phase may be assumed to be random so that the latter dependence may be removed by averaging over many repeated measurements. However, modal energy distribution, modal reflection co-efficients and modal mean free paths generally are unknown and the effect of averaging on them is unclear. The problem of analysis is complicated further by the fact that modal energy distribution during reverberation decay can not be the same as in steady state since the various modes within a narrow frequency measurement band decay at different rates determined by their individual modal losses. Never the less standard procedures for room calibration implicitly proceed on the assumption that modal losses during reverberation decay are the same as the modal losses during steady state. This assumption is not true as will be demonstrated.

In this paper a modal decay model is developed and explored with the aid of a computer program. The analysis is supported by measurements made in a 180 m^3 reverberation room on which the computer model is based. Statistical obsorption co-efficients are investigated using the computer program. It is shown that the answers depend critically upon the assumed initial energy distribution among excited room modes.

The written portion of the paper presented here contains a summary of the analysis. The results of experiments, using the computer model, will be discussed during the oral presentation. Supporting experiments carried out in a 180 m^3 reverberant room also will be discussed where appropriate during the oral presentation.

MODAL MEAN FREE PATH

We consider a rectangular room of dimensions ℓ_x, ℓ_y and ℓ_z. The following useful quantities may be defined in terms of the room dimensions,

$$a_1 = \ell_y \ell_z \quad ; \quad a_2 = \ell_z \ell_x \quad ; \quad a_3 = \ell_x \ell_y$$

and

$$V = \ell_x \ell_y \ell_z$$

To proceed we define dimensionless wavenumber components as follows,

$$x = m_x c/2f\ell_x$$
$$y = m_y c/2f\ell_y$$
$$z = m_z c/2f\ell_z$$

Satisfaction of the wave equation requires that these quantities be related by the following equation.

$$x^2 + y^2 + z^2 = 1 \tag{1}$$

It may be shown that the mean free path of any mode i is [2]

$$\ell_i = V/(x_i a_1 + y_i a_2 + z_i a_3) \tag{2}$$

If we introduce spherical co-ordinates and we assume that the dimensionless mode numbers are continuous rather than discrete then integration over all possible values of the mode numbers in wavenumber space gives the following familiar expression for the mean free path.

$$\ell = 4V/S \tag{3}$$

where

$$S = 2(a_1 + a_2 + a_3) \tag{4}$$

MODAL MEAN REFLECTION CO-EFFICIENT

To proceed the walls of the rectangular enclosure are assumed locally reactive; for example, the walls may be described by an admittance factor dependent only upon location and frequency then, following Morse-Bolt, the expected reflection co-efficient for any wave incident upon any wall may be calculated in terms of the admittance factor and the angles of incidence and reflection [3]. However, these angles are completely determined by the mode numbers so that the appropriate reflection co-efficients may be calculated for every mode. For example, for an oblique mode six reflection co-efficients are required but for computational purposes these may be replaced by their geometric mean. Similarly all tangential and axial modes may be considered. Thus in general,

$$\beta_i = (\beta_{i_1}\beta_{i_2}\cdots\beta_{ij})^{1/j} \tag{5}$$

where

$$\beta_{ij} = \beta_{ij}(x_i y_i z_i) \tag{6}$$

For axial modes the value of j is 2, for tangential modes 4 and for oblique modes 6. Expressions given by Morse-Bolt are used for the above calculations.

To enable consideration of the standard sound absorption test one wall will be characterized by two areas of differing admittance, whereas all other walls are characterized by uniform admittance over their entire surfaces.

At the exceptional wall the mean wall admittance for the i th mode is,

$$\bar{\beta}_{ij} = 1 - (\Sigma S_i(1 - \beta_{ij})/\Sigma_{i=1}^{N} S_i) \qquad (7)$$

No effort has been made to consider possible diffration effects at the jth wall.

TRANSIENT RESPONSE

Decaying Sound Field

A sound field initially in steady state is considered subsequent to shutting off the source of sound. The intensity of the ith modal component at time t after source shut off travelling around its modal circuit is

$$I_i = I_{io}\beta_i^{ct/\ell_i} \qquad (8a)$$

Alternatively equation (8a) may be written as

$$I_i = I_{io}\ \exp[-c\alpha_i t/\ell_i] \qquad (8b)$$

where $\alpha_i = \ln\beta_i$ (9)

Modal analysis implicitly assumes standing waves which in turn implies that for every wave of intensity I_i there will be an opposite travelling wave of intensity $-I_i$ resulting at any instant and place in the sound field in energy density E_i/V. The sum of all modes within a narrow frequency band is

$$E = \Sigma_{i=1}^{N} E_i\ \exp[-c\alpha_i t/\ell_i] \qquad (10)$$

The sum is overall excited modes assumed N in number.

The matter of energy distribution among modes may be considered. The simplest assumption, and the assumption implicit in all classical analysis, is that energy is distributed equally among all of the excited modes. In this case equation (10) takes the form

$$E = E_o\Sigma_{i=1}^{N}\ \exp[-c\alpha_i t/\ell_i] \qquad (11a)$$

An alternative assumption not considered in the classical literature and perhaps not at all, is that of equal power flow in steady state into every mode. With the assumption of equal power flow equation (10) takes the form

$$E = E_o\Sigma_{i=1}^{N}(\ell_i\alpha/\ell\alpha_i)\exp[-c\alpha_i t/\ell_i] \qquad (11b)$$

In equation (11b)

$$\alpha = \frac{1}{N}\Sigma_{i=1}^{N}\alpha_i \qquad (12)$$

The well known Norris-Eyring equation follows from equation (11a) if all α_i and ℓ_i of the terms in the sum are replaced with assumed mean values. In this case equation (11a) becomes

$$E = E_T\ \exp[(ctS/4V)\ln(1-\bar{\alpha})] \qquad (13)$$

In the above equation $E_T = NE_o$ has been used.

It is to be noted that all of the assumptions required to derive equation (13) may be questioned. (1) The assumption of equal energy per mode cannot be correct; oblique modes near grazing incidence at any wall will suffer large absorption no matter what wall construction is used. (2) The replacement of the modal mean free path and modal mean reflection coefficients with assumed mean values is questionable. For example while it may be shown that the mean of all possible modal mean free paths is given by equation (3) no similar demonstration exists for the mean absorption $\bar{\alpha}$. Large errors can certainly be expected using equation (13) or its equally well known approximate form due to Sabine.

STEADY STATE

A sound field in steady state equilibrium with its source is described by the following equation

$$W = \Sigma_{i=1}^{N} E_i c\alpha_i/\ell_i \tag{14}$$

In equation (14) the modal energies become narrow frequency bands of energy rather than single frequencies.

In the case of equal energy per mode equation (14) becomes

$$W = -E_o \Sigma_{i=1}^{N} c\alpha_i/\ell_i \tag{15}$$

while for the case of equal power flow into every mode equation (14) takes the form

$$W = -E_T c\alpha/\ell \tag{16}$$

In equation (16) α is given by equation (12) and ℓ by equation (3).

EXPERIMENTAL WORK

An experimental program is underway on two fronts to investigate the ideas presented above. On the one hand work is being conducted in a 180 m^3 reverberant room in the low frequency range of the 63 Hz and 80 Hz one third octave bands. In this range individual modes can be distinguished with care and investigated. On the other hand a computer program has been constructed to simulate the 180 m^3 reverberant room and allow investigation of the model. Results of these studies will be described in the oral presentation.

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SYMBOLS

a_1, a_2, a_3	wall areas
c	speed of sound
E	stored energy in acoustic field
E_i	stored energy in ith mode
E_o	modal energy
E_R	energy in reverberant sound field
E_T	stored energy in acoustic field in steady state; at zero time
I_i	intensity of wave travelling in ith modal circuit
I_{io}	initial intensity of wave travelling in ith modal circuit
f	frequency in hertz
i	(integer) number of reflections
ℓ	mean free path averaged over all modes
ℓ_i	mean free path of ith mode
ℓ_x, ℓ_y, ℓ_z	room dimensions
N	total number; modes in narrow frequency band, area subsections
n_x, n_y, n_z	mode numbers
S	room wall total surface area
S_i	area of ith subsection of one wall
t	time
V	room volume
W	acoustic power introduced into field of enclosure
x, y, z	dimensionless wave number components
x_i, y_i, z_i	dimensionless wave number components of ith mode
α	mean value of the quantities α_i averaged over all modes in a narrow frequency band
α_i	negative of the natural logarithm of the ith reflection coefficient
$\bar{\alpha}$	Norris-Eyring absorption coefficient; Sabine absorption coefficient
β_i	geometric mean reflection coefficient of ith mode
$\bar{\beta}_{ij}$	mean reflection coefficient of wall composed of area subsections

SOUND TRANSMISSION THROUGH RIBBED WALLS

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ABSTRACT

The need to design the walls of industrial buildings to satisfy a specified exterior noise level at some distance from the building raises the question of the transmission loss achievable by light-weight sheet claddings. The purpose of this paper is to re-examine the influence of the sheet profile on the coincidence frequencies and the directivity of the transmission coefficient. It is argued that the rib direction will play a role in the noise level experienced but that the conventional transmission loss tests will not predict this effect. A discussion of panel resonances is also included.

INTRODUCTION

Much information is available on the sound transmission properties of walls and the underlying principles are well understood. The special features of the transmission behaviour of orthotropic and ribbed panels have been described in the literature (see, for example, Chapter 11 of Reference 1), particularly with respect to the subpanel resonances and coincidence or critical frequencies (2). While the overall transmission loss of these panels, as would be measured in coupled reverberation chamber tests, is in many cases important, there are other circumstances where detailed information concerning the directionality of the radiation on the receiver side may be useful.

When an industrial building encloses noisy machinery, the exterior walls of the building will radiate sound which, in the absence of barriers, etc, will be transmitted as freely propagating waves. Both waves radiated downwards from the wall which are partially absorbed and reflected upwards (depending on the ground conditions), and waves radiated skywards will not contribute to the sound at a far field receiver position in the horizontal plane. The difference between this situation and the one of a barrier between two enclosed spaces is quite obvious when it is recalled that in the latter case the surrounding walls tend to reflect all the waves to produce a diffuse field in the receiving space.

If the walls of a building do not have any directionality in their sound radiation properties there is little significance in the fact that certain transmitted wave directions do not contribute to the horizontal sound field. However, when walls, such as those constructed of ribbed panels, are directional in their radiation there is an additional feature of the "interior diffuse" to "exterior free" transmission problem to be considered.

PROPERTIES OF A RIBBED PANEL

A by-product of profiling sheets in order to increase their rigidity and, therefore, minimise the extent of the supporting structure, is an increase in the speed of structural transverse waves which travel in the rib direction. As a consequence the coincidence frequency for those waves occurs at a much lower frequency than the coincidence frequency of waves in an unstiffened sheet. For example, the coincidence frequency for a sheet of 0.6 mm thick steel sheet is 20 kHz but when the material is used to produce the LBI Spandek profile the coincidence frequency for waves in the rib direction becomes only 400 Hz. In general the deeper the profile the lower this latter frequency becomes and the more potentially damaging will be its effect on the wall transmission loss. Table 1 lists for illustrative purposes the relevant properties of some common cladding profiles in the LBI product range.

An additional acoustic effect of profiling a sheet is the introduction of various subpanels. Although the spans of the supporting structure are generally sufficiently large for basic panel resonances to be unimportant in sound transmission calculations (see Table 1), the subpanel resonances may be, depending on the profile geometry, in the audible frequency range. Low order subpanel resonances will cause a localised decrease in the transmission loss and, at higher frequencies, a likely modal effect due to the dipole sources produced by lack of cancellation at subpanel edges (1). However the precise effect on the transmission loss is known to be difficult to predict because of the way in which different elements of the ribbed profile interact.

COINCIDENCE EFFECTS

The transverse deflection, w, of a ribbed panel may be described by the orthotropic plate equation,

$$(1-i\eta)(D_x\partial^4 w/\partial x^4 + 2H\partial^4 w/\partial x^2\partial y^2 + D_y\partial^4 w/\partial y^4) + m\partial^2 w/\partial t^2 = \Delta p, \qquad (1)$$

where D_x, H, D_y are the appropriate panel flexural stiffnesses with respect to an orthogonal co-ordinate system oriented such that the x-axis is aligned with the rib direction, m is the surface mass of the panel, Δp is the localised pressure difference, t is the time variable, and η is the panel loss factor.

A simple analysis of the behaviour of an acoustic wave at frequency, f (Hz), incident angle, θ , and angle, ϕ , with respect to the rib direction (see Figure 1), when it is incident on an infinite panel, whose response is described by equation (1), leads to the derivation of the transmission coefficient, τ , which may be written as

$$\tau^{-1}(\theta,\phi)=\{1 + \eta M\cos\theta(f/f_{c\ell})^3\sin^4\theta(\cos^4\phi + 2\beta\cos^2\phi\sin^2\phi+ \alpha\sin^4\phi)\}^2$$
$$+ M^2\cos^2\theta(f/f_{c\ell})^2\{1-(f/f_{c\ell})^2\sin^4\theta(\cos^4\phi+ 2\beta\cos^2\phi\sin^2\phi+ \alpha\sin^4\phi)\}^2, \qquad (2)$$

where the lower coincidence frequency, $f_{c\ell}$, is given by

$$(4\pi f_{c\ell})^2 = mc^4/D_x,$$

and $$M = \pi m f_{c\ell}/(\rho c),$$

$$\alpha = D_y/D_x \ll 1,$$

$$\beta = H/D_x \ll 1,$$

with c being the speed of sound in air of density, ρ. (See Table 1 for representative values of these parameters.)

Although equation (2) is incapable of describing the effects of the panel supports and does not provide subpanel resonant response information, it does illustrate many features of practical importance and, in particular, shows the effects of acoustic and panel wave matching which occurs at and above the lower coincidence frequency.

For isotropic panels there is, for each frequency greater than the coincidence frequency, an incident wave angle which would give, in the absence of panel damping, a transmission coefficient of unity, i.e. to waves incident at that particular angle the panel is completely transparent. For the orthotropic panel, equation (2) indicates that for each frequency greater than the lower coincidence frequency, $f_{c\ell}$, there is, over a range of polar angles, ϕ, an angle of incidence, θ, for which the transmission coefficient approaches unity (η small). The range of polar angles extends from zero (the rib direction) to an angle which is the same as the angle of incidence at zero polar angle giving the greatest transmission. A typical locus of the high transmission condition may be seen in Figure 1.

The transmission behaviour of each component of a diffuse sound field will be described by equation (2) and, thus, in order to evaluate the overall transmission properties, $\tau(\theta, \phi)$ should be integrated over all wave angles to obtain a mean transmission coefficient, $\bar{\tau}$, i.e.

$$\bar{\tau} = \frac{\int_0^{2\pi}\int_0^{\theta_{lim}} \tau(\theta,\phi)\cos\theta \sin\theta \, d\theta \, d\phi}{\int_0^{2\pi}\int_0^{\theta_{lim}} \cos\theta \sin\theta \, d\theta \, d\phi} . \qquad (3)$$

It has been found that the use of an upper limit on θ slightly less than the grazing incidence angle, $\pi/2$, gives better agreement between theory and experiment. Reference 1 suggests that $\theta_{lim} = 78^{\circ}$ be used. The transmission loss, R, is defined by

$$R = -10 \log \bar{\tau} . \qquad (4)$$

The use of equation (3) is justified when the ensemble of transmitted waves contribute to the sound field on the receiving side of the panel. This is the case when waves of all angles reflected from the enclosing walls in the receiving space contribute to the field. If a ribbed panel were placed in the aperture between coupled reverberation chambers (as described in (3) for the laboratory testing of transmission loss properties of barriers), it would be expected that the most appropriate integration limits are those of equation (3). Even the field loss transmission measurements (4) are designed to achieve a similar result by an averaging of pressure readings in the near field of the test panel.

For the theoretical examination of the sound field caused by the wall radiation from an industrial building, the integration limits on equation (3) should be modified so that only the relevant receiver positions are included in the summation. (This approach is similar to that used in the study of the influence of directivity patterns of aircraft jet engines on, for example, the sideline noise.) If the panels are fixed on the building so that the ribs run vertically then, in order to obtain a measure of the received noise in the horizontal plane, ϕ should be limited to angles close to $\pi/2$. The resulting transmission loss calculated from equation (4) is defined as $R_{\pi/2}$. In the results which are presented in Figure 2 the bounds on ϕ are $[80^o, 100^o]$ and $[260^o, 280^o]$ but the use of a $\pm 10^o$ or a $\pm 20^o$ range has an insignificant effect on the calculated value of $R_{\pi/2}$. When the ribs run horizontally the bounds on ϕ become $[-10^o, 10^o]$ and $[170^o, 190^o]$; the transmission loss value in this case is defined by R_0.

The significance of these limits can be gauged immediately by reference to Figure 1: the integration over the $\pi/2$ region will not give a coincidence peak contribution, unlike that around zero, until the frequency is sufficiently large for the coincidence transmission effects to have penetrated to large values of ϕ. In fact, coincidence only becomes important for waves running perpendicularly to the rib direction when the frequency reaches the upper coincidence frequency (effectively the plane sheet coincidence frequency).

Figure 2 shows a comparison of the overall transmission loss, R, and the transmission loss values of R_0 and $R_{\pi/2}$ for the LBI Spandek profile (0.6 mm base metal thickness). As might be expected, from the discussion of the previous paragraph, both R and R_0 exhibit the coincidence "dip" at the lower coincidence frequency while $R_{\pi/2}$ closely follows the field incidence "mass loss" result (1). As may be seen from equation (2), the transmission coefficient at frequencies and angles which cause the second term to become zero is influenced by the magnitude of the loss factor, η, ("resistance controlled"), and this is reflected in the size of the dip in the transmission loss beyond the coincidence frequency. (For $\eta = 0.01$, instead of 0.001, R and R_0 would be increased by 1 dB at 500 Hz but by 4 dB for R and 10 dB for R_0 at 9 kHz.)

The general features illustrated in Figure 2 are typical of other profiled sheet materials used to clad buildings.

CONCLUSIONS

Two conclusions follow from the foregoing analysis:

(i) the rib direction on walls clad with profiled sheets is an important factor in determining exterior sound levels; and

(ii) the wall transmission loss, as measured by established test methods (3,4), will not be a good predictor of the far field noise levels unless consideration is given to the significance or otherwise of any coincidence dip apparent in the measured transmission loss values.

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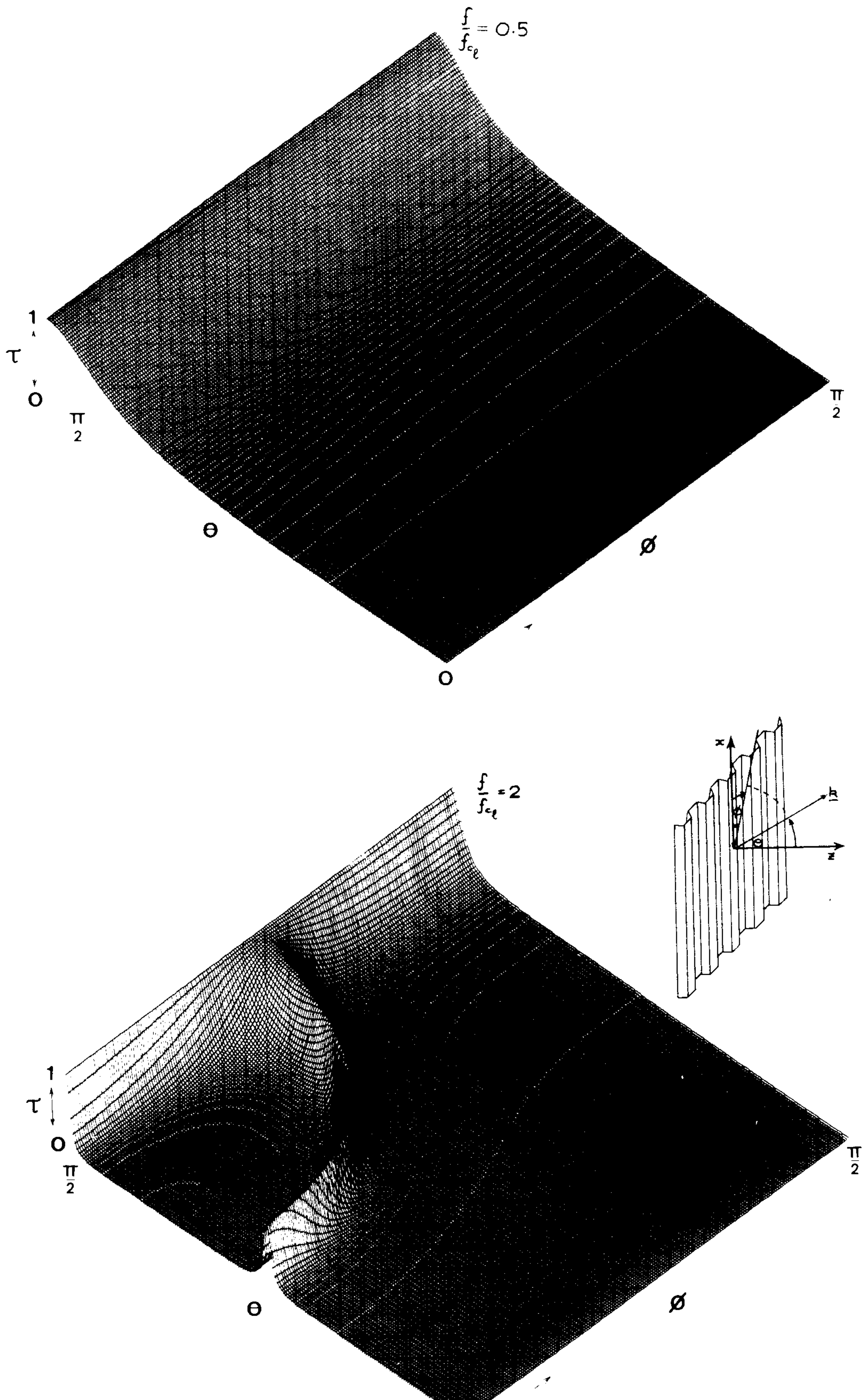
TABLE 1 - PROFILE PROPERTIES

	Custom Orb 	Spandek 		Deep Rib 600 
Base metal thickness (mm)	0.42	0.48	0.60	0.75
Surface mass, m (kg/m^2)	4.4	5.4	6.7	7.9
Stiffness in the rib direction, D_x (Nm)	3000	11 000	13 600	130 000
$\alpha = H/D_x$	3.7×10^{-4}	1.4×10^{-4}	2.2×10^{-4}	4.3×10^{-5}
$\beta = D_y/D_x$	5.2×10^{-4}	1.7×10^{-4}	2.5×10^{-4}	5.8×10^{-5}
$f_{c\ell}$ (Hz)	700	400	400	150
M	24	17	21	8.8
Basic resonance frequency using recommended wall support spacing (mm) given in parentheses (Hz)	22 (1350)	12 (2400)	10 (2700)	10 (4500)
Subpanel resonance frequency, based on element length (mm) given in parentheses (Hz)	-	1200 (30)	1500 (30)	200 (90)



<u>Figure 1:</u> The Variation of the Transmission Coefficient, τ, with Angle of Incidence, θ, and Angle, ϕ, with Respect to the Rib Direction for the LBI Spandek Profile.
(Base Metal Thickness, 0.6 mm: η 0.001)

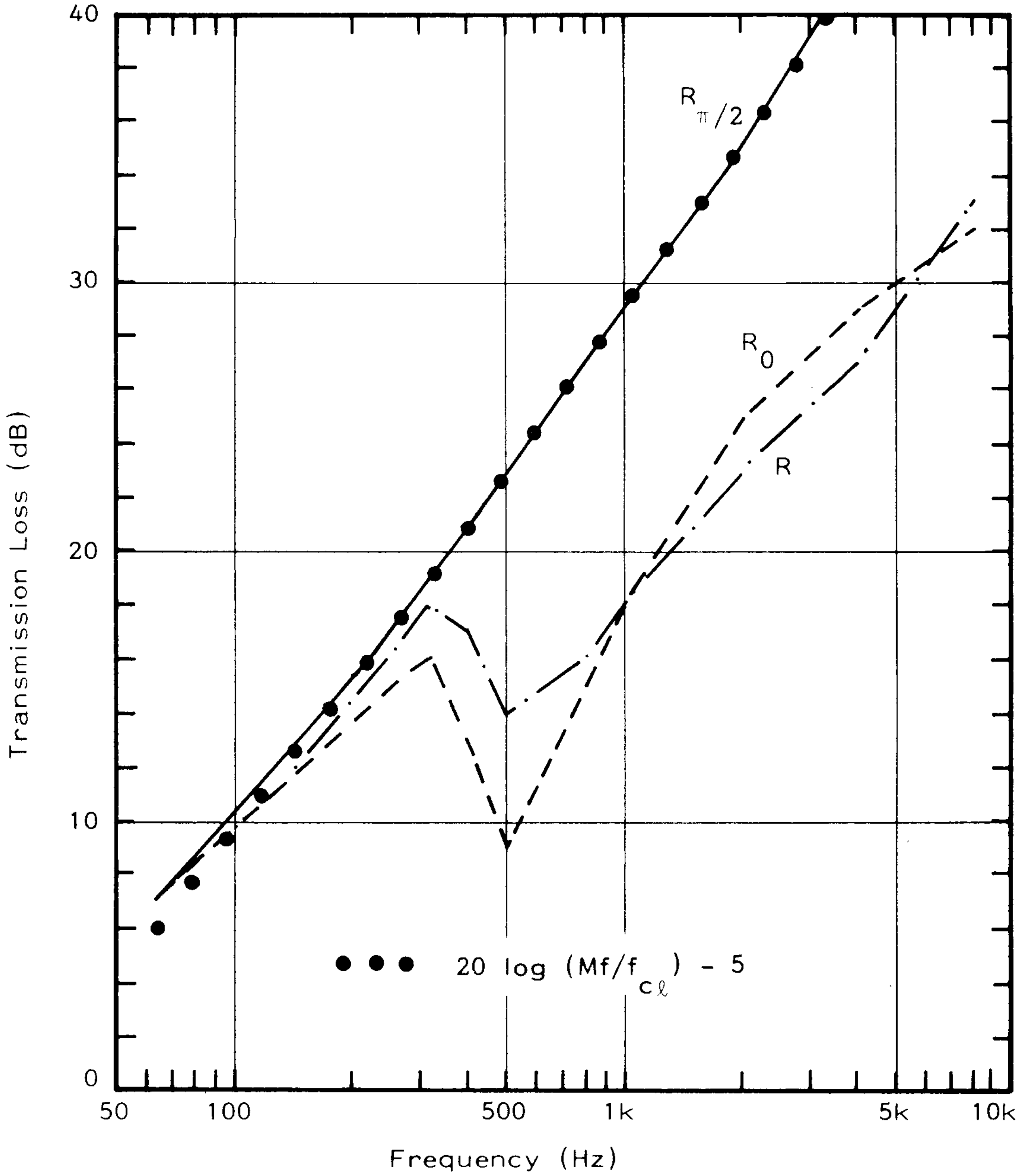


Figure 2: A Comparison of Theoretically Calculated Values for the Transmission Loss of the LBI Spandek Profile. (Base Metal Thickness, 0.60 mm; η = 0.001)

"A NON-STATISTICAL SOUND FIELD DECAY ANALYSIS USING GENERALIZED ZETA FUNCTIONS"

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A B S T R A C T

Riemann's Zeta Functions are made more general to account for the absorption of energy and for different degrees of divergence in acoustic wave radiation. A comparison between the decays of steady and impulsive sound fields is made using these Generalized Zeta Functions.

A seven place table of the Generalized Zeta Functions is presented.

INTRODUCTION.

In a previous paper [1] the author described the relationship between the sound power leaving a source and the sound power arriving at a receptor in the following way:

Put W^*_{Ai} = sound power emanating from A at time, i; i = 0,1,2,..

W_{Bi} = sound power arriving at B at time, i

α_j = attenuation coefficient for ray j ; j = 0,1,2,..

and let the interval between successive times, i, be a (say the greatest) common factor of the times for all rays for which α_j is not zero.

The sound power arriving at B at successive times, 0,1,2,.... are:-

$$\left.\begin{aligned} W_{B0} &= \alpha_0 W^*_{A0} + \alpha_{-1} W^*_{A-1} + \alpha_{-2} W^*_{A-2} + \text{------} \\ W_{B1} &= \alpha_0 W^*_{A1} + \alpha_{-1} W^*_{A0} + \alpha_{-2} W^*_{A-1} + \text{------} \\ &\\ W_{Bi} &= \alpha_0 W^*_{Ai} + \alpha_{-1} W^*_{A(i-1)} + \alpha_{-2} W^*_{A(i-2)} + \text{---} \end{aligned}\right\} \qquad (1)$$

or $$W_{Bi} = \sum_{j=0}^{-\infty} \alpha_j \cdot W^*_{A(i-j)}. \qquad (2)$$

In the present paper, a case is discussed in which the values of α_0, α_{-1} etc. are related in a particular way. Furthermore the sequence W_{Bi} resulting from the case, $W^*_{A0} = W^*_{A-1} = \ldots = 1$, is compared with the sequence, $\alpha_k \, W^*_{Ak}$.

For the case discussed, this corresponds with a comparison of two common methods of determining reverberation times. It is shown that the methods are not equivalent.

A Special Case.

Suppose, $W^*_{A0} = W^*_{A1} \doteq \ldots = 1$

and, the αs, representing coefficients for rays of lengths of equal increments, may be represented thus:

$$\alpha_k = \frac{\beta_k \; \delta_k}{(k+1)^i}$$

where, $\delta = 1$ or 0

Furthermore, in an idealised situation, the βs may be represented thus:

$$\beta_k = \beta^{(k+1)}$$

A series for the (reverberant) power arriving at B at time 1 is then ($\delta = 1$),

$$U_1 = V_1 + V_2 + V_3 + - - - -$$

$$= \sum_{V=1}^{\infty} \frac{\beta^V}{V^i}$$

This series is a generalization of the well known Riemann Zeta Functions,

$$\zeta\,(i) = \sum_{V=1}^{\infty} \frac{1}{V^i} \qquad i = 2,3,----$$

Accordingly, we designate U_1 thus:-

$$\zeta\,(i,\beta) = \sum_{V=1}^{\infty} \frac{\beta^V}{V^i}\,. \tag{3}$$

in which i and β are not restricted to integers.

Should the source be discontinued, after time 1, then the power arriving at B at time 2 is less than that at time 1 by the value of the leading term; and so on. The proximate series are:-

$$\zeta(i,\beta,1) = U_1 = V_1 + V_2 + V_3 + - - -$$

$$\zeta(i,\beta,2) = U_2 = V_2 + V_3 + - - -$$

$$\zeta(i,\beta,t) = U_t = V_t + V_{t+1} + - - - \tag{4}$$

This is our Generalized Zeta Function.

Numerical Values.

Values of the Riemann Zeta Function, $\zeta(i,1,1)$ to 20 decimal places for integral values of i have been published elsewhere [2].

Values of the Generalized Zeta Function, $\zeta(i,\beta,1)$ to 7 decimal places are given, in Table 1 of this paper, for a range of values of i and β.

The series, $\sum_{V=1}^{\infty} \frac{\beta^V}{V^i}$ was found to converge too slowly for convenience. By an application of Kummer's Transformation [3], an equivalent series was derived which converges much more rapidly. This is shown in equation (5).

$$\zeta(i,\beta,t) = \frac{\beta^t}{(1-\beta)t^i} - \frac{\beta}{1-\beta} \sum_{V=t}^{\infty} \frac{\beta^V}{V^i} \left\{ 1 - \left(\frac{V}{V+1}\right)^i \right\} \tag{5}$$

The tabulated values were determined from this equation, putting t=1.

Comparison of the Sequences U and V.

The sequence, V, may represent the decay of an impulsive sound and U the decay of a steady sound abruptly stopped; corresponding with two well known methods of determining reverberation times.

Figure 1 gives values of 10 log V_V/U_V for the first 11 values of V for a variety of values of β and i. If the slope of a curve in Figure 1 be zero, then the above two methods of determining reverberation time would be equivalent.

It will be seen that this is very nearly true for $\beta = 0.99$, $i = 0$; i.e. for small absorption and no divergence. The disparity is greatest in cases of high absorption ($\beta = 0.1$).

In actual buildings, the replete functions, $\zeta(i,\beta,t)$ do not represent the case because many of the δ_ks = 0. That is, the series are lacunose. Furthermore, the sound power arriving at β is represented by the sum of Generalized Zeta Functions with different values of β, and i also because i depends somewhat on β. In some cases, the reverberation times determined by the two methods are similar. This outcome is commonly due to lacunae, but may also be due to low absorption and low diffusion as previously mentioned.

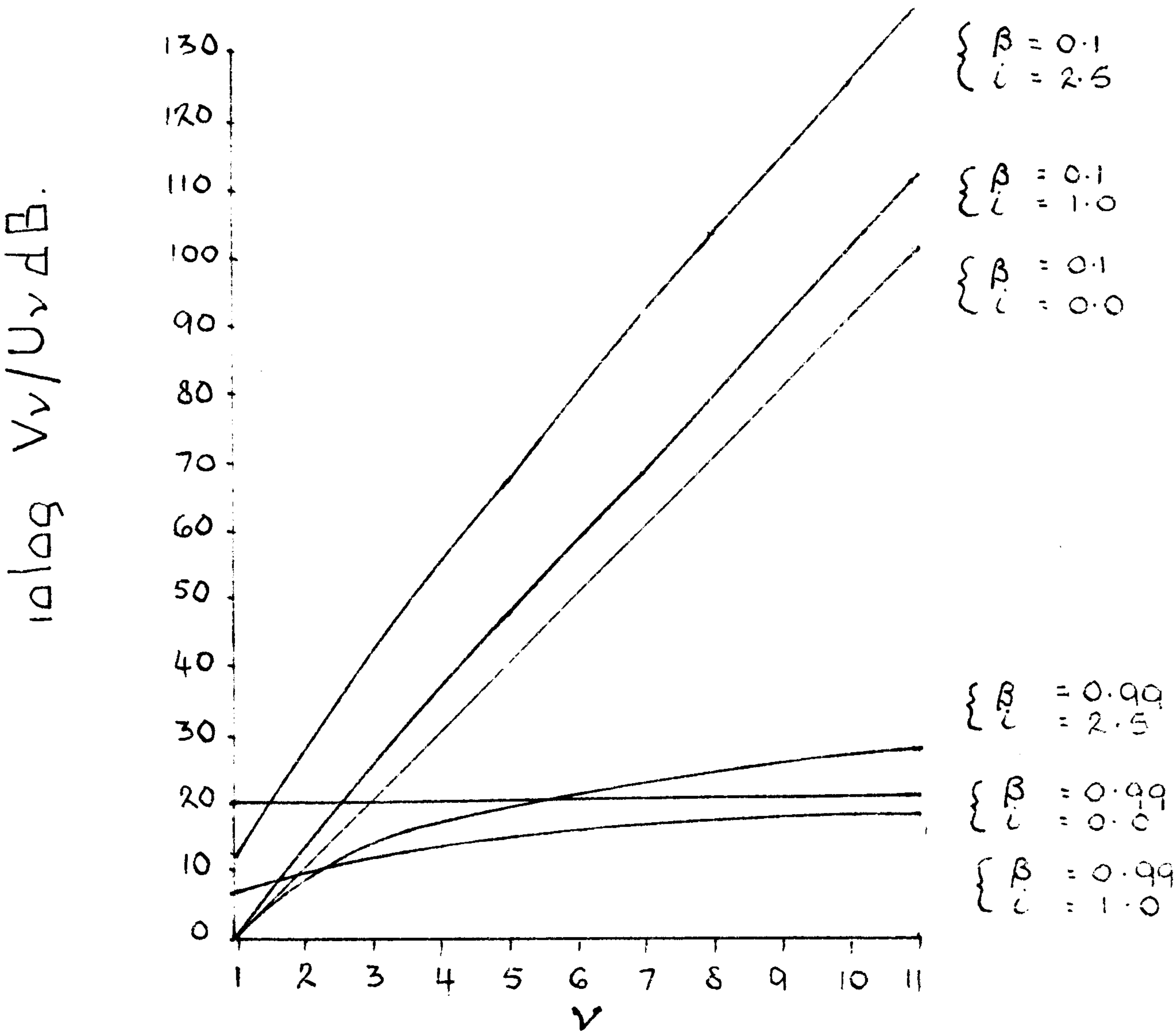

FIGURE I.

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[3] Knopp, K., "Infinite Sequences and Series". Dover Publications.

TABLE 1

GENERALIZED ZETA FUNCTIONS, $\zeta(i,\beta,1)$

$$\zeta(i,\beta,1) = \sum_{V=1}^{\infty} \frac{\beta^V}{V^i}$$

β \ i	0.0000000 00	1.0000000-01	2.0000000-01
0.99	9.9000000 01	6.6529892 02	4.5430325 01
0.9	9.0000000 00	7.5063303 00	6.3241375 00
0.8	4.0000000 00	3.5413756 00	3.1582890 00
0.7	2.3333333 00	2.1353727 00	1.9649972 00
0.6	1.5000000 00	1.4040774 00	1.3198177 00
0.5	1.0000000 00	9.5203835-01	9.0925573-01
0.4	6.6666667-01	6.4328395-01	6.2216933-01
0.3	4.2857143-01	4.1814927-01	4.0864233-01
0.2	2.5000000-01	2.4622093-01	2.4274396-01
0.1	1.1111111-01	1.1032278-01	1.0959205-01

β \ i	3.0000000-01	4.0000000-01	5.0000000-01
0.99	3.1588189 01	2.2398066 01	1.6221831 01
0.9	5.3829597 00	4.6292080 00	4.0219504 00
0.8	2.8369781 00	2.5663766 00	2.3375564 00
0.7	1.8179124 00	1.6905502 00	1.5799383 00
0.6	1.2456308 00	1.1801638 00	1.1222635 00
0.5	8.7102617-01	8.3680696-01	8.0612672-01
0.4	6.0307837-01	5.8579569-01	5.7012118-01
0.3	3.9996252-01	3.9203107-01	3.8477745-01
0.2	2.3954317-01	2.3659505-01	2.3387826-01
0.1	1.0891454-01	1.0828620-01	1.0770334-01

β \ i	6.0000000-01	7.0000000-01	8.0000000-01
0.99	1.2017381 01	9.1162871 00	7.0860769 00
0.9	3.5297849 00	3.1285130 00	2.7994027 00
0.8	2.1432908 00	1.9777101 00	1.8360315 00
0.7	1.4835943 00	1.3994387 00	1.3257254 00
0.6	1.0709446 00	1.0253631 00	9.8479468-01
0.5	7.7857527-01	7.5379497-01	7.3147346-01
0.4	5.5591678-01	5.4300373-01	5.3126012-01
0.3	3.7813839-01	3.7205713-01	3.6648265-01
0.2	2.3137342-01	2.2906287-01	2.2693058-01
0.1	1.0716253-01	1.0666063-01	1.0619474-01

TABLE 1 (Cont'd).

β	i 9.0000000-01	1.0000000 00	1.5000000 00
0.99	5.6444008 00	4.6051702 00	2.2716601 00
0.9	2.5278870 00	2.3025851 00	1.6144385 00
0.8	1.7143437 00	1.6094379 00	1.2585704 00
0.7	1.2609840 00	1.2039728 00	1.0031228 00
0.6	9.4861628-01	9.1629073-01	7.9820885-01
0.5	7.1133740-01	6.9314718-01	6.2483702-01
0.4	5.2056880-01	5.1082562-01	4.7341217-01
0.3	3.6136904-01	3.5667494-01	3.3831110-01
0.2	2.2496191-01	2.2314355-01	2.1591554-01
0.1	1.0576219-01	1.0536052-01	1.0374145-01

β	i 2.0000000 00	2.5000000 00	3.0000000 00
0.99	1.5886255 00	1.3175394 00	1.1858329 00
0.9	1.2997147 00	1.1290030 00	1.0496590 00
0.8	1.0747946 00	9.7168653-01	9.1060586-01
0.7	8.8937762-01	8.2179287-01	7.8006393-01
0.6	7.2758631-01	6.8385316-01	6.5600251-01
0.5	5.8224053-01	5.5499728-01	5.3721319-01
0.4	4.4928297-01	4.3343732-01	4.2287782-01
0.3	3.2612951-01	3.1794897-01	3.1240018-01
0.2	2.1100378-01	2.0764083-01	2.0532420-01
0.1	1.0261779-01	1.0183523-01	1.0128868-01

β	i 3.5000000 00	4.0000000 00
0.99	1.1133741 00	1.0703241 00
0.9	9.9677127-01	9.6400537-01
0.8	8.7287681-01	8.4882119-01
0.7	7.5351168-01	7.3621724-01
0.6	6.3786795-01	6.2585167-01
0.5	5.2541231-01	5.1747906-01
0.4	4.1575747-01	4.1091056-01
0.3	3.0860595-01	3.0599454-01
0.2	2.0372038-01	2.0260558-01
0.1	1.0090609-01	1.0063775-01

THE QUALIFICATION OF A REVERBERATION ROOM FOR PURE-TONE SOUND POWER MEASUREMENTS

by

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ABSTRACT

This paper describes the qualification of a 607 m^3 reverberation room at the CSIRO Division of Building Research for pure-tone sound power measurements according to International Standard 3742. To qualify the reverberation room it was necessary to design and construct a large rotating diffuser, to install a rotating microphone boom, and to install low frequency absorbers. The qualification procedure is outlined and the results of four runs of the qualification procedure will be presented. The modification of the rotating diffuser to reduce the noise radiated by it into the reverberation room is described.

INTRODUCTION

Until recently sound power measurements in reverberation rooms were restricted to measurements on sound sources which emitted only broad -band noise. However, during the sixties and seventies our knowledge of the nature of reverberant sound fields was greatly expanded. In 1975, the International Organization for Standardization published a Standard [1] which allowed sound power measurements on discrete-frequency and narrowband sources in reverberation rooms. This Standard requires the use of a large number of source positions or the use of a reverberation room which passes the qualification procedure laid down in the Standard. It is now known that multiple source positions will not necessarily solve all the problems, and hence it is desirable that all reverberation rooms which are to be used for sound power measurements should pass the qualification procedure.

There are two problems to be overcome in the measurement of the sound power of a pure-tone sound source in a reverberation room. The first problem is the measurement of the mean square sound pressure of the standing wave interference pattern set up in the room by the sound source. Unlike a broad-band reverberant field, whose mean square pressure varies only slightly with position, a pure-tone reverberant field has a mean square pressure which varies greatly with position. This problem can be overcome by using a large number of microphone positions or by using a moving microphone.

This problem applies at all frequencies since the spatial variance of the mean square sound pressure is only slightly greater at low frequencies than at high frequencies. However, it is easier to overcome at high frequencies since the number of equivalent independent microphone positions given by a moving microphone is equal to the number of half wavelengths covered by its traverse, and the wavelength of sound is inversely proportional to its frequency. With multiple microphones, independent microphone positions must be at least a half wavelength apart so that it is possible to fit more independent microphone positions into a reverberation room for meaurements at high frequencies.

The second problem is the large variation of input impedance presented to a pure-tone sound source by a reverberation room as a function of source position, frequency and room shape. This variation is due to low values of modal overlap. The modal overlap is the product of modal bandwidth and modal density. The modal bandwidth is inversely proportional to the reverberation time of the room and the modal density is proportional to the square of the frequency. Thus the variation of input impedance is only a problem at low frequencies.

This problem can be overcome in three different ways. The first solution is to use a large moving diffuser to modify the "shape" of the reverberation room and take average values with different room "shapes". The second solution is to add low frequency absorbers to the room. This decreases the low frequency reverberation time and thus increases the modal bandwidth and hence the modal overlap at low frequencies. The third solution is the use of multiple source positions. This solution only partially solves the problem since it reduces the spatial component of the variation but not the frequency and room shape components.

THE QUALIFICATION PROCEDURE

The pure-tone qualification procedure requires the excitation of the reverberation room at twenty-five different frequencies in each third-octave band using a loudspeaker whose near-field sound pressure level response has previously been measured at the same twenty-five frequencies when placed on a reflecting floor in an anechoic room. At each frequency, the mean square sound pressure level is measured and corrected using the near-field pressure response of the loudspeaker. The standard deviation of these twenty-five corrected sound levels is then calculated. If the standard deviation for a third-octave band is less than the value tabulated for that band in the Standard then the room passes the qualification procedure in that third-octave band.

The Standard states that if a continuous microphone traverse of length ℓ metres is used in a room of volume V cubic metres the qualification procedure needs to be carried out only at frequencies below the larger of $6000/\ell$ and $5000V^{-1/3}$. We were seeking to qualify the large reverberation room at the CSIRO Division of Building Research which has a volume of 607 cubic metres. We rotated the microphone in a circle of radius 2.2 m, a radius determined by the standard length of cable on a Bruel and Kjaer microphone preamplifier. Initially, we used an arm mounted on a Bruel and Kjaer turntable to rotate the microphone with a period of 80 seconds, but it was necessary to mount the microphone power supply upon the turntable because of the limited number of sliprings in the turntable. Later, use of a Bruel and Kjaer rotating microphone boom enabled the microphone preamplifier to be powered from the control room and a shorter rotation period of 32 seconds to be used. There was no advantage in using the fastest rotation speed of the microphone boom since the Standard requires that the sound pressure level measurements in the lowest three third-octave bands be made over at least 30 seconds.

The two limiting frequencies mentioned at the start of the last paragraph are 434 and 590 hertz. Since 590 Hz is in the 630 Hz third-octave band, our first attempt at qualification was made in the 100 to 630 Hz third-octave bands. Later measurements also included the 800 Hz third-octave band. The qualification test is a lengthy procedure. If measurements are made in the ten third-octave bands from 100 Hz to 800 Hz, then a total of 250 measurements have to be made in both the reverberation room and the anechoic room. If we average over 32 seconds and allow 28 seconds for the operator to set the frequency to within 0.3% of the required value, which is what the Standard demands, then a single run takes 500 minutes which is 8.3 hours. Use of the turntable with a period of 80 seconds raises the time to 15 hours.

On our first attempt the reverberation room qualified in the 500 and 630 Hz third-octave bands but not in the seven lower third-octave bands. We then built a large rotating diffuser consisting of a rectangular panel 2.25 m high and 4 m wide, rotating about its vertical axis of symmetry with a period of 32 seconds. It was built by covering a metal frame with 10 mm thick plywood, giving a surface density greater than 5 kg/m^2. With the rotating diffuser, the room qualified in all third-octave bands except the 100, 125 and 200 Hz third-octave bands.

We then added three low frequency absorbers to the room. These absorbers measured 1.8 x 1.4 x 0.15 m and were filled with 85 kg/m^3 mineral wool. The back face was 6 mm hardboard and the front face was 2 layers of 0.2 mm polythene. In this state, the room qualified in all the third-octave bands tested.

With the diffuser stationary and the low frequency absorbers still in the room, the room did not qualify in the 100 or 200 Hz bands and only just qualified in the 160 Hz band.

The results of these four qualification tests and the qualification limits are shown in Figure 1. We conclude that the rotating diffuser and the low frequency absorbers both separately improve the performance of the room, and that both are needed to qualify the room for pure-tone soundpower measurement.

NOISE REDUCTION OF ROTATING DIFFUSER

Initially, we had trouble with noise from the rotating diffuser, but this problem was eventually overcome. A 60 tooth sprocket fixed on the diffuser shaft is driven by an 18 tooth bicycle freewheel via a bicycle chain. The freewheel is used to stop overrun destroying the 40:1 reduction worm drive gearbox which drives the freewheel. The reduction gearbox requires an input speed of 250 rpm. The shaft has a slotted disk with 60 slots. The slotted disk rotates between a light emitting diode and a phototransistor which gives a 250 Hz electrical signal when the speed of the shaft is correct.

The 250 rpm speed was initially provided by a V-belt drive from a continuously variable speed gearbox and a half horse power 1440 rpm induction motor mounted on the base of the diffuser. This was a noisy arrangement which we quietened by replacing the variable speed gearbox and induction motor with a DC electric motor whose speed was controlled by varying the armature voltage. To reduce brush noise this motor was run at 83 rpm by using a 225 mm pulley on the motor shaft and a 75 mm pulley on the 40:1 reduction gearbox. This made the diffuser much quieter (see Figure 2), and the qualification tests described in this paper were conducted with the diffuser in this configuration.

Recently we have reduced the noise from the rotating diffuser still further by placing the 40:1 reduction gearbox and the DC electric motor outside the reverberation room. The connection between the bicycle freewheel and the 40:1 reduction gearbox is made with a 13 mm diameter flexible drive shaft through an existing hole in the 300 mm concrete wall of the reverberation room. The DC motor is coupled directly to the reduction gearbox and run at 250 rpm since there is now no need to control the brush noise.

The octave band sound power levels of the rotating diffuser with its three different drives are shown in Figure 2, together with the background octave band sound power levels due to the acoustical background noise in the reverberation room and the electrical background noise in the measuring system. It should be pointed out that these background levels are dependent on the ambient noise outside the reverberation room.

The noise from the rotating diffuser with the flexible drive is now so low that there are very few measurements with which it would interfere and no further noise reduction measures are planned.

CONCLUSION

The $607m^3$ reverberation room at the CSIRO Division of Building Research has been qualified for pure-tone sound power measurements according to International Standard 3742. The initial problem of high background noise-level caused by the rotating diffuser, which is necessary to qualify the room, has been solved by changing to a quieter drive system for the diffuser.

REFERENCE

[1] International Standard. ISO 3742-1975, Acoustics - Determination of sound power levels of noise sources - Precision methods for discrete-frequency and narrow-band sources in reverberation rooms.

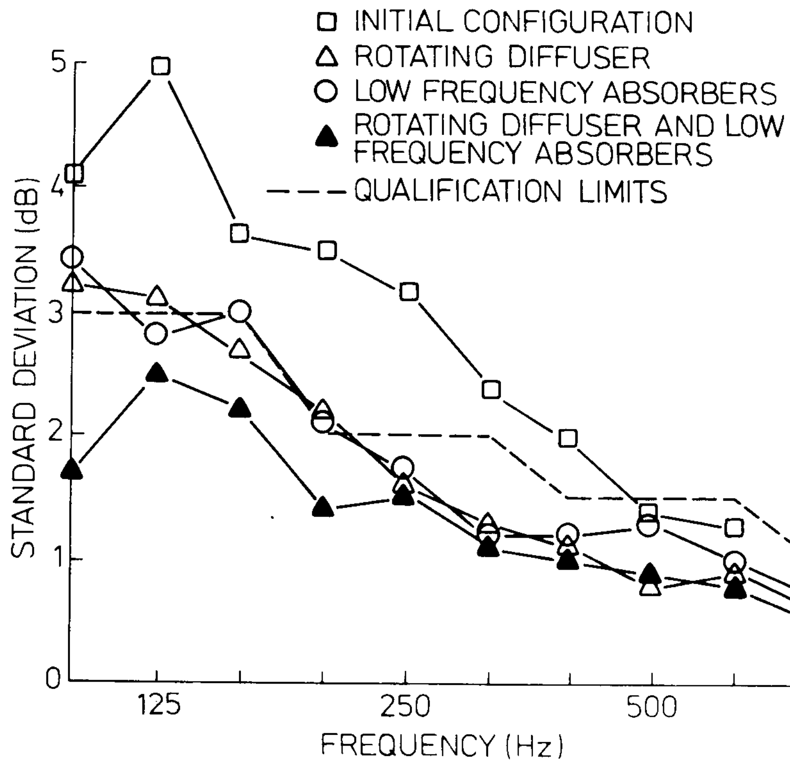

Figure 1. Standard deviations obtained in four different attempts to qualify a 607m^3 reverberation room for pure-tone sound power measurements. Also shown are the upper limits of standard deviation allowed if the room is to qualify.

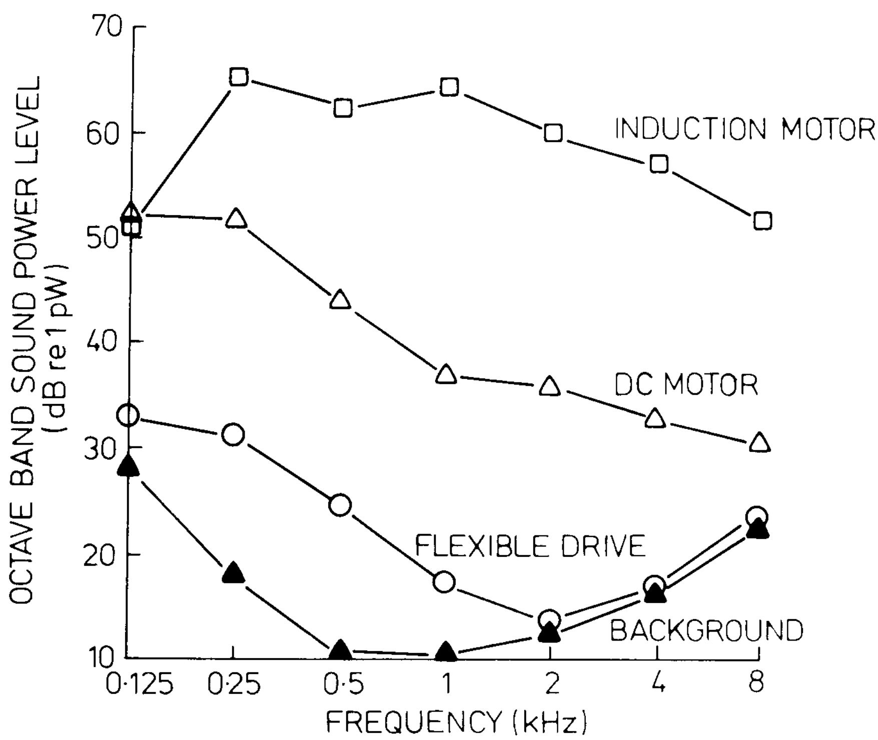

Figure 2. Octave band sound power levels of the rotating diffuser with three different drive systems. Also shown are the equivalent octave band sound power levels of the background electrical and acoustical noise.

A UNIFIED THEORY OF SOUND ABSORPTION IN FIBROUS POROUS MATERIALS

by

David Alan Bies

ABSTRACT

It will be shown that two simply related analytic expression exist in terms of the fiber diameter, bulk density and gas properties which allow calculation of all acoustical properties of fibrous porous materials and quantitative prediction of results for all common uses.

INTRODUCTION

Rayleigh first considered the problem of sound absorption by a porous medium. He considered viscous losses of sound propagating in a single narrow pore. Later Zwikker and Kosten greatly extended the analysis to include not only viscous but thermal effects as well.[1] They considered a model composed of an assembly of narrow pores and they introduced the concept of the structure factor, to be determined empirically, as a means for averaging the effect of pores oriented randomly with respect to an advancing sound wave. Most importantly they showed how the problem could be separated into two distinct parts; a viscous part, that of determining a complex density, and a thermal part, that of determining a complex compressibility

In more recent times Bies studied the acoustical properties of steel wool and he showed that the complex density was a well defined function of Rayleigh's parameter $r^2\rho_o f/\mu$. His effort to define the complex compressibility was not successful but he did show that it should scale on Rayleigh's parameter multiplied by an expression dependent upon the ratio of specific heats.[2] (Symbols are defined at the end of the text.)

Still later Delaney and Bazley investigated the acoustical properties of a large range of commercially available fibrous porous materials for a single gas (air) at room temperature. They used a scaling parameter of $\rho_o f/R_1$ and they determined well defined empirical expressions for the complex density and the complex compressilility of their gas-porous- material systems. They provided a very large collection of data for a range of their frequency parameter from 0.01 to 1.0 which may be used for comparison with any theoretical prediction scheme.[3]

In the following paper an expression for the flow resistance of a fibrous porous material will be derived which depends upon one empirical number. The expression shows explicitly the dependence of flow resistance on the fiber diameter, bulk density and gas properties and these dependencies are confirmed by experiment. Following that it will be shown that the microscopic scaling parameter of Rayleigh is directly related to the macroscopic description in terms of the physical details of a small scale model without the necessity, as encountered by Zwikker and Kosten, of having to introduce the empirical structure factor for this purpose.

The problem addressed by Zwikker and Kosten is reconsidered. It is shown that in dimensionless form the differential equations and boundary conditions for the complex density and complex compressibility are identical. The scaling parameter for the complex compressibility is $f\rho_o C_p/\beta$ which, however, is simply related to Rayleigh's scaling parameter. Finally it is shown that two simply related functions exist in terms of which the complex density and complex compressibility may be written. The characteristic impendance and complex propagation constant in turn may be written in terms of the latter quantities so that finally analytic expressions can be written for all quantities.

In this paper the characteristic functions are determined empirically from complex density data. However, it is argued that it should be possible to derive analytically the equations as well as to determine the one unknown constant of the flow resistance expression from consideration of an appropriate model which fits the boundary conditions. Such models are currently under investigation.

FLOW RESISTIVITY

It has been shown that a general expression for the flow resistance of a fibrous porous material including both linear and nonlinear behaviour may be written as [4]

$$Rd/\ell\rho_o c = (\mu/d\rho c)\, f_1(\rho_m/\rho_f) + (u/c)\, f_2(\rho_m/\rho_f) \tag{1}$$

Following convention we will define the flow resistance per unit length in the limit of very small particle velocity as the flow resistivity R_1.

$$R_1 = \lim_{u\to o} (R/\ell) \tag{2}$$

Empirically the flow resistivity is found to depend upon bulk density, fiber diameter and gas properties as follows.

$$R_1 d/\rho_o c = 27.3\, (\mu/d\rho c)\, (\rho_m/\rho_f)^{1.53} \tag{3}$$

By straight forward physical argument it may be shown that the following expression holds for the flow resistivity.

$$R_1 d/\rho_o c = (\sqrt{2}K/\pi)(\mu/d\rho c)(\rho_m/\rho_f)^{3/2}(1 - \rho_m/\rho_f)^{-3/2} \tag{4}$$

If the dimensionless bulk density ρ_m/ρ_f is very much less than unity, which is the case for which equation (3) holds, then equation (4) becomes a very good approximation to equation (3) provided the unknown constant K is set equal to 3.03.

SCALING PARAMETERS

We consider a plane surface embedded in a fiberous material. By physical argument it may be shown that the characteristic dimension r_c of unblocked area per penetrating fiber in the surface is

$$r_c = d/\sqrt{2})(\rho_f/\rho_m)^{1/2}(1 - \rho_m/\rho_f)^{1/2}\,. \tag{5}$$

In a unit length ℓ of the material the volume to surface area ratio V/S is

$$V/S = (\pi r_c^2 \ell / 2\pi r_c \ell F) = r_c/2F \quad , \tag{6}$$

where F is a number less than one. On the other hand in a fibrous material the volume to surface area ratio V/S is

$$V/S = (d/4)(\rho_f/\rho_m) \tag{7}$$

Thus the fraction F of unblocked area per-penetrating-fiber perimeter bounded by a fiber surface is from equations (5), (6) and (7)

$$F = \sqrt{2}\,(\rho_m/\rho_f)(\rho_f/\rho_m - 1)^{\frac{1}{2}} \tag{8}$$

We now consider possible equality of scaling parameters

$$\rho_o f r^2/\mu = \rho_o f/R_1 \tag{9}$$

Substitution of equation (4) into equation (9) gives after a bit of algebra

$$r^2 = (4\pi/K)\left(\frac{d}{4}\,\frac{\rho_f}{\rho_m}\right)^2 \left[\sqrt{2}\,\left(\frac{\rho_m}{\rho_f}\right)\left(\frac{\rho_f}{\rho_m} - 1\right)^{\frac{1}{2}}\right]\left[1 - \frac{\rho_m}{\rho_f}\right] \tag{10}$$

Consideration of equation (10) and use of equations (7) and (8) allows the following to be written

$$r^2 = (4\pi/K)(V/S)^2\,[F]\,[u/v] \tag{11}$$

The last term has been rewritten based on the observation that the volume velocity between fibers v must be greater than the external volume velocity per unit area u because of the presence of the fibers. The last term shows that the effect of finite porosity is accounted for in the scaling parameter and need not be considered explicitly. Equation (11) shows that the Rayleigh characteristic dimension r is a constant of the fibrous material system.

MEASURABLE QUANTITIES

As shown by Zwikker and Kosten the characteristic impendance of a porous material may be written in terms of a complex density ρ and a complex compressibility κ as follows

$$Z/\rho_o c = \sqrt{\kappa\rho} \quad . \tag{12}$$

On the other hand the complex propagation constant may be written in terms of the same quantities as follows

$$(c\ /\omega)(2\pi/\lambda_m - i\alpha_m) = \sqrt{\rho/\kappa} \quad . \tag{13}$$

The quantities on the left of equations (12) and (13) may be measured using an impendance tube. For example using porous samples of finite thickness but alternately rigidly backed and backed with a quarter wave cavity the quantities may be determined from measurement and analysis of the standing wave in front of the sample. Alternatively if a deep sample is used so that internal reflections from the sample termination may be neglected then standing wave analysis and direct measurements within the sample will suffice. In any case as shown by equations (12) and (13) the quantities ρ and κ may be calculated from physical measurements. Alternatively the equations show how measurable quantities may be predicted in terms of the complex density and complex compressibility and as shown elsewhere these quantities are sufficient to completely determine analytically all of the common acoustical uses of such materials.

COMPLEX DENSITY

The Navier-Stokes equation simplified to one dimensional viscous incompressible flow is

$$\rho_o(\partial v/\partial t) = - (\partial P/\partial z) + \mu\nabla^2 v \tag{14}$$

We assume that the volume velocity v is dependent upon frequency as $\exp(i\omega t)$ then equation (14) takes the following form.

$$\nabla^2 v = (i\omega\rho_o/\mu)(v - i/\omega\rho_o)(\partial P/\partial z)) \tag{15}$$

We introduce the following definitions.

$$\sigma = - (1 + i\omega\rho_o v/(\partial P/\partial z)) \tag{16}$$

$$\delta = \sqrt{\omega\rho_o/\mu} \tag{17}$$

Substitution of equations (16) and (17) in equation (15) gives the following equation for the reduced velocity σ.

$$\nabla^2\sigma - i\delta^2\sigma = 0 \tag{18}$$

If we consider a volume bounded in part by fibers and in between by an imaginary boundary then at the fiber surfaces we require the volume velocity v to be zero or

$$\sigma = - 1 \tag{19}$$

On the other hand at the neutral boundary between fibers we require that the gradient of the velocity normal to the boundary be zero. Thus at the neutral boundary between fibers we require

$$\partial\sigma/\partial n = 0 \tag{20}$$

Finally it may be shown that the complex density takes the form

$$\rho = 1/(1 + \overline{\sigma}) \tag{21}$$

where

$$\overline{\sigma} = \overline{\sigma}(r_B\delta) \tag{22}$$

The dimensional length r_B of equation (22) is presumed equal to r_c of equation (5) or related directly to it and also related directly to r of equation (11).

COMPLEX COMPRESSIBILITY

The thermal equation for a confined gas subjected to a time varying pressure is [6]

$$\nabla^2 T = i\alpha^2(T - K) \tag{23}$$

where

$$K = [PT_o(\gamma - 1)]/P_o\gamma \tag{24}$$

and

$$\alpha = \sqrt{\omega\rho C_p/\beta} \tag{25}$$

In equations (23) and (24) it is assumed that both pressure P and temperature T vary with time as $\exp(i\omega t)$.

To proceed we introduce the following definition.

$$\tau = (TP_o\gamma/PT_o(\gamma - 1)) - 1 \tag{26}$$

We also note in passing that the following approximate relation holds.[2]

$$\alpha = \delta\sqrt{4\gamma/(9\gamma - 5)} \tag{27}$$

Substitution of equation (26) in equation (23) gives the following differential equation for the reduced temperature τ.

$$\nabla^2\tau - i\alpha^2\tau = 0 \tag{28}$$

We again consider a volume bounded in part by fibers and in part by a imaginary boundary between fibers. At the surfaces of the fibers we require that the variational temperature T be zero so that at the fiber surfaces

$$\tau = -1 \tag{29}$$

On the other hold at the neutral surface between fibers we require that the normal gradient of the variational temperature be zero. Thus

$$\partial\tau/\partial n = 0 \tag{30}$$

Finally it may be shown that the expression for the complex compressibility of the gas contained in our small imaginary volume may be written as follows:

$$\kappa = 1/[1 + (1 - \gamma)\bar{\tau}] \tag{31}$$

where

$$\bar{\tau} = \bar{\tau}(r_B\alpha) \tag{32}$$

We note the formal identity of equations (18) through (22) with equations (28) through (32). This suggests that a solution for the density will define a solution for the compressibility or vice versa and as will be shown this is indeed the case.

CHARACTERISTIC FUNCTIONS

We assume that the quantities $\bar{\sigma}$ of equation (22) and $\bar{\tau}$ of equation (31) may be written as follows.

$$\bar{\sigma}\,(r_B\delta) = a_1(x) + i\,b_1(x) \tag{33}$$

and

$$\bar{\tau}\,(r_B\alpha) = a_2(x) + i\,b_2(x) \tag{34}$$

where

$$x = \rho_o f/R_1$$

We may write for the complex density $|\rho|$ exp $i\phi$ the following:

$$|\rho| = [(1 + a_1)^2 + b_1{}^2]^{-\frac{1}{2}} \tag{36}$$

$$\phi = \tan^{-1}[-\,b_1/(1 + a_1)] \tag{37}$$

Similarly we may write for the complex compressibility $|\kappa|$exp $i\theta$

$$|\kappa| = [(1 - (\gamma - 1)a_2)^2 + (\gamma - 1)^2 b_2^2]^{-\frac{1}{2}} \tag{38}$$

$$\theta = \tan^{-1}\,[(\gamma - 1)b_2/(1 - (\gamma - 1)a_2)] \tag{39}$$

Delaney and Bazley provide empirical straight line approximations for their data which may readily be used to calculate the complex density and compressibility and thus the quantities a_1, b_1, a_2 and b_2. It is found that to a very good approximation the following is true,

$$a_2 = a_1 (0.592) \tag{40}$$

and

$$b_2 = b_1$$

Alternatively an empirical equation may be constructed which accurately describes the data for the complex density and has the correct limiting behaviour at both high and low frequencies. This equation may be used to calculate a_1 and b_1. These quantities may then be used to calculate all other quantities. Comparisons between the data of Delaney and Bazley and the predictions of these equations will be shown to be quite good during the oral presentation.

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SYMBOLS

c	Speed of sound	u	Unit surface Nolume velocity
C_p	Constant pressure Specific heat	v	Internal volume volocity
d	Fiber Diameter	V	Volume
f	Frequency	z	Length co-ordinate
ℓ	Material thickness	Z	Characteristic impedance
P	Acoustic pressure	α_m	Porous medium attenuation constant
P_o	Static pressure	β	Thermal conductivity
r, r_B, r_C	Charateristic radius	γ	Ratio of specific heats
R	Flow resistance	λ_m	Wavelength in porous medium
R_1	Flow resistivity	μ	Gas viscosity (air)
S	Surface area	ρ_o	Density of gas (air)
t	Time	ρ_f	Density of fibers
T	Temperature (variational part)	ρ_m	Fiberous material bulk density
T_o	Static temperature	$\omega = z\pi f$	Angular frequency

The Measurement of Noise Containing Impulsive Characteristics

by L.C. Kenna and J.A. Rose

National Acoustic Laboratories - Sydney

Abstract:-

The importance of impulsive noise and problems related to measuring it in ways which will permit assessment of its possible effects on people, particularly when it is embedded within a broad-band background noise are discussed, steps being taken to improve the situation are outlined and experimental results reported.

Introduction:-

Impulsive noise is of importance because of its potential to inflict damage to hearing and for its particularly annoying characteristics.

Most attempts to legislate for reduced noise exposure at the work-place or in the general community have placed impulsive noise either in the "too-hard" basket requiring undefined special equipment, or treated it with arbitrary corrections based on subjective evaluations.

Because of the wide diversity of characteristics of impulsive noise and the difference in levels of impulses which are just detectable and those which dominate the noise environment, we believe that arbitrary corrections are not appropriate for either field.

In an attempt to overcome these deficiencies N.A.L. has been researching the effects on people and developing related measurement systems.

Since only the peak level of impulsive noise is measurable with simple instrumentation and some events occur only rarely and without adequate warning, a number of approaches have been tried to capture impulsive signals so that critical parameters such as rise-time, rate of decay, repetition rate etc. can be determined.

A major problem concerning impulsive noise is that it is rarely found uncontaminated by other forms of noise and therefore not only do we need to know how its own characteristics affect people, but also how it combines with other types of noise to create overall effects.

The ultimate aim is therefore to derive relatively simple systems for measuring the overall characteristics of noise with sufficient precision to enable assessment of effects with reasonable accuracy.

Effect on Hearing

Legislation in Australia and the S.A.A. Hearing Conservation Code (AS1269-1979) are not specific in how to deal with impulse noise and the method offering most promise for objectively rating the possible damaging effects of such noise is that proposed by Atherley and Martin (ref. 1) in which they divide impulse noise into three categories based on the repetition rate of the impulses.

For rates exceeding 10 per second, they postulate that the noise may be treated as broad-band in the usual manner (i.e. SLM, "Slow"-response, eye-averaging, energy summation).

For rates between 1 per second and 10 per second, they propose two sub-categories depending on the rate of decay of sound between impulses. Under their proposed system these require a complex and tedious method of analysis. N.A.L. has developed an attachment to a peak reading sound level meter (ref. 2) which permits easy determination of this correction. Fig. 1 shows its performance characteristics.

Since hearing damage is thought to result from "A" weighted noise energy, it would seem logical to use instruments such as L_{Aeq} meters, dosemeters, noise analysers etc. to evaluate the potential for harm to hearing of impulsive/broad-band noise.

A few representative items were subjected to a series of 46 test runs during which impulses of varying peak level, duration and repetition rates were presented embedded in broad-band noise in a diffuse field. L_{Aeq} was varied between 89 and 98 with peak levels from 109 dB to 124 dB. These levels were within the claimed capacity of all instruments which were calibrated by normal methods and by white noise.

Items selected for test were a commercial statistical noise analyser, a N.A.L. long-term integrating device (ref. 3), two integrating noise dosemeters and two sampling dosemeters.

A common microphone fed the N.A.L. (A. & M.) system, which was used as a reference, the commercial analyser (sampling at 10 Hz with rms detector set at "slow" and "instant level") and the N.A.L. integrator while the other devices used their own individual microphones placed close to the reference microphone.

Table 1 below shows the differences between the reference device and the means of the various meter readings after periods ranging from 15 to 60 minutes.

Measurement System	Broad-band Ambient Level		
	35 dBA	80 dBA	90 dBA
Commercial Noise Analyser	- 5.7	- 6.0	- 4.3
N.A.L. Long-term Integrator	- 5.8	- 6.0	- 4.6
Integrating Dosemeter No. 1	- 8.4	- 8.3	- 6.3
Integrating Dosemeter No. 2	- 6.4	- 6.2	- 4.9
Sampling Dosemeter No. 1	- 19.2	- 12.1	- 6.6
Sampling Dosemeter No. 2	- 17.7	- 10.6	- 4.6

All the test items give results which are lower than the reference device and therefore use of these figures could lead to increased possibility of hearing damage if the situation was close to critical limits.

There is good agreement between the commercial noise analyser, the N.A.L. long-term integrator and one of the integrating dosemeters but the other I/D would be near to tolerance limits. The sampling dosemeters (4 times/second) were far too low and to investigate whether the slow sampling rate was the cause, the commercial analyser was reduced to sampling twice per second without changing the result.

The above test is not exhaustive but is presented to illustrate the degree of errors which can occur when using devices which are primarily intended for broad-band noise to evaluate the combination of impulses and broad-band noise.

Effects on Communities

The position in relation to annoyance from impulsive noise suffers from all the deficiencies connected with effects on hearing but has in addition psychological and sociological factors.

Attempts to apply relatively simple systems to Commonwealth Governmental problems have been unsatisfactory and we have decided to investigate problems where impulsive noise is clearly the major component of noise disturbing residential areas.

The first study involving both noise measurements and social survey has been completed (ref. 4) and a larger study using a wider variety of noise sources and residential areas is underway.

To enable unattended gathering and identification of noise impulses a system has been devised whereby the acoustic signal is delayed until a high-quality tape recorder starts up to register the signal then automatically shuts down (see Fig. 2). It is hoped that the noise data plus social survey correlations will enable the derivation of a system of wider application than presently available methods.

One of the few areas where this has been attempted concerns helicopter noise (ref. 5) which specifies a fairly complex noise measurement and analysis system to provide an adjustment of plus 0 to 6 dB to broad-band noise values to allow for the impulsive contribution to annoyance. Significantly, the latest version of I.S.O. Draft Proposal (1996 - Part 2) dealing with community noise gives two fairly simplistic methods for determining the presence of impulsive noise but does not specify how the results of these measurements are to be used to assess the effects on people.

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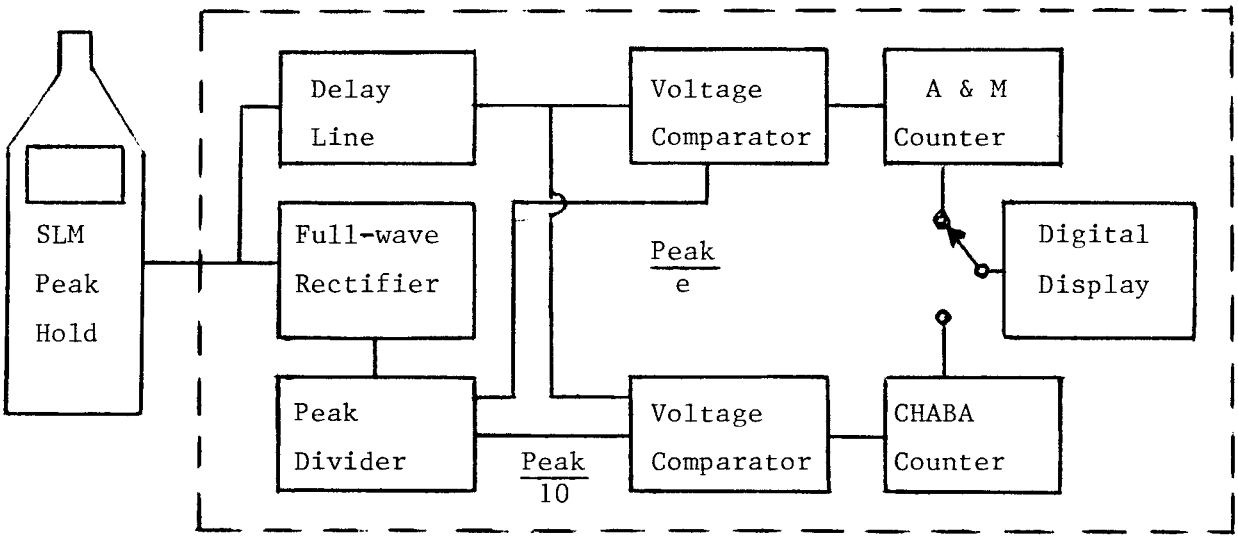
NAL IMPULSE DURATION METER FIGURE 1



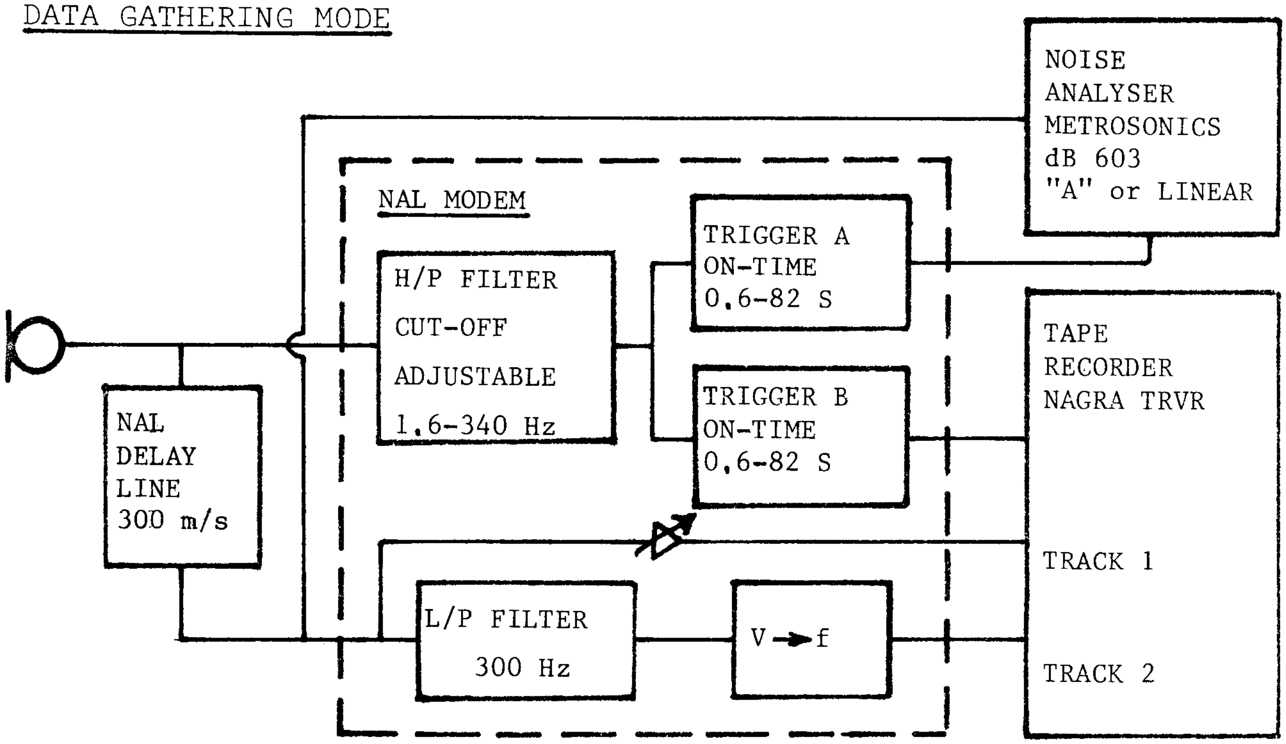
Operation

The peak level is read from the SLM which also permits range setting to ensure that the signal sent to the IDM is in its optimum dynamic range.

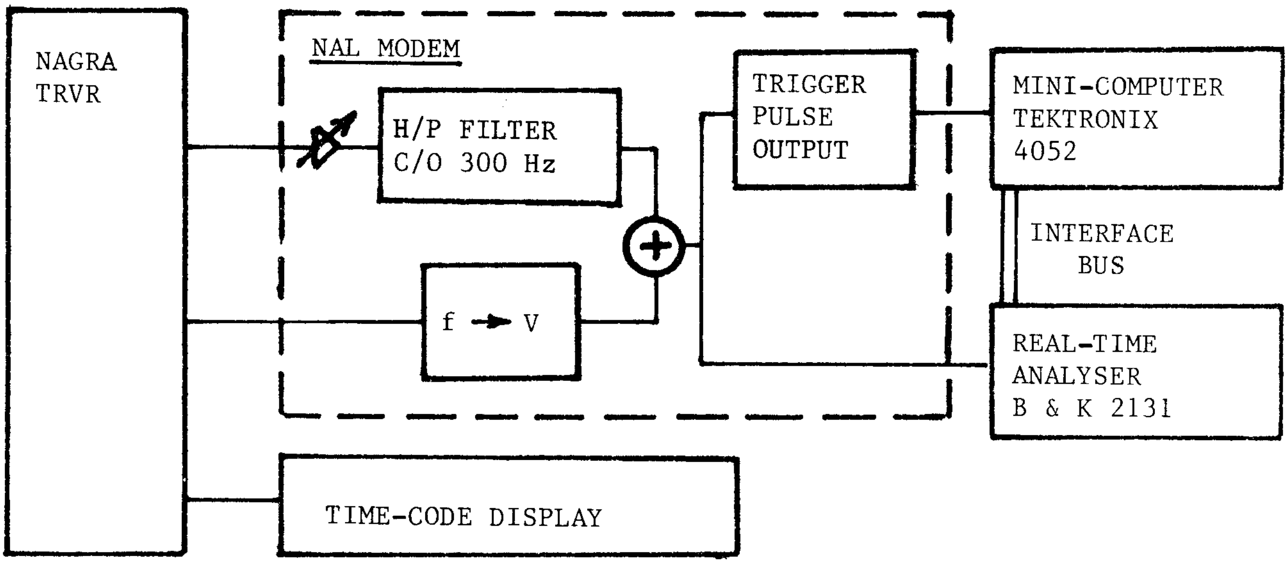
Within the IDM the signal divides with one channel delaying the original wave-form for later comparison with the output of the other channel in which the peak is divided by both "e" and 10. The digital counters record the time it takes for the peak to decay to the $\frac{P}{e}$ or $\frac{P}{10}$ levels respectively. The duration for whichever criterion is desired ie, A and M or CHABA is displayed digitally to enable easy determination of the exposure permissable per day for the impulse noise involved.

NAL IMPULSE NOISE MEASUREMENT SYSTEM FIGURE 2

DATA GATHERING MODE



ANALYSIS MODE



Operation:

The noise-analyser stores time-coded peak and sound-exposure levels, the H/P filter eliminates false-starts due to wind, the TRVR stores the delayed pulse overall on AM/FM with time-coding on Track 3.

Analysis combines AM and FM, recording start triggers the mini-computer which commands the real-time analyser to measure the energy in 1/3 octave bands over selected periods. All relevant site, instrumental and pulse data are kept for later processing.

Manual operation substitutes a peak-reading device for the dB 603, the TRVR is operated manually (no delay needed) but no replay a preset-delay permits the RTA to operate correctly.

"NOISE GENERATION BY WALL IMPACTS"

BY

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ABSTRACT:

Results are presented on the sound pressure waveforms radiated by impacts on brick and plasterboard walls, and are analysed to identify the main noise generation mechanisms. For valid objective comparisons of different impacts, the importance of the subjective response to annoying impulsive sounds is stressed, and deficiencies in the existing literature pointed out.

INTRODUCTION:

The trend to lightweight partition walls within commercial and domestic constructions gives increased emphasis to a little-studied type of intruding noise : noise generated by wall impacts and transmitted to the receiving space.

There is clear motivation to seek standardized tests for comparing the impact transmission behaviour of different constructions for rating purposes, as well as to improve the performance of deficient constructions given an understanding of the noise generation and transmission mechanisms. The problem has much in common with the transmission of footstep noise by floors, for which the existing tests are under active investigation and revision, but there are important differences in the constructions and the nature and variety of possible impacts.

Fundamental to the study is an understanding of how the ear responds to impulsive noise, and the quantification of the factors contributing to the annoyance of such sounds; the literature on these aspects is reviewed, and deficiencies highlighted.

Results obtained to date on the impact forces generated, the resultant wall responses, and the sound pressure signals observed, are presented. The results are interpreted in terms of existing models of impact noise generation and the important radiation mechanisms are identified.

Human Response to Impulse Noise.

Any intended standardized comparison of the noise generated by impacts on walls or floors requires objective measurements of sound pressure which correlate with the subjective response of humans to impulsive noise. We require the specification of such instrument parameters as frequency weighting, averaging time and peak level or energy level detection to be derived from pschoacoustic experiments.

Regrettably, much of the available literature deals with idealized transient waveforms and yields information about pulse duration, envelope wave peak and rise time, or continuous noise level, which is not easily transferred to practical measurements.

There is an important distinction between the damage and annoyance due to impulse noise, since the brain's response time affects the latter. This is reflected in the different averaging times used for objective assessments of impulse loudness and hearing risk. There is also the complication that long impulses, such as sonic booms, allow the brain to cause tensor muscle contraction in response to intense stimuli; there may need to be different objective measurements of long and short impulses, relative to the neurological response time.

The time constant for objective loudness assessment has been standardized to be 35 ms despite a wide spread in experimental values (ref 3.) However a recent study (ref 4) favours a time constant of 5 ms, after a search for the best correlation between peak SPL and subjective response.

The literature is also equivocal on whether subjective loudness best correlates with the total energy or the peak level of the impulse; the meaning of peak level is not consistent across the different studies but should probably be taken as the peak of the averaged signal. The studies favouring the energy hypothesis (e.g. ref 5) do so mainly on the basis of the impulse duration dependence, whilst the peak hypothesis is favoured by Schultz (ref 6), Kumagai et al (ref 4) and also neurological response studies on guinea pigs (ref 7).

Previous experiments also suffer from the alteration of their signal spectra whilst varying their rise and fall times or carrier frequency. In consequence there is little evidence favouring any particular frequency weighting for objective loudness assessment. Sonic boom and artillery noise studies tend to favour C weighting, but this may not be relevant for more common impulses which have shorter durations and cause less low frequency mechanical vibration.

Impact Noise Generation Mechanisms.

Akay (ref 1) has recently reviewed impact noise generation mechanisms and classified these into five different categories : air ejection, rigid body deceleration , radiation due to rapid surface deformation, pseudo steady state radiation and fracture radiation.

Commonly occurring wall impacts are unlikely to cause fracture, and air ejection can also be ignored since most impactors are at least partially rounded. For planar walls the rigid body radiation becomes modified by panel reflection, and this has been treated using an image source (ref 2) assuming negligible panel motion.

Wall impacts are inherently inelastic, energy being readily transferred to bending waves in the panel, and hence pseudo steady state radiation is expected to contribute significantly on both sides of the wall. However this ringing noise is more rapidly attenuated in typical walls than in metallic panels which have low structural damping.

RESULTS.

The measurements performed to date have focussed on typically occurring wall impacts with emphasis on those likely to cause greatest annoyance. A steel hammer of mass 0.3 kg has been instrumented to measure impact force and acceleration and used to impact both brick and stud plaster walls. Impact velocities are typically around 0.2 m/s.

The impact duration has been estimated from the force pulse duration to be around 0.5 ms for brick walls and 6 ms for plaster walls. However when a plaster wall is struck directly on a stud the duration decreases to around 2 ms and there are substantial modifications of the pressure waveform and spectrum as well.

A typical pressure waveform obtained on the impact side of a plaster wall is shown in Fig. 1. Considerable pseudo-steady state radiation is evident and is clearly quite complicated due to the large number of discontinuities of real partition walls. The ringing causes the observed sound pressure level to fall quite slowly after the initial peak, the level having only decreased by ~20 dB after 100 ms.

The pressure waveform commences with a large rarefaction pulse and must be generated by the rapid acceleration of the plaster panel in the vicinity of the impact point. There is no suggestion of any preceding compression pulse radiated by the deceleration of the impactor.

The pressure waveform obtained on the impact side of a brick wall is shown in Fig. 2. The sound pressure levels generated are around 20 dB lower than for plaster walls struck with the same impact velocity, and the spectrum peaks at around 2 KHz, as opposed to 200 Hz for plaster wall impacts. The initially observed pressure pulse is compressive, indicative of impactor radiation but pseudo steady state radiation is of comparable amplitude.

Sound pressure levels have been calculated using exponential averaging with time constants in the range 2 - 35 ms, for both plaster and brick wall impacts (Figs. 3 and 4). The peak levels attained depend strongly on the time constant : for example, changing the time constant from 35 ms to the more recently favoured 5 ms increases the peak by around 10 dB. It is clearly important to establish the correct time constant, as well as other factors mentioned previously, to allow further progress in the objective comparison of impact noise.

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FIG.1 PLASTER WALL, SOUND PRESSURE (I)

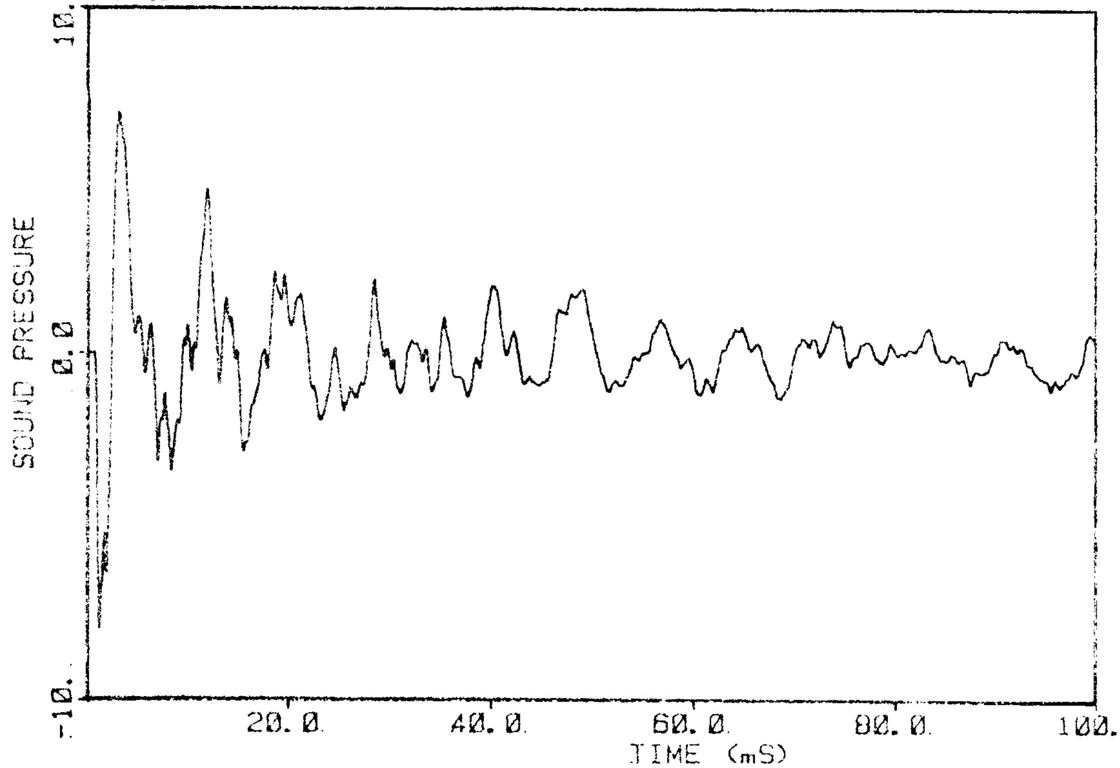


FIG.2 BRICK WALL, SOUND PRESSURE (I)

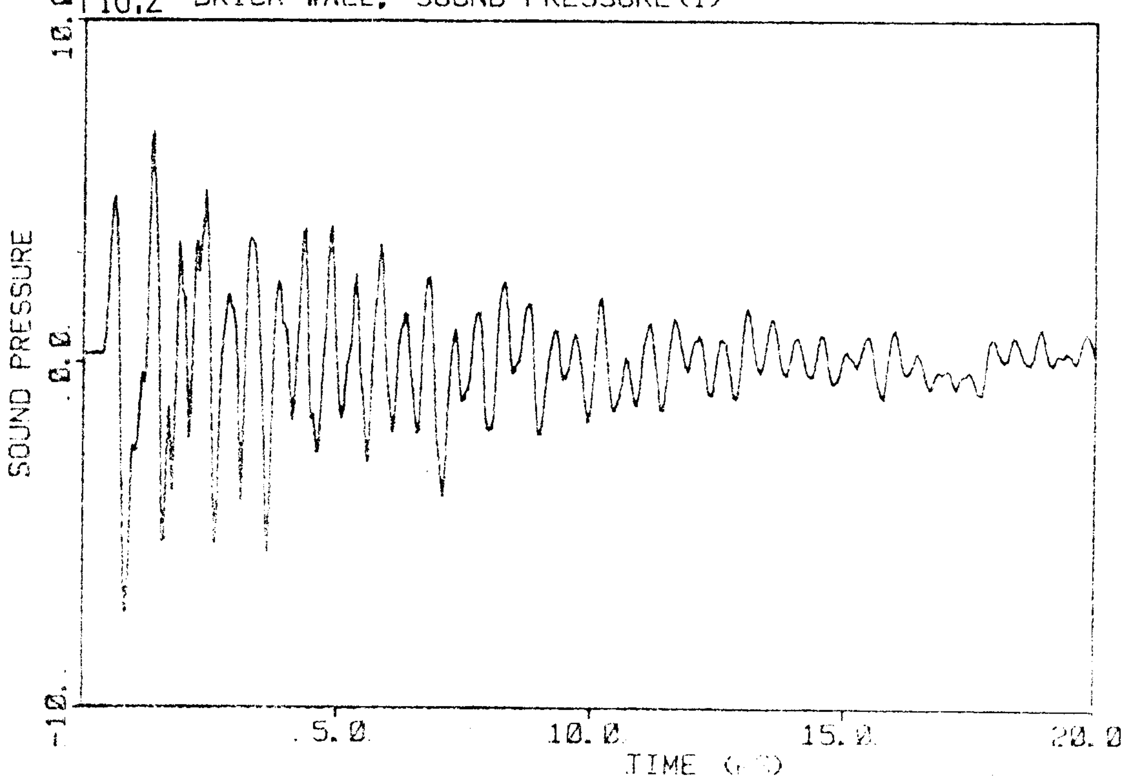


FIG.3 AV -BRICK WALL, SOUND PRESSURE(I)

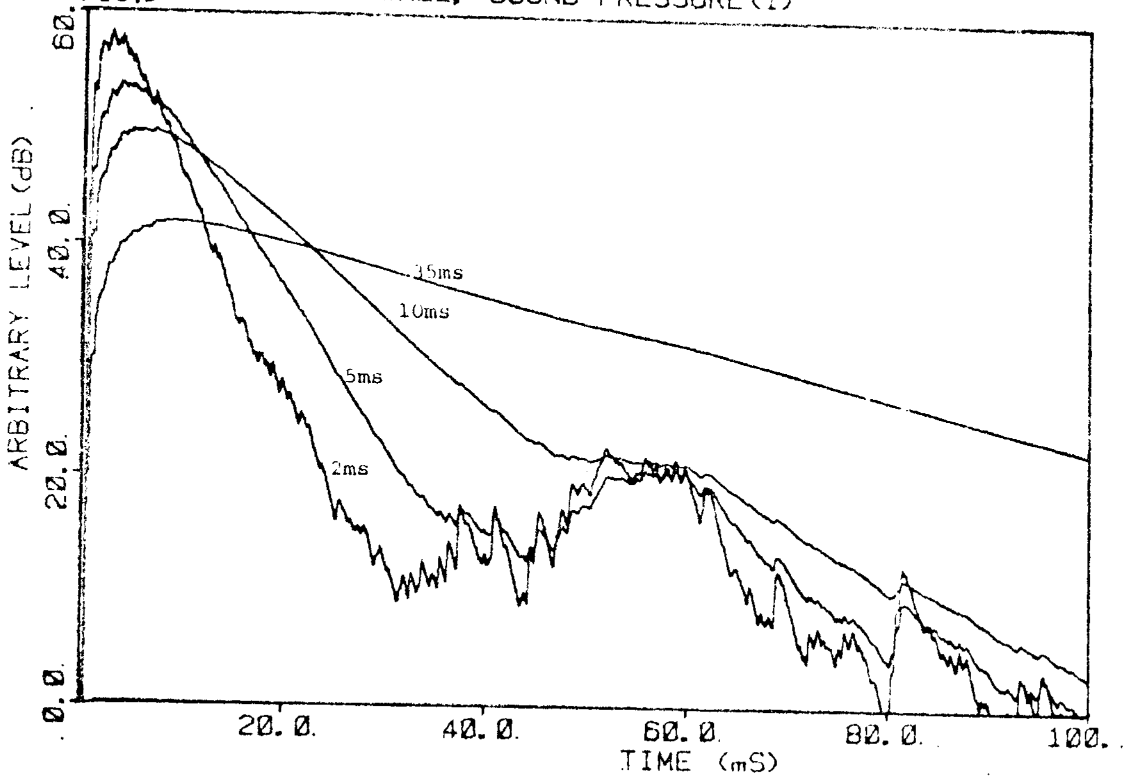
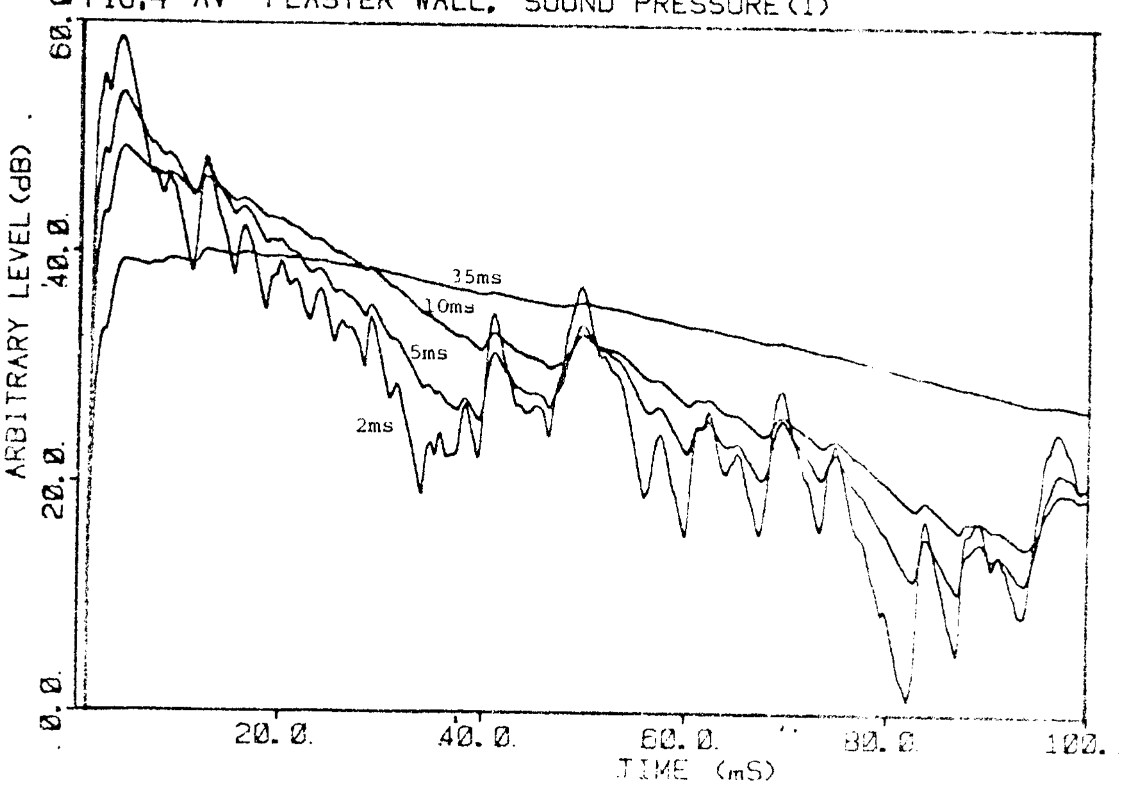


FIG.4 AV -PLASTER WALL, SOUND PRESSURE(I)



ABSORPTIVITY AND THE EFFECTS OF PLACEMENT

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MODEL EXPERIMENTS WERE DESIGNED TO INVESTIGATE OBSERVED ANOMALIES IN REVERBERATION TIMES WHICH SEEM TO BE ASSOCIATED WITH ABSORBENT PLACEMENT. THE RESULTS SHOW SYSTEMATIC DEPARTURE FROM SABINE ESTIMATES, RTS BEING DOUBLED OR EVEN TRIPLED BY EFFECTS OF PLACEMENT IN RELATION TO AXIAL LENGTHS. PRACTICAL PLACEMENT RULES & EFFICIENCY FACTORS ARE DESCRIBED.

INTRODUCTION These investigations were inspired by a realisation that the acoustic behaviour of rooms is not always consistent with expectation. For example, long corridors, with heavily absorbent ceilings and floors, can be very lively spaces. Even on analysis, some spaces which should be dead, because of the contained Sabines of absorbent, are nevertheless, quite reverberant. Sometimes these anomalies seem to be related to the placement rather than the total amount of absorbent material exposed.

Unfortunately the conventional room equations - except those we use for the calculation of room modes and eigentones - cannot cope with directions or position. They relegate all Room geometry to a simple Volume to Surface ratio.

The following work investigates anomalies, reviews some of the equations and recommends useful, practical procedures.

I would like to record my gratitude to the Commonwealth Experimental Building Station at Ryde for providing their equipment and their laboratory, also personally to Ted Weston, John Whitlock, Con Demos and many others for advice, encouragement and assistance with the following experiments.

AIM The aim of the investigation was (i) to isolate, identify and quantify room field effects due to placement of absorbent material and (ii) to test and subsequently analyse the efficiency of conventional and unconventional formulae as estimators of reverberation behaviour.

METHOD Model analysis was used. Most measurements were made in a rectangular box approx. 2100 x 1200 x 900 mm internally, constructed from 18 mm Pyneboard kindly donated by C.S.R. The box contained a loudspeaker and microphone.

1: BOX PARAMETERS

ITEM	IN AXIS XX	IN AXIS YY	IN AXIS ZZ	TOTALS
Axial Lengths	X = 2.13m	Y = 1.22m	Z = 0.91m	4.26m
Axial surfaces	2YZ= Sx	2XZ= Sy	2XY= Sz	
	= $2.23m^2$	= $3.90m^2$	= $5.20m^2$	$11.33m^2$
Volume	-	-	-	$2.36m^3$

NOTE THAT, IN ALL THAT FOLLOWS, X > Y > Z

The absorbent material used was 25 mm thick fibre glass batts applied in units of 0.125 m^2. Placement is indicated by the array matrices (p^X, q^Y, r^Z) - meaning P units arrayed on the surfaces perpendicular to axis X etc.etc. Such arrays are drawn from the parent group ("ALL p + q + r units").

APPARATUS

The sound source was 1/3 Octave pink noise at centre frequencies of 1250, 2000 and 3150 H_Z; chosen to be non-harmonic and to be eigentone free. Wavelengths were less than any box dimension but not less than 100 mm.

Measurements were made using CEBS' computer-controlled, integrated measuring and recording equipment. Each individual test run, once initiated, is automatically repeated until the continuously averaged reading of the Reverberation Time reaches a predetermined level of significance. Results, in full statistical detail are then printed out and the decay is charted.

In this way, it was possible to make approximately 8000 readings at 3 set frequencies on 34 basically different arrays before the equipment (designed for larger scale experiments!) finally bottomed out. The results were then subjected to intensive statistical analysis.

RESULTS As the graphs show, the results do not appear to conform to the Sabine formula.. Every group of n array samples produced a wide scatter (usually) about the Sabine curves. In the most extreme case, RT varied from about 300 up to almost 1000 milliseconds for the same amount of absorbent. On the basis of inspection alone, position of the absorbent seemed to exert a considerable effect in every case.

STATISTICAL TESTS. Several hypotheses were tested:-

1) IS THE SCATTER OF RESULTS MERELY RANDOM? Could the individual R/T readings for the several different array positions of (say) "n" pieces be merely random samples drawn from the pooled parent population of "ALL n"?

Student's "t" test was used to check each of the 99 array mean R/Ts for statistical independence. The results clearly reject the Null Hypothesis and show that:- Placement exerts a real and an extremely significant effect in every case.

2: SCATTER TEST

PROBABILITY - ESTIMATED FROM STUDENTS "T" - THAT THE SCATTER OF OBSERVED ARRAY RT'S ABOUT A PARENT GROUP MEAN WERE STOCHASTIC EFFECTS RATHER THAN SYSTEMATIC VARIATION DUE TO THE PLACEMENT.

Parent and arrays	at 1250 HZ: pr.stochastic	at 2000 HZ: pr.stochastic	at 3150 HZ: pr.stochast	Pooled .no.samples
"ALL 1": 3x3 Arrays	$10^{-10} \leqslant pr. \leqslant 10^{-5}$ 243 RTS	$10^{-10} \leqslant pr. \leqslant 0.5$ 256 RTS	$10^{-10} \leqslant pr. \leqslant 10^{-9}$ 164 RTS	663 RTS
"ALL 2": 3x6 Arrays	$10^{-10} \leqslant pr. \leqslant 10^{-5}$ 294 RTS	$10^{-10} \leqslant pr. \leqslant 0.05$ 364 RTS	$10^{-10} \leqslant pr. \leqslant 10^{-1}$ 435 RTS	1093 RTS
"ALL 3": 3x4 Arrays	$pr. \leqslant 10^{-10}$ 319 RTS	$pr. \leqslant 10^{-10}$ 216 RTS	$10^{-10} \leqslant pr. \leqslant 10^{-3}$ 191 RTS	726 RTS
"ALL 4": 3x3 Arrays	$10^{-5} \leqslant pr. \leqslant 10^{-2}$ 236 RTS	$10^{-10} \leqslant pr. \leqslant 0.01$ 280 RTS	$10^{-3} \leqslant pr. \leqslant 10^{-1}$ 300 RTS	816 RTS
"ALL 6": 3x7 Arrays	$pr. \leqslant 10^{-10}$ 618 RTS	$pr. \leqslant 10^{-10}$ 403 RTS	$10^{-10} \leqslant pr. \leqslant 0.7$ 265 RTS	1286 RTS
"ALL 9": 3x10 Arrays	$10^{-10} \leqslant pr. \leqslant 10^{-2}$ 1067 RTS	$10^{-10} \leqslant pr. \leqslant 0.01$ 809 RTS	$pr. \leqslant 10^{-10}$ 796 RTS	2672 RTS
ALL GROUPS: 3x33 Arrays	2777 RTS	2328 RTS	2151 RTS	7256 RTS

Note that the "t" ratio exceeds a value of 5 for 82 of the arrays; in only two cases are there less than 10 degrees of freedom. Consequently the placement effect must be rated as "Highly Significant" in more than 80% of all the observations.

2) ARE PLACEMENT EFFECTS SYSTEMATIC? Are there similar array-related effects operating at each of the three frequencies?

A Rank Concordance test was used because of its distribution-free flexibility and its freedom from the effects of absolute magnitude. The resultant Coefficient of Concordance ($W = 0.945$ on a scale range between 0 and 1) was then subjected to Snedecors F test for statistic significance. The result showed that there are extremely significant systematic effects due to placement which are independent of any frequency effect. (pr.appr. 10^{12}:1). See Table 4.

3) ARE PLACEMENT EFFECTS SYSTEMATIC WITHIN EACH "n" SAMPLE GROUP OF ARRAYS? This was difficult to test because of the small size of some sample groups such as "ALL 1" which, although based on more than 660 separate measurements, only contains 3 arrays. Consequently this is not a statistically significant group.

It was thus necessary for this test to regroup the arrays into four larger units as:- "ALL 1 plus ALL 2"; "ALL 3 plus ALL 4"; "ALL 6" and "ALL 9". The Concordance Test was then used again (with resultant coefficients of $W = 0.721$, 0.921, 0.859 and 0.793). The subsequent "F" tests showed that there are extremely significant systematic effects due to array placements within the above groups irrespective of any frequency effects. See Table 4.

4) DO THE EXPERIMENTS DEMONSTRATE ANY CONSISTENT PATTERN IN THE RESULTS? The absolute values (RTs in milliseconds) for each array of "n" fibre-glass units were converted to percentage R/T values relative to their associated Average R/T for "All n" groups. All frequencies were pooled.

These transformed relative values were then treated as a new, pooled, variate. This new variate was found to have a mean which differed by less than one half a Standard Error from the three original means.

The new relative variables were regrouped into the seven divisions of all three Singlets; all three Doublets and the Triplet group as tabled below. These groupings were found to display quasi-normal distribution about seven mean values - The X singlets were the most variable and had the highest mean Relative R/T while the Triplets were the most tightly grouped and had the lowest mean Relative R.T. Pearson Coefficients of Variability were calculated. See Table 3 and Diagram 4.

An Array Efficiency Factor was also established with a nominal unit value for the XYZ triplet arrays, grading to 0.51 for the inefficient X00 Singlets.

3: RELATIVE ARRAY EFFICIENCIES

ARRAY TYPE (STANDARDISED POOLED DATA)		NO.OF ARRAYS	NO.OF TEST RUNS	MEAN REVERB. TIME %	STAND. ERROR %	VARIATION COEFF. V %	ARRAY EFFICIENCY FACTOR ϵ
SINGLETS	X00	15	1057	141	11.2	31	.51
SINGLETS	0Y0	15	1051	107	3.3	12	.67
SINGLETS	00Z	15	708	95	2.1	9	.76
DOUBLETS	XY0	15	1315	92	3.6	15	.78
DOUBLETS	X0Z	15	1228	87	4.0	18	.83
DOUBLETS	0YZ	15	1310	85	3.5	16	.85
TRIPLETS	XYZ	9	508	72	2.6	11	1.00

Thus 1) Triplets are the most efficient arrays
2) Singlets are the least efficient arrays.

3) Material placed in the shortest axis exerts more absorbent effect than equal amounts in longer axes.

The observations appear to be consistent with an axial transit-time factor which affects absorbent efficiency.

The small calculated Standard Errors are most significant and the analysis shows that there is a very clear and consistent pattern in the effect of the arrays on absorbent efficiency.

5) ARE THE OBSERVED REVERBERATION TIMES CONSISTENT WITH ANY REVERBERATION ESTIMATOR? Besides the Sabine Eyring equations, comparisons were made with the equations of Millington/Sette (JAES Vol.18 No.6 1979 and JASA Vol.31 No.12 1950), Gomperts. (Acustica Vol.16 No.5 1965) and Fitzroy (JASA Vol.31 No.7 1959). Indices were also constructed and compared for four theoretical axially-biased estimators and finally Fitzroy's equation was recalculated with new coefficients for αg and αp in an unsuccessful attempt to optimise fit.

Attempts to correlate these ten variates with the observations by using Regression Analysis proved unfruitful because of insufficient discrimination which is inherent in this method. However a Spearman's Rank Correlation method proved most suitable and twenty one separate comparisons were then calculated with Spearman's R coeff's ranging between 0.43 and 0.95 (these on a scale between 0 and 1). Significance was tested using Fisher's Z transform.

The results (Table 5) showed that Fitzroy's equation produced the most conformable results and is capable of reproducing axial variations.

CONCLUSION This investigation is incomplete and needs extension. However it would seem reasonable to conclude at this stage that:-

(i) It is unwise to concentrate all absorbent into one axis,

(ii) For maximum efficiency (or minimum cost) and for minimum irregularity in effect, it is best to divide the absorbent up between the three axes.

(iii) Equal consistency can be attained, requiring less diffusion, from tri-axial placement as opposed to single axis placement,

(iv) That the Sabine type (Volumetric) equations do not seem to be as accurate as the Fitzroy type (Axial) equations in estimating reverberations for uneven placements.

4: CONSISTENCY OF PLACEMENT EFFECTS

PROBABILITY - USING CONCORDANCE CO-EFFICIENT (W) - THAT THE OBSERVED DEPARTURES FROM CONVENTIONAL SABINE TYPE R/T ESTIMATES WERE CONSISTENT WITHIN GROUPS AND WERE DUE TO SYSTEMATIC EFFECTS.

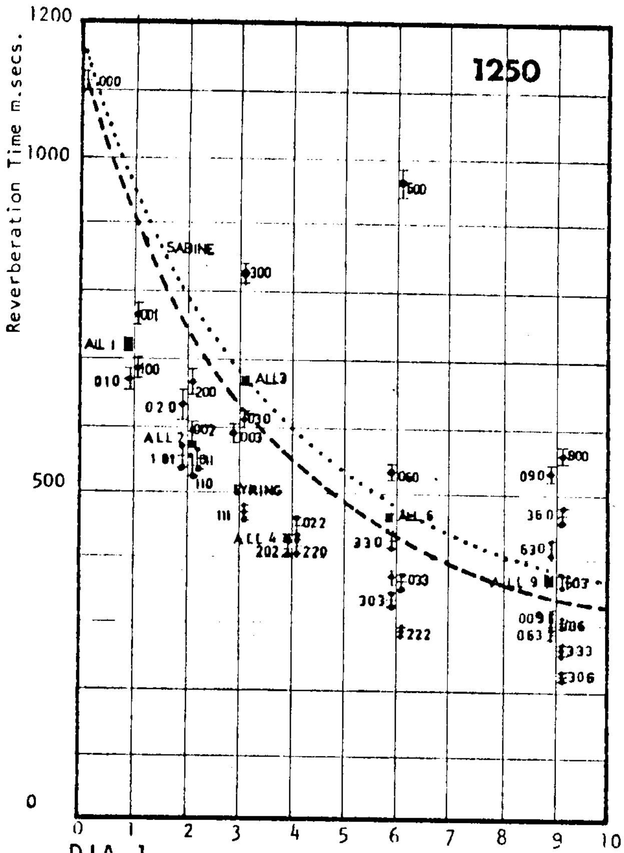
(Note All three frequencies pooled)

VARIATE	TOTAL R/TS.	NUMBER OF ARRAYS	CONCORDANCE COEFFICIENT	SNEDECORS F	Pr. $\geqslant$ W
Within "ALL 1" and "ALL 2"	1756	3 x 9	0.721	5.176	$<4.3\times10^{-3}$
Within "ALL 3" and "ALL 4"	1542	3 x 7	0.921	23.2	$<1.5\times10^{-4}$
Within "ALL 6"	1286	3 x 7	0.859	12.19	$<8.8\times10^{-4}$
Within "ALL 9"	2672	3 x 10	0.793	7.67	$<4.5\times10^{-4}$
Within ALL GROUPS	7256	3 x 33	0.945	51.87	$<3.5\times10^{-11}$

5: COMPARISON OF REVERBERATION ESTIMATORS

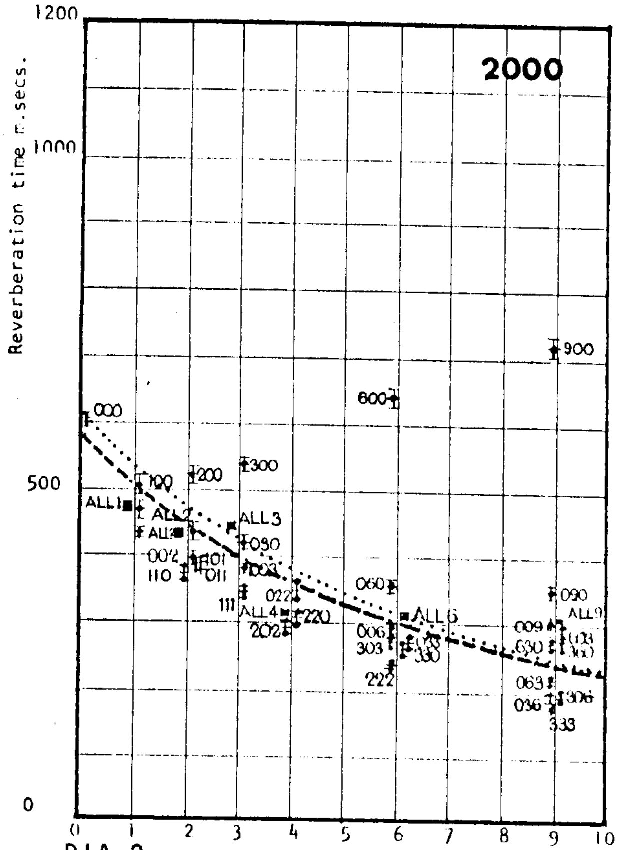
"NULL" TEST - USING SPEARMAN'S RANK CORRELATION COEFF (R) - OF FOUR CONVENTIONAL (VOLUMETRIC) AND FIVE UNCONVENTIONAL (AXIAL) ESTIMATORS FOR SIGNIFICANCE OF COMFORMITY WITH THE OBSERVED REVERBERATION TIMES

COMPARISON BETWEEN OBSERVATIONS AND RT ESTIMATES BY:-	AT 1250 HZ		AT 2000 HZ		AT 3150 HZ	
	R	Pr. $\geqslant$ R	R	Pr. $\geqslant$ R	R	Pr. $\geqslant$ R
VOLUMETRIC						
1. Sabine Simplex						
2. Sabine Eyring	.72	$<5\times10^{-7}$	.68	$<5\times10^{-5}$	.63	$<5\times10^{-5}$
3. Millington Sette						
4. Gomperts						
AXIAL						
5. $\sum_i^{xyz} \frac{L_{TOT}}{L_i} \times$ Sab.RT	.43	$<10^{-2}$	.43	$<10^{-2}$	.46	$<5\times10^{-3}$
6. $k\sum_i^{xyz} \frac{\mathrm{Log}(1-\bar{\alpha}_i)}{L_i}$	.56	$<5\times10^{-4}$	.58	$<2\times10^{-4}$	.43	<01
7. $\sum_i^{xyz} \frac{S_i\bar{\alpha}_i}{L_i}$ xSab.Rt.	.83	$<10^{-9}$	.79	$<10^{-8}$	.70	$<10^{-6}$
8. $\prod_i^{xyz}\left(\frac{S_i\bar{\alpha}_i}{L_i}\right)^{1/3}\times$Sab. RT.	.75	$<5\times10^{-9}$	.72	$<5\times10^{-7}$	.69	$<5\times10^{-5}$
9. $\sum_i^{xyz} \frac{S_i}{S^2} \times \frac{0.161V}{-\ln(1-\bar{\alpha}_i)}$ FITZROY	.93	$<10^{-10}$	.95	$<10^{-11}$	.89	$<10^{-10}$
10. ditto with new coefficients	.93	$<10^{-10}$	.95	$<10^{-11}$	.89	$<10^{-10}$



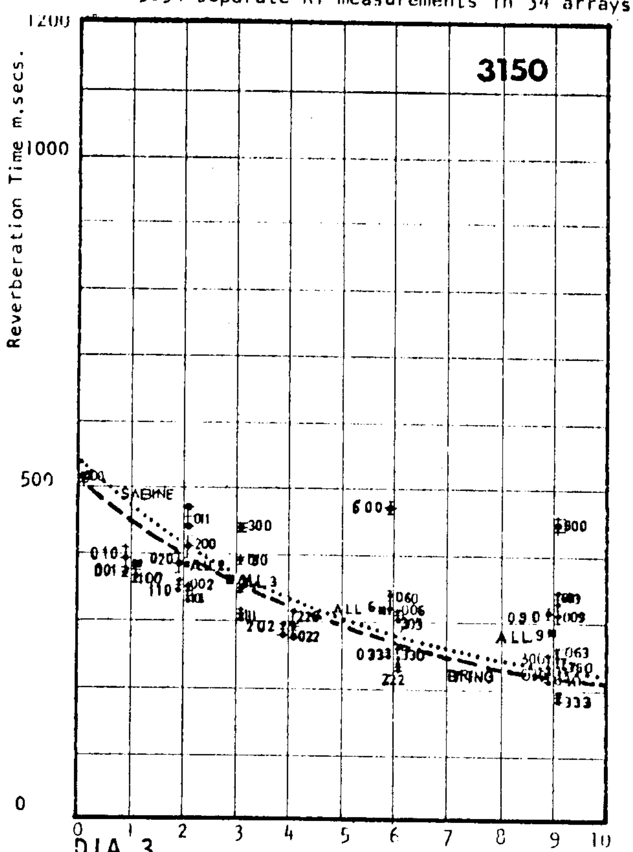
DIA.1

Singlets Doublets Triplets
REVERBERATION TIMES IN MILLISECONDS FOR ARRAYS
Plots are means ± 2SE. Arrays are XYZ F/gl.units
From 3034 separate RT measurements in 34 arrays



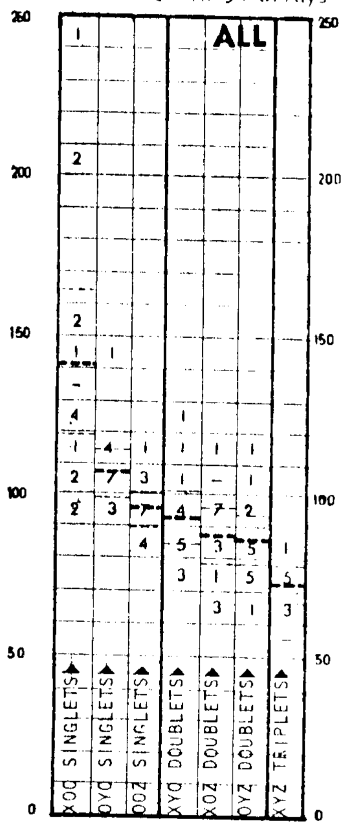
DIA.2

Singlets Doublets Triplets
REVERBERATION TIMES IN MILLISECONDS FOR ARRAYS
Plots are means ±2SE. Arrays are XYZ F/gl.units
From 2544 separate RT measurements in 34 arrays



DIA.3

Singlets Doublets Triplets
REVERBERATION TIMES IN MILLISECONDS FOR ARRAYS.
Plots are means ±2SE. Arrays are XYZ F/gl.units
From 2265 separate RT measurements in 34 arrays



DIA.4

Means Stand.Errors
DISTRIBUTION OF THE PERCENT-RELATIVE REVERBERATION TIMES
7843 RTS • 99 arrays