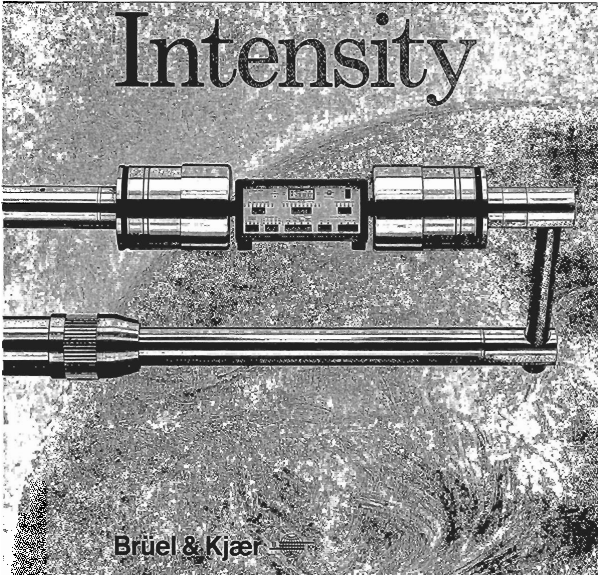
Acoustics Australia

AUSTRALIAN ACOUSTICAL SOCIETY

VOL. 16 No. 2 AUGUST, 1988



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Sylomer

Consists of a cellular polyurethane elastomer. It is used mainly for vibration isolation, shock absorption, and noise abatement. By choice of bearing surface area, material type and material thickness, the static and dynamic characteristics of the elastic bearing can be adapted to the respective application.

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- *Elongation at break 250% to 500%, depending on type.*
- *Ratio of vertical to horizontal stiffness approximately 2,5.*
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- *High dynamic load capacity.*
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The Advantages of Elastic Bearings with Large Surface Area

For Sylomer-mats the bearing surface can be adjusted to the application by choosing the material type and the thickness. In many cases reasonable advantages result from elastic supporting the full bearing area of machinery, plants, foundations or building components.

- *Reduced specific load on foundation.*
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- *Reduced excitation of natural vibrations and bending vibrations in the supported component.*
- *Reduced sound radiation thanks to "counter-drone" effect.*
- *High slip resistance.*
- *No risk of dirt accumulation or "noise bridges".*
- *Sylomer can be used like "permanent shuttering".*
- *Simple construction.*

Application Examples for Sylomer

- *Elastic beds for plant and machinery.*
- *Vibration isolation of building foundations.*
- *Sound isolation inside buildings.*
- *Vibration isolation in railway track construction.*

Standard Range

Sylomer material type	R 12	R 25	L 12	L 25	M 12	M 25	S 12	S 25
Thickness (mm)	12	25	12	25	12	25	12	25
Colour of the material	blue		green		brown		black	
Recommended max. bed for vibration isolation	0,035 N/mm²		0,08 N/mm²		0,15 N/mm²		0,50 N/mm²	

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Acoustics Australia

Vol. 16 No. 2 August 1988

CONTENTS

In place of their usual inside front cover advertisement, Bruel & Kjaer have supplied the illustration for the front cover. Featured in this issue is a review article by K. B. Ginn and J. N. Smith of Bruel & Kjaer, Denmark.

Subscription Rates (1988):

	Surface Mail	Airmail
1 year	A$36.00	A$45.00
2 years	A$64.80	A$82.80
3 years	A$94.50	A$121.50

Address all correspondence to:
The Chief Editor
PO Box 180
Gymea, NSW 2227

Acoustics Australia is published by the Australian Acoustical Society (Incorporated in N.S.W.)
35-43 Clarence Street, Sydney, N.S.W. 2000, Australia.

Responsibility for the contents of articles rests upon the authors not the Australian Acoustical Society.

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Printed by
Cronulla Printing Co. Pty. Ltd.,
16 Cronulla Street, Cronulla 2230.
(02) 523 5954.

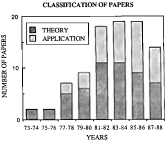
ISSN 0814-6039

NEWS

Editorial

The papers and notes in this issue are all on the topic of sound intensity; the theory, measurement and application. The principle of determining intensity from pressure and velocity measurements was first published in the form of a patent by H. F. Olsen in 1932 (US Patent 1 892 644 "System responsive to energy flow in sound waves"). In the subsequent decades there were sporadic reports on measurements of sound intensity using a variety of instruments but it was not until the late 1970s that there was a sudden increase in interest in the direct measurement of sound intensity. This was brought about by the possibility of using two closely spaced and matched pressure microphones to give both the pressure and the particle velocity associated with the sound field coupled with the availability of advanced computer analysing systems. The commercial production of probes and complete sound intensity analysing systems also led to an increase in the amount of research being undertaken.

The accompanying figure shows the number of papers on sound intensity which have been published in English language Journals since 1973. The classification of the



papers, in terms of theory and application, shows that while the emphasis in the recent years has been on the applications there are still many aspects of the theory of sound intensity which require investigation. It is currently just over half the way through the period 1987-88, yet a substantial number of papers have already been published. The first International Conference on Acoustic Intensity was held in Senlis in France in 1981. Interest in the topic has continued and at the Second International Conference in 1985, 80 papers were presented by researchers from over 20 countries. Most acoustics conferences now have a proportion of papers on some aspects of sound intensity.

Included in this issue of Acoustics Australia is a review paper by Bernard Ginn and J. N. Smith from Bruel & Kjaer, Denmark, and the paper by A. Cops and F. Wijnants held over from the April special issue. The other papers and reports are applications of sound intensity measurements for investigations of sound power of sources, insertion losses and noise reductions.

Marion Burgess
Associate Editor

ACT

April Meeting

The April meeting of the ACT Group was a tour of the National Recording Studios in Canberra. The meeting commenced with an outline of the acoustics aspects of the design of the complex by **Eric Taylor,** from Godfrey and Spowers. The group were then taken to the various studios and control rooms by **Ralph Ortner,** senior audio engineer from NRC, who explained the process of audio and video production.

The National Recording Studios are Canberra's first major independent video and audio production complex. The facilities cater for all types of production including:

- documentaries and corporate videos
- advertising for television and radio
- educational and audio-visual productions
- audio post-production for video
- music recording for bands and radio.

The audio-studio ranks among the best in Australia and its acoustically designed control room includes a 36-channel mixing console and a master synchroniser which locks with the video facilities for frame-accurate sound tracking. Ralph Ortner gave a demonstration of the replacement of a small part of a taped commentary with a corrected version which had been recorded subsequent to the initial recording.

Following the inspection a small group adjourned to a nearby Turkish restaurant for an enjoyable and satisfying dinner.

Marion Burgess

NSW

March Technical Meeting

On Friday, 18th March, **Graham Caldersmith,** who is a violin maker from Canberra, spoke on "Physics and Tradition in Violin Making".

In spite of substantial and continuing research into the physical behaviour of the violin, and persistent attempts at innovation in its design, the violin's essential sounding body has remained unexcelled for 300 years, even while more recent woodwind instruments and piano types have undergone important developments since industrialisation.

The violin's dramatic ascent in the Renaissance - Baroque musical expansion may have impressed into succeeding romantic and modern cultures a "violin stereotype" which remains unbroken, but analyses of the violin's vibrating and sound-generating characteristics suggest a special relationship between the violin's "voice" and human speech/hearing characteristics which is probably optimal.

It is also likely that traditional methods of adjusting the shape of the separate tops and backs, and their wood thicknesses gave the "original master" makers useful information about the musical properties of the complete instruments.

Graham provided a survey of principal physical analyses of violin behaviour preceded by a demonstration of traditional and derived modern methods of "tuning" violin tops and backs. The evening was most interesting as well as informative.

May Technical Meeting

Mr. Keith Davidson, of M. B. and K. J. Davidson Pty. Ltd. was the speaker at a Technical Meeting of the New South Wales Division of the Society, held at the National Acoustic Laboratories, Chatswood, on the 17th of May, 1988. The subject of the talk was "Short Leq - a new acoustic measuring technique".

The use of computer technology has given a new lease of life to the old 'outbox' processing technique of recording the acoustic signal and analysing it on replay. Using short Leq the raw data can be stored and used to re-create any acoustic situation, where the actual peak value of the signal is not involved.

A lively discussion of the role of short Leq in relation to older methods of acquiring and analysing acoustical data followed Mr. Davidson's talk and his contributions to the discussion were greatly appreciated by those present.

AHLBORN of Holzkirchen, West Germany, makers of temperature measurement and control equipment, have appointed **AUSTRALIAN METROSONICS Pty Ltd** as exclusive agents for their products in Australia and New Zealand. For details of the range of AHLBORN products available contact Australian Metrosonics Pty Ltd, PO Box 120, Mt Waverley, Victoria 3149. Phone: (03) 233 5889.

Digital Audio 1

No doubt Roy Caddy's paraphrase[1] of Michael Flanders' "Song of Reproduction" was well meant: I cannot recall who "Hutch" was, either. But the result was disaster. Not only did he do a grave injustice to Flanders' verse, but he translated "bel canto" into the wrong language doing it.

The original version, as recorded from a live performance of "At the Drop of a Hat" on Parlophone PMCO 1033, went like this:

"With a tone control at a single touch,
I can make Caruso sound like Hutch.
Then, I never did care for music much;
It's the High Fidelity."

Reference

1. Caddy, R., "Digital Techniques in Audio Equipment", Acoustics Australia, Vol 16 No 1, pp 20-22.

Dennis Gibbings
7 May 1988

Digital Audio 2

Dennis is right in his quotation — but so am I.

Flanders and Swan released a mono version of "At the Drop of a Hat", cited by Dennis, in which that line appeared.

Later they issued a stereo recording called "At the Drop of a Hat" which included many omissions and commissions compared with the original mono recording. For example, one "Mud Mud" chorus was sung in Russian.

By this time I suspect that Hutch's star had sunk below the horizon. (I do not know who Hutch was, either.) The essence of revue is topicality so Caruso and Hutch was changed to belle canto and Dutch. With both recordings to choose from I chose the latter as being more appropriate to Australia — Joan Sutherland is a bel canto singer.

I do stand corrected though in that I should have cited "Parlophone PCSO 3001" at the end of the article, possibly a unique reference in AA.

May I also admit to another error in my article. I miscalculated and put the notch at C sharp. It should have been the B flat just below four octaves above middle C and B sharp to E flat should have read A flat to C natural.

The good news is that the Bureau of Standards has vetoed the notch idea on musical and other grounds. The notch will not be inserted in CDDs.

The issue of the two records with the same title but with minor alterations in content reminds me of seeing the same research information published in a science journal and, with a few alterations and omissions, republished in a non-science journal. Two papers for the price of one?

Roy Caddy
4 June 1988

Reaction to Jaffe

Every era has its own special vulgarity. Christopher Jaffe in "Application of advanced electronic systems to concert halls and auditoriums" (Acoustics Australia, Vol 16 No 1) gives an account of what might pass as a remedy for bad architecture, but what should never be designed for in serious music. The author mentions, inter alia, Dr Beranek and Marshall. I think Marshall's discovery that the first reflections should be from the side-walls and not overhead reflections goes a long way to explain the disappointing results with a number of buildings (such as shown in the paper) which were designed only on Beranek's principles.

In the 1930s all the churches and town halls tried to make organ pipes do the work of symphony orchestras — with the result that there is now no organ in Sydney fit to play Bach, etc. Today's ambition is to make cone loudspeakers do the work of every conceivable instrument and human or any other voice.

Organ pipes still sound like organ pipes and loudspeakers still sound like loudspeakers.

Campbell Steele
13 May 1988

Ed:

Campbell Steele's comment on organs and Bach would be correct if this were 1958 instead of 1988. Today there are many fine new organs throughout Australia designed especially to play Bach's organ music.

Speakers Needed

In my role as President of the Otago Branch of the New Zealand Institute of Physics, I would like to draw to the attention of Acoustical Society members that our Branch has a small amount of money (which can sometimes be supplemented from our University of Otago Science Dean's Fund) for bringing speakers to meetings in Dunedin. It is not enough to pay Trans-Tasman fares, but can be used to pay travel for persons passing through other areas of New Zealand, such as Christchurch or Auckland, on their way to more distant places. If any scientists would be interested in giving our members a seminar on a fairly general topic, particularly one on acoustics or some other aspect of building science, we would be interested to hear from them to see if we could work out a suitable arrangement for a visit.

KEITH R. DAWBER,
President of Otago Branch NZIP.
PO BOX 56, DUNEDIN, N.Z.
17 June 1988

QLD

April and June Meetings

In April a visit was made to the **School of Audio Engineering** at **Milton.** Some twenty people attended and gained a valuable insight into acoustics associated with studios.

Later that month a visit was made to the factory of **Associated Building Panels** at **Alderley.** The company provided a luncheon and visitors were shown the manufacturing process of perforating panels, and were informed about the range of perforated panels available.

In June two meetings were held. The first one dealt with **noise associated with leisure activities** and was attended by around 30 people. Four topics were discussed, these being (i) noise from raceways, (ii) noise from open-air concerts, (iii) noise from indoor cricket centres and (iv) noise from pubs and clubs. A stimulating discussion ensued for each of these topics.

The other June meeting was on the topic of "Industrial Silencers and their Design" by **Mr. Tim Marks** of N.A.P. Silentflo and attended by about twenty people. Tim spoke on silencers in the context of large industrial applications and gave examples on how silencing of of large gas compressors and fans on power station stacks could be achieved. He also gave examples on the characteristics of "thin" and "thick" splitters for reducing noise levels.

Both these meetings were held at the premises of the Division of Noise Abatement and Air Pollution Control and thanks go to them for providing the venue and supper.

Frits Kamst

New Members

• Admissions

We have pleasure in welcoming the following who have been admitted to the grade of Subscriber while awaiting grading by the Council Standing Committee on Membership.

ACT
Mr M. L. Evenett.

New South Wales
Mr D. L. Bout, Mr J. W. Cotterill, Mr G. S. J. Glazier, Mrs S. H. McLain, Mr N. I. Oprya, Ms S. A. Ridler, Mr D. J. Spearitt.

New Zealand (NSW register)
Mr Effendi.

• Graded

We welcome the following new members whose gradings have now been approved.

Affiliate

Tasmania (Vic register)
Mr R. N. Stone.

Student

New South Wales
Mr F. J. Weatherall.

Subscriber

ACT
Mr R. Ortner.

New South Wales
Mr. A. J. Madry

Member

Victoria
Mr G. P. Benke, Dr M. Podlesak, Mr E. D. Sceney.

Tasmania (Vic register)
Mr B. L. Doolan.

New South Wales
Mr B. G. Marston

Western Australia
Mr J. D. Macpherson

Sound Intensity: The State of the Art

K.B. Ginn & J.N. Smith

Bruel & Kjaer A/S
2850 Naerum, Denmark

ABSTRACT: The advent in 1981 of readily available instrumentation capable of measuring sound intensity has heralded a new era in the world of acoustics. Since that date the continuous progress in the application of the sound intensity technique has been the subject ot numerous review articles and international conferences [1] [2] [3]. This paper summarises the present state of instrumentation, standardisation and applications for sound intensity techniques.

HISTORICAL BACKGROUND

The theoretical background for intensity measurements was described as early as 1932 by H.F. Olson. Several attempts were made in ensuing years to develop practical instrumentation. Methods tried included two closely spaced microphones (Clapp & Firestone 1941) and a hot-wire anemometer with a pressure microphone (Baker 1955). The subject was approached with renewed interest in 1977 by F.J. Fahy and J.Y. Chung who showed independently that the sound intensity function could be calculated from the imaginary part of the cross-spectrum function using a dual channel FFT analyser. O. Roth in 1981 [2] showed how real-time digital filtering techniques could be used for the calculation of sound intensity in standardised octave and third octave bands. From this date onwards sound intensity ceased being a laboratory curiosity and became a practical tool with a large number of applications.

WHAT IS SOUND INTENSITY?

Sound intensity, or sound-energy flux density, is a vector quantity which is in contrast to sound pressure which is a scalar quantity. Sound intensity describes the net amount and direction of flow of acoustic power at a given point in space. Hence the dimensions are energy per time per area and the units are W/m^2.

Intensity is normally presented as a level in dB with a reference of 1 pW/m^2. This reference level is chosen so that for a plane wave in a free field the sound intensity level, L_I is equal to the sound pressure level, L_P (1pW/m^2 is, in fact, an approximation; by using this value there will be a small, insignificant error.)

The plane wave propagating in a free field, mentioned above, is an example of a purely active sound field. A standing wave is an example of a purely reactive sound field where there is no net propagation of sound energy and hence the sound intensity level is zero.

Between these two extremes of a plane wave in a free field and a standing wave in a tube, lie a host of acoustical situations involving sound fields with both active and reactive components where the sound intensity technique can be applied to great effect.

SOUND INTENSITY MEASUREMENTS

Generally speaking, sound intensity measurements do not, and cannot, replace sound pressure measurements. Sound intensity and sound pressure measurements complement each other.

One of the properties of sound intensity meters is that they only measure the propagating, or active, part of the sound field. A sound pressure meter responds to the total sound energy, i.e. the sum of the propagating and non-propagating parts. The difference between the sound pressure and sound intensity levels in a sound field is called the pressure-intensity index and it has gained wide acceptance in the sound intensity community as an indicator for the difficulty in obtaining accurate intensity measurements.

The pressure-intensity index is a field indicator and reveals much about the sound field in which sound intensity measurements are to be taken. However, even if a sound intensity meter has been fully calibrated, sound intensity measurements cannot be taken with confidence until the residual pressure-intensity index of the probe and meter has been measured.

The residual pressure-intensity index of an intensity-measuring system is an indicator of the phase matching of the whole system. Good phase matching is important because active intensity measurement involves the measurement of a phase gradient. When no phase gradient is detected between the two microphones, no active intensity exists, however, inevitable minor phase mismatches between the measurement channels cause the detection of so called residual intensity. For a discussion of how residual intensity affects sound intensity measurements, see [4].

When taking sound intensity measurements both the pressure-intensity index spectrum of the sound field and the residual pressure-intensity index spectrum of the measuring system must be considered. The more diffuse a sound field is, the greater the difference between the sound pressure levels and sound intensity levels. It is in such diffuse fields that the sound intensity technique can be used to maximum effect, hence it becomes essential to know accurately the residual pressure-intensity index spectrum of the system. Furthermore, a knowledge of the residual pressure-intensity index spectrum enables corrections to be made to the intensity measurements for errors due residual intensity. These corrections can be made in real time with the analyzer described in [5].

SOUND INTENSITY INSTRUMENTATION

The main elements of a sound intensity measurement system are a probe, an analyser, a calibrator, a storage medium and some post-processing capablity, see *Fig. 1*.

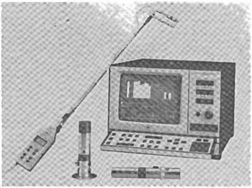


Fig. 1. Complete sound intensity system: probe, analyzer and calibrator.

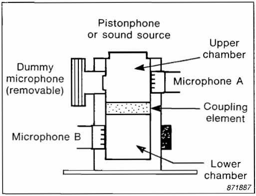


Fig. 2. Simplified cross section of intensity coupler.

SOUND INTENSITY PROBES

Several probe designs are currently in use. The most common is a two-microphone probe which enables the mean sound pressure to be calculated midway along the probe axis and the particle velocity to be calculated by using the finite difference approximation method. An alternative approach is based on an ultrasonic technique for measuring particle velocity. In [3] it was demonstrated that the state of the art design for an intensity probe is based on two "face to face" microphones fitted with phase correctors.

INTENSITY ANALYZERS

The first dedicated intensity analyser made its debut in 1981 and was based on ⅓ octave parallel digital filters. Since then several instrumentation manufacturers have produced similar systems. Small, battery-operated intensity systems first became available in 1986 in the form of a 1/1 octave analogue filter, serial analyser. At the beginning of 1988 an analyser was released [5] capable of measuring both active and reactive intensity in 1/1, ⅓ and 1/12 octaves. Other quantities related to intensity measurements such as particle velocity, mechanical power and surface intensity can be measured by connection the requisite transducers to the analogue inputs. The storage and post-processing requirements have been built into the analyser itself in the form of a large RAM memory and a 3½" disc drive. Software for intensity mapping is being developed so that the analyser will be entirely self-contained, no computer will be necessary.

CALIBRATOR

The calibrator is the last of the elements of an intensity system to be mentioned here although it is arguably the most important. The instrument described in [6] enables intensity systems to be calibrated against simulated sound intensity and particle velocity levels. A simplified cross-section of the intensity coupler is shown in *Fig. 2*. When used for sound intensity and particle velocity calibrations a pistonphone is used as the sound source (*Fig. 3*). When used for measuring the residual intensity and residual pressure intensity index of the measurement system a broad band source is employed (*Fig. 4*). The measured residual pressure-intensity index can be stored on a disc and used to correct the actual intensity measurement for the phase mismatch introduced by the instrumentation. In fact the post-processing in the new B & K analyser Type 2133 is fast enough to enable pressure-intensity index and correction for phase mismatch to be calculated and displayed in real-time.

Fig. 3. Pistonphone placed on coupler for sound intensity and particle velocity calibrations.

APPLICATIONS

Sound Power Determination

One of the principal applications of sound intensity measurements is the *in situ* determination of sound power radiated by noise sources. The radiated sound power can be determined from intensity measurements on a suitable control surface enclosing the source, since the intensity

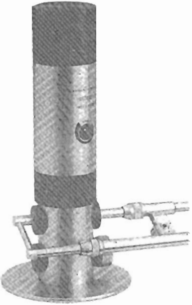


Fig. 4. Broadband source placed on coupler for measuring residual intensity and residual pressure-intensity index.

describes the power passing through an area.

$$W = \int\int_S \vec{I} \cdot d\vec{A} = \int\int_S I_n \, dA$$

The integration (in practice the summation) over the control surface of the intensity component normal to the surface will directly give the power W of the source. Some of the advantages of using intensity rather than sound pressure measurements for determining sound power are:

1. The method excludes any influence from steady background noise provided there is no absorption within the enclosed surface.
2. No anechoic or reverberation test chamber is needed. Measurements can be performed in ordinary rooms since reflections can be regarded as background noise.
3. Near field as well as far field measurements are acceptable. Near field measurements improve the signal to noise ratio and require less space, but the number of measurement points may have to be increased due to the sound field close to a source being rather complex.
4. There is no restriction upon the control surface. Any shape can be used.

Source Location

The second main application of sound intensity measurements is source location. The methods can be divided into two main groups, namely source ranking and intensity mapping. Source ranking is used when it is desirable to compare the sound power radiated by various components of an engine. Both the overall sound power level and the sound power level in different frequency bands

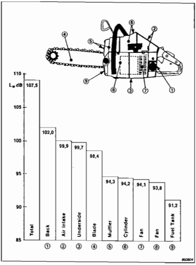


Fig. 5. Source ranking of a chainsaw.

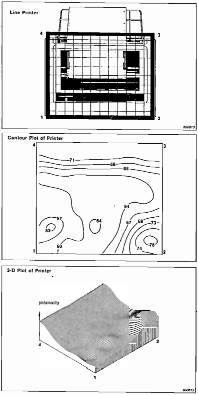


Fig. 6. Contour and 3-D plots of sound intensity for a line printer obtained with a measurement grid.

can be compared. Control surfaces are defined around the various components of interest and the sound power of the individual parts is determined from intensity measurements as described earlier. As an aid to source location an intensity map can be produced over the offending machine. With suitable software, number plots, contour plots or 3-D plots can be made of active and reactive intensity.

Fig. 5 shows how by defining small control surfaces, the parts of the chainsaw were ranked in terms of their sound output. *Fig. 6* shows how by using a grid, contour plots and 3-D plots of sound intensity can be obtained for a commercial printer.

OTHER APPLICATIONS

Many other applications are the subject of active research including transmission loss, sound absorption and radiation efficiency [1] [6] [8].

STANDARDIZATION

There are at present two international bodies working on the preparation of standards for the determination of sound power from sound intensity measurements. These are ISO TC 43/SC WG 25 and ANSI S12-21. Both groups have produced draft standards which stress the importance of "field indicators" (such as the pressure-intensity index, described above) to quantify the quality of the measurement. Two further groups from IEC and ANSI respectively are engaged in studying the requirements for the standardization of sound intensity instrumentation and calibration.

SUMMARY

Complete sound intensity systems are now available including probes, analyzers and calibrators. Forthcoming standards will certainly accelerate the promulgation of the intensity techniques which in turn will add to the growing data base. What does the future hold? We do not need to gaze into a crystal ball to predict that the intensity technique will have a long and active life in the service of acoustics.

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(Received 22 June 1988)

APPENDIX

Theory of Sound Intensity

Pressure Velocity Relation

It can be shown that in a medium without mean flow, the intensity vector component in the direction r equals the time averaged product of the instantaneous pressure *p* and the corresponding instantaneous particle velocity *u*, at the same position.

$$I_r = \overline{pu_r} \tag{1}$$

Pressure can be measured easily, it is the most commonly measured quantity in acoustics. The particle velocity is estimated from a measurement of the pressure gradient. This method rests on the linearized Euler equation, which is the equivalent of Newton's second law. In the direction r we have

$$\rho \frac{\delta u_r}{\delta t} = \frac{\delta p}{\delta r} \tag{2}$$

or solving for u_r

$$u_r = -\frac{1}{\rho} \int \frac{\delta p}{\delta r} dt \tag{3}$$

where p is the density of air.

Finite Difference approximation

The pressure gradient can be approximated by using two identical pressure microphones.

$$\frac{\delta p}{\delta r} = \lim_{\Delta r \to 0} \left(\frac{\Delta_p}{\Delta_p} \right) \approx (p_B - p_A) / \Delta r \tag{4}$$

The instantaneous pressure can be taken as the mean value of the two pressure signals.

$$p \approx \frac{(p_B + p_A)}{2} \tag{5}$$

Δr is the separation between the two measurement points. These approximations are valid provided that Δr is small compared to the wavelength. A practical sound intensity probe consists of two closely spaced pressure microphones, allowing measurement of both pressure and the component of the pressure gradient along a line joining the centres of the microphones. Hence, the magnitude of the component of the intensity vector along this line is measured. Modern sound intensity measurements are performed using the following equation:

$$I_r = \overline{pu_r} \approx -\overline{\left[(p_B + p_A) / 2\rho\Delta r \right] \int (p_B - p_A)\, dt} \tag{6}$$

The reactive intensity, which is the product of the pressure and the quadrature component of the particle velocity can be obtained from

$$J_r = \overline{pu_{\text{quad}}} = -\overline{\left[(p_B + p_A) / 2\rho\Delta r \right] (p_B - p_A)} \tag{7}$$

Signal Processing

For processing the signals from the transducers, two approaches are in current use. The digital filter approach is a direct method, by which sound intensity is calculated in true real time in the time domain. In the FFT approach the intensity is calculated in the frequency domain from the imaginary part of the cross-spectrum function.

$$I_r(f) = -(ImG_{AB} / \omega\rho\Delta r) \tag{8}$$

The reactive intensity can be obtained from

$$J_r(f) = \frac{-1}{2\omega\rho\Delta r} (G_{BB} - G_{AA}) \tag{9}$$

Laboratory Measurements of the Sound Transmission Loss of Glass and Windows— Sound Intensity versus Conventional Method

A. Cops and F. Wijnants
Laboratorium voor Akoestiek en Warmtegeleiding
K.U. Leuven
Celestijnenlaan 200 D
3030 Leuven, Belgium

ABSTRACT: When designing buildings with a maximum of sound insulation against traffic noise it is important to know the different sound transmission paths of the facades, which are generally complex. First of all the design of the weakest parts in the facade, namely the windows, has to be improved. With the conventional measuring methods it is only possible to measure the general sound insulation values of the complete facades, not the individual parts separately. In this research the sound intensity technique has been used. With this technique it is possible to measure different parts of building constructions separately. Research has been done on glass panels and on practical window constructions in laboratory conditions. The reason why measurements in laboratory circumstances have been preferred is the better accuracy and reproducibility which can be obtained with the different measuring methods.

1. INTRODUCTION

In recent years a new technique has been developed to measure the sound radiation of sources and structures. This technique, known as the sound intensity technique, has been primarily applied to measure the noise power of sources but is used also extensively to measure the sound transmission loss of building structures [1-7].

In order to validate this method it is important to compare the results with the still existing methods and eventually with models like the Statistical Energy Analysis technique (7). In order to increase the measuring speed, comparisons have been made between the faster scanning method and the slower fixed points measurements of the radiated sound intensity. Also of great importance is the value of the pressure-intensity index which can be due either to more or less diffuse field conditions in the receiving room or to the special energy vortices near strongly radiating panels [8,9]. In order to clear this point, measurements have been done with the sound intensity probe both at a short distance from the glass panels and windows and at a greater distance.

The most important advantage of the intensity method over the conventional two-room method is the possibility to measure the sound transmission loss of individual parts of a facade or a facade element separately. This procedure has been used to measure the sound transmission loss of the main frame, the opening frame which contains the glass panel and the glass panel itself, of a PVC, a wooden and an aluminium window separately and to calculate later on the total sound transmission loss of the complete windows. These last values have also been compared with the results obtained with the conventional method.

2. THE TWO MICROPHONE TECHNIQUE

It is well known that the sound intensity radiated by a structure in a direction r can be calculated from the sound pressure measurement at two closely spaced points A and B from

$$I_r = [(p_A + p_B)/2] \,.\, (1/\varrho) \int [(p_A - p_B)/\Delta r] dt \qquad (1)$$

or from

$$I_r = (2\pi \varrho \Delta r)^{-1} \int (\mathrm{Im}\{G_{AB}\}/f) df \qquad (2)$$

with:

p_A and p_B the microphone pressures
Δr the microphone separation
$\mathrm{Im}(G_{AB})$ the imaginary part of the cross-spectrum between the two microphone signals A and B.

While the sound intensity measured with this two-microphone technique is a function of the phase difference between the signals in the points A and B, a phase error between both measuring channels will cause an error in the sound intensity level. It can be shown that the error is given by [10]:

$$L_\epsilon = 10 \log [1 \pm 10^{(L_K - L_{K,0})/10}] \qquad (3)$$

with:

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

This residual pressure-intensity index is a measure of the bias error that may be present between the two channels of the equipment and thus is a measure of the quality of the system.

Table 1—

Residual pressure-intensity index $L_{K,0}$ of the measuring sound intensity equipment and the pressure-intensity index L_K at different probe distances to the test objects

1/3 Oct. Band Freq'cy (Hz)		Glass pane 6mm thick			Double glass panel (10/12/4) mm thick		
	$L_{K,0}$	L_K	L_K	L_K abs.mat in niche	L_K (10 mm)	L_K (4 mm)	L_K
		5 cm	28 cm	5 cm	5 cm	5 cm	28 cm
100	8	2.4	10.8	3.3	10.6	2.2	6.1
125	9	2.8	19.7	3.3	5.9	2.4	9.1
160	10	11.0	4.2	3.6	4.9	2.8	1.7
200	11	4.0	2.2	3.2	2.4	1.7	1.1
250	12	5.1	0.2	3.7	4.4	2.0	1.8
315	13	5.2	1.1	3.6	6.2	3.1	1.9
400	14	4.6	1.5	3.6	7.5	4.6	2.9
500	15	6.5	3.1	4.0	7.5	5.3	5.8
630	16	8.3	4.6	4.2	8.2	6.1	7.3
800	17	7.9	4.8	4.6	9.5	6.4	6.0
1000	17	8.5	5.2	5.0	11.6	8.4	4.5
1250	17	9.8	5.5	5.8	12.7	9.5	6.0
1600	17	16.6	6.3	7.3	9.1	9.8	5.7
2000	17	14.6	6.9	6.2	6.1	11.7	4.6
2500	17	12.3	6.1	3.7	3.8	10.5	4.0
3150	17	4.3	4.8	2.6	3.1	8.5	3.6
4000	17	4.0	3.5	2.2	3.4	4.4	3.5
5000	17	3.8	3.2	1.7	1.5	3.4	0.7

A possible way to determine $L_{K,0}$ is to apply the same signals to both measuring channels. Due to a phase error between the channels a residual intensity level is measured. The difference between this level and the measured sound pressure level is by definition the residual pressure-intensity index. During this measurement the Bruel & Kjaer sound intensity analysing system, Type 3360 has been used and an estimation of the pressure-intensity index is given in Table 1.

If the intensity is not measured in one point, but will be given as a mean value of N points, it can be shown that Equation 3 will also give the error, on the mean sound intensity level caused by the phase error, if L_K is considered as the global pressure-intensity index:

$$L_K = 10 \log \sum_N 10^{L_p/10} - 10 \log \sum_N 10^{L_I/10} = L_p - L_I \quad (4)$$

Whereas the measured phase is proportional to the microphone distance Δr between A and B, an increase of this distance will result in a decrease of the measuring error. However at higher frequencies a larger microphone distance will cause another error, namely the error due to the finite distance approximation [11]. As a consequence the microphone distance is chosen in that way that this last error will be smaller than 1 dB in the frequency interval 100-5000 Hz ($\Delta r = 12$ mm). In order to verify the accuracy of this probe distance, measurements with Δr = 50 mm were done. The results between 100 Hz and 1000 Hz corresponded within 1 dB.

Another measuring error which will be shortly discussed in relation to this two-microphone technique is the statistical error. By taking the mean value of the sound intensity between N points, this measuring error can be divided into two independent parts, ϵ_1 the statistical error obtained by defining the sound intensity itself and ϵ_2 caused by the spread of the sound intensity between the N measuring points. If the statistical error ϵ of a quantity is defined as the ratio of the square root of the variance of this quantity to the quantity itself, one has

$$\epsilon(I) = [\epsilon^2_1(I) + \epsilon^2_2(I)]^{1/2} \quad (5)$$

$$\epsilon_1(I) = [(1/2NBT)\{1 + (1/\gamma^2_{AB})\} + \{(1/\gamma^2_{AB}) - 1\} \cot^2\theta_{AB}]^{1/2} \quad (6a)$$

$$\epsilon_2(I) = [var(I)]^{1/2}/NI \quad (6b)$$

with:

B = the used bandwidth

T = the measuring time

γ^2_{AB} = the coherence between the two signals A and B

θ_{AB} = the phase difference between A and B

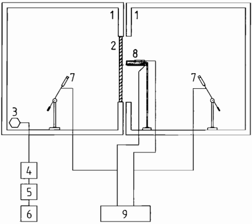
3. MEASUREMENT EQUIPMENT AND RESULTS

The laboratory facility to measure the sound transmission loss of test objects with either the sound intensity or the conventional method, as well as the complete measuring equipment set is shown in Figure 1. The absorbing wedges surrounding the measuring opening during the sound intensity measurements were placed to optimise the sound field conditions into the receiving room. Measurements have been done in an opening with dimensions of 1.50 m by 1.25 m and a niche depth of 0.40 m as prescribed in DIN 52 210 [12]. The opening is staggered as shown in Figure 2. The measured glass panels and windows were fixed as shown in this figure at a position 2/1 within the niche.

The two-microphone technique can be applied in two ways to determine the sound intensity radiated by a test object. By fixed point measurements, the test object is subdivided in a large number of equally sized areas and the sound intensity level is measured in the centre point of these areas. When using the scanning method, the test object is divided into a smaller number of equal areas. The measuring probe is swept with a speed of about 1 cm/s over each of these areas during a well-defined time. The scanned method is a faster and easier way in obtaining results. The sound transmission loss calculated from the sound intensity method is given by

$$R = Lp_1 - 10 \log \{(1/N) \sum_{i=1}^{N} (I_i/I_o)\} - 6.2 + 10 \log \{1 + (\lambda S_1/8V_1)\} \quad (7)$$

VERTICAL SECTION OF SOUND TRANSMISSION ROOMS



1 Transmission Rooms
2 Panel or Wall
3 Loudspeaker System
4 Amplifier
5 Filter GenRad 1925
6 White Noise Generator B&K 1402
7 Microphone B&K 1/2" 4134
8 Sound Intensity Probe B&K 3519
9 Sound Intensity Analysing System B&K 3360

Figure 1: Vertical section of the sound transmission rooms together with the equipment used to measure the sound transmission loss of building constructions

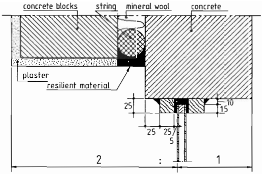


Figure 2: Mounting of the test specimen on the staggered opening, following DIN 52 210 (dimensions in mm)

with:
Lp_1 = the mean value of the sound pressure level in the transmitting room
I_i = the radiated sound intensity by the ith surface
N = the number of the subdivided surfaces.
The last term in the equation is the correction for the Waterhouse-effect [4,13,14].

Results obtained with the fixed points and the scanning method have been compared for a 6 mm thick glass panel. During the fixed points measurements the panel was subdivided into 120 areas of equal size. The scanning method was applied twice: once with 30 and once with 4 areas of equal size. During all the measurements the distance of the centre point of the intensity probe was kept constant at 5 cm from the panel. The deviation of the measuring results obtained with the fixed point method and those from the scanning method are represented in Figure 3. The agreement between the results obtained with both measuring approximations is, over the whole frequency region, within the measuring accuracy. Nevertheless the agreement with the fixed points method is better if the number of scanned surfaces is rather large. In all other measurements with the sound intensity method the scanning approximation, with a large number of measuring areas, is used. The sound transmission loss of a test object can also be measured using the well-known conventional two-room method and is in this case calculated with the conventional formula corrected for the Waterhouse-effect [4,13,14]:

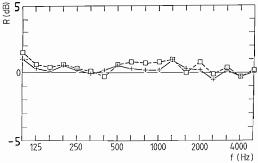


Figure 3: Deviation of the sound transmission loss results obtained with the scanning method from the fixed points method:
□– – –□ scanning over 4 surfaces
+———+ scanning over 30 surfaces

$$R = Lp_1 - Lp_2 + 10 \log (S/A_2) + + 10 \left[\{1 + (\lambda S_1/8V_1)\} / \{1 + (\lambda S_2/8V_2)\} \right] \quad (8)$$

with:
Lp_1 and Lp_2 the mean value of the sound pressure level in the transmitting and receiving rooms respectively
S the surface area of the panel
S_1 and S_2 the total surface area of the transmitting and receiving rooms respectively
V_1 and V_2 the volume of the transmitting and receiving rooms respectively
A_2 the absorption of the receiving room

Whereas both measuring rooms have equal volumes of 87 m³ the last term in this expression vanishes.

The results of the conventional sound transmission loss measurements of the glass panel with thickness 6 mm are compared in Figure 4 with the results of the sound intensity method. This figure shows discrepancies of 2 dB and more below 250 Hz and in the third octave band frequencies 1600, 2000 and 2500 Hz, which is the region around the coincidence frequency of the glass panel. The deviations below 250 Hz can be explained by a lack of diffusivity of the transmitting room. Around the coincidence frequency the difference between the results of both measuring methods is caused by a measuring error due to the phase error on the sound intensity measurements, as is made clear in Table 1. This table gives the measured pressure-intensity index of the sound field close to the 6 mm thick test object. At 1600, 2000 and 2500 Hz where

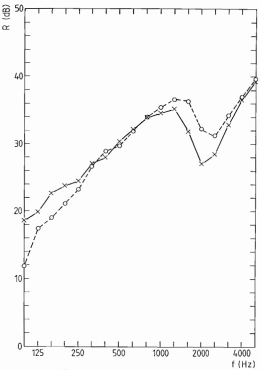


Figure 4: Comparison of the sound transmission loss of a 6 mm thick glass panel:
x———x conventional method
o– – –o intensity method, distance centre of the probe to the measuring surface 5 cm

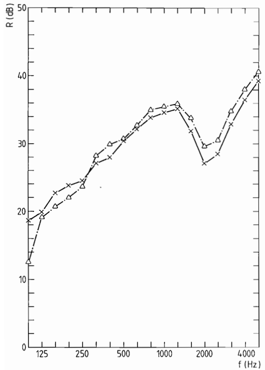


Figure 5: Comparison of the sound transmission loss of a 6 mm thick glass panel:
x———x conventional method
△—·—·—△ intensity method, distance centre of the probe to the measuring surface 28 cm

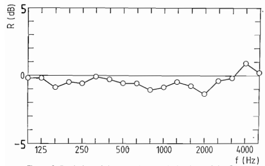


Figure 6: Deviation of the sound transmission loss of the 6 mm thick glass panel with absorbing material from the values without absorbing material in the niche

the observed discrepancies become important, the difference between L_K and the estimated residual pressure-intensity index $L_{K,0}$ of the measuring equipment becomes rather small; which, according to formula 3, indicates that the phase error is important at this frequency.

In order to verify whether this phase error is caused by the semi-diffuse sound field conditions in the receiving room or by the complexity of the sound field very close to the radiating panel, sound intensity measurements are performed with the centre of the probe at 28 cm from the radiating panel. This

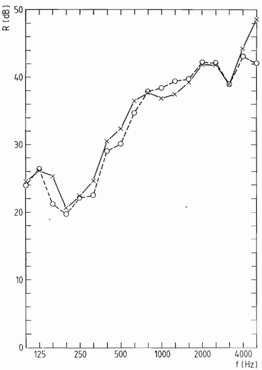


Figure 7: Comparison of the sound transmission loss of a double (4/12/10) mm glass panel:
x———x conventional method
o– – –o intensity method, distance centre of the probe to the measuring surface 5 cm

measuring area is situated at the edge of the niche. The results of these measurements are shown in Figure 5. As is clear, they are in bettter agreement with the conventional method results. This can be explained because of a smaller phase error in the vicinity of the coincidence frequency with the sound intensity measurements, as is shown in Table 1: the pressure-intensity index and hence the phase error is much smaller at a larger distance of the panel, except at 100 and 125 Hz. It can be concluded that the higher L_K values close to the panel are due to the complexity of the radiating sound field. No explanation is found for the large L_K at 100 and 125 Hz. In an attempt to reduce this complexity of the sound field, due to standing waves within the niche, absorbing material was placed at the edges in the niche of the measuring opening in the receiving room.

The sound transmission loss of the panel was measured using the sound intensity technique and keeping the distance of the probe centre to the panel constant at 5 cm. As shown in Table 1, the L_K and thus the phase error is reduced. However, the sound transmission loss of the panel has increased by 1 to 2 dB (Figure 6) especially in the mid frequencies where standing waves occur with reflecting niches and vanish with absorbing niches. In this case a comparison with the conventional method is not significant due to the change in niche configuration. It is expected that the L_K becomes smaller, and thus the results are more accurate, if a heavier test object is tested, because this will have a less complicated vibrating pattern than the 6 mm glass panel. In order to investigate this assumption, a second series of measurements was performed on a double glass panel composed of two panes with thicknesses 4 mm and 10 mm which are separated by a 12 mm air space. The critical frequencies of the panes are 3050 Hz and 1220 Hz respectively.

The results of the measurements of the sound transmission loss using the conventional and the sound intensity method, with the probe close to the surface, are shown in Figure 7. The deviations between the results are generally less than for the 6 mm pane measurements. The reason is that, for all the frequencies, except near the coincidence frequency of the 10 mm pane, the pressure-intensity index of the sound field near the object is much smaller than $L_{K,0}$ and therefore the phase error on the intensity results is rather small. It can be noticed that the coincidence dip of the 10 mm panel is not observed with the intensity method, which can be explained by the higher L_K at these frequencies (Table 1). As seen from Table 1, L_K becomes maximal at the critical frequency of the 10 mm glass pane. This can be explained by the fact that during the measurements the double glass panel was placed with the 10 mm pane facing the receiving room. A second intensity mapping was done, with the panel turned around so that the 4 mm pane faced the receiving room. Again the results matched those obtained with the conventional method and therefore are not shown here. Now the maximum values of the pressure-intensity index were found at 2000 Hz, 2500 Hz and 3150 Hz, which corresponds with the critical frequency region of this 4 mm pane (Table 1).

In Figure 8 the results of the sound transmission loss at the same double panel obtained with the intensity method, with the probe centre at 28 cm, are compared with those of the conventional method. Now the agreement between the results is very good over the whole frequency region and the pressure-intensity index has decreased to such a low level that the phase error is negligible.

The most important advantage of the sound intensity method compared with the conventional one is the possibility to measure the STL of individual parts of a facade element separately. This procedure has been followed to measure the sound transmission loss of the main frame, the opening frame which contains the glass panel and the glass panel itself. Measurements have been done on three different windows: a PVC, a wooden and an aluminium window containing a 6 mm thick glass panel. The main frame and the opening frame are subdivided in eight equal surfaces and the radiated sound intensity level is obtained by scanning over a well-defined time. The glass panel is subdivided in at least 30 equal parts and the same scanning time per surface is used. The measurements have been done with the centre point of the intensity probe at 5 cm from the measuring object. Figure 9 represents the STL results obtained for the three different parts of the PVC window. Over the whole frequency region the main frame and the opening frame give better results than those of the glass panel. It is amazing that in the coincidence frequency region of the glass panel the STL values of the frames also have a tendency to drop. This is, expecially for the opening frame, due to the strong connection with the glass panel. Also in this figure the overall calculated STL of the window is presented.

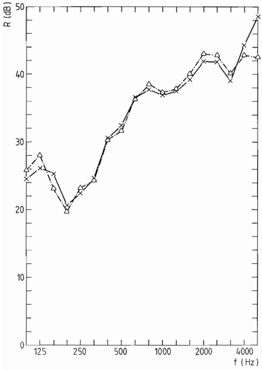


Figure 8: Comparison of the sound transmission loss of a double (4/12/10) mm glass panel:
x——x conventional method
△—·—·—△ intensity method, distance centre of the probe to the measuring surface 28 cm

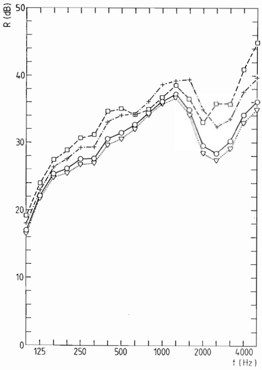


Figure 9: Sound transmission loss of the PVC-window obtained with the intensity method (probe distance 5 cm):
□———□ main frame
+—·—·—+ opening frame
△··········△ 6 mm glass panel
o——o overall STL of the window

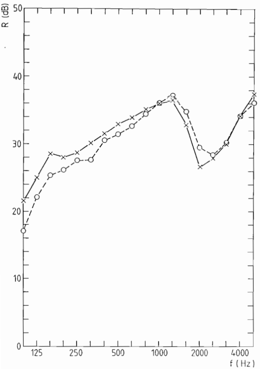


Figure 10: Comparison of the sound transmission loss of the PVC-window with a glass panel of thickness 6 mm:
x———x conventional method
o— — —o intensity method, distance of the centre of the probe to the measuring object 5 cm

In Figure 10 the overall STL obtained with the intensity method is compared with the conventional method results. Also measurements with the intensity technique have been executed with the centre of the probe at a distance of 28 cm from the window. At this distance only the total STL can be measured. As before the results between the intensity and the conventional method are in better agreement than with the short distance measurements. The results obtained in the wooden and the aluminium windows show the same tendency. This is the reason why these figures are not included within the text. It can be concluded that with the intensity method it is possible to measure different parts of composed structures separately. This is only possible with the centre of the intensity probe at short distances to the measuring object. Nevertheless the overall results show some small discrepancies compared with the conventional method especially for strong radiating objects. It is expected that with less radiating surfaces the agreement between both measuring methods will be still better.

4. CONCLUSIONS

The accuracy and the validity of sound transmission loss measurements with the sound intensity technique depends on the difference between the measured pressure-intensity index L_K and the residual pressure-intensity index $L_{K,0}$, which is a measure of the bias error of the equipment. From Equation 3 it is clear that for small differences the measuring accuracy is bad. If the differences are larger than −7 dB an accuracy better than 1 dB is obtained. Since $L_{K,0}$ is determined by the phase-matching of the system it can be controlled by adjustment of the microphone spacing Δr. The reactivity near the surface of the object depends on the amount of absorption in the receiving room, on the radiation pattern of the measuring object and on the standing waves within the niche as is clearly shown with the experiments on the 6 mm thick glass panel.

If the sound transmission loss of the measuring object is higher and, as a consequence, the radiating pattern of the surface is less complicated, these inconveniences are less important as shown in the experiments with the double glass panel with thickness (10/12/4) mm.

The intensity technique is especially valuable to measure the STL of individual parts of composed building constructions like windows or other constructions. This gives the possibility to detect the weakest parts in the constructions and to redesign them.

(Received 8 September 1987)

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Determination of the Insertion Losses of Acoustic Laggings by Intensity Measurements

K.P. Byrne

School of Mechanical and Industrial Engineering
The University of New South Wales
P.O. Box 1
Kensington NSW 2033

ABSTRACT: The development of a rig which was used with acoustic intensity measurements to experimentally determine the insertion losses produced by typical industrial acoustic laggings is described. The experiments were designed to test the validity of theoretical predictions of the insertion losses. It is shown how flanking transmission in the rig had to be controlled to allow meaningful measurements to be made.

INTRODUCTION

Acoustic laggings are used to inhibit the transmission of the sound radiated from vibrating surfaces such as those of ducts, pipes and machines. The basic components of acoustic laggings are porous layers, impervious barriers and air spaces. The porous layers are often fibreglass or mineral wool blankets and the impervious barriers are often metal cladding sheets or loaded plastic sheets. Sometimes "damping layers" are attached to the impervious barriers or the vibrating surface. Frequently, laggings consist of nothing more than a blanket of a porous material which is laid over the vibrating surface and then covered with a metal cladding sheet. However, more complex laggings which incorporate several porous layers and several impervious barriers are also used.

The transmission of sound through plane multi-layer structures has been studied for many years, [1] to [3]. Much of this work has been directed towards understanding and predicting sound transmission through building elements such as double wall partitions. Surprisingly, there has been very little interest in studying the performance of acoustic laggings despite their widespread industrial use. The Australian Electrical Research Board acknowledged this point and sponsored a research project which was directed towards improving the understanding of how acoustic laggings, particularly those of the type used in the power generation industry, function. The work, which was carried out in the School of Mechanical and Industrial Engineering at The University of New South Wales, involved both a theoretical and an experimental component. The purpose of the experimental component was to verify the accuracy of the theoretical predictions. Details of the theoretical work are given in reference [4] along with comparisons between predicted and experimental results. An outline of the development of the rig used for the experimental work is given in this paper.

THE EXPERIMENTAL RIG

A method of measuring the insertion losses of an acoustic lagging in various frequency bands involves the measurement of the sound reduction indices of a base structure, for example, a metal plate, and those of the base structure plus lagging. The sound reduction indices of an acoustic construction are usually measured by mounting the construction between two special purpose rooms, one of which contains a noise source, and measuring the sound pressures in the rooms. The University of New South Wales does not possess such facilities and a specially constructed rig was used. The important features of the basic rig are shown in *Figure 1*. The lagging was supported on a grid of fine tensioned wires which were located approximately 20mm above the

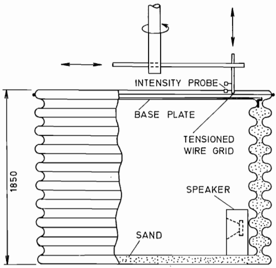

Figure 1: Initial rig configuration.

base plate. This arrangement was adopted to avoid the mechanical excitation of the lagging by the vibrating base plate. It is possible, in principle at least, to determine with such a rig, the insertion losses produced by a lagging by measuring the sound pressures in the room containing the rig with and without the lagging present. However, it is necessary to achieve a situation in which the bulk of the acoustic power flow into the room is through the base plate or the lagging when it is present. It was suspected that this situation could not be achieved when the lagging was present, as flanking transmission through the side walls of the rig would be important. In view of this suspicion, which was subsequently confirmed, it was decided to try to overcome the problem by using intensity measurements to determine the space averaged acoustic intensity radiated from the base plate and the lagging. These two sets of acoustic intensity measurements would allow the insertion loss produced by the lagging to be determined. Several workers [5] — [7] have described how intensity measurements can be used to measure the transmission losses of panels and so some basis for measuring the insertion losses in this way was available. The acoustic intensity measurements were made with a B & K Sound Intensity Probe Type 3519 and a B & K Dual Channel Analyser Type 2032. The probe was fitted with ½" microphones spaced 12mm apart which, under reasonable conditions, would allow measurements to be made from 80 Hz to 5000 Hz. The narrow band intensity measurements made with this system were processed with a HP 300B computer to give ⅓ octave band intensity measurements. A discussion of this type of measurement system is given by Ginn and Upton [8]. Acoustic intensity measurements were made at approximately 100 locations uniformly distributed over a circular area 1000mm in diameter above the base plate or the lagging. The intensity probe was located normal to and 200mm from the surface of interest. These 100 measurements were averaged to give the final result. Similar processing was also applied to determine the space averaged ⅓ octave sound pressures at the probe.

The 2380mm diameter base plate was made of 6mm thick steel plate. This material and thickness were chosen as they are representative of plates used in many ducts in thermal power plant. The critical frequency for 6mm steel plate is approximately 2000 Hz. At higher frequencies the propagation velocity of flexural waves in this thickness plate is greater than the velocity of sound in air at ambient conditions. The base plate was bolted to a heavy rolled angle ring which was attached to the top of the inner corrugated iron tank shown in Figure 1. This inner tank, which was approximately 2230mm in diameter and 1600mm deep, contained four box mounted 275 watt speakers which were driven by band limited random noise. This system was used to acoustically excite the base plate. The transmission of sound from the walls of this inner tank was inhibited by placing it in an outer tank and filling the cavity between the two tanks with sand. The rig in this configuration is subsequently referred to as the "initial rig configuration" and was as shown in Figure 1.

Unfortunately, it was found that flanking transmission was a problem with the initial rig configuration. Even with a simple lagging there apparently was, in some frequency bands, a nett flow of acoustic energy into the lagging from the region above it. The intensity probe was used to locate the sources of the contaminating acoustic energy and it was found that this acoustic energy came from the annular gap between the tops of the inner and outer tanks and the outer surface of the outer tank. It was decided in view of this and the possible ambient noise problem to enclose the intensity probe in a lined chamber and to lag the outer tank with 50mm of high density mineral wool and 0.7mm thick sheets of flat galvanised steel. The rig in this configuration is subsequently referred to as the "final rig configuration" and was as shown in *Figure 2*.

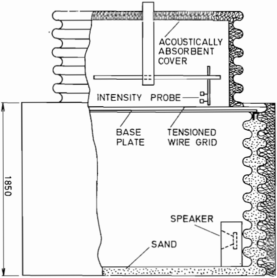

Figure 2: Final rig configuration.

TYPICAL RESULTS

Results with initial rig configuration

The results of measurements made to establish the insertion loss of a "typical" lagging with the test rig in its initial configuration as shown in Figure 1 are plotted in *Figures 3 and 4*. The typical lagging was formed of a 20mm airspace,

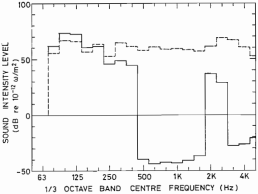

Figure 3: Space averaged sound intensity measurements with initial rig configuration. --- Bare plate. — With "typical" lagging.

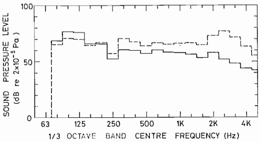

Figure 4: Space averaged sound pressure measurements with initial rig configuration. --- Bare plate. — With "typical" lagging.

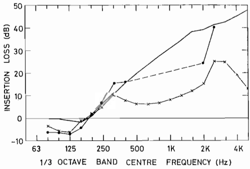

Figure 5: Predicted and measured ⅓ octave band insertion losses for "typical" lagging. Measured values with initial rig configuration. — Theoretical values. — Derived from measured intensities. — Derived from measured pressures. [--------Associated with negative intensity with lagged plate].

a 50mm fibreglass blanket with a flow resistivity of 17,600 Rayls/m and a 0.81mm thick aluminium sheet. The insertion losses derived from the space averaged intensity measurements plotted in Figure 3 and those from the space averaged pressure measurements plotted in Figure 4 are plotted in *Figure 5*. The theoretically predicted insertion losses derived by the theory described in reference [4] are also shown. The "negative" intensities in some ⅓ octave bands as shown in Figure 3 reveal the shortcomings of the rig. The negative intensity can be interpreted as a flow of acoustic energy into the lagging from the region above it. The insertion loss measurements derived from acoustic pressure measurements, although not showing this gross error, obviously do not agree well with the predicted result.

Results with final rig configuration

The results corresponding to those shown in Figures 3 to 5 are shown in *Figures 6 to 8* for the rig in the final configuration. The features of the rig in the final configuration are shown in Figure 2.

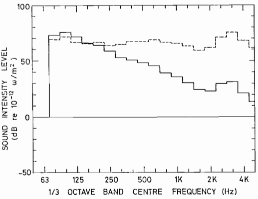

Figure 6: Space averaged sound intensity measurements with final rig configuration. --- Bare plate. — With "typical" lagging.

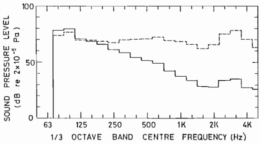

Figure 7: Space averaged sound pressure measurements with final rig configuration. --- Bare plate. — With "typical lagging.

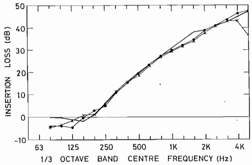

Figure 8: Predicted and measured ⅓ octave band insertion losses for "typical" lagging. Measured values with final rig configuration. — Theoretical values. — Derived from measured intensities. — Derived from measured pressures.

COMMENTS OF THE RESULTS

It is well known that it is difficult to make reliable acoustic intensity measurements in an acoustic field which is highly reactive or contains a significant diffuse field component. It can be seen, with reference to Figures 3 and 4 that in the 400 Hz ⅓ octave band the sound pressure level was 60dB and the sound intensity level was 44dB giving a Reactivity Index of 16dB. The sound pressure level and the sound intensity level were 58dB and 40dB respectively in the 500 Hz ⅓ octave band and so the Reactivity Index was 18dB. It was decided, in view of these high Reactivity Index values, that the initial rig shown in Figure 1 would need modification to reduce the Reactivity Index of the sound field in which the intensity probe operated. The initial rig was modified to produce the final rig shown in Figure 2. It can be seen from Figures 6 and 7 that the Reactivity Indices in the 400 and 500 Hz ⅓ octave bands were reduced to 3dB as a result of the modifications. Satisfactory measurements then could be made. The excellent agreement between the predicted and measured insertion losses can be seen in Figure 8. It is also noteworthy, that over much of the frequency range, the insertion losses derived from pressure and intensity measurements are identical. The main implication of this result is that the methods used to control the flanking transmission were too effective and so over much of the frequency range of interest, acoustic intensity methods were not in fact necessary to measure the insertion loss produced by the lagging. The zealous application of effective methods for controlling flanking transmission arose from the difficulties encountered initially.

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The Usage of Sound Intensity Techniques

for Studying the Effects of Bounding Surfaces on the Radiated Sound Power of Sound Sources

M.P. Norton and J. Soria

Department of Mechanical Engineering
University of Western Australia
Nedlands, 6009, Western Australia

ABSTRACT: Sound intensity measurement techniques are utilised to demonstrate that bounding surfaces can affect the radiated sound power of sound sources at frequencies where the distance, d, from the acoustic centre of the source to the reflecting plane is less than the corresponding acoustic wavelength, λ. Theoretical upper limits, based on a constant volume source model, for the variations in radiated sound power indicate that if the source is located at the centre of a large flat reflecting surface, or at the intersection of two large flat reflecting surfaces, or at the intersection of three large flat reflecting surfaces, the radiated sound power is ~ 3, ~ 6, or ~ 9 dB respectively greater than what it would be in free space. These values are in addition to the well known directivity factor. In practice, the general trend is for the increases to be somewhat less than that predicted by the constant volume model because the effects on non-perfect reflections from the bounding surfaces will reduce the effect of the image source. These observations are very important for engineering noise control — they are generally not well understood by noise control engineers and are therefore often overlooked when estimating noise radiation levels from sound sources.

1. INTRODUCTION

It is a common assumption in most noise control texts that the radiated sound power of a source is constant, irrespective of the source location within the environment. However, for certain types of sources, the close proximity of rigid reflecting surfaces can significantly increase the sound power radiated at low frequencies. Practically, this means that sources situated close to rigid boundaries such as walls, floors and corners may produce sound pressure levels greater than for free-field operation, and consequently exceed specified noise criteria. The assumption of a constant power sound source is based upon the approximation that the acoustic radiation impedance of a source in a free-field remains the same when the source is relocated in some environment other than a free-field. This is not always the case and, often for vibrating and radiating structures, a better approximation is to assume that the sources are constant volume sources, i.e. the motion of the vibrating surface is unaffected by the acoustic radiation load, implying an infinite internal impedance. Bies[1] discusses the effects of variations in acoustic radiation impedance on the sound power of various types of sound sources.

This paper reviews the theoretical concepts associated with three different sound power models (constant power sources, constant volume sources and constant pressure sources) and reports on some experiments that have been conducted using a custom-built portable, sound intensity measurement system to analyse the effects of bounding reflecting surfaces on the radiated sound power of a small domestic vacuum cleaner motor and a pneumatic hand drill.

2. THEORETICAL CONCEPTS

Most industrial noise sources are mounted on a ground plane or in close proximity to it. In the far-field, they can often be approximated as single point sources. The effects of the ground plane have to be accounted for though. These effects are particularly pronounced when the sound source is less than one acoustic wavelength (λ) from the ground plane. The analysis which follows is not to be confused with the concept of directivity which the reader might be familiar with. The directional effects of floors, intersecting walls, corners, etc. on the sound radiation characteristics of an omni-directional noise source are well known and documented. This section relates to a point which is often omitted in the literature on noise control engineering — that the radiated sound power of a source can be affected by rigid, reflecting planes.

Consider the case of a monopole near a rigid, reflecting, ground plane. At some point in the far-field, the sound pressure will be the sum of two sound waves — i.e. a direct and a reflected wave. The reflected wave can be modelled by an image monopole below the reflecting surface. The problem thus reduces to that of two interfering monopoles. In practice, the ground plane will have some finite reflection coefficient (not all the sound will be necessarily reflected) and there will be some finite phase difference between the two waves. If, as an upper limit, one assumes that the ground plane is a hard reflecting surface, then the reflection coefficient is unity and the phase difference between the two waves is zero. The problem thus reduces to two in-phase monopoles of equal source strength (a dipole is modelled as two out-of-phase monopoles of equal source strength).

The combined velocity potential at the observer position (some point, X, in space) can be obtained from the monopole velocity potential, Φ, with the source strength $Q_1(t)$ and $Q_2(t)$ being of equal strength and phase. It is[2]

$$\phi(r,\theta,t) = -\frac{Q_p e^{i(\omega t - kr)}}{4\pi r} 2\cos(kd\cos\theta) \quad , \qquad 1$$

where Q_p is the peak value of the velocity potential, r is the radial distance, θ is the angular position, k is the wavenumber and d is the distance from the acoustic centre of the source to the ground plane. When $d << \lambda$, $kd << 1$ and the above equation simplifies to

$$\phi(r,t) = -\frac{2Q_p e^{i(\omega t - kr)}}{4\pi r} \quad . \qquad 2$$

Equation 2 is simply double the far-field velocity potential for a monopole sound source! The hard, reflecting ground plane has resulted in a doubling of the velocity potential which in turn produces a fourfold increase in the sound intensity. There is only a twofold increase in the radiated sound power because the intensity only needs to be integrated over half space (the other half is baffled by the rigid ground plane). The sound intensity is[2]

$$I(r) = \frac{Q_{rms}^2 k^2 \rho_0 c}{4\pi^2 r^2} \quad , \qquad 3$$

and the radiated sound power is

$$\Pi = 2\pi r^2 I(r) = \frac{Q_{rms}^2 k^2 \rho_0 c}{2\pi} \quad . \qquad 4$$

The interesting result to come out of this limit analysis is that the radiated sound power of the monopole has been doubled. This is essentially because while the r.m.s. strength, Q_{rms}, and the surface vibrational velocity of the source have not changed (from when it is radiating into free space), the reflecting plane has produced a doubling of the velocity potential and the acoustic pressure. So, instead of having a constant sound power, the source has a constant volume velocity. These concepts of constant volume velocity sources as opposed to the more commonly referred to constant power sources are discussed from an engineering noise control point of view in this paper. They can be regarded as an upper limit — in practice the effects on non-perfect reflections from the ground plane will reduce the effect of the image source. Norton[2] discusses a more rigorous analysis for the effects of a reflecting plane on a monopole.

Now, for a simple omni-directional sound source,

$$\Pi = \frac{4\pi r^2 I}{Q} \quad , \qquad 5$$

where I is the sound intensity, r is the distance from the source, and Q is the directivity factor (note that Q is not a source strength here). For a constant power source, $\Pi = \Pi_0$ = a constant; hence, as Q increases, p_{rms}^2 and I increase. If, for argument, the source were a constant pressure source, p_{rms}^2 = a constant and as Q increases Π would decrease. The concept of a constant pressure source is a theoretical one and, as will become evident shortly, it represents the lower limit of variations in radiated sound power. If the source were a constant volume source, Π would increase as Q increases; thus an increase in p_{rms}^2 and I is a function of both Q and Π.

From the preceding discussion it can be seen that for a constant power source, the effect of the ground reflector is to fold the sound field back on to itself; for a constant volume source, in addition to this, the pressure is doubled. By considering velocity potentials and analysing the problem from fundamentals, it is evident that instead of a twofold increase in intensity (as would be expected if a directivity factor of two was allocated to the baffled source) there is an additional factor to be accounted for — the radiated sound power of the source has increased! The velocity potential (and hence the acoustic pressure) everywhere has now doubled. Thus, for a constant volume source, $\Pi = \Pi_0 Q$ and equation 5 becomes

$$I = \frac{\Pi_0 Q^2}{4\pi r^2} \quad . \qquad 6$$

By taking logarithms on both sides

$$L_p = L_{\Pi_0} + 10\log_{10} Q^2 - 10\log_{10} 4\pi r^2 \quad . \qquad 7$$

Based on the arguments presented in the preceding paragraphs, in principle, three sound power models can be postulated: constant power; constant volume; and, constant pressure. The effects of source position on these sound power models are summarised in *Table 1*. From the table it can be seen that if a sound source is modelled as a constant power source, the source position does not affect its radiated sound power; if a sound source is modelled as a constant volume source, reflecting surfaces increase the radiated sound power of the source; if a sound source is modelled as a constant pressure source, reflecting surfaces decrease the radiated sound power of the source. The constant volume model is a conservative model and represents an upper limit. In reality, most practical sources fall somewhere in between the constant power model and the constant volume model — i.e. hard reflecting surfaces do have an effect on the sound power radiated by the source at frequencies where the distance, d, from the acoustic centre of the source to the reflecting surface is smaller than the acoustic wavelength ($d << \lambda$). As mentioned in the introduction, constant power and constant volume sources can be thought of in terms of acoustic radiation impedances: the widely used constant power source model is based upon

TABLE 1 **Variations in radiated sound power for different sound power models**

Source Position	Directivity Q	Sound power model Const. power $\Pi = \Pi_0$	Const. volume $\Pi = \Pi_0 Q$	Const. pressure $\Pi = \Pi_0/Q$
Free space	1 (+0 dB)	Π_0	Π_0	Π_0
Centre of a large flat surface	2 (+3 dB)	Π_0	2 Π_0 (+3 dB)	$\Pi_0/2$ (−3 dB)
Intersection of two large flat surfaces	4 (+6 dB)	Π_0	4 Π_0 (+6 dB)	$\Pi_0/4$ (−6 dB)
Intersection of three large flat surfaces	8 (+9 dB)	Π_0	8 Π_0 (+9 dB)	$\Pi_0/8$ (−9 dB)

the approximation that the acoustic radiation impedance of a source in a free-field remains the same when the source is relocated close to reflecting surfaces, whereas the constant volume source model accounts for the fact that the motion of many vibrating surfaces (machine covers, engine blocks, small motors, pumps, etc.) is unaffected by the acoustic radiation load, implying an infinite internal impedance. The constant pressure source model is only a theoretical concept, representing a lower limit. There is some discussion amongst researchers that certain aerodynamic noise sources can be modelled as constant pressure sources (Bies[1]), but this point needs to be quantified.

The experiments reported on in this paper, using sound intensity measurement techniques, illustrate that the radiated sound power of a small domestic vacuum cleaner motor and a pneumatic hand drill is dependent upon the environment. When the distance, d, from the acoustic centre of the source to the reflecting plane is less than the acoustic wavelength, λ, the radiated sound power is not constant. The general trend is for the increases to be somewhat less than that predicted by the constant volume model. Typical increases in radiated sound power for sound sources positioned in a corner, over the corresponding free-field values, are of the order of 6-8 dB at those dominant source frequencies where $d \ll \lambda$.

3. DETERMINATION OF SOUND POWER USING THE SOUND INTENSITY TECHNIQUE

3.1 General comments

The principle of sound intensity (and sound power) measurement using the two microphone technique is well established and will therefore only be briefly outlined here. Progress in the application of sound intensity techniques to noise control engineering can be found in a recent paper by Maling[3].

The sound power, Π, of a noise source can be approximated by a finite sum of the form

$$\Pi \approx \sum_{k=1}^{N} I_{n,k} \, \Delta S_k \quad , \qquad 8$$

where $I_{n,k}$ is the average normal sound intensity component over the surface area ΔS_k. This is usually achieved by dividing the imaginary surface into smaller discrete surfaces and measuring the normal sound intensity at a fixed position on these smaller surfaces. The sound intensity is then given by the sum of Equation 8. An alternative, is to sweep the sound intensity probe over a larger representative surface area. The sound intensity is thus averaged during the sweeping process. The multiplication of the average normal sound intensity by the swept surface area results in the radiated sound power through that representative surface area. The sweep technique was used to determine the radiated sound power in this investigation.

3.2 The custom-built signal processing unit

Radiated sound power estimates, using the sound intensity technique, can be obtained practically via analogue or digital techniques. The custom-built signal processing unit used in this work is an analogue one. The processing unit is small, portable, relatively inexpensive and easy to construct. The basis of the processing unit is a quarter-square multiplier configuration. Some of the prerequisites for this unit were that it should include: (i) the setting of the calibration constants for the two microphones; (ii) analog outputs after the signal processing stage to allow for tape recording or digital analysis; (iii) external octave or one-third-octave filters and (iv) true mean-square averaging and immediate display of the overall axial sound intensity vector. The term "axial" refers to the direction which is collinear with the two microphone membranes.

The two microphone output voltages, V_1 and V_2, are applied to the individual inputs into the unit where they are multiplied by their respective calibration constants, i.e. α_1 [pressure/volts] and α_2 [pressure/volts]. The gain in each channel can be set to the particular calibration constant of the microphone by using an internal reference source. At this stage, both microphone signals are also ac-coupled and low-pass filtered. This process is achieved by using matched broadband amplifiers with individual variable gains in conjunction with matched low-pass filters. This section of the circuit has less than 1.5° phase variation between both channels in the frequency range 100 Hz — 12.8 kHz. The difference between the two pressure signals is then fed into an integrator stage. A considerable amount of time was spent in optimising the integrator. The final version which is installed in the signal processing unit has a gain accuracy which is better than 0.1 dB, and a phase which is accurate to within 2° in the frequency range 100 Hz — 12.8 kHz. The signal from the integrator is fed separately into a sum and difference amplifier. In the sum amplifier, the sum of the two microphone pressures is added to the integrating amplifier output signal. In the difference amplifier, the sum of the two microphone pressures is subtracted from the integrating amplifier output signal. The sum and difference amplifiers in the signal processor unit have a gain error of less than 0.06 dB, and the relative phase difference between them is less than 1° in the frequency range 100 Hz — 12.8 kHz. It should be pointed out that the error values given in the preceding paragraph are the worst case values. These worst case values occur at the high frequency end of the operating range for all the components with the exception of the integrator. The low frequency phase error for the sum and difference amplifiers and the filters is approximately ±0.1°. These points are verified with the accurate results obtained during the calibration procedure which was conducted in a standing wave tube (see section 3.3 and Table 2).

The signal analysis required to evaluate the sound intensity component I_x in terms of the signals I_1 and I_2 from the second set of sum and difference amplifiers is quite straightforward. The output from the final summing amplifier is given by

$$I_1 = \int (p_1(t) - p_2(t))\, dt + (p_1(t) + p_2(t)) \quad , \qquad 9$$

and the output from the final difference amplifier is given by

$$I_2 = \int (p_1(t) - p_2(t))\, dt - (p_1(t) + p_2(t)) \quad . \qquad 10$$

In practice, the signals are usually stationary and the expected square values of Equations 9 and 10 can be time-averaged and subtracted from each other. Denoting the time-averaged value by an overbar, the process yields,

$$\overline{I_1^2} - \overline{I_2^2} = 4\, \overline{(p_1(t) + p_2(t)) \int (p_1(t) - p_2(t))\, dt} \quad . \qquad 11$$

Hence, from the two microphone definition of sound intensity, the sound intensity vector component in the x-direction is

$$I_x = \frac{\overline{I_1^2} - \overline{I_2^2}}{8 \rho \Delta x} \quad . \qquad 12$$

The mean square value of a signal can also be evaluated by the area under the auto spectrum of the signal. Thus, between the frequencies f_1 and f_2,

$$I_x(f_1, f_2) = \int_{f_1}^{f_2} \frac{\left[G_{11}(f) - G_{22}(f) \right]}{8 \rho \Delta x} df \quad , \qquad 13$$

where $G_{11}(f)$ and $G_{22}(f)$ are the one-sided auto spectra of $I_1(t)$ and $I_2(t)$ respectively.

3.3 System calibration

A Bruel & Kjaer (B & K) type 3519 sound intensity probe system was used. This system can be used with 6.35 mm diameter type 4135 microphones or 12.7 mm diameter type 4165 microphones. The microphones are arranged in a face-to-face configuration, separated by a plastic spacer of either 6 mm or 12 mm for the 6.35 mm micrphones and 12 mm or 50 mm for the 12.7 mm microphones. The probe microphones are calibrated using a B & K pistonphone type 4220. The calibration constants are then set on the sound intensity signal processing unit. This minimises any error due to gain differences between the two microphones. Both the 12 mm spacing/6.35 mm microphone and the 50 mm spacing/12.7 mm microphone configurations were used in the experiments reported on here, to extend the useful frequency range, as recommended by Gade[5].

The complete sound intensity measurement system was calibrated in a pipe with a standing sound wave (with a standing wave ratio of ~ 3-17 dB). The sound intensity of a plane standing wave in a pipe can be estimated by measuring the minimum and maximum mean square pressure along the pipe. It can be shown[5] that using these two values, the sound intensity can be calculated from

$$I_x = \sqrt{\frac{\overline{p^2_{min}}}{\rho c} \frac{\overline{p^2_{max}}}{\rho c}} \quad , \qquad 14$$

where I_x is the sound intensity in the axial direction and ρc is the characteristic impedance. A horizontally suspended steel pipe 3.050 m long with an inside diameter of 203 mm was used for this purpose. A loudspeaker was fixed at one end, whilst the other end was left open. The first higher order acoustic cut-off frequency was calculated to be at 940 Hz. Sinusoidal excitation was applied to the loudspeaker below the first acoustic cut-off frequency at 480 Hz, 547 Hz, 716 Hz, 775 Hz and 836 Hz. The frequencies used for the sinusoidal excitation were in close proximity to the organ pipe mode frequencies. Using a 1 m long extension rod, the sound intensity probe was inserted into the pipe to determine the minimum and maximum sound pressure level for each particular excitation frequency. Equation 14 was then used to determine the sound intensity. The sound intensity was also measured with the sound intensity probe and the signal processing unit. The results of both methods are summarised in *Table 2*.

It should be pointed out, that the outputs I_1 and I_2 from the signal processing unit were fed into a two-channel spectrum analyser, where the sound intensities from the two auto spectra G_{11} and G_{22} and Equation (14) were evaluated. The results in Table 2 are very encouraging with the largest discrepancy being only 1.3 dB at 547 Hz, all other measurements being within 1 dB. This is particularly so because such a sound field is purely reactive, and it is well known that differential phase shifts in the microphone channels lead to an overestimation of sound intensity in highly reactive fields.

TABLE 2
Sound intensity measurements in a pipe with sinusoidan excitation of organ pipe modes

Frequency (Hz)	I (using eqn 14) (dB)	I_x (S.I. system) (dB)
480	95.5	95.5
547	86.5	87.8
716	95.3	95.6
775	97.5	97.8
836	98.1	98.0

The residual reactivity index, $L_{k,o}$, (Gade[5] discusses reactivity indices in some detail) of the complete sound intensity system (12.7 mm microphones & 50 mm spacer) was estimated to be −13.1 dB at 250 Hz and −17.0 dB at 500 Hz. The reactivity indices, L_k, of the test sources were typically −3.5 dB at 250 Hz and −2.3 dB at 500 Hz. Hence, the low frequency dynamic capability of the system is well in excess of the recommended 7 dB[5].

As a further qualitative test, sound pressure level measurements were obtained at distances of 350 mm, 700 mm and 1400 mm from the acoustic centres of the sources. In each case, there was a clear decrease in sound pressure level (~ 3-5 dB) with each doubling of distance, indicating that the field was not highly reactive. Also, the experimental results obtained with the custom-built signal processing unit compared very favourably with similar experimental results obtained with a Data Precision DATA 6000 digital signal analyser (using the cross-spectral technique) the only phase error present in the latter case being the phase error associated with the B&K sound intensity probe.

4. EXPERIMENTAL CONSIDERATIONS

Two common appliances (a small domestic vacuum cleaner motor and a pneumatic hand drill) were chosen for the experimental measurements because their compact size, output noise spectra and lack of strong directionality meant that they could reasonably be approximated as monopole sound sources at low frequencies.

Sound power measurements were first made for the vacuum cleaner motor. For this machine, the major noise sources are the electric motor itself, suction fan noise and vibration and structure borne noise radiated by the plastic casing. It has been shown in the theoretical section of this paper and in other recent investigations (for example Zhao and Zheng[6]) that the increase in radiated sound power caused by bounding surfaces is significant only at frequencies where $d << \lambda$. For a practical source placed in contact with a floor or the junction of intersecting surfacs, the dimension d is the effective radius of the course. This dimension is approximately 85 mm for the vacuum cleaner motor, and a sound wave with this wavelength corresponds to a frequency of ~ 4 kHz. Increases in radiated sound power would therefore only be expected at frequencies somewhat less than ~ 1 kHz for the vacuum cleaner motor section. From the sound pressure level spectrum in *Figure 1* (uncalibrated sound pressure level scale relative to 2×10^{-5} Pa) it was observed that the noise output peaked at ~ 2800 Hz, but that identifiable components existed at frequencies as low as 800 Hz. This suggests that the overall radiated sound power of the vacuum cleaner motor should not be affected by the bounding surfaces — i.e. only the sound

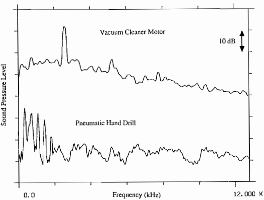

Figure 1: Sound pressure level spectra of the vacuum cleaner motor and the pneumatic hand drill (relative units, dB re 2 × 10^{-5} Pa)

power associated with frequency components less than ~ 1kHz should be affected.

A small pneumatic hand drill was chosen as the second source because of its very small size and the ability to adjust the running speed to give a noise output with high sound levels at low frequencies. A primary peak of ~ 350 Hz was selected with a number of strong harmonics falling in the frequency range of interest, resulting in a noise source with distinctly different spectral characteristics to those of the vacuum cleaner motor. This is clearly evident from the second sound pressure level spectrum which is also presented in Figure 1. The effective radius of the hand drill is approximately 40 mm. This corresponds to an acoustic wave with a frequency of ~ 8.6 kHz, implying that increases in radiated sound power could be expected at frequencies below ~ 2kHz. Here, since the dominant frequencies in the sound pressure spectrum are less than 2 kHz, the overall radiated sound power of the hand drill should be affected by the bounding surfaces.

An important consideration concerning the relative sound powers radiated in the various measurement locations is to ensure that other factors do not alter the running condition of the source over the time period required for the experiments. In the case of the electric vacuum cleaner motor, no significant fluctuations in input electrical power or operational efficiency would be expected to occur as all measurements were taken consecutively once the motor had been 'warmed up' to a steady state. For the pneumatic hand drill, a large constant pressure air supply was used, and the control valve used to regulate the running speed was securely taped to prevent accidental adjustment of the setting. It may be concluded therefore that any variation in radiated sound power that did occur was due to the effect of the nearby reflecting surfaces, and that no other sound power variations occurred.

The sound power radiated by the two experimental sources was determined using the sound intensity measurement system discussed earlier in this paper. Signal digitisation, sampling and data processing were performed with a Data Precision DATA 6000 digital analyser controlled by an HP-86 microcomputer. Experimental measurements were conducted in a large room which provided an environment in which the sound field was not diffuse for the types of sound sources used. Four source measurement locations were used within the room, these being: (i) the source suspended by a boom 1600 mm above the tiled linoleum floor; (ii) on the floor in the centre of the room; (iii) at the junction of the floor and a plaster coated brick wall; and (iv) in a corner.

Sound intensity measurements were made using a cubic surface enclosing the source. The sweep technique was used to scan the sound intensity probe over the measurement surfaces with an extension arm and a hand grip whilst the microphone signals were being sampled and analysed. A cube with 515 mm sides was selected to define the surface over which the sound intensity was measured, and the total radiated sound power was determined by summing the contributions for all the cube faces which were not coincident with a reflecting plane. The cube dimension was chosen such that the source could be kept at some distance from any one measurement face (and hence from the probe), as recommended by Rasmussen [7] and Wu and Crocker[8]. Six scan lines per surface were used, resulting in a characteristic sampling length, l_s, of nearly 85 mm. This dimension (l_s) is small compared to both the distance from the acoustic centre of the source to the scan surface, and an acoustic wavelength in the frequency range of interest. This relates to the accuracy of approximating the true surface integral by a number of discrete measurements and is discussed in some detail by Pope[9].

Care was taken during the experimental program to use a constant and consistent a scanning speed as possible to improve the precision for hand-swept scanning, as recommended by Bockhoff[4]. In addition, a slow sweep rate was used so that at least two samples were taken per scan, and a total of 50 averaged samples were taken for each surface. Due to the airflow generated by both appliances, a spherical windshield was used with the sound intensity probe. The effect of a windshield is to reduce the accuracy of the individual sound intensity measurements. However, it has been demonstrated by Rasmussen[10] that this effect cancels for relative measurements made under similar conditions. Thus, the differences between the sound power radiated for a source in free space and in contact with various reflecting surfaces (e.g. a corner) can be used with confidence.

It is a recognized fact that reflections due to microphone clips, extension arms and so on can cause errors in the determination of sound intensity. However, the B & K sound intensity probe and extension have been designed to minimize such errors. Similarly, it might be expected that the reflecting surfaces (i.e. walls and floor) could cause measurement errors when the sound intensity probe is in close proximity to them during a scanning process. Investigations conducted by Thompson and Huynh[11] and Thompson and Tree[12] indicate that for $d << \lambda$ (i.e. low kd), the errors associated with such reflections are insigificant. Parameter limits of $0.1 \leq k\Delta r \leq 1.3$ and $0 \leq \Delta r/r \leq 0.5$ are suggested for a "worst-case" design with a maximum inaccuracy of ± 1.5 dB ($k = \omega/c$ is the wavenumber, Δr is the microphone separation, and r is the distance between the source and the measurement point). The system used in the experiments reported on here is well within these operational guidelines.

5. DISCUSSION OF RESULTS AND CONCLUSIONS

The experimental results for the radiated sound power of the domestic vacuum cleaner motor and the pneumatic hand drill are presented in *Figures 2, 3 and 4*. Narrowband (100

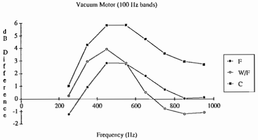

Figure 2: Relative radiated sound power levels (dB difference) for the vacuum cleaner motor. (F — difference between floor and free space values; W/F — difference between wall/floor intersection and free space values; C — difference between corner and free space values.)

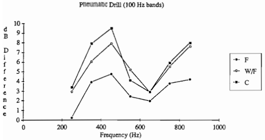

Figure 3: Relative radiated sound power levels (dB difference) for the pneumatic hand drill. (F — difference between floor and free space values; W/F — difference between wall/floor intersection and free space values; C — difference between corner and free space values.)

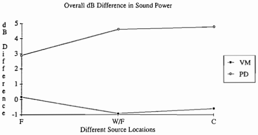

Figure 4: Relative overall radiated sound power levels (dB difference) for the vacuum cleaner motor (VM) and the pneumatic hand drill (PD) for different source locations. (F — difference between floor and free space values; W/F — difference between wall/floor intersection and free space values; C — difference between corner and free space values).

Hz frequency bands) and overall sound power measurements were obtained. The results in Figures 2 and 3 are relative radiated sound power levels (dB differences between the different locations and the free space value) for the narrow band measurements at low frequencies where $d << \lambda$ - i.e. the results are presented in 100 Hz frequency bands with center frequencies ranging from 250 Hz to 950 Hz. The significant increases in radiated sound power for the different locations are clearly seen. The results presented in Figures 2 and 3 clearly illustrate that the radiated sound power of a sound source is dependent upon its environment when the distance, d, from the acoustic centre of the source to the reflecting plane is less than the acoustic wavelength, λ — i.e. reflecting surfaces can affect the sound power characteristics of a sound source. The results presented in Figure 4 are relative overall radiated sound power measurements (dB differences between the different locations and the free space value). The overall sound power radiated by the vacuum cleaner motor is not affected by the bounding, reflecting surfaces, whereas the overall sound power radiated by the pneumatic hand drill is. This is because the dominant source frequencies for the former (see Figure 1) are greater than ~ 2 kHz — i.e. in the region where $d > \lambda$, whilst the dominant source frequencies for the latter (see Figure 1) are less than ~2kHz — i.e. in the region where $d << \lambda$.

The results in Figure 2 demonstrate that increases in low frequency narrowband radiated sound power are observed even though the dominant source of sound is at a much higher frequency (see Figure 1). However, because one is measuring sound pressure levels that are some 15 dB below the peak level at ~ 2800 Hz one would expect the errors in the measurement system to be larger due to a lower signal to noise ratio. The negative dB differences of ~ 1.5 dB that were recorded in some frequency bands are probably a consequence of this. What is clear is that whilst the reflecting surfaces do not have an effect on the overall radiated sound power (Figure 4), they do have a noticeable effect on the radiated sound power in the low frequency bands.

The results in Figure 3 pertain to a situation where the dominant source of sound is within the low frequency region. The peaks at 350 Hz, 450 Hz, 750 Hz and 850 Hz correspond, in general terms, with the maxima in the sound pressure level spectra (Figure 1), and the reflecting surfaces do have a distinct effect on the overall radiated sound power (Figure 4).

Two main conclusions come out of this study. Firstly, reflecting, bounding surfaces do indeed affect the radiated sound power of sound sources at frequencies where the distance, d, from the acoustic centre of the source to the reflecting plane is less than the acoustic wavelength, λ. Theoretical upper limits, based on a constant volume source model, for the variations in radiated sound power indicate that if the source is located at the centre of a large flat reflecting surface, or at the intersection of two large flat reflecting surfaces, or at the intersection of three large flat reflecting surfaces, the radiated sound power is ~ 3, ~ 6 or ~ 9 dB respectively greater than what it would be in free space. Secondly, in practice, the general trend is for the increases to be somewhat less than that predicted by the constant volume model because the effects on non-perfect reflections from the bounding surfaces will reduce the effect of the image source.

As already stated earlier on in this paper, these results are very important for engineering noise control — they are generally not well understood by noise control engineers and are therefore often overlooked when estimating noise radiation levels from sound sources.

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ACKNOWLEDGEMENTS

The financial support provided by Worksafe Australia (National Occupational Health and Safety Research Grant) is gratefully acknowledged together with the assistance provided by Mr. S.J. Drew in parts of the experimental programme, and Mr. R. Greenhalgh in the design and construction of the custom-built, portable sound intensity measurement system.

(Received 20 April 1988)

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REPORT

Some Experiences with Sound Intensity Measurements

Joseph Lai and Marion Burgess
Department of Mechanical Engineering
University College, University of New South Wales
Australian Defence Force Academy
A.C.T. 2600

ABSTRACT: Sound intensity methods in a normal environment are compared with standard laboratory methods for the measurement of (a) sound power from a small hand drill, (b) sound transmission loss of a wall.

1. INTRODUCTION

The Australian Defence Force Academy, incorporating a University College of the University of New South Wales, received its first intake of students in 1986. In the following year, an extensive range of items of equipment for use in the acoustic facilities within the Department of Mechanical Engineering were purchased and the anechoic room commissioned (Lai, 1987). This equipment included a Bruel & Kjaer Dual Channel FFT Analyser, type 2032, with a sound intensity probe, type 3519 and a dedicated micro-computer. Over the last year experience with the use of the sound intensity system has been obtained in a number of areas. This note outlines some of the findings associated with the measurement techniques.

2. SOUND POWER

With the use of sound intensity measurements, the determination of sound power from machinery and other noise sources does not have to be made in a special acoustic environment, such as an anechoic room. The sound power is determined directly from the measurement of the sound intensity normal to a surface forming an hypothetical enclosure for the source. The sources of error associated with the measurements of sound intensity have been given by Gade (1985). Basically the lower frequency limit is related to the phase mismatch of the microphones and the upper frequency limit is determined by the finite difference approximation used in the derivation of the relationship between the particle velocity and the pressure at the two microphones.

In theory, for measurements of the sound power of a source, the effect of external noise should be negligible as any sound which passes into the enclosing surface through one area will pass out through another. In practice there is a limit to this external noise suppression and this was examined by measuring over a surface enclosing no sound source. The external noise was provided by a loudspeaker source having a sound power of 89 dB. The apparent sound power of the nonexistent source range from −78.4 to 77.8 dB. The results, similar to those of Stirnemann et al (1985), indicate an external noise suppression of around 10 dB for the sound intensity system used. A simplified model, developed to examine the sensitivity of the apparent power measured to the error, ϵ, associated with the intensity measurement showed that for low values of ϵ the noise suppression changed rapidly, e.g. for ϵ from 0 to ± 0.5 dB the noise suppression ranged from >50 dB to 20 dB (Lai and Dombek,1987). For an error ϵ, of ± 1 dB, the noise suppression was of the order of 10 dB which agrees with most measurements reported to date.

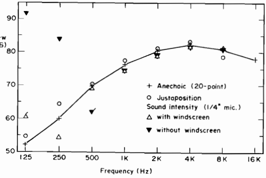
Good agreement has been obtained between determinations of the sound power of sources from measurements of the sound pressure level over a hemisphere in a free field (anechoic room) and from sound intensity measurements in a normal environment (a room of volume 85 m³ with a reverberation time of 0.8 sec at 500 Hz). The comparisons for a small hand-drill can be seen on Figure 1. For the juxtaposition measurements a B&K reference sound source, type 4205, was used in the anechoic room. As the motor in the hand drill sets up air circulation, the measurements at the low frequencies without a windscreen led to very high values. The measurements with the windscreen indicate suppression of the effects of the mean air flow.

3. ACOUSTIC PROPERTIES OF BUILDING MATERIALS

Measurements of sound intensity can be used for determination of the sound transmission loss (STL) of building materials. The STL is defined as:

$$STL = 10 \log (W_i/W_r) \quad (1)$$

where W_i is the incident sound power
W_r is the radiated sound power



A direct measure of the sound power radiated by the partition can be determined from the sound intensity measurements on the receiving side of the test wall. The incident sound power is determined from measurements of sound pressure in the source room. The conventional methods for determination of STL are based on measurements of the sound pressure levels in reverberant source and receiving rooms. Results obtained using intensity measurements have been shown to agree with those obtained using the conventional methods in traditional testing facilities (e.g. Cops, Minten and Mynke, 1987). The errors and limitations associated with such measurements have been investigated by van Zyl, Erasmus and Anderson (1987).

There have been few reports in the literature of field measurements of transmission loss where it would seem that the use of sound intensity for a direct measurement of the transmitted sound power would reduce some of the problems inherent in determination of field transmission loss by the conventional methods. One of the difficulties has been that while the probes are relatively small and compact, the analysing equipment and associated computers have been bulky and somewhat difficult to move around. Nielsen (1986) has reported on field measurements made at the Building Research Establishment with a battery operated analyser.

Recent experiences by the authors with measurements of the field transmission loss of some walls in a building highlighted some of the practical considerations necessary. Some of the walls to be tested comprised the facade of the building. It was considered that the area surrounding the building would be an ideal receiving space as it would be non-reverberant. Background noise was a potential problem and as the walls were likely to have a high noise reduction the measurements were made at times when the ambient noise was minimum.

The first results indicated large amounts of negative intensity indicating that the sound energy was travelling into the wall, not out from it! It soon became apparent that the fluctuations of the wind, which was less than 5m per sec, were having a significant effect on the results even though the windshield was used. As it was impossible to continue with the measurements by avoiding the wind gusts, some freestanding screens were used to provide windscreening. These screens were constructed from a lightweight timber framework, 1.2 m wide by 2.2 m high, clad with fibreglass and a thin cloth covering. With these screens in position, about 1 m either side of the area of wall to be scanned, the effect of wind fluctuations was minimised.

For some of the one-third octave bands of interest, the difference between the sound pressure levels and the sound intensity levels was high (of the order of 15 dB) which indicated that the accuracy of the intensity measurements was limited. An examination of the narrow band data for these frequency ranges showed that there were significant amounts of negative intensity. This most likely resulted from the effects of flanking transmission from elements of the construction other than those under test. A procedure for minimising the limitations to the use of sound intensity measurements arising from flanking transmission by adjusting the absorption in the receiving space has been examined by van Zyl and Erasmus (1987).

The field transmission loss values obtained for one of the test walls are shown on Figure 2. Only the data for which the difference between the pressure and intensity levels was within an acceptable range is plotted. The agreement with published data for a similar construction can be seen from this figure.

4. CONCLUSIONS

Experiences with a sound intensity system have indicated that sound power data can be obtained with a minimum of effort. However it is important that the inherent errors and limitations of the system are appreciated at the time of the measurements. The pressure-intensity difference provides a good indication when the measurement conditions are such that accurate values for the sound intensity are not likely to be obtained. An examination of the narrow band data can also indicate problem areas, hence the advantage of an FFT analysis system over a real time analysis system.

Future areas for investigation will include studies of the radiation patterns of vibrating objects and further work on investigations of the acoustic properties of building materials both inside an anechoic room and under field conditions.

(Received 26 April 1988)

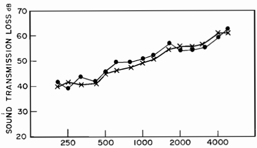


Figure 2: Comparison of sound transmission loss data
•——• *Field tests using measurements of sound intensity*
x——x *Published data from conventional tests in laboratory*

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REPORT

The Influence of Background Noise on Sound Power Determination by Measuring Sound Intensity in Different Environments

N. Tandon
ITMMEC, Indian Institute of Technology
New Delhi 110016 India

ABSTRACT: The effect of the background noise levels on the sound power determination by sound intensity technique is investigated. The scanning method of measurement is used in an ordinary room and in an acoustic chamber. The results indicate that the sound power can be determined accurately even if the background noise level is higher than the source level, in both the rooms. Background noise levels of about 5 dB higher than the source level can be accepted for an accuracy of ± 1 dB in the sound power levels.

1. INTRODUCTION

One of the advantages of determining sound power of sound sources by sound intensity measurements is that the measurements can be performed even in the presence of steady background noise. The measurements are conducted with the probe held normal to a hypothetical surface, called the control surface, enclosing the source and assuming that there is no sound absorption within this surface, so that the background noise entering the surface is equal to the noise leaving the surface. So it is even possible to perform measurements when the extraneous noise levels are higher than the source levels. But in practice limitations are imposed by the dynamic range of the instruments, sound field distribution and the measurement distance from the source [1-2]. The present work is aimed at investigating the extent of the influence of the background noise on sound power obtained from sound intensity. The measurements were performed in two different environments of an ordinary room (96 m³ volume) and an acoustically damped room (66 m³ volume) whose walls and ceiling have been treated with sound absorbing materials [3]. The sound absorption coefficient for the acoustic room varies between 0.83 and 0.91 in the frequency range of 125 Hz to 8 kHz. The average reverberation time of the ordinary room is 0.93, 1.06, 1.07 and 1.01 sec. at 250, 500, 1k and 2k Hz respectively.

2. EXPERIMENT

All the intensity measurements have been performed over a 1 metre cubical box shape control surface enclosing a reference sound source (SS). A speaker used as a source of producing background noise was placed at a distance of 1.5 m from one edge of the base of the control surface box. Sound intensity was measured normal to the five surfaces of the box using a face to face intensity probe with 12 mm spacer, a dual channel FFT and a desk top computer. A random noise signal was fed to the speaker through a power amplifier. Sound intensity was measured by sweeping (scanning method) the probe manually with an approximate speed of 0.3 m/s and with the distance between sweeps being 20 cm (approx.). The data were stored in the computer and the sound power determined in the frequency range 125 to 5k Hz.

Before determining the sound power of the reference source, SS, with background noise, the sound power of the source alone without speaker was determined six times in each room to study the reproducibility of the measurements. The results of these measurements are given in Table 1. As seen in the table quite good reproducibility was obtained in both the rooms, with a lower standard deviation for the acoustic chamber.

The sound power obtained without the speaker in each room is taken as the reference sound power of the source. The difference between the background and the source noise, ΔL_p, is defined as

$$\Delta L_p = L_{p,BN} - L_{P,RS}$$

where $L_{P,BN}$ is the sound pressure level of the speaker, fed with a random noise signal, measured at the base edge of the control surface box, facing the speaker and $L_{P,RS}$ is the sound pressure level of the reference source at the same point. ΔL_P levels of −10, −5, 0, 10, 15 and 20 dB were used for obtaining sound power of the source in both the rooms. The difference in the sound power levels of the source obtained with the background noise and the sound power level of the source without background noise (speaker off) is denoted as ΔL_w

$$\Delta L_w = L_{w,BN+RS} - L_{w,RS}$$

Table 1— Reproducibility of Measurements

Measurement No.	Sound Power Measured, dB	
	Ordinary room	Acoustic chamber
1	79.6	79.6
2	79.5	79.5
3	79.7	79.5
4	79.7	79.5
5	79.7	79.5
6	79.7	79.6
Standard deviation, σ_n	0.076	0.047

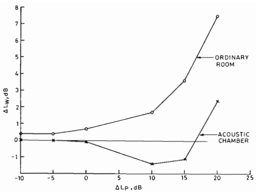

Figure 1: Influence of the difference of background and source noise level, ΔL_P on the accuracy of sound power determination.

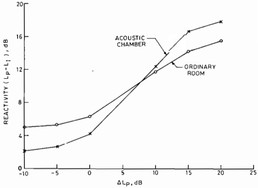

Figure 2: Reactivity vs background and source sound pressure level difference, ΔL_P

3. RESULTS AND DISCUSSION

The values of ΔL_w obtained for both the rooms are plotted against ΔL_p in Figure 1. It is seen that the error in sound power determined, rises rapidly when $\Delta L_p > 10$ dB in the ordinary room. In the acoustic chamber ΔL_w becomes negative and then again positive. Bockhoff [2] also reports ΔL_w becoming negative in the free field conditions after certain ΔL_p levels indicating that the measured power levels with background noise become lower than the sound power of the source. The results obtained indicate that sound power can not be determined accurately if ΔL_p is higher than about 10 dB. ΔL_p levels of approx. 5 dB are acceptable for ± 1 dB accuracy in sound power, for both the rooms. This means that sound power can be determined by sound intensity technique even if the background sound pressure level is higher than that of the source.

Figure 2 shows the average reactivity, the difference between the measured value for pressure (L_P) and intensity (L_I), around the control surface for different values of ΔL_P. As expected, the accuracy of sound power determination in Figure 1 suffers with increasing value of reactivity. The speed of the sweep was kept constant throughout the measurements whereas it should be slower for high values of reactivity. So probably better results (i.e. higher levels of ΔL_P being acceptable) can be expected if the scanning speed is slower for higher ΔL_P levels.

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(Received 26 April 1988)

BOOK REVIEW

FREQUENCY ANALYSIS

R. B. Randall

Bruel & Kjaer Publication, 1987, 344 pp. Price $60.

The newly published third edition of B & K's frequency analysis book covers both signal analysis, often referred to as one channel analysis, and system analysis, often called two channel analysis. Rather than focusing on the traditional distinction between analog and digital analysis, the author, R. B. Randall, prefers to distinguish between analysis performed using FFT (fast fourier transformation) and analysis performed using filters, where digital filtering is only regarded as a particular case of the latter.

What characterises this book is its balance between in-depth-going background theory on signal processing, explained in a pictorial and easy-to-understand manner, and examples of applications ranging from the electro-acoustic to the machine diagnostic fields.

In it the reader will find all necessary definitions from the simplest, such as power spectral density and averaging process, to more elaborate ones such as impulse response, cross-correlation or frequency response. Less straight forward analysis (such as the analysis of short or long transients or the analysis of non stationary signals) are explained in detail. The merits of different techniques are discussed in a manner which enables easier selection of the most appropriate. Examples cover reciprocating machine cycle analysis, fast or slow run-up and coast-down, and speech analysis.

Newer techniques, such as Hilbert transform and Cepstrum analysis, are extensively covered. Examples of application of the Hilbert transform for amplitude demodulation in the diagnostics of rolling element bearing faults, as well as phase demodulation for analysis of torsional vibration in reciprocating machines, are illustrated. The various applications of Cepstrum analysis for echo removal, evaluation of reflecting properties of surfaces, detection of voiced speech and determination of voice pitch as well as determination of harmonic and side band patterns for diagnostic in rotating mach'nes are described. The use of complex Cepstrum for echo removal in time domain and general deconvolution is also mentioned.

Besides many original examples of measurements, this book contains an extensive bibliography. It can be used as a text book or, by use of its very complete index, as a reference book.

The author, Bob Randall, after working as an application engineer in signal analysis and machine diagnostics for 17 years in Bruel & Kjaer, is now appointed as Senior Lecturer in Mechanical Engineering at the University of New South Wales, Sydney, Australia.

Joelle Courrech

NEW PRODUCTS

Echotech

Tec Smart Meter

The Tec Smart Meter is a predictive maintenance analyser which is totally automated. Predictive maintenance is a relatively new concept in the field of machinery maintenance. Predictive maintenance is a systematic programme of regularly monitoring machinery to determine the actual mechanical condition while under operating conditions. By continuous or periodic monitoring of machine parameters such as vibration, electrical current, temperature, pressure or other process variables, and by comparing the results to previous or normal operation "baseline" readings, developing problems can be detected. With this advanced early warning of developing machinery problems, dramatic improvements can be made in maintenance planning.

In addition to the route scanning survey functions, the SMART METER displays and stores complete signatures for on-the-spot trouble shooting or for fault diagnostics/root cause analysis back at the host computer. The features include selectable FFT resolution from 100 lines up to 1600 lines, making it especially useful for analysing more complex machine trains. With this expanded capability, TEC offers a Plot Expand Function which permits the user to more readily identify a frequency region of the plot by expanding the region to fill the entire display. TEC also offers a back-lit display for use in dimly lit areas.

The SMART METER can be employed as a useful teaching apparatus for Mechanical, Electrical, Civil and Chemical Engineering departments. The Model 1320 second generation SMART METER offers a 16 bit 8086 microprocessor, a dynamic range of better than 65db, automatic selection of the correct input amplifier gain, automatic spurious data rejection, and the ability to measure phase which enables the dynamic balancing of rotating machinery.

Further information: Echotech Pty Ltd, 6/22 Bridge St, Eltham, Vic 3095 (tel (03) 439 5222).

Bruel & Kjaer

Software for Machine Fault Diagnosis

Bruel & Kjaer Type 7616 Application Software is designed to help the maintenance engineer co-ordinate the vibration monitoring activities of a maintenance team using one or more Type 2515 Vibration Analysers. The Type 7616 organises the collection of vibration data and process parameters, such as temperature, speed and load, and automatically reports changes in the vibration spectrum. Special attention is paid to machine process parameters in order to make relevant comparisons, since vibration spectra vary with these parameters. The Type 7616 distinguishes between 30 different process parameters per machine.

By using a three-dimensional plot (showing the vibration increase of several spectra simultaneously) and performing a trend analysis of vibration increase, the maintenance engineer can schedule maintenance in advance of a predicted breakdown. It is also possible to make trends in any combination of process paramaters.

The Type 7616 runs on the IBM XT or AT Personal Computers. Access to the different functions of the Type 7616 can be limited to suit the responsibilities of individual members of the maintenance team. The routines for the day-to-day collection of data are designed to be used by any member of the team.

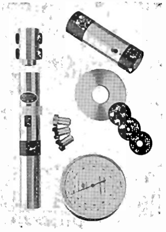
New Sound Intensity Probe

Sound Intensity Probe Type 3545 is a lightweight two-microphone probe for measuring sound intensity in the frequency range 20 Hz to 10 kHz. It has a single cable, terminated by an 18-pin plug, which is specially designed for connection to Dual Channel Real-time Frequency Analyser Type 2133.

Type 3545 is supplied in an attache case containing Microphone Pairs Types 4178 and 4181, a ¼" dual preamplifier, spherical and ellipsoidal windscreens and a telescopic rod for holding the probe.

The probe can also be fitted to Remote Control ZH 0354, which services all measurement and control functions. Extension cables are available to enable measurements to be performed up to 100 metres from the analyser. The dual preamplifier is connected to the tip of the Remote Control Unit via an 18-pin plug, which both allows signals to pass directly to the analyser and carries the polarisation voltage for the microphones.

Complete calibration of sound intensity measurement systems which use the Type 3545 can be conveniently made with Sound Intensity Calibrator Type 3541. This permits simultaneous sensitivity adjustment of both channels of the analyser (in pressure, particle velocity or intensity mode) and allows determination of the Residual Pressure-Intensity Index of the microphone-preamplifier-analyser combination.



Sound Intensity Calibrator

Bruel & Kjaer's new Sound Intensity Calibrator Type 3541 enables users to calibrate fully their intensity-measuring equipment. Probe microphones are inserted into the unique coupler which, in conjunction with a pistonphone, simulates a plane wave passing along the axis of the microphone probe.

Type 3541 is supplied with a calibration chart which states the levels of sound pressure, sound intensity and particle velocity which are to be detected in the coupler. Correction terms for the calibration levels when conditions are different to the original calibration are also given on the calibration chart, together with instructions for using the Type 3541.

In addition to the calibrations, Type 3541 can be used to measure the residual pressure-intensity index spectra of intensity-measuring equipment. This is important if the equipment is to be used accurately. Residual pressure-intensity index spectra are measured by using the coupler and a broad-band sound source.

Fully Automated Monitoring System

Bruel & Kjaer introduce the answer to machine-condition monitoring, the fully automatic monitoring system based on software package Type WT 9118. The system combines the security of a permanently installed broadband monitoring scheme with the powerful detection capabilities of spectrum comparison monitoring, the ultimate two in one. All signal channels, up to a maximum of 512, are monitored via a Type 2505 Multipurpose Monitor which compares their overall vibration level against three preset limits. Any violation of these limits will trigger alarms via Trip Relay Box WB 0376 and, if necessary, shut down the machine.

The spectrum comparison system monitors each channel sequentially. It produces a proportional-band frequency spectrum of the machines vibration levels for comparison with a similar spectrum taken when the machine was in a known "healthy" condition. As soon as a fault begins to develop, the shape of the frequency spectrum changes and this change will be detected by the spectrum comparison. By analysis of the "fault" spectra via the Type 2033 High Resolution Signal Analyser the cause of the fault can be diagnosed.

All fault spectra detected by the spectrum comparison system are retained on disc and a short fault warning is given via an on-line printer. Spectra for each channel may be stored under six separate speed classifications, allowing different machine operating conditions to be taken into account and minimising erroneous fault warnings. The machine speed is detected via Tacho Interface WB 0915. If a fault is identified, its rate of increase with running time can be displayed. A least-squares extrapolation can then be made, giving a prediction of when a pre-defined "danger" limit will be exceeded in the form of a trend analysis.

Further information: Bruel & Kjaer Aust, 24 Tepko Road, Terrey Hills, NSW 2084.

Envelope Analysis

— the key to rolling-element bearing diagnosis

Diagnosing faults in rolling-element bearings is made easier with the latest in vibration-analysis instrumentation from Bruel & Kjaer. By combining Vibration Analyzer Type 2515 with Envelope Detector WB1048, envelope analysis can be effectively used to identify and diagnose bearing faults.

Envelope Detector WB 1048 contains an eight-position bandpass filter and has three gain ranges. It can also be bypassed during analysis. The instrument is housed in a splash-proof aluminium box and will fit into one of the pockets of the leather case of the Vibration Analyzer Type 2515. Use of the Envelope Detector adds yet another dimension to the battery-operated Type 2515, making it the most powerful, portable fault-detection and -diagnostic tool on the market today.

Metrosonics

RS232 Interface

Metrosonics Inc. announces availability of the Model dt-435 Data Translator that enables the company's noise and industrial hygiene data loggers to interface to an RS-232C device such as personal or mainframe computers, digital recorders and modems.

The dt-435 gives users of the Metrosonics db-301 Noise-Profiling Dosimeter, d1-331 Universal Data logger, db-653 Metroreader and Interscan 5000 series Toxic Gas Dosimeter the flexibility of writing their own programmes for analysing and archiving occupational exposure data, and the opportunity to utilise computers not currently supported by the company's software.

The dt-435 is the size of a pocket calculator and operates on an internal battery or external power. Internal switches allow selection of baud rate, parity and word size, to ensure proper communication with the receiving device.

AUSTRALIAN METROSONICS Pty Ltd is presenting a family of new AHLBORN portable temperature measuring instruments, microprocessor controlled, as seen recently at the Hannover-Fair.

Technical description and further details clarify that this new series THERM 2280 is based on the very modern technology of microprocessing, C-MOS-technology and a 30-year-old experience.

These instruments are based on a homogeneous concept, for example, microprocessor controlled technology, digital linearisation, autocalibration as well as a precise reference-junction compensation. Different kinds of thermocouples or even NTC-, Pt 100- and infrared-sensors can be connected.

The instruments offer temperature ranges of —200 to +1760,0°C with a resolution of 0,1 K or 0,01 K.

All instruments can be run by an alternative power supply: 9 V battery or mains adapter.

Further information: Metrosonics, P.O. Box 120, Mt Waverley 3149, Victoria (tel (03) 233 5889).

ICA — 1989

Dr. Neville Fletcher attended the recent ICA meeting in Bordeaux at which the arrangements for the 13th Congress in Belgrade (Aug. 24-31, 1989) were discussed.

He reports that the planning has advanced to the stage that invitations have been sent to speakers for plenary sessions, acceptances are in, and the programme is in proof stage. The Congress Circular will be posted out in about a month's time and there is a rather heavy emphasis on nonlinear acoustics. Planning is being based on an attendance of about 1000 at Belgrade. The SAVA Centre, in which the Congress is to be held is a large new convention centre, with attached hotel, rather like the venue of the 12th ICA in Toronto, and should be very convenient. The weather in Belgrade at this time of year is likely to be quite warm — high 20s.

For the satellite symposia in Zagreb (September 1-3) on Electroacoustics and in Dubrovnik (September 4-6) on Sea Acoustics the attendance is expected in each case to be about 200. Dubrovnik, in particular, is most attractive from the tourist point of view.

Robert Angus has returned to Queensland to take up a new position at Vipac. Robert, whose special areas of interest are machine and structural vibration, has been working in Vipac's Melbourne office for the past two years.

☆ ☆ ☆

Noela Eddington and **Warren Renew,** both of the Division of Noise Abatement and Air Pollution Control, as well as **Frits Kamst** of Winders, Barlow & Morrison Pty. Ltd. will attend the Noise '88 Conference in Stockholm and the Inter-Noise '88 Conference in Avignon, France. All three are presenting papers.

POLMET 88

Polmet 88 is the second in a series of international conferences and exhibitions on Pollution in the Urban Environment, to be held in Hong Kong from 28th November to 2nd December 1988.

The conference is being organised jointly by the H.K. Institution of Engineers and the H.K. Government Environmental Pollution Advisory Committee. The main theme will be Pollution in the Metropolitan and Urban Environment, with emphasis on issues of relevance to countries in Asia and the Pacific Region.

Prominent organisations and corporations concerned with environmental protection and management have already confirmed their participation in both the exhibition and the conference. The conference has already attracted more than 300 delegates from many countries.

Further information: POLMET 88 Secretariat: 9/F, Island Centre, No. 1 Great George Street, Causeway Bay, HK. Telephone: 5-8954446, Telex: 74679 BINHK HX, Fax: 5-777791.

WESTPAC III

The 3rd Western Pacific Regional Conference is to be held from November 2-4, 1988 in Shanghai, China. The Australian Acoustical Society is a founding member society of Westpac and AAS members are encouraged to participate in this Conference, which will contain presentations of invited and contributed papers on all topics of acoustics plus scientific visits and a technical exhibition. A number of post-conference tours to other parts of China are also planned.

Registration fees are $US150 prior to August 31st and $US180 afterwards. If any AAS member is planning to attend, the Society's Council would be interested to know — could you please inform Mr. R. A. Piesse, General Secretary.

Further information: Secretary of Westpac III c/- Institute of Acoustics, Academia Sinica, 17 Zhongguancun Street, P.O. Box 2712, Beijing, China.

FASE

The 8th Symposium of the Federation of Acoustical Societies of Europe (FASE) will be held during the week of 24th April, 1989. The theme will be **Environmental Acoustics** and the call for papers has been distributed. Provisional title, and abstract, should be forwarded to the Secretariat:

Viajes el Corte Ingles, Dpto. Congresos, Avda, Cesar Augusto, 14, 2a planta, 50004 Zaragoza, Spain.

Standards

A new American National Standard is now available. It is ANSI S12.8-1987 and titled **"Methods for determination of insertion loss of outdoor noise barriers".**

This standard presents methods for determination of outdoor noise barrier insertion loss. It does not standardise methods to compare the performance of different barriers or to generalise or certify the performance of a particular barrier for different situations. Where background noise prevents determination of full insertion loss, determination of a lower bound to that value may be feasible. The recommended noise descriptors are the time averaged A-weighted level or octave band sound pressure level, or A-weighted sound exposure level, although use of other appropriate descriptors is not precluded.

ADVERTISER INDEX

Readers are asked to mention this publication when replying to advertisements.

FUTURE EVENTS —

● Indicates an Australian Conference

1988

September 5-7, CRACOW

CONFERENCE ON NOISE CONTROL 88
Details: Dr. R. Panuszka, Organising Committee Conference Noise Control 88, Inst. of Mechanics & Vibroacoustics AGH, A1. Mickiewicza 30, 30-059 Krakow, Poland.

October 3-5, CHICAGO

IEEE ULTRASONICS SYMPOSIUM
Details: Univ. Illinois, Bioacoustics Research Lab., Atten.: W. D. O'Brien Jr., Urbama, Illinois 61801, USA.

October 4-7, HIGH TATRA

ELECTROACOUSTICS
27th Conference.
Details: House of Technology, Eng L. Goralikova, Skultetyho ul. 1, 832 27, Bratislava, Czechoslovakia.

October 15-16, WASHINGTON

HISTORY OF ULTRASOUND
Details: American Institute of Ultrasound in Medicine, 4405 East-West Highway, Suite 504, Bethesda, MD20814, USA.

October 17-21, WASHINGTON

WFUMB/AIUM MEETING AND
2nd CONGRESS OF SONOGRAPHERS
Details: AIUM, Conventions & Education, 4405 East-West Hwy., Suite 504, Bethesda, MD 20814, USA.

November 2-4, SHANGI

WESTPAC III

Developments of Acoustics in the Western Pacific Region.
Details: Secretariat Westpac III, Institute Acoustics, Academia Sinica, 17 Zhongguancun St, Beijing, China.

November 14-18, HONOLULU

2nd JOINT MEETING OF ACOUSTICAL SOCIETIES OF AMERICA AND JAPAN
Details: Secretariat ASA-ASJ Joint Meeting, Ac.Soc.Japan, Ikeda Bldg 4F, Yoyogi 2-7-7, Shibuya, Tokyo 151, Japan.

November 14-17, KOBE

9th INTERNATIONAL ACOUSTIC EMISSION SYMPOSIUM
Details: Prof. Dr. I. Kimpara, Dept. Naval Architecture, Faculty of Eng., University of Tokyo, 3-1, Hongo-7, Bunkyo-ku, TOKYO 113, JAPAN.

● November 24-25, VICTOR HARBOUR

NOISE INTO THE NINETIES
Details: R. P. Williamson, School of Built Environment, SAIT, Nth. Terrace, SA 5000.

November 25-27, WINDERMERE

AUTUMN CONFERENCE — IOA
Noise In and Around Buildings
Details: Institute of Acoustics, 25 Chambers Street, Edinburgh, EH1 1HU, U.K.

November 28 - December 2, HONG KONG

POLMET 88
Pollution in the Metropolitan and Urban Environment.
Details: Polmet 88 Secretariat, c/- Hong Kong Institution of Engineers, 91F, Island Centre, No 1, Great George St, Causeway Bay, Hong Kong.

1989

March 7-10, HAMBURG

86th AES CONVENTION
Details: Herman Wilms, Exhibition Director, Zevenbunderslaan 142/9, Brussels, Belgium 1190.

April 3-5, LIVERPOOL

MODERN PRACTICE IN STRESS AND VIBRATION ANALYSIS
Details: Meetings Officer, Institute of Physics, 47 Belgrave Square, London, SWIX 8QX, U.K.

• April 10-14, PERTH

1989 NATIONAL ENGINEERING CONFERENCE
Developing Australia's Resources
Details: Conference Manager, 1989 Nat. Eng. Conf., Institution of Engineers, 11 National Circuit, Barton, ACT 2600.

April 24-28, ZARAGOZA

8th FASE SYMPOSIUM
Environmental Acoustics
Details: Viajes el Corte Ingles, Dpto Congresos, Avda. Cesar Augusto, 14, 2a planta, 5000 4 Zaragoza, Spain.

April 25-29, GLASGOW

INTERNATIONAL CONFERENCE ON ACOUSTICS, SPEECH AND SIGNAL PROCESSING
Details: Inst. Elect. & Electronic Eng., Conference Co-ordinator, 345 E 47th St., New York, NY 10017, USA.

May 22-26, SYRACUSE

MEETING OF ACOUSTICAL SOCIETY OF AMERICA
Details: Murray Strasberg, ASA, 500 Sunnyside Blvd., Woodbury, New York 11797, USA.

May 23-27, GDANSK

4th SPRING SCHOOL ON ACOUTO-OPTICS
Details: Prof. A. Sliwinski, Inst. of Experimental Physics, University Gdansk, Wita Stwosza 57, 80 952 Gdansk, Poland.

June 7-10, PECS

6th SEMINAR ON NOISE CONTROL
Details: Optical, Acoustical & Film-technical Soc., FO u. 68, H-1027, Budapest II, Hungary.

August 16-18, SINGAPORE

INTERNATIONAL CONFERENCE NOISE & VIBRATION 89
Details: The Secretariat, International Conference Noise & Vibration 89, c/- School of Mechanical & Production Engineering, Nanyang Technological Institute, Nanyang Ave., Singapore 2263.

August 19-22, MITTENWALD

INTERNATIONAL SYMPOSIUM ON MUSICAL ACOUSTICS
Details: Sekretariat des ISMA 1989, c/- Muller-BBM, Robert-Koch-Str 11, 8033 Planegg, W. Germany.

August 24-31, BELGRADE

13th ICA

September 4-6

SYMPOSIA
Sea Acoustics — Dubrovnik.
Electroacoustics — Zagreb.
Details: 13 ICA Secretariat, Sava Centre, 11070 Belgrade, Yugoslavia.

October 4-6, MONTREAL

IEEE/UFFCS
Ultrasonics Symposium.
Details: Allied-Signal Inc., Atten.: H. van de Vaart, PO Box 10221R, Morristown, NJ 07960, USA.

October 18-19, BARCELONA

II WORLD CONGRESS OF CHRONICAL RONCOPATHY
"Snore and OSAS Syndrome."
Details: Prof. E. Perello, Facultat de Medicina, Universitat Autonoma de Barcelona, Passeig de la Vall D'Hebron, S/N 08035 Barcelona, Spain.

November 6-10, ST LOUIS

MEETING OF ACOUSTICAL SOCIETY OF AMERICA
Details: Murray Strasberg, ASA, 500 Sunnyside Blvd., Woodbury, New York 11797, USA.

December 4-6, NEWPORT BEACH

INTER-NOISE 89
Details: Inter-noise 89, Inst. Noise Control Eng., PO Box 3206, Poughkeepsie, NY 12603, USA.

1990

May 21-25, PENNSYLVANIA

MEETING OF ACOUSTICAL SOCIETY OF AMERICA
Details: Murray Strasberg, ASA, 500 Sunnyside Blvd., Woodbury, New York 11797, USA.

November 26-30, SAN DIEGO

MEETING OF ACOUSTICAL SOCIETY OF AMERICA
Details: Murray Strasberg, ASA, 500 Sunnyside Blvd., Woodbury, New York 11797, USA.

AUSTRALIAN ACOUSTICAL SOCIETY

1988 ANNUAL CONFERENCE

NOISE INTO THE NINETIES

VICTOR HARBOUR 24 - 25 NOVEMBER, 1988

Venue: Apollon Motel, Victor Harbour

Registration: Registration will commence on the evening of 23rd November, 1988. Registration will be on a limited basis. Invitations will be distributed during July.

Registration Fees: Full-time delegates - $185.00
Accompanying delegates - $80.00
(excluding accommodation and tours).

Further Information: Mr. R. P. Williamson,
School of Built Environment
S.A.I.T.
North Terrace, S.A. 5000
Tel: (08) 236 2227

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