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acoustique canadienne

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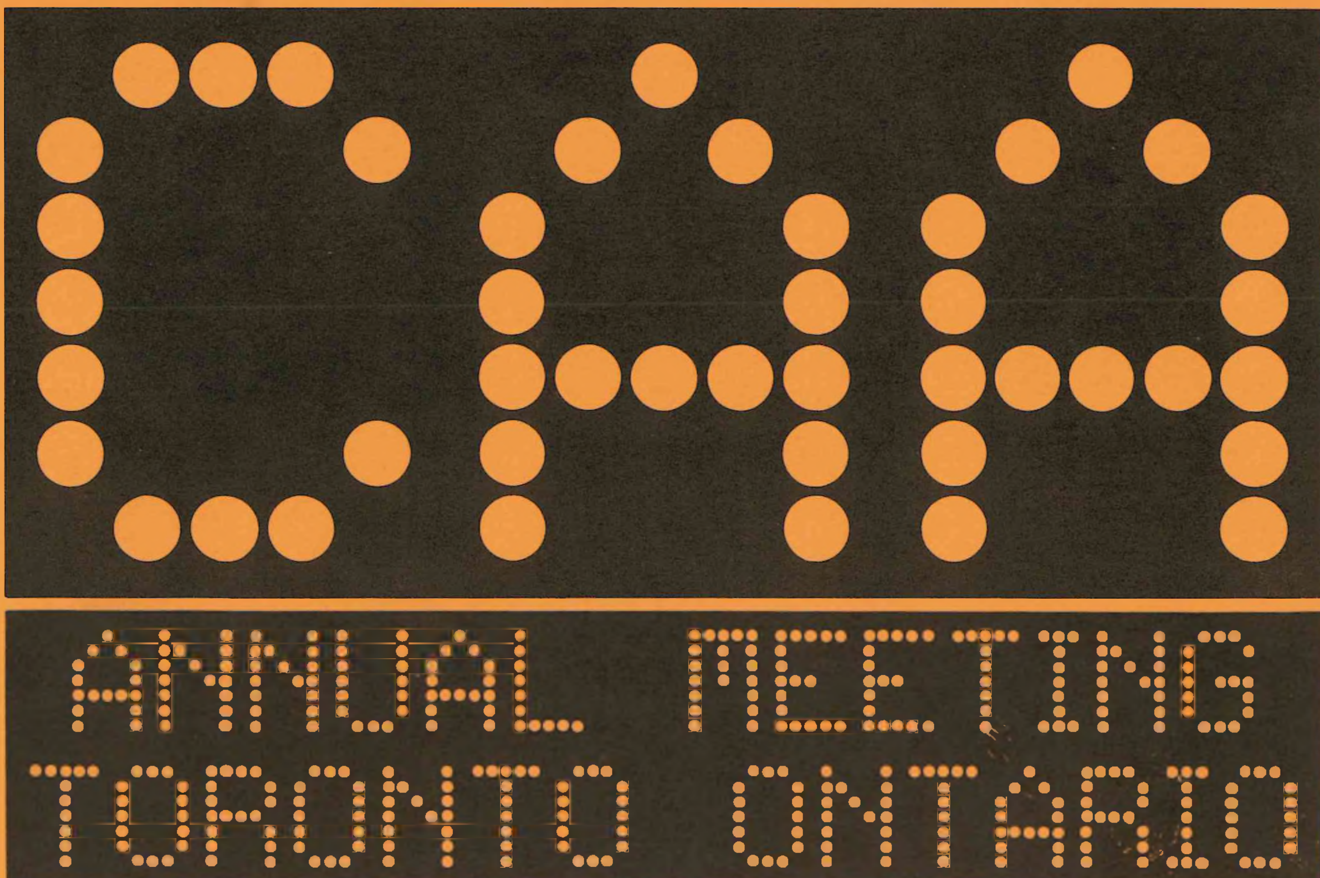

Simon Tuckett

canadian acoustics

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Articles on all aspects of Acoustics and Vibration in English and French are welcome, and should be addressed to any editor. All papers are reviewed.

Vous êtes invités à soumettre des articles sur tous les domaines de l'Acoustique et des Vibrations, en français ou en anglais. Prière de les envoyer à un des rédacteurs. Tout article est révisé.

EDITORIAL

With the publishing of the last 1982 issue of Canadian Acoustics, one is tempted to reflect a little on progress over the year. The name "CANADIAN ACOUSTICS", adopted January 1982, has become established as is the publication itself, now referred to internationally. Our thanks to the authors and referees for the high quality of the papers published. They also reflect the diverse nature of Canadian acoustics ranging from music to underwater sound and from noise control to acoustic reflex response.

We look now to the forthcoming Canadian Acoustical Association meeting in Toronto, October 20th, 1982, to help set the pace for 1983. CAA is heavily involved in preparations for the International Congress on Acoustics (ICA) meeting in Toronto in 1986. We will all have to work to make this a successful meeting and a useful one for Canada. One way you can help is by telling your colleagues about Canadian Acoustics. They may wish to subscribe too.

In addition to 3 seminars, October 18-20, the CAA business meeting October 20, and the CAA Symposium October 21-22, the CSA Committee on Acoustics and Noise Control is holding their annual meeting on Tuesday, October 19, beginning at 4:00 p.m. All interested parties are generally allowed to attend, and it is a most useful meeting for those with an interest in acoustics standards.

EDITORIAL

A l'occasion de la publication de ce dernier numéro de l'Acoustique Canadienne, l'on est tenté de réfléchir un peu sur le progrès accompli au cours de cette année. Le titre "ACOUSTIQUE CANADIENNE" a été adopté en janvier 1982, et l'on s'y refère internationalement. Nous voudrions remercier nos auteurs et nos réviseurs de l'excellent travail qu'ils ont fait. Ce travail reflète la diversité de l'acoustique au Canada qui porte de la musique aux ondes ultrasoniques et du contrôle de bruit jusqu'à la réponse acoustique réflexive. Maintenant, nous attendons la prochaine Réunion Annuelle de l'Association Canadienne de l'Acoustique à Toronto le 20 octobre 1982 afin de nous préparer pour l'année 1983. L'ACA est en pleine activité pour la préparation du Congrès International sur l'Acoustique (CIA) à Toronto en 1986. Nous sommes invités tous afin de faire réussir ce congrès. Une façon de contribuer est de faire part à vos collègues de notre publication "Acoustique Canadienne". Peut-être sont-ils intéressés à s'abonner aussi.

En octobre 1982, les activités suivantes auront lieu: les trois séminaires du 18 au 20 octobre, la réunion d'affaires de l'ACA le 20 octobre et le symposium de l'ACA du 21 au 22 octobre. En plus le comitée de l'ACN sur l'Acoustique et le contrôl de bruit tiendra sa réunion annuelle le mardi 19 octobre à partir de 16.00 h. La réunion est ouverte pour tous ceux qui s'intéressent aux standards acoustiques.

NEWS

ACOUSTICS COURSE IN SICILY

The International School of Physical Acoustics announces their 1st Course on "Fundamental Principles and Applications of Acoustic Waves", November 30 - December 10, 1982. The subscription fee, which includes board and lodging, is $500(US). Closing date for applications is October 31, 1982. For more information contact (1) Prof. W.G. Mayer, Georgetown University, Washington, D.C., U.S.A. 20057 or (2) Prof. A. Alippi, Istituto di Acustica-CNR, Via Cassia, 1216, 00189, Rome, Italy.

JOURNAL OF LOW FREQUENCY NOISE AND VIBRATION

This new journal is edited by Dr. H.G. Leventhall, University of London, U.K., and published by the Multi-Science Publishing Co. Ltd., 42145 New Broad Street, London, EC2M 1QY, U.K. The aim and scope of this new quarterly publication, Annual Subscription $80.00(US), are as follows: -

The considerable and growing interest in the phenomena of low frequency noise and vibration and their powerful effects of man, animals, and the environment, spreads across several disciplies; studies and investigations of these topics are to be found at present in the periodical literature of acoustics, geophysics, architecture, civil and mechanical engineering, medicine, psychology and zoology.

The purpose of this quarterly Journal is to bring scattered material together between one set of covers for the benefit of those working in the relevant areas.

Topics that the Journal will include are: sources of infrasound, low frequency noise and vibration: detection, measurement and analysis; propagation of infrasound and low frequency noise in the atmosphere; propagation of vibration in the ground and in structures; perception of infrasound, low frequency noise and vibration by man and animals; effects on man and animals; interaction of low frequency noise and vibration: vibrations caused by noise, radiation of noise from vibrating structures; low frequency noise and vibration control: problems and solution.

Sample copies are available from the publishers upon request.

NEW RESEARCH CONTRACTS

To Dr. G.R. Branton, Dept. of Chemistry, University of Victoria, Victoria, B.C., $9602, for "Development of a fibre-optic acoustic detector." Awarded by the Dept. of Fisheries and Oceans.

To Northern Seismic Analysis, Echo Bay, Ont., $10,000, for "Analysis of Beaufort Sea seismograms". Awarded by the Dept. of Energy, Mines and Resources.

To Geophysical Service Incorporated, Calgary, Alta., $74,800, for "Processing refraction seismac data". Awarded by the Dept. of Energy, Mines and Resources.

To J.A. McCallum, Medicine Hat, Alta., $27,000, to "Develop the theory of blast wave propagation". Awarded by the Dept. of National Defence.

To Huntec-Lapp Systems Limited, Scarborough, Ont., $23,840, for "Feasibility study for motion sensing of a towed acoustic array". Awarded by the Dept. of National Defence.

To Atlantic Information Systems Limited, Ottawa, Ont., $49,423, for "Instrumentation studies for ship noise reduction - phase II". Awarded by the Dept. of National Defence.

To Arrowsmith Computing Limited, Nanaimo, B.C., $35,000, for "Development and implementation of procedures to analyse acoustical data". Awarded by the Dept. of Fisheries and Oceans.

To Dr. E. Jeffries, Faculty of Medicine, University of British Columbia, Vancouver, B.C., $1000, for "Report on a seminar on vibration induced diseases". Awarded by the Dept. of the Environment.

To Dr. M.J. Clouter, Dr. H. Kiefte, Dept. of Physics, $32,559, to "Study the acoustic properties of Structure I and II gas hydrates by brillouin spectroscopy". Awarded by the Dept. of Energy, Mines and Resources.

To Huntec-Lapp Systems Limited, Scarborough, Ont., $20,170, for "Synthetic aperture sonar study". Awarded by the Dept. of National Defence.

To Norpak Limited, Kanata, Ont., $248,207, for "Development of an integrated voice and graphic display terminal and related test bed compatible with Telidon". Awarded by the Dept. of Communications.

POSITIONS WANTED

Electronic Engineer, P. Eng., with comprehensive research and development-oriented experience in acoustics and related areas, is looking for a challenging position in the field of air or underwater acoustics. Fluent in English, French, Russian, Italian, Dutch.

For complete resume and references, contact:

Philippe de Heering,
22 Ferris Road,
Toronto, Ontario.
M4B 1G3

Tel.: (416) 759-0321

KARL OMAN is looking for a position with an environmental or experimental acoustics team. He is a McGill Architecture Graduate with Physics and Physiology Training (B.Sc., McGill), 2 years architecture experience, environmental and architectural acoustics training, with academic excellence. For further information please contact Mr. Oman directly at 558 16th Avenue, Lasalle, Québec, H8P 2S3 (telephone: 514-366-0399).

B.Sc in Psychology and Physiology
B.A. (Hon.) in Psychology

Trained in Canada, with experience in interior environmenta analysis, major areas including; acoustics, illumination, therma comfort, with some knowledge in radiation, vibration, energy, ai quality, air circulation,and office layout and planning.

I have helped set out a standards and procedures manual in acoustics dealing with topics including; ambient noise, noise criteria, room absorption, noise reduction, transmission loss, standard transmission curve, articulation index, reverberation time, and sound masking.

For a complete resume and references, contact:

Anita Rosenfeld
60 McLeod St., Apt. # 904
Ottawa, Ontario
K2P 2G1
Canada
Phone: (613) 232-7586

POSITION WANTED

M.Eng. in Occupational Health & Safety

Industrial Hygienist is looking for a suitable position in the Occupational Health an Safety field. He holds a Masters of Engineering in Occupational Health and Safe as well as a Bachelors of Science in Physic Chemistry both from the University of Toron

His experience over the past two years includes the prediction and control of work place noise in the design stages of a major electrolytic zinc reduction plant, the prediction of occupational health hazards and control measures for the same plant, noise surveys of power generating stations, noise dosimetry of the construction industry, ventilation studies, hazard assessment of asbestos, formaldehyde, heat stress, wood dust, etc.

For a complete resume and references contac

Paschal Dranitsaris
57 Anewen Drive
Toronto, Ontario
M4A 1R9
Telephone: (416) 757-0314

CANADIAN ACOUSTICAL ASSOCIATION

Outstanding 1982 membership fees must be paid before October 20, 1982, in order to preserve your eligibility to participate in shaping the future path of the Association. Kindly forward your cheque to the Executive Secretary, the address is on the inside back cover.

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Simonsen, K.	Taylor, S.	Walton, T.	

NOTICES

Ontario Model Municipal Noise Control By-Law

The technical publications and provincial standards supporting the by-law have been revised and reprinted and are currently being circulated for peer review. Ninety environmental noise control by-laws have been approved to date by the Ontario Minister of the Environment. It is now proposed to re-draft the existing document into a regulatory format. Comments are invited. The deadline for comments is October 15, 1982.

Sound Attenuation in Lined Ducts. Simplified Design Procedures

Recipients of this publication are advised that an errata sheet is now available. The helpful comments of reviewers is gratefully acknowledged.

Single free copies of either of the abovementioned documents is available on request from John Manuel. See inside back cover for address & telephone.

REPORT ON JOINT CAA/MOE TRAINING COURSES, 1982

At the 1981 CAA Annual Meeting held in Edmonton, Alberta, the membership agreed to cooperate with the Ontario Ministry of Environment in presenting acoustics training courses in Toronto during the week of March 29, 1982.

1. Forty-four trainees successfully participated in three of the six four-day acoustics technology training courses initially advertised.

2. Training was given in:
   - Noise control in the workplace. Emphasis was given to the application of the proposed Ontario noise regulation.
   - Environmental acoustics technology for noise control officers, Level 1.
   - Noise control in land use planning.

3. The training program was enthusiastically supported by local CAA members. The teaching faculty, listed below, will be honoured at the President's Luncheon on October 20, 1982 in Toronto. We applaud their interest and support.

MOE Staff & CAA Members	External CAA Faculty Members	
D. Beach*	J.E.Coulter	BarmanCoulterSwallow
S.D.Benner	S.Gewurtz	Ont.Min of Labour
L. Butko*	A.D.Lightstone	Valcoustics
D.F.Choy*	H. Lofgreen*	Bilsom
J. Clifford	A. McKee*	Bruel & Kjaer
D.Fumerton*	M. Merritt	John Bain & Assoc.
H. Gidamy	D. Ostler*	Bruel & Kjaer
D. Guscott	M. Pike	Ont.Min of Labour
C.Krajewski*	M. Sacks	Tacet Engineering
J.Manuel*	J. Zeidler*	Bilsom
G.Murphy*		
V.Schroter*		

*Workshops, field trip and other forms of peripheral support.

4. The financial report on this activity will be presented at the 1982 Annual General meeting. The funds accumulated are intended to be used towards providing interim financing for the 1986 ICA Canada.

5. It is recommended that this cooperative training effort be expanded to include other jurisdictions in major centres across Canada.

John Manuel, Convenor

Some Simple Formulae for Normal Mode Wave Numbers, Cutoff Frequencies, and the Number of Modes Trapped by a Sound Channel*

Dale D. Ellis

Defence Research Establishment Atlantic
P.O. Box 1012, Dartmouth, Nova Scotia, Canada, B2Y 3Z7
(June 10, 1982)

ABSTRACT

To a good first approximation acoustic propagation in an underwater sound channel is dominated by a finite number of trapped modes. However, exact solutions are known for only a few special cases, making it necessary in general to use numerical methods to solve the normal mode equation. But often one is interested only in the gross features, such as the number of modes or cutoff frequencies, and one does not need the detail provided by a complete normal mode calculation. Even if a normal mode calculation is desired, the computation time can be reduced considerably if the mode wavenumbers can be estimated in advance. In such a case, the WKB method can be used to obtain formulae which, although they are approximate, are given in closed form. In this paper formulae based on exact and WKB solutions are presented for the number of modes trapped in some simple sound channels and for the wave numbers and cutoff frequencies associated with these modes. The number of trapped modes is shown to depend on the gross features of the sound channel, while the distribution of modal wave numbers depends to a greater degree on the details of the sound speed profile shape.

RESUME

Dans une bonne première approximation, la propagation acoustique dans un canal de son sous-marin est dominée par un nombre fini de modes piégés. Cepandant, des solutions exactes ne sont connues que pour quelques cas spéciaux, ce qui oblige en général à utiliser des méthodes numériques pour résoudre l'équation du mode normal. Souvent pourtant, le chercheur ne s'intéresse qu'aux caractéristiques brutes comme le nombre de modes ou les fréquences de coupure et il n'a pas besoin de la quantité de détails fournie par un calcul complet du mode normal. Même, lorsqu'un calcu du mode normal est voulu, le temps de calcul peut être considérablement réduit si les nombres d'onde du mode peuvent être estimés auparavant. Dans un tel cas, la méthode WKB peut servir à obtenir des formules qui, bien qu'elles soient approximatives, se présentent sous une forme fermée. Dans cette communication, l'auteur présente des formules basées sur des solutions exactes et sur des approximations WKB pour le nombre de modes piégés dans des canaux de son simple et pour les nombres d'onde et les fréquences de coupure liés à ces modes. Il démontre que le nombre de modes piégés depend des caractéristiques brutes du canal de son, tandis que la distribution des nombres d'onde modaux dépend dans une plus grande mesure des détails de la forme du profil de la vitesse du son.

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* Presented at the 101st meeting of the Acoustical Society of America in Ottawa, 18-22 May, 1981.

1 Introduction

Acoustic propagation in an underwater sound channel is dominated by a finite number of trapped modes whose wavenumbers depend on the sound speed profile in the channel. Exact solutions are known for only a few special profiles, making it necessary in general to use numerical methods to solve the normal mode equation. But often one is interested only in the gross features, such as the number of modes or the cutoff frequencies, and one does not need the detail provided by a complete normal mode calculation. Moreover, even if a normal mode calculation is desired, the computation time can be reduced considerably if the mode numbers can be estimated in advance. In such cases the WKB (after Wentzel-Kramers-Brillouin and others) method can be used to obtain formulae which, although they are approximate[1], are given in closed form.

In this paper, formulae based on WKB solutions are presented for the number of modes trapped in some simple sound channels and for the wave numbers and cutoff frequencies associated with these modes. The number of trapped modes is shown to depend on the gross features of the sound channel, while the distribution of mode wavenumbers depends to a greater degree on the details of the profile shape. Results are presented for the square (isovelocity-channel) profile, the parabolic profile, and the bilinear profile. An example shows how the simple formulae can be applied to a realistic ocean environment.

While the analysis is presented in terms of underwater acoustics, the results are applicable to other areas, such as transmission in an inhomogeneous waveguide or to the solution of the Schrodinger equation. Most of the results presented here have been known for some time[2], but what is new is that the formulae for some of the more complicated waveguides can be put in the same functional form as the well known formulae for the ideal waveguide. The physical parameters, such as frequency, depth, and sound speed are easily distinguished from the details of the shape of the sound speed profile, which can be treated as a dimensionless quantity. The very simple form of the expressions makes them useful for back of the envelope calculations or for use with a pocket calculator. Moreover, using the same functional form for the expressions allows the effect of the shape of the sound speed profile to be easily seen.

2 The normal mode equation

The normal mode equation can be written as,

$$u_n''(z) + [\omega^2/c^2(z)-k_n^2]u_n(z) = 0 \qquad (1)$$

where,

z is the depth coordinate (increasing with depth from the surface),

$\omega = 2\pi f$ is the angular frequency,

$c(z)$ is the sound speed as a function of depth,

k_n is the wave number or eigenvalue,

$u_n(z)$ is the normal mode function,

and u'' denotes the second derivative of u with respect to z.

In general one wants to determine the normal mode wave numbers k_n and the associated mode functions $u_n(z)$, subject to certain boundary conditions. Notice the quantity $[\omega^2/c^2(z)-k_n^2]$ which will be important in the discussion later; in particular, it is equal to zero at a turning point, where $\omega/c(z) = k_n$.

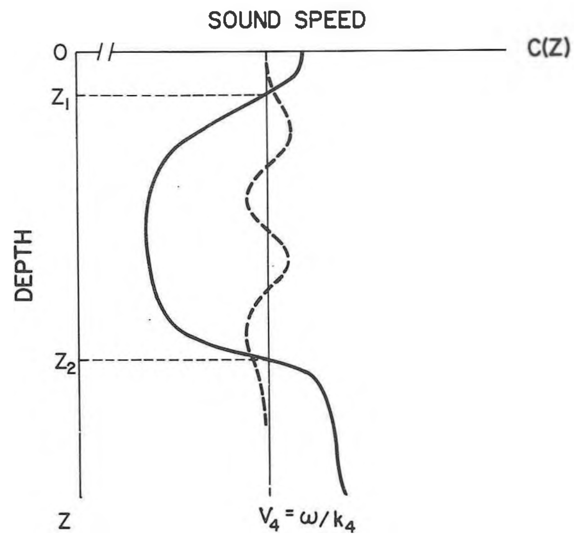

Figure 1. Normal mode solution (heavy dashed line) superimposed on a sound speed profile (heavy solid line).

Figure 1 shows a sound speed profile (heavy solid line) with a normal mode function (heavy dashed line) superimposed (with arbitrary amplitude) at the appropriate phase velocity $v_n = \omega/k_n$. Notice a number of things about the mode function:

1) At the turning points z_1 and z_2, $\omega/c(z_1) = \omega/c(z_2) = k_n$. These are the classical turning points for the equivalent ray travelling in the sound channel.

2) At the air-water interface the pressure, and hence the mode function, is zero; i.e. $u_n(0) = 0$.

3) Between 0 and z_1 the normal mode function has an increasing exponential type of behaviour.

4) Between z_2 and ∞ the solution has a decreasing exponential type of behaviour.

5) Between z_1 and z_2 there are three zero crossings since this is the fourth mode; (in general the n-th mode will have n-1 zero crossings).

Furthermore, note the sinusoidal behaviour between z_1 and z_2; note also that $u_n \simeq \sin(\pi/4)$ at the upper turning point and $u_n \simeq \sin(n-1/4)\pi$ at the lower turning point.

3 The WKB method

If the sound speed profile is changing slowly with respect to an acoustic wavelength, the WKB approximation[1] allows the solution to be written in terms of a slowly varying amplitude r(z) and a monotonically increasing phase $\phi(z)$:

$$u_n(z) = M\, r(z) \sin[\phi(z)] \tag{2}$$

where

$$r(z) = [\omega^2/c^2(z) - k_n^2]^{-1/4} \tag{3}$$

$$\phi(z) = \int_{z_1}^{z} [\omega^2/c^2(z') - k_n^2]^{1/2} dz' + \delta_1, \quad z_1 < z < z_2 \tag{4}$$

and M is a normalization constant. The integral in Eq. (4) will be referred to as the phase integral. Note several points:

1) the term in square brackets is the same term that appeared in Eq. (1).

2) near a turning point r(z) is singular.

3) $\phi(z)$ is well behaved, however, and can be used to determine the WKB eigenvalues k_n.

4) $\phi(z_1) = \delta_1$ at the upper turning point.

5) $\phi(z_2) = n\pi - \delta_2$ at the lower turning point.

Figure 2 shows how the phase δ_1 at the upper turning point depends on boundary effects. If the surface is a pressure release one, the pressure is zero, $u_n(0) = 0$ and the phase $\delta_1 = 0$; if the surface is rigid, the normal derivative of the pressure is zero, i.e. $u_n'(0) = 0$, and the phase $\delta_1 = \pi/2$. At a turning point the phase δ_1 is between 0 and $\pi/2$; $\delta_1 = \pi/4$ is the usual choice[1]. The same comments apply to the phase at the lower turning point.

In the WKB method the total phase change between the turning points is given by:

$$\int_{z_1}^{z_2} [\omega^2/c^2(z) - k_n^2]^{1/2} dz + \delta_1 + \delta_2 = n\pi. \tag{5}$$

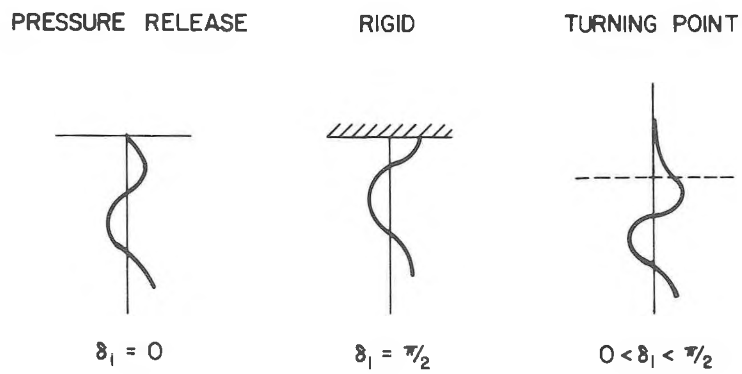

Figure 2. Effect of various boundary types on the phase δ.

where δ_1 and δ_2 are the phases at the upper and lower turning points. With the usual choice of $\delta_1 = \delta_2 = \pi/4$ the WKB eigenvalue equation becomes

$$\int_{z_1}^{z_2} [\omega^2/c^2(z) - k_n^2]^{1/2} dz = (n-1/2)\pi \tag{6}$$

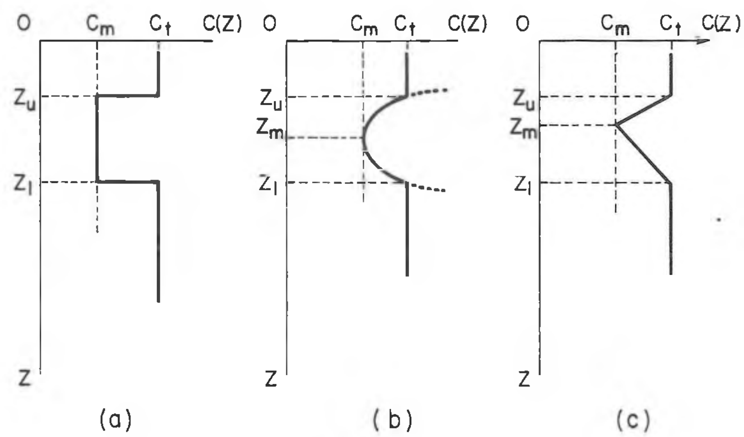

Figure 3. Three simple sound channel shapes: (a) the square isovelocity-channel profile, (b) the parabolic sound channel, and (c) the bilinear sound channel.

One wishes to solve equation (6) for k_n which appears explicitly in the integrand and implicitly in the limits z_1 and z_2. However, for certain sound speed profiles c(z) the integral can be evaluated analytically and an expression obtained for k_n. Three such sound speed profiles are shown in Figure 3 : (a) the square isovelocity-channel profile, (b) the parabolic sound channel, and (c) the bilinear sound channel. Note that it is actually c^{-2} rather than c which is parabolic or linear. Some notation is also introduced at this point: c_m = the minimum sound speed in the channel, c_t = the maximum sound speed in the channel, and $h = z_l - z_u$ = the maximum vertical extent of the sound channel.

4 The number of trapped modes

The WKB eigenvalue equation can be solved for the number of trapped modes N if the phase integral can be evaluated either analytically or numerically. Using $n \to N$, $z_1 = z_u$, $z_2 = z_l$, $k_N = \omega / c_t$ and rearranging Eq. (6) gives $h = z_l - z_u$ and $x = z/h$:

$$N = 1/2 + (2hf/c_m)[1-c_m^2/c_t^2]^{1/2} \int_0^1 \{[c_m^2/c^2(x)-c_m^2/c_t^2]/[1-c_m^2/c_t^2]\}^{1/2}\, dx \tag{7}$$

By introducing the dimensionless quantities

$$a = [1-c_m^2/c_t^2]^{1/2} \tag{8}$$

and

$$\alpha = \int_0^1 \{ [c_m^2/c^2(x)-c_m^2/c_t^2]/[1-c_m^2/c_t^2]\}^{1/2}\, dx \tag{9}$$

equation (7) can then be written as

$$N = 1/2 + (2hf/c_m)\, a\, \alpha \tag{10}$$

The quantity a is introduced strictly for notational convenience. The quantity α, however, is related to the shape of the sound speed profile, but contains none of the physical parameters such as the frequency, depth or sound speeds. Note that in the case of an isovelocity or square profile $\alpha = 1$, and Eq. (10) gives the classical formula for the number of trapped modes. Figure 4 graphically illustrates the significance of the integral in Eq. (9), where the vertical extent of the channel $z_l - z_u$ gets mapped into the range 0 to 1, and where the sound speeds between c_m and c_t get mapped into 0 to 1 and where the integrand of Eq. (9) (denoted by g(x)) is enclosed in a square box of unit size. The integral α is given by the shaded area, which can be calculated analytically or numerically or even estimated by eye.

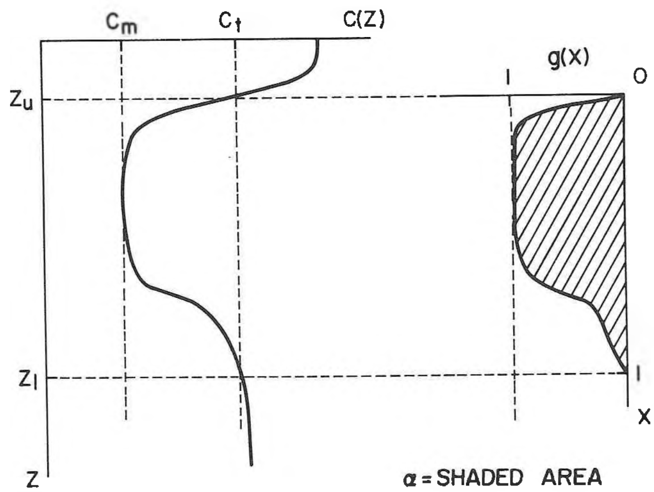

Figure 4. The sound channel and the associated function g(x); the shaded area α is defined in Eq. (9).

5 Cutoff frequencies

The cutoff frequency for the n-th mode can be obtained by rearranging Eq. (10) to give

$$f_n^{cut} = [c_m(n-1/2)] \;/\; [2h a\alpha] \qquad (11)$$

6 Wave numbers

For the three sound speed profiles shown in Figure 3, the eigenvalue equation (6) can be solved for the wavenumbers k_n, giving results of the form

$$k_n^2 = \omega^2/c_m^2 - A(n-1/2)^p \qquad (12)$$

where the specific values of A and p are given in Table 1. For the three profiles considered p varies between 2/3 and 2, while the corresponding values of the shape parameter α varies only between 2/3 and 1. Moreover, α enters equations (10) and (11) in a linear fashion, while p appears as an exponent in equation (12). Thus, the wave numbers are much more sensitive to the profile shape than are the trapped modes and the cutoff frequencies.

Table 1

Specific values of α, p and A for three simple profiles.

Profile Shape	α	p	A
Square	1	2	$(\pi/h)^2$
Parabolic	$\pi/4 = 0.79$	1	$4a\omega/(c_m h)$
Bilinear	2/3	2/3	$[3\pi\omega^2 a^2/(2c_m^2 h)]^{2/3}$

7 Example

Table 2 shows an example of how the formulae might be applied to a realistic ocean environment, and compares the wave numbers with those obtained from a normal mode calculation. The sound speed profile approximates a typical summer sound speed profile in 100 m of water on the Scotian Shelf: a 20 m isovelocity layer of speed 1520 m/s at the surface, a minimum sound speed of 1460 m/s at a depth of 40 m, and a speed of 1490 m/s at the bottom. The table compares the wave numbers, or phase velocities, obtained using the bilinear formula with those from a complete normal mode calculation at 200 Hz.

Table 2

Comparison of the phase velocities obtained from equation (12)

with those from a complete normal mode calculation.

Mode Number	Phase Velocities (m/s)		
	Normal mode	Equation (12)	Difference
1	1465.72	1465.46	-0.26
2	1471.34	1471.43	0.09
3	1475.99	1476.15	0.16
4	1480.13	1480.29	0.16
5	1484.94	1484.09	0.15
6	1487.48	1487.63	0.15

The WKB method gives a good approximation to the normal mode wave numbers; four or five digits accuracy as in the above example is not unreasonable. In fact for the parabolic profile the WKB and exact calculations give the identical results. Provided that this is of sufficient accuracy, the value of the analytic formula is obvious from a computational point of view.

8 Discussion

Equations (10)-(12) together with the values of α, p and A given in Table 1 summarize the results of this paper: simple analytical formulae for normal mode wave numbers, cutoff frequencies, and the number of modes trapped in a sound channel of simple shape. The factor of 1/2 appearing in Eqs. (10)-(12) can be generalized to a δ which depends on the boundaries of the sound channel as well as the type of turning point that the mode "sees". Equations (10) and (11) for the number of modes and the cutoff frequencies are useful for more general profiles provided that the integral of Eq. (9) can be estimated.

The results show the sensitivity of the modes to the shape of the sound speed profile: the number of modes depends on the shape parameter α of the sound speed profile, while the distribution of mode numbers is more sensitive to the details of the profile shape.

ACKNOWLEDGEMENT

Useful discussions were held with my colleagues at D.R.E.A., in particular with David Chapman who made a number of valuable suggestions.

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STATISTICAL ENERGY ANALYSIS

A BRIEF INTRODUCTION

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ABSTRACT

Statistical energy analysis (SEA) in a powerful tool in the study of high frequency vibration. A brief review is given of the basic ideas of SEA by describing some sample calculations and listing some areas where SEA has been applied successfully.

SOMMAIRE

L'analyse statistique d'énergie est un puissant instrument de travail pour l'étude des vibrations à haute fréquence. Une brève revue des idées de base de cette analyse est présentée à l'aide d'exemple de calculs et d'applications couronnées de succès.

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INTRODUCTION

Statistical energy analysis (SEA) is a way of studying dynamical systems. The method was developed in the 1960's by R. H. Lyon and others at the research and consulting firm Bolt, Beranek and Newman. A description of the basic theory and of typical methods of application of SEA is given in the text by Lyon [1]. SEA can provide reliable estimates of vibration levels on complicated systems by using very simple models. Its use is most appropriate at high frequencies in complicated structures with very many degrees of freedom. Despite its power, however, the technique has not been adopted as wholeheartedly as one might expect by design engineers [2].

The intent of this paper is to draw the attention of readers of Canadian Acoustics/Acoustique Canadienne to SEA by describing the basic ideas of SEA through a sample application, and by listing some typical areas where SEA has been used very successfully.

BASIC SEA

As a sample application we shall discuss the transmission of vibration from the turbulent-boundary-layer excited skin of a small aerospace vehicle to an internal instrument shelf (Figure 1).

SEA uses power and energy as the basic variables to describe systems. A system is divided into two or more subsystems each characterized by an appropriate energy variable. A basic SEA model that describes the skin and shelf vibration of the aerospace vehicle is shown in Figure 2. We use two energy variables, one each for the skin and shelf, respectively. The skin and shelf can also each be represented by groups of resonant modes of vibration. One of the major advantages of using SEA is that detailed information on the modes of vibration is not always necessary. SEA is usually applied at high frequencies to obtain vibration information in octave or third octave bounds. In order to discuss the energy of a subsystem we use only the total mass of the subsystem, m, an average mean-square velocity, $\langle v^2 \rangle$, and the number, N, of vibration modes that have resonant frequencies in the frequency band of interest. (This information is available in the literature for a wide variety of structures.) The relation $E_{tot} = m\langle v^2 \rangle$ relates the total energy to the mean square vibration velocity. Also $E_{tot} = NE$, where each of the N modes characterizing the subsystem has vibrational energy E. Usually it is the $\langle v^2 \rangle$ for each part of the system that we need to know, and are the answers we hope to find from the SEA calculations.

Power describes the rate of flow of energy into or out of a system and from one part of a system to another. In our model (Figure 2) the power flows of interest are the total input power from the turbulent boundary layer, the power dissipated by damping each of the subsystems, and the power transmitted from one subsystem to the other. In steady state conditions, the net power in to each subsystem must equal the net power out. The basic power balance equations for each subsystem are:

$$P_1 = P_{1diss} + P_{12} \tag{1a}$$

$$P_{12} = P_{2diss} \tag{1b}$$

Dissipated power in each subsystem is related to the energy by the damping loss factor, so that

$$P_{1diss} = \omega\eta_1 N_1 E_1 \tag{1c}$$

$$P_{2diss} = \omega\eta_2 N_2 E_2 \tag{1d}$$

Finally, and by analogy with (1c) and (1d), we introduce a coupling loss factor η_{12} and write

$$P_{12} = \omega\eta_{12} N_1(E_1 - E_2)$$

We assume that the input power P_1 to the skin and the loss factors η_1 and η_2 can be measured or otherwise estimated. Independent power input to the shelf P_2 could easily be included if it were present. The numbers of resonant modes $N = n\Delta\omega$ where n is the modal density and $\Delta\omega$ is the frequency bandwidth (usually octave or third octave) must also be measured or calculated. P_{12} represents the power transmitted from the skin to the shelf. By analogy with the dissipated power P_{diss}, P_{12} is written in terms of a coupling loss factor η_{12} which satisfies a symmetry relation $N_1\eta_{12} = N_2\eta_{21}$. Calculations of the parameters required have been made for a number of structural and acoustic couplings [1]. These calculations have formed an important part of the development of SEA.

Equation (1e) has a straightforward thermodynamic analog. In thermodynamic terms the energy E of the individual modes of oscillation represents temperature. What equation (1e) states for vibration is thus analogous to the statement that the heat transfer between two bodies is proportional to the difference in their temperatures.

The basic steps in SEA consist of (i) formulating a model in terms of two or more groups of resonant modes of vibration, noting that occasionally non-resonant motion must be included, and also that more than one group of modes, for example both flexural and torsional modes, may be needed to describe the motion of part of a structure, (ii) evaluating the parameters required and (iii) solving the power flow equations (1) to obtain the energies and hence the mean square vibration levels on various parts of the structure. Other vibration parameters such as stress can subsequently be obtained as required.

SAMPLE APPLICATION

Sample calculations made by H. G. Davies are included as a chapter in the text by Lyon [1]. These calculations draw heavily on earlier work and data taken J. E. Manning. The example is discussed briefly here.

Figure 1 shows a schematic of part of a vehicle designed to gather data during reentry into the earth's atmosphere. The skin of the vehicle is in essence a conical shell of length about 12 ft. and diameter varying from 6.4 to 39 in. A stiff ring attached to the skin supports an instrument shelf. The shelf itself is a fairly involved structure having several flanges. The skin is subjected to very intense turbulent boundary layer excitition as it reenters the earth's atmosphere. We try to estimate the vibration level on the instrument shelf during this part of the flight. This is by no means a trivial example. But we shall see that a very simple SEA model gives quite adequate agreement with experimental data.

Below about 50 Hz the vibration is dominated by large scale flexural modes of the whole vehicle. The usual analytical or numerical techniques of vibration analysis may be used in this frequency range and no advantage is gained by using SEA. However, in the third octave band centered for example at 2000 Hz we estimate that there are 64 resonant modes of vibration. It is in this type of situation that SEA can be particularly useful.

We first formulate a suitable SEA model. We could treat the skin, stiffeners, ring connector, various parts of the instrument shelf and the interior acoustic spaces as separate interacting systems each described by one or more groups of similar modes. Very many parameters are required for this complicated model. Experience suggests that the simple two system model shown in Figure 2 will give adequate results. One of the advantages of SEA is that groups of modes that are treated separately in complicated models often show up merely as additional dissipative mechanisms in a simpler model, and this added dissipation is included implicitly in any experimentally determined loss factor for the simple model. Thus a simple model will often suffice.

We next estimate the values of the parameters needed for our SEA model. A combination of calculated and measured values is usually needed. As SEA is a statistical approach detailed information about the structure is not always necessary, and may in fact be redundant. This aspect of SEA modeling is one of the advantages of SEA particularly when it is used during the early design stages of a structure or vehicle. Ways of estimating or measuring the parameters involved are described in detail in reference [1]. Typical values are given below for the 2000 Hz third-octave band to provide a feeling for the sorts of numbers involved in the calculations.

Modal densities

Information on the modal densities (numbers of resonant modes per unit frequency) of simple structures is to be found for example in references [1] and [3]. In our example no analytic solution exists for the modal densities of a cylindrical, let alone conical, shell. Values of N_1 can be obtained by using an average diameter of the conical shell and empirical results from [4] for cylindrical shells. A better estimate is obtained by dividing the conical shell into a number of cylindrical shells and adding the modal densities for each section. Manning obtains, for example, 64 modes in the 2000 Hz third octave band. Well above the ring frequency $c_\ell/2\pi a$ (in this example about 1000 Hz) the modal density asymptotes to that of a flat plate of the same area, $n(\omega) = (\text{area})/(4\pi h c_\ell/\sqrt{12})$. Suitable values for use in our example are diameter = 0.95 ft., length = 12 ft., c_ℓ = 10,000 ft/sec (an experimental value for the sandwich panel type of construction used), and h = 3/8 in.

The instrument shelf is a small fairly stiff structure and a detailed analysis does not seem feasible. One approximate estimate at high frequencies is obtained by treating the shelf as an equivalent flat plate, adding the areas of each flange. Since modal densities of coupled structures are additive this approach is reasonable. Suitable values are area = 4 ft^2 and $(hc_\ell/\sqrt{12})$ = 200 ft^2/sec, giving $n(\omega) = (200\pi)^{-1}$ and hence 4.6 modes in the 2000 Hz third-octave band.

At low frequencies (100 to 1000 Hz) Manning's experiments suggested that some third-octave bands contain resonant modes of the shelf, while others do not. In this case, when the modal density is very low a better model of the shelf is a one degree of freedom system attached by massless moment arms to the shelf. We take $N_2 = 1$ for this range of frequencies. This model is discussed below.

Coupling loss factor

This is usually the most difficult parameter to evaluate. Typical values and expressions are given in references [1] and [3]. In the present example the ring connector provides a stiff connection that preserves the angle between the shelf and skin. At high frequencies, vibration on the shell is purely in surface bending modes. The power transmission in this case may be modelled by that between two flat plates one attached at right angles to the other (Figure 3). The coupling loss factor for this case is given in reference [1]. For the parameter values already quoted we find $\eta_{21} = 2f^{-\frac{1}{2}}$ and $\eta_{12} = N_2\eta_{21}/N_1$. At 2000 Hz the numerical values are $\eta_{21} = 0.045$ and $\eta_{12} = 0.0032$.

To develop a suitable low frequency model we again have recourse to some of the experimental data of Manning. It was noted that at low frequencies typical vibration levels on the skin were at least 20 dB higher than the level at the ring connector, showing that the connector has a considerable stiffening effect. Power is transmitted to the shelf primarily by a moment at the connector. We are thus justified in considering only axial motion of the shelf. Figure 4 shows our low frequency model. Because of axial symmetry we may replace the shell by a beam of width $2\pi a$. We make the additional simplication of treating the beam as infinitely long. We may then use the input impedance for an infinite beam given, for example, in reference [3] and the result [1]

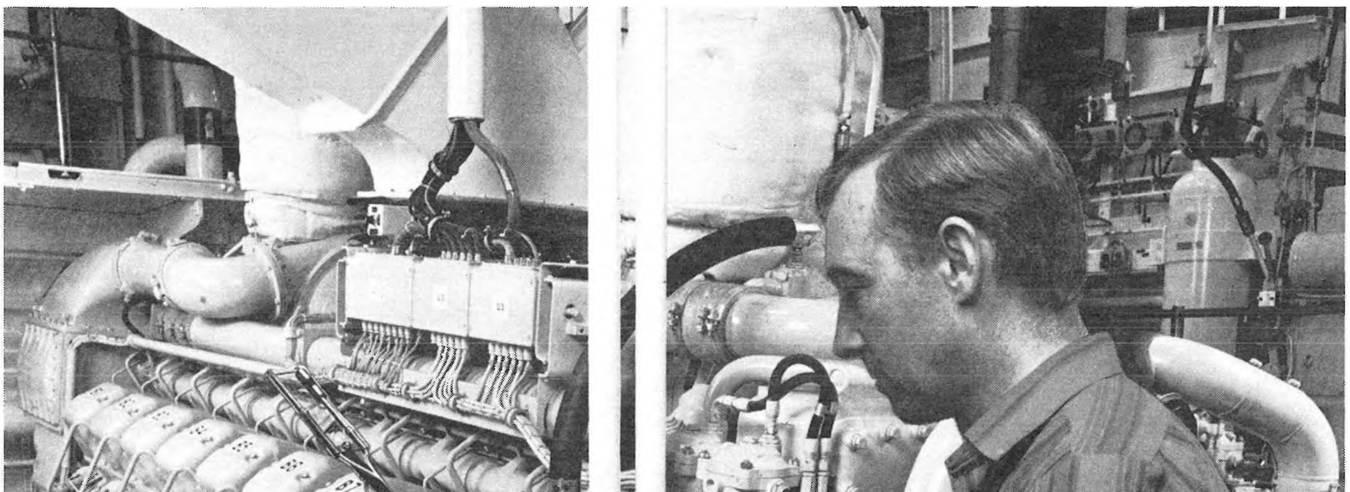
$$\eta_{21} = \text{Real}(Z)/\omega M_2 \tag{2}$$

where Z is the total impedance of the mass, spring and beam. For our parameters, we find $\eta_{21} = 20\, f^{-\frac{3}{2}}$ and $\eta_{12} = \eta_{21}/N_1$.

Experimental values of coupling loss factors can be obtained in cases where the coupling is too complicated to be modelled accurately analytically. Mean square vibration levels on two coupled structures may be measured and equations (1) used to find η_{21}. Details of these techniques are given by Lyon [1].

Loss factor

Values of loss factors can usually be obtained only by experiment. The



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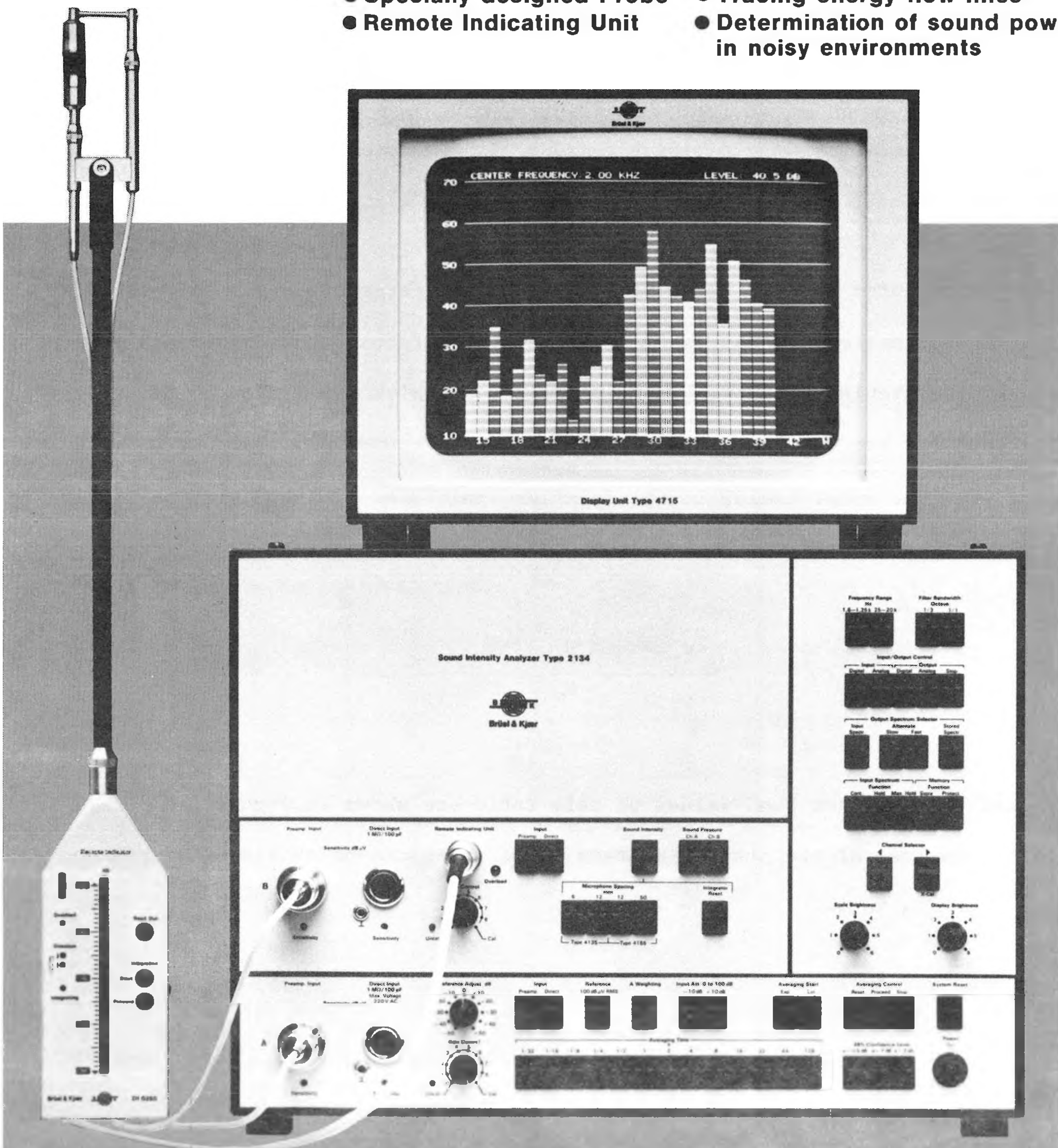

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values used in the present calculations are $\eta_1 = 0.025$ and $\eta_2 = 0.1$.

Power input

Calculation of the power input from the turbulent boundary layer to the conical shell is by no means trivial. Various theoretical and empirical methods may be used to estimate either the input power directly or the mean square acceleration of the skin. These values are related by

$$P_{IN} = \omega\eta_1 \, E_{1tot} = \frac{\eta_1 M_1}{\omega} \langle a_1{}^2 \rangle \tag{3}$$

where it is noted that η_1 is usually considerably larger than η_{12} so that to a first approximation at least we may neglect the effect of the coupling when calculating the input power. These calculations, although discussed in reference [1] are outside the scope of this paper. However, what our SEA model provides specifically is the relative amplitudes of the vibration levels on the skin and shelf. From equations (1) we obtain

$$\frac{\langle v_2{}^2 \rangle}{\langle v_1{}^2 \rangle} = \frac{M_1}{M_2} \frac{\eta_{12}}{\eta_2 + \eta_{21}} \tag{4}$$

Typical third-octave band values of this ratio are shown in Figure 5. Specific values of the shelf vibration level $\langle v_2{}^2 \rangle$ can be obtained, of course, once $\langle v_1{}^2 \rangle$ is known.

Results of sample calculations

Figure 5 compares theoretical values using equation (2) and experimental values obtained by Manning. Manning's values are for a full-sized shelf inside a cylindrical shell, with the values adjusted theoretically to account for the different modal density of the actual conical shell. Agreement between theory and experiment is seen to be adequate.

APPLICATIONS OF SEA

SEA was developed initially for aerospace applications and has been used very successfully in that area for calculating the vibratory response of complicated structures involving structure to structure vibration transmission as discussed in our sample application, and also for structure/acoustic field interactions for estimating sound radiation and noise levels. In this latter case a simple SEA model might consist of one group of modes representing the vibrating structure and a second group representing the acoustic field with which the structure interacts. SEA models involving both structural and airborne transmission have also been used in noise control on board ships. Similar structure/acoustic interaction models were used in the design stages of a U.K. gas-cooled nuclear reactor to estimate the fatigue life of the reactor shell excited by the very intense internal acoustic field caused by large axial flow compressors.

Perhaps the best recent overview of the scope of SEA and its applications was given at the 100th meeting of the Acoustical Society of America in 1979. One session at that meeting was developed entirely to SEA and included among other papers invited review papers on the application of SEA to building acoustics, vibration of internal combustion engines, shipbuilding, and wave excited motion of off-shore structures.

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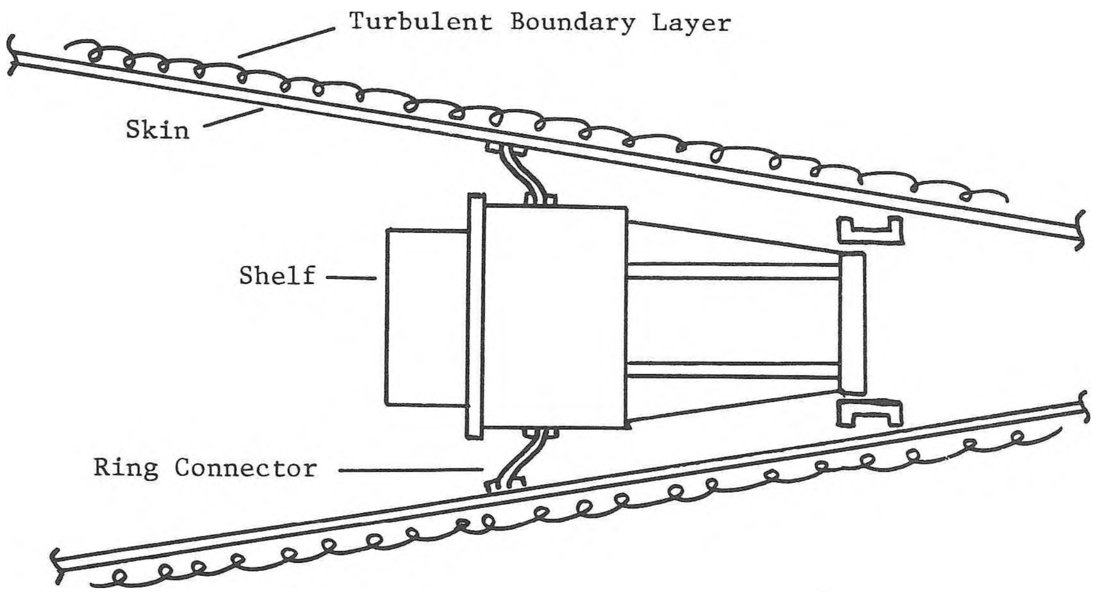

Figure 1

Schematic of Skin and Shelf

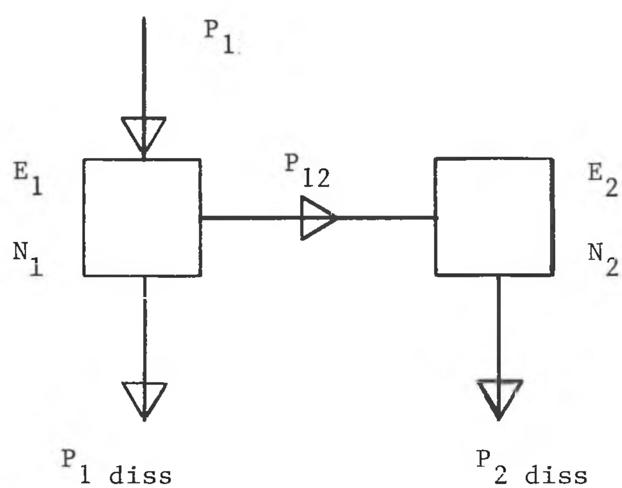

Figure 2

SEA Model of Skin and Shelf

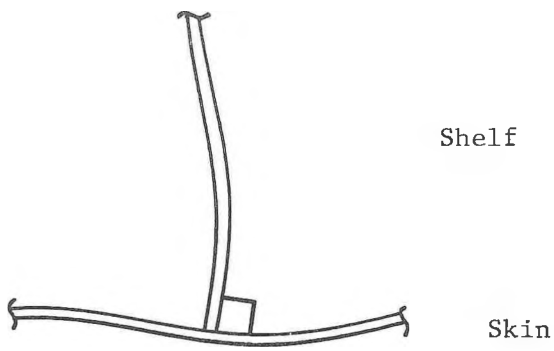

Figure 3

Plate model for high

frequency y_{12}

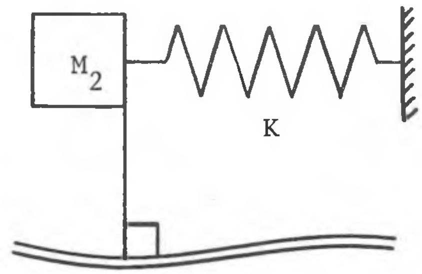

Figure 4

Lumped parameter model

for low frequency y_{12}

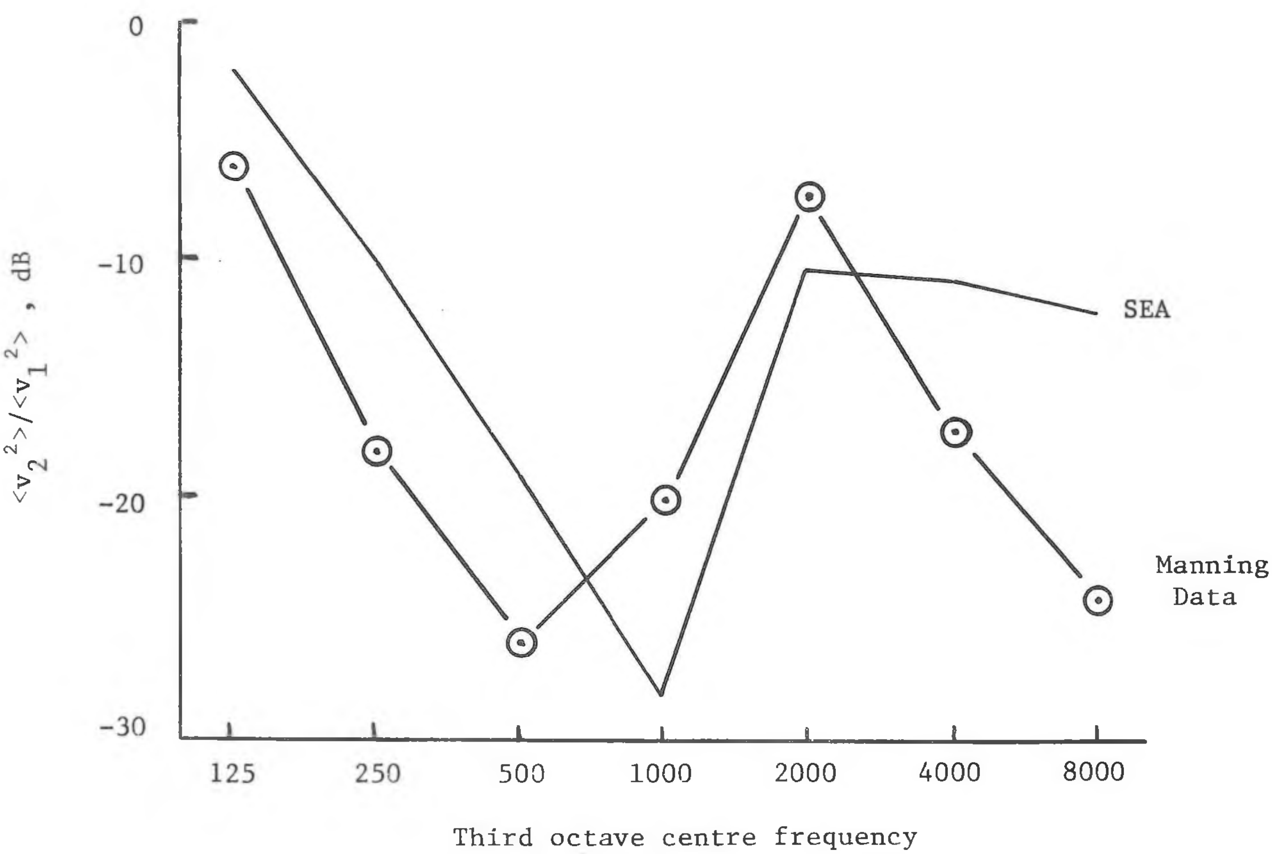

Figure 5

NOISE EXPOSURE SURVEY IN A NUCLEAR GENERATING STATION

by

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ABSTRACT

Measurement techniques and some results from the survey are presented. The concept of L_{Trade} as the best estimator for noise exposure is introduced. It was also shown how the use of dosimeters may help to localize noise sources (plastic suits in our case). Finally, the close cooperation between workers, supervisors and surveyors was found to be fundamental for the success of the survey.

SOMMAIRE

Les techniques de mesure et certains résultats de l'étude sont rapportés. Le concept de $L_{métier}$ est présenté comme la meilleure façon d'évaluer l'exposition au bruit. Il a aussi été démontré comment l'utilisation de dosimètres peut aider à localiser les sources sonores (survêtements en plastique, dans notre cas). Enfin, l'étroite collaboration entre les travailleurs, les superviseurs et les enquêteurs a été essentielle au succès de l'étude.

1.0 INTRODUCTION

Workers in nuclear generating stations are exposed to varying noise levels during the shift. Consequently, their daily noise exposure is unpredictable and its evaluation has to be performed using noise dosimeters. In this note, we will present some of the techniques and procedures used during the exposure survey performed at the Pickering Nuclear Station "A", where besides the individual we were also interested in obtaining average noise exposure of workers from each trade.

During a previously performed noise level survey, workers from five different trades were found likely to be at risk from noise exposure. Therefore, it was decided workers from these trades were to be included in the exposure survey. Their number was decided to be as large as to represent, yet not to interfere with, the normal activities performed at the station.

The selection of the individuals was done by station management. Each worker's exposure was measured during five consecutive days to account for the inevitable exposure variation between days. This procedure is also in accord with the new Ontario Health and Safety Act proposed Regulations (June 1981) where emphasis is on weekly rather than daily exposures.

Noise doses were measured during both normal operations and shutdown (when most of the maintenance work takes place) periods.

2.0 INSTRUMENTATION

Dosimeters GenRad, type 1954-9710 were used for this survey. They were preset to a threshold level of 80 dBA, criterion level of 85 dBA and exchange rate of 5 dBA.

Doses were read on a Reader GenRad, type 1954-9720, which was also used as battery tester and calibrator.

A sound level meter B&K type 2215 was used for spot checks of high noise level areas as a coarse means of controlling dosimeter readings.

Dosimeters were calibrated at the beginning and at the end of each shift according to manufacturer's instructions. In addition, on several occasions another test was performed, where dosimeters were exposed to noise of constant level, controlled with a Sound Level Meter. The measured dose was then compared to that calculated from the SLM readings, so to reassess dosimeters performance in a real-life situation. (This procedure is now performed in our laboratory by using a reverberant diffuse enclosure where microphones of all dosimeters are exposed to 90 dBA pink noise.)

3.0 MEASURING PROCEDURE

At the beginning of the study each participant was briefed on the purpose of the test and his cooperation was sought to obtain meaningful results. Participants were asked to assist in switching dosimeters on and off and filling forms detailing types of jobs and areas they were working in during the day.

At the beginning of the shift a technician calibrated all dosimeters and checked their batteries. Then he handed them to the workers, who switched the instruments on and placed them in their shirt pockets. Microphones were attached to shirt collars and positioned upwards. Workers were instructed not to remove dosimeters during the whole shift.

At the end of the shift, the workers had to switch off the dosimeters and to return them to the technicians. Each individual filled in the above-mentioned form. The technician then performed the readings and checked the calibration and the batteries of each dosimeter. He also discussed measurement results with the workers, trying to relate dose readings to jobs being performed during the shift.

4.0 CALCULATIONS

With the information of dose and exposure duration, the daily L_{OSHA} was calculated for each individual using a computer program. Then the mean weekly noise exposure for each individual was obtained as:

$$\overline{L}_{OSHA} = 16.61 \log \frac{1}{n} \Sigma\, 10^{L_i/16.61}, \quad \text{dBA}$$

where n = number of readings (usually 5)
L_i = daily L_{OSHA}

Noise exposure for each trade was obtained from the $\bar{L}_{OSHA}$ of individuals from the trade, using the equation

$$\bar{L}_{Trade} = \frac{\Sigma \bar{L}_{OSHA}}{n}, \text{ dBA}$$

where η = number of individuals (see Table 1).

5.0 RESULTS AND COMMENTS

Two sets of results, from normal operation and shutdown conditions were obtained. Student's T test showed that their differences were not statistically significant. Thereafter all $\bar{L}_{OSHA}$ from each trade were pooled and $\bar{L}_{Trade}$ obtained. They are shown in Table 1, together with the range of the individual's $\bar{L}_{OSHA}$ and standard deviation.

To perform tasks in some areas workers must wear plastic suits that are complete coveralls, where the air is supplied through a hose connected to a source of approximately 620 kPa (90 psi) air. The air flow acts as an additional source of noise. During the survey it was noted that noise dose readings were constantly higher when plastic suits were worn.

To test how significant was the contribution of the noise generated in plastic suits, daily $\bar{L}_{OSHA}$ from workers wearing suits were separated from these of the rest. Mean $\bar{L}_{OSHA}$ and standard deviations were then calculated from both populations.

Results shown in Table 2 indicate that $\bar{L}_{OSHA}$ is much higher when using plastic suits and that the difference between both means is statistically significant. Independent sound level measurements confirmed the above findings.

High noise levels in plastic suits could be due to several factors such as suit design, high air pressure and damaged parts of the suit. A much quieter suit of new design is now gradually being introduced so L_{Trade} of trades currently wearing these devices is expected to be reduced in the near future. This study shows how an additional benefit such as assessing a noise source may be obtained from the exposure survey.

This survey proved to be a satisfactory experience resulting from the close cooperation between workers, supervisors and surveyors. Not a single dosimeter was damaged during the survey period that lasted for over four months, nor were abnormally high or low $\bar{L}_{OSHA}$ values recorded, thus confirming worker's positive attitude toward this exercise.

Because of the sampling technique used and because of possible changing working environments (noise levels in some parts of the station are also related to power output) exposure surveys should be repeated periodically every two to three years.

Table 1. Noise Exposure by Trade

Trade	Number of Workers		$\bar{L}_{Trade}$, dBA		
	On Site	Tested	Mean	St. Dev.	Range
Operators*	330	18	88	5.4	79-97
Mechanical Maintainers	160	24	90	3.8	81-97
Service Maintainers	40	2	82	-	77-85
Chemical Technicians	25	3	80	3.9	76-84
Control Technicians	125	8	92	5.2	81-96

*First operators excluded because they are exposed to noise levels below 80 dBA.

Table 2. Noise Exposures of Workers With and Without Plastic Suits (L_{OSHA}, dBA)

Trade	In Plastic			No Plastic			
	n*	Mean	St. Dev.	n*	Mean	St. Dev.	Mean Increase
Operators	9	95.1	9.84	57	86.4	5.62	8.7
Control Techn.	9	98.1	7.96	26	89.4	9.60	8.7
Mech. Maintainer	11	95.5	8.64	104	89.0	7.00	5.5

n* = number of readings

TEMPORARY THRESHOLD SHIFT AND THE TIME PATTERN OF NOISE EXPOSURE*

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ABSTRACT

Temporary threshold shift (TTS) is related to the time pattern of noise exposure in a relatively complex fashion. This relationship essentially involves the on-fraction, the period of intermittence and the sound level during noise bursts and quiet intervals. Data from recent studies on the influence of these exposure parameters and more specifically of their mutual interaction is reviewed. Unpublished data is presented on the effect of the interaction between the on-fraction and the period of intermittence on the growth of TTS and also on the effect of the ambient sound level on the recovery from TTS. The predictability of TTS is examined in the context of both laboratory and real life conditions. The implication for the recommended exchange rate between sound level and time in occupational noise exposure limits and for administrative control of exposure is discussed.

SOMMAIRE

Le découpage temporel de l'exposition à un bruit intense joue un rôle important dans l'acquisition et la récupération de la fatigue auditive (DTS: décalage temporaire des seuils d'audition). Les facteurs en cause sont la fraction temporelle, la période d'intermittence, le niveau sonore qui prévaut durant les expositions et les pauses. Un examen critique des données scientifiques disponibles de même que la présentation de données originales ont permis de définir les limites dans lesquelles on peut prédire le DTS dans des conditions d'exposition en laboratoire et en usines bruyantes. De cette analyse ont été tirées des implications à l'égard de l'adoption d'un coefficient de pondération durée-intensité dans le contexte des limites d'exposition au bruit en milieu de travail.

Introduction

Noise-induced temporary threshold shift (TTS), sometimes called auditory fatigue, is a reversible loss of hearing sensitivity following exposure to high sound levels. This phenomenon has been studied extensively under laboratory conditions as a possible predictor of occupational hearing loss (also termed "noise-induced permanent threshold shift", PTS) for a variety of noise exposure conditions. During the last 25 years, a fairly large number of experimental studies have been conducted on the influence of the time pattern of noise exposure on growth and recovery of TTS. They all had a common justification, that is to contribute to the identification of hearing damage risk criteria for intermittent exposure to industrial noise. This contribution was further emphasized by the scarcity of reliable data on occupational hearing loss following intermittent and variable noise exposures (Radcliffe, 1970). Because these studies shared a common aim of predicting TTS in real life situations (and from that point, the possible growth of permanent threshold shift-PTS), they are reviewed in this specific context.

The rationale of the present analysis is that, apart from considering the possibility of predicting PTS from TTS, one must establish how TTS can <u>actually</u> be predicted

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for a number of real life conditions for which it is specifically set forth as a guideline for limiting noise exposure.

The variables that are involved in predicting TTS from intermittent exposures are the on-fraction, the period of intermittence, the sound level during noise bursts and during quiet intervals and finally the total time of exposure. Their respective influence and their possible mutual interactions will be briefly examined and implications on different exchange rates between sound level and time will be considered.

1. The influence of the on-fraction:

Because of the lack of a simple model relating the variables just mentioned, the influence of the exposure pattern on TTS is usually described essentially in terms of the on-fraction rule (Burns, 1969, 1973; Kryter et al., 1966; Melnick, 1978, 1979; Sulkowski, 1980; Ward, 1963, 1973). This rule states that TTS from intermittent noise is less than if the exposure is continuous. The reduction in the amount of TTS_2 (TTS_2: TTS measured 2 min after the end of exposure) is proportional to the ratio of the time notoccupied by noise to the total time of exposure (Ward et al., 1958). This was said to hold for burst durations between 250 msec and 2 min and for sound energy above 1000 Hz.

Evidence show that, for a constant total amount of sound energy, the magnitude of TTS is smaller as the on-fraction is decreased. This is exemplified by the data presented in Figure 1. These are results from an experiment conducted in our laboratory: 8 normal hearing subjects were exposed to a pink noise during 128 minutes; the on-off period was equal to 1 min, the on-fractions (R) were 0.1, 0.5, 0.9 and 1 and the corresponding sound pressure levels (L_{pA}) were 106, 99, 96.5 and 96 dBA respectively. These L_{pA} were selected in order to obtain a constant total amount of energy across the 4 noise conditions, the sum of energy being equivalent to 90 dBA for an 8 hour exposure.

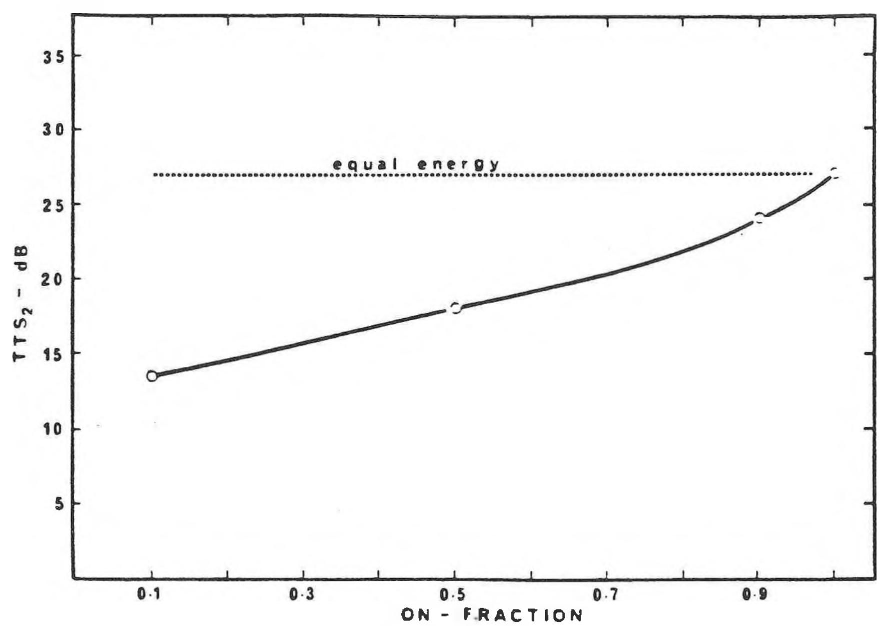

Fig. 1. TTS_2 at 4 kHz as a function of the on-fraction for a constant cumulated sound energy equivalent to 90 dBA for 8 hours; period of intermittence = 1 min; total exposure duration = 128 min (N = 8)

It follows clearly fron the results presented in Figure 1, as well as from those from other studies (Ward, 1976, 1981) that, for a constant final TTS_2 at 4 kHz, the exchange rate between sound level and exposure time varies inversely with the on-fraction, provided that the period of intermittence is in the range of one or two minutes.

But despite a relatively great care of all reviewers in stating the limits of the on-fraction rule (see Ward, 1963, 1973; Burns, 1969, 1973; Melnick, 1978, 1979), its interaction with other exposure variables is rather neglected, possibly because of the lack of parametric studies on these interactions. Nevertheless, some indications are provided by the presently available data as shown in Figure 2. The on-off period, the sound level and the frequency characteristics of the noises studied are identified at the bottom of this figure. All data were obrained after 8-hour exposures, except for line A for which it was 1.7 hour. It is worth mentioning that the data points connected by line A were part of the original data set that served as the actual basis for the on-fraction rule.

When comparing the slope of the different lines in this figure, one can see that TTS_2 is certainly not a simple function of the on-fraction of the noise. The relationship between TTS_2 and the on-fraction appears to change

- with the sound level: comparing line B to C and also line D to line E shows that, for similar on-off periods, different sound levels are associated with different slopes.
- with the period of intermittence: comparing line B to line D for identical sound levels.
- possibly with the total exposure time: as one compares line A, fitting data for 1.7-hour exposures with line E, describing results for 8-hour exposures; but, here, the comparison is further complicated with a variation in the period.

With all these interactions, is it still appropriate to describe the influence of the exposure pattern strictly or mainly in terms of the on-fraction rule?

2. The influence of the period of intermittence:

The influence of the period of intermittence alone can be described by the results from 4 different parametric studies, as shown in Figure 3. Line A refers to 8-hour exposures to an octave-band level of 85 dB and line B to 95 dB. Line C corresponds to data for 2-hour exposures to a broadband noise at a sound pressure level of 101 dB. Line D refers to TTS_2 from an octave band level of 112 dB during 52 min. It must be emphasized that all studies bear strictly on exposure involving an on-fraction of 0.5 (exactly 0.46 in the case of line D).

Generally speaking, the amount of TTS_2 is exponentially related to the period of intermittence. For each doubling of the period, a 1.5 to 2 decibels increase in TTS_2 is observed. But, this does not seem to hold for very short periods and very high sound levels (line D). Besides, it is not clear whether differences in the slopes of the lines relating TTS_2 to the logarithm of the period merely reflect sampling errors or interactions with the sound level or other variables.

Moreover, the period can dramatically influence the pattern of recovery from TTS, if one recalls the results reported by Ward (1970, fig. 5); cycles of exposure and quiet of 10-sec duration, producing relatively small amounts of TTS_2, required more than 2 days for complete recovery in a number of subjects. It is surprising that this observation did not give rise to more research on the exact exposure parameters that were responsible for such an effect. Finally, the effect of periods longer than 40 min have not been studied as such. For what length of the period does an intermittent exposure produce the same amount of TTS_2 as a continuous exposure of equal cumulated energy and total exposure time is still an open question.

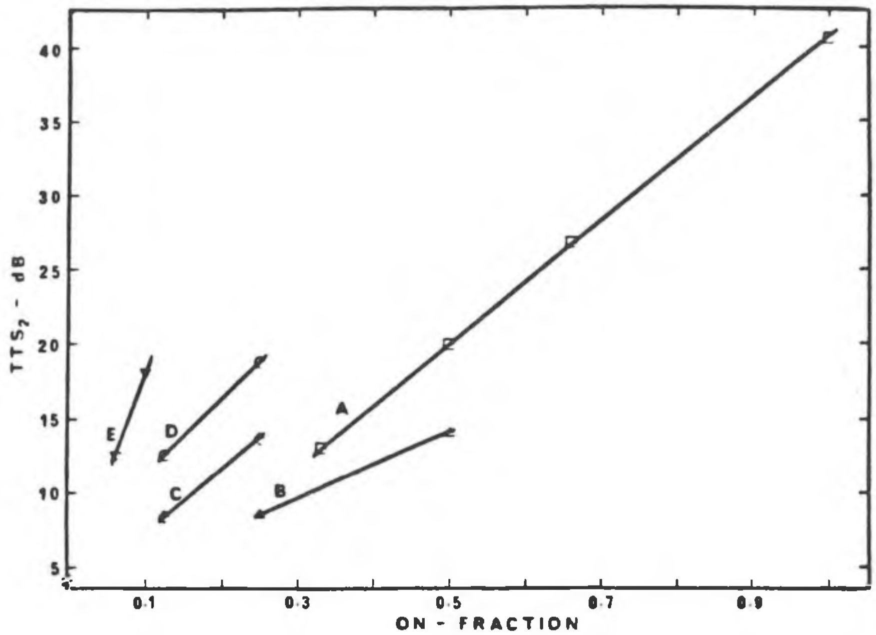

Fig. 2. TTS_2 at 4 kHz as a function of the on-fraction
Line A: 1-1.5 min on-off, 106 dB, pink noise (Ward et al., 1958)
Line B: 1.5 min on-off, 90 dB, 2800-5600 Hz (Ward, 1976)
Line C: 1.5 min on-off, 95 dB, 2800-5600 Hz (Ward, 1976)
Line D: 40 min on-off, 90 dB, 2800-5600 Hz (Ward, 1976)
Line E: 50 min on-off, 105 dB, 1400-2000 Hz (Ward, 1970)

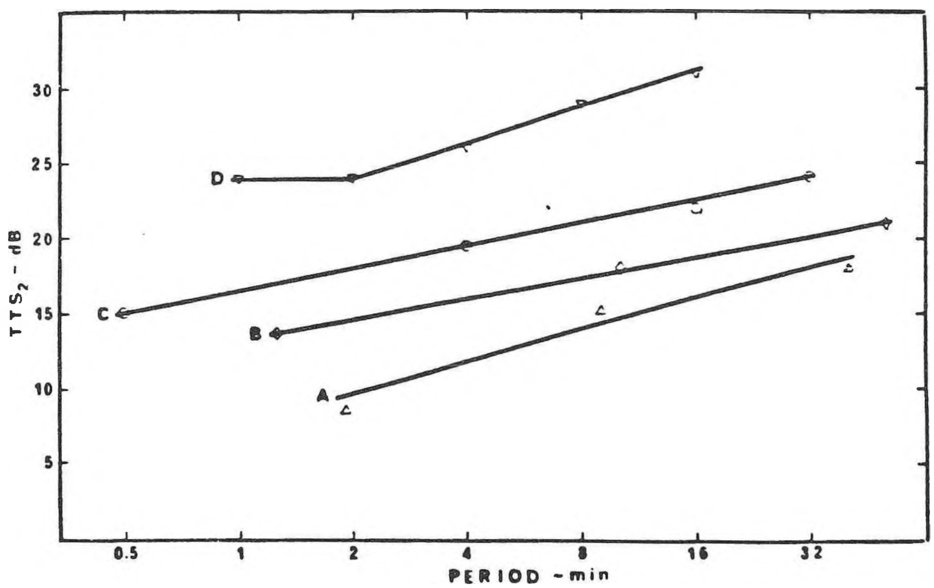

Fig. 3. TTS_2 at 4 kHz as a function of the period of intermittence
Line A: Duration - 480 min, 85 dB, 1200-2400 Hz (Ward, 1976)
Line B: Duration - 480 min, 95 dB, 1400-2000 Hz (Ward, 1970)
Line C: Duration - 128 min, 101 dB, pink noise (Hétu & Trémolières, 1977)
Line D: Duration - 52 min, 112 dB, 1700-3400 Hz (Selters & Ward, 1962)

3. The influence of the sound level during the so-called quiet intervals

Recovery from TTS has generally been assumed to take place at an optimal rate when the noise level falls below the critical level for the growth of TTS. This level, defined as "effective quiet" was first estimated to be 85 dB SPL for a broadband noise (Ward et al., 1958) and between 65 and 75 dB for the octave bands for which the ear is more sensitive (Ward et al., 1959). Surprisingly, octave band levels between 85 and 89 dB were adopted in the CHABA criteria (Kryter et al., 1966) and the Intersociety Committee (Radcliffe, 1970) defined "no noise" as levels below 90 dBA.

A number of studies has been performed since that time, using essentially two kinds of procedures: (a) measures of the influence of the sound level during the quiet intervals of an intermittent exposure and (b) measures of the rate of recovery from a given TTS_2 as a function of the ambient continuous sound level.

Results from the studies belonging to the first group (i.e.: intermittent exposures) clearly showed that broadband noise levels near or above 70 dBA during "quiet" intervals did increase the final amount of TTS_2 (Klosterkötter et al., 1970; Schmideck, Henderson & Margolis, 1972, 1975). Results from the other group of studies performed by Austrian researchers (Lehnhardt and Bucking, 1968; Schwetz et al., 1970; Dopler et al., 1973) indicate that recovery from TTS is actually slower in a 70 dB SPL broadband noise than in a quieter condition. No such difference was observed in a 65 dB octave band level centered at 2000 Hz.

In a recent study, Ward, Cushing and Burns (1976) by combining the two procedures mentioned above (using an intermittency period of 1.5 min and a post-exposure period of 2 hours) obtained octave band levels of effective quiet that were similar to those estimated for the most sensitive frequencies in the study performed in 1959 (see: Ward et al., 1959), these levels falling between 65 and 75 dB per octave.

In our laboratory, we have studied the influence of the level of a continuous broadband noise on the recovery from TTS with 2 groups of 12 subjects.

When comparing, in a first series of experiment (Figure 4 A) the effect of a 75 dBA to a 40 dBA recovery condition, significant differences ($P < 0.01$ - Wilcoxon) were obtained between the two conditions after 120 min of exposure. In a second series of experiments, four recovery conditions were compared (50, 60, 70 and 80 dBA), as shown in Figure 4 B. At 50 dBA, a significant recovery ($P < 0.006$ - Friedman X^2), was obtained between the 60th and the 120th minute of exposure, while no such difference could be observed at 60 and 70 dBA. A significant increase in TTS was obtained at 80 dBA for the same period. Thus, in both series of experiments, difference between conditions essentially occured during the second hour in the recovery environment.

These results, as well as those from the studies mentioned earlier, clearly imply that the definitions of effective quiet or "no noise" in damage risk criteria based on TTS (i.e.: the CHABA and the Intersociety Committee proposals) as well as in the most recent formulation of the OSHA standard (OSHA, 1981) do not correspond to conditions of optimal recovery from TTS. It also implies that the prediction of the growth and the recovery of TTS under intermittent exposure must take into account the effect of the actual sound level associated with what is considered as a quiet interval, if such a prediction intends to apply to real life situations.

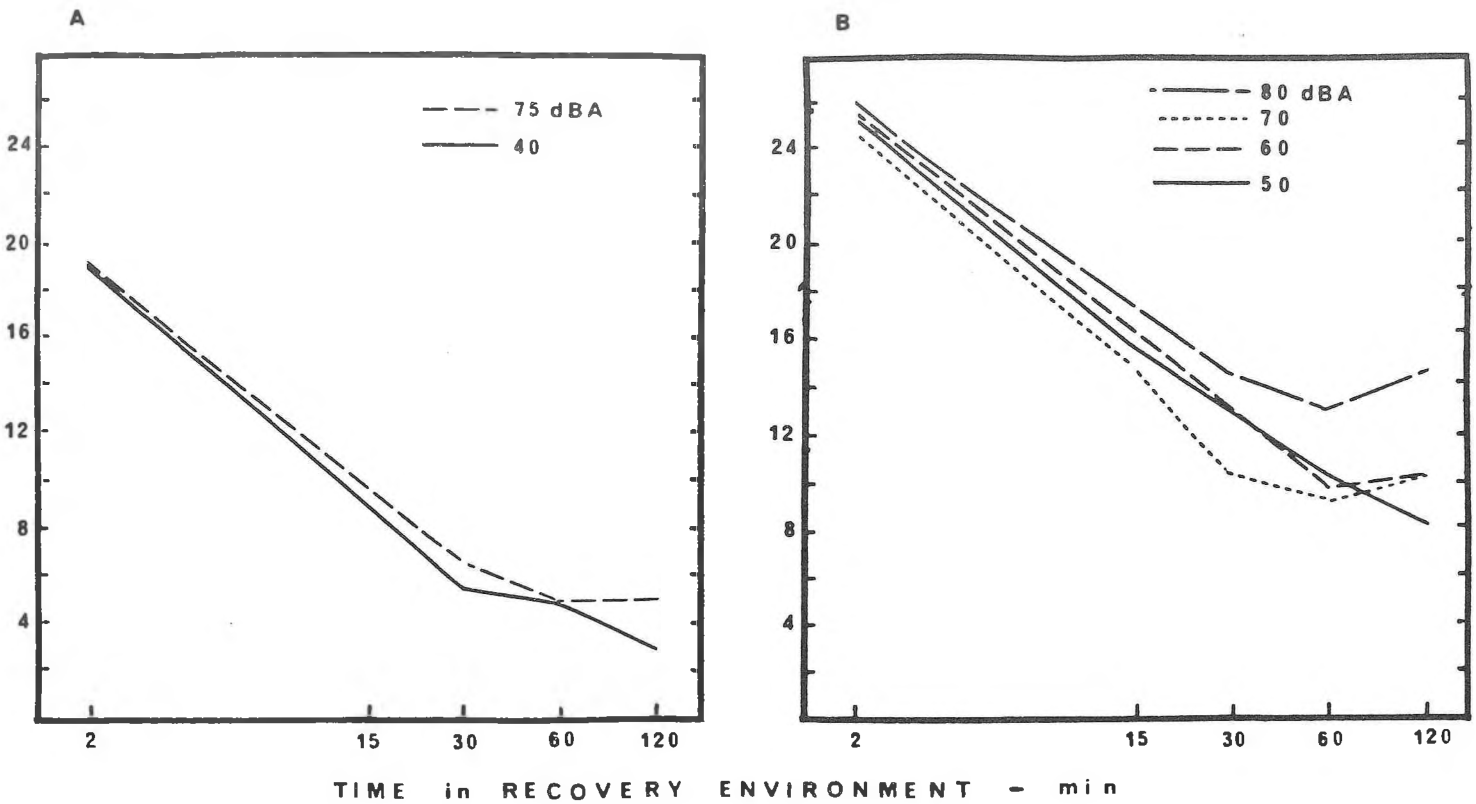

Fig. 4. Recovery from TTS at 4 kHz as a function of the ambiant sound level (parameter) in two separate experiments A and B, each involving 12 subjects.

4. Implications for a generalized time-intensity trade-off for various exposure patterns

The majority of present North America legislations concerning noise exposure limits and PTS rely on a 5-dBA time-intensity trading relationship. This was based on a simplification of criteria derived from TTS studies on intermittent exposures, although the link between TTS and PTS is notwell established. As shown in Figure 1, such a trade-off is certainly conservative if the on-fraction is smaller or equal to 0.5 together with a on-off period of 2-3 minutes and with a sound level below 65 dBA during quiet intervals.

Now, the question is: does such a condition ever occurs in industry? Few representative statistics on the exposure patterns for typical noisy jobsites are presently available; studies on the industrial noise environments have been more frequently concerned with the spectral characteristics of the noise (see: Royster and Stephensen, 1976; Bostford, 1967). Experience show however:

- that the actual quiet intervals (rest periods) are very short as compared to the length of the exposures to high sound levels
- that the rest periods are not frequent in a normal workday
- and that they are frequently characterized by sound levels clearly above 65, even above 75 dBA.

In other words, the on-fractions are larger than 0.5, the intermittency periods longer than a few minutes and the time intervals of low noise levels do not represent effective quiet conditions. The effects of these exposures are more probably nearer those from varying noise conditions or continuous exposures of less than 8 hours than those from intermittent noise.

In this context, a uniform time-intensity trade-off of 5 dBA is certainly inappropriate, more especially as the original CHABA proposal comprised a 2 dB sound level increase for reducing the duration of exposure from 8 to 4 hours and a 3 dB increase for a decrease in exposure time from 4 to 2 hours or from 2 to 1 hour.

The situation is even more paradoxical when one considers that the 5-dBA trade-off is used to set limits of daily exposures which last more than 8 hours (see: OSHA, 1981). Again, to my knowledge, no statistics on overtime and on compressed workweek schedules are presently available. But, overtime is not an unlikely event in a number of jobsites for a majority of noisy plants. The 5-dBA trade-off was based on the fact that for an equal amount of energy, TTS was smaller when the exposure was intermittent. Now it is used to define permissible exposures that produce asymptotic threshold shifts (ATS; see: Melnik, 1976; Nixon et al., 1977). Recovery from ATS generally requires more than 16 hours (Mills et al., 1970; Mosko et al., 1970) while the rest period between two days work in this condition is smaller than 16 hours. One would have expected that ATS data would have justified exposure limits for durations longer than 8 hours that were more conservative.

5. Conclusion

A fairly large number of studies have been conducted on the influence of the time pattern of noise exposure on TTS. The amount of data presently available allows one to predict the growth and the recovery of TTS for a number of laboratory conditions. The use of exponential models for predicting TTS has been shown by Keeler (1976) to be highly accurate for a variety of exposure patterns. However, these models need to take into account the influence of the exact sound level during the so-called quiet intervals as well as the interaction between the on-fraction and the intermittency period, in order to state valid predictions for a majority of real life situations. More data are needed to clarify the exact contribution of the on-fraction for extreme values of the period length, that is a few seconds and an hour or so.

Looking back to the way TTS studies were used to set guidelines for industrial noise exposure control (i.e.: The Intersociety Committee proposal, see: Radcliffe, 1970) provides a good example of oversimplification and premature generalization of empirical rules bases on a limited number of laboratory observations. Studies of the actual exposure patterns in noisy industry are seriously needed before claiming being able to predict the growth and the recovery of TTS in these particular conditions. Whatever may be the predictive value of TTS for the growth of PTS, there is a need to know how the auditory perception of millions of industrial workers is subject daily to serious disturbances because of occupational noise exposure.

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Every three years an international Congress on Acoustics is convened under the double patronage of the International Commission on Acoustics and the Acoustic Society of the host country.

The 11th Congress will be held in Paris (Hotel SOFITEL, Paris) July 19-27, 1983. The venue for the opening session will be the main theatre at the Sorbonne, in Paris, GALF (a group of French speaking acousticians) will be wholly responsible for the organization of the Congress. The Congress will deal with every aspect of acoustics and will be heralded by three smaller "Satellite" Symposia, held in:

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TOULOUSE: also July 15-16 on oral communications.

Tous les trois ans, un Congrès International d'Acoustique se tient sous la double égide de la Commission Internationale d'Acoustique (ICA) et de la Société Acoustique du pays hôte.

Le 11e Congrès se tiendra en France, à Paris (Hôtel SOFITEL-PARIS), du 19 au 27 juillet 1983. La Séance inaugurale se tiendra dans le Grand Amphithéâtre de la Sorbonne. Le GALF (Groupement des Acousticiens de Langue Française) assume l'organisation de ce Congrès.

Il concernera tous les domaines de l'acoustique, et sera précédé de Symposia satellites:

- à Marseille les 12 et 13 juillet sur l'absorption acoustique active et les asservissements acoustiques;
- à Lyon les 15 et 16 juillet sur le rayonnement acoustique des structures vibrantes;
- à Toulouse les 15 et 16 juillet également, sur la communication parlée.

July 13-15, 1983. Edinburgh — Scotland

INTER-NOISE '83

For further information contact:
Mrs. Cathy MacKenzie
Secretary
Institute of Acoustics
25 Chambers Street
Edinburgh, EH1 1HU
Scotland, U.K.

September 4-7, 1983. London — Gr. Britain

4th CONFERENCE OF THE BRITISH SOCIETY OF AUDIOLOGY

October 1983. High Tatra — Czechoslovakia

22nd ACOUSTICAL CONFERENCE ON ELECTROACOUSTICS AND SIGNAL RECORDING

November 7-11, 1983. San Diego, California — U.S.A.

MEETING OF THE ACOUSTICAL SOCIETY OF AMERICA

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ULTRASONICS INTERNATIONAL 83

Call for papers

In line with the Ultrasonics International conference's policy of alternative venues between the UK and overseas, it has now been arranged to hold this conference, the 14th in the series, in Halifax, Nova Scotia. The arrangements made before this possibility arose, of holding the conference at the University of Surrey, are therefore regrettably postponed, but it is intended that the next event will be held there in 1985.

Halifax, a historic port and capital city of Nova Scotia, is a large university town, founded in 1749. It has many tourist attractions, and also modern conference facilities. The Bedford Institute of Oceanography is nearby.

The scope of the conference includes:

underwater ultrasonics	ultrasonics in aerospace	ultrasonics and the Arctic
acoustic microscopy	photoacoustic spectroscopy	ultrasonic tomography
transducer materials	acoustic emission	high power ultrasonics
calibration of transducers	data handling	optoacoustics
defect and tissue characterization	cavitation	visualization
		non-linear acoustics

It is intended to keep the conference fee low - it is hoped that no increase will be made from UI 81. Some funds will be available for students and other applicants.

Papers are invited on recent developments in any field of ultrasound, especially on practical aspects and applications. Papers are also invited for the poster sessions, a major feature of the conference, where an opportunity is given for informal presentation and discussion. All contributions will be reviewed by the scientific panel. They should cover recent developments and not have been published previously or presented at a major conference. The language of the conference is English.

Authors offering papers for presentation should send abstracts of 200-300 words with one illustration to the Conference Organizer. The deadline for receipt of abstracts is 31 December 1982.

There will again be a commercial exhibition of ultrasound equipment concurrent with the conference. Companies interested in exhibiting should contact the Conference Organizer.

The official carrier is Air Canada. Details of discount fares may be obtained from the Conference Organizer. The airline flies from most major cities.

Delegates from North America and Japan may find it convenient to attend the conference before continuing to the International Congress on Acoustics in Paris the following week.

Conference Organizer: Dr. Z. Novak, Butterworth Scientific Ltd.
P.O. Box 63, Westbury House
Bury Street, Guildford, Surrey GU2 5BH England

Important Canadian Standards for CAA Members

Published CSA Standards		Price
*Z107.1-1973,	Specification for Sound Level Meters (Adopted ANSI S1.4-1971)	$11.00
*Z107.2-1973,	Methods for the Measurement of Sound Pressure Levels (Adopted ANSI S1.13-1971)	11.50
Z107.3-1974,	Methods for the Determination of Sound Power Levels of Small Sources in Reverberation Rooms (Adopted ANSI S1.21-1972)	10.00
Z107.4-1975,	Pure Tone Audiometers for Limited Measurement of Hearing and for Screening (4a, 4e)	4.75
Z107.5-1975,	Octave, Half-Octave, and Third-Octave Band Filter Sets (Adopted ANSI S1.11-1966 (R1971))	10.50
Z107.21-M1977	Procedure for Measurement of the Maximum Exterior Sound Level of Pleasure Motor Boats	8.00
Z107.22-M1977	Procedure for Measurement of the Maximum Exterior Sound Level of Stationary Truck with Governed Diesel Engines	8.00
Z107.23-M1977	Procedure for Measurement of the Maximum Interior Sound Level in Trucks with Governed Diesel Engines	7.75
Z107.51-M1980	Procedure for In-Situ Measurement of Noise from Industrial Equipment	11.00
Z107.53-M1982	Procedure for Performing a Survey of Sound Due to Industrial, Institutional, or Commercial Activities	10.00
Z107.71-M1981	Measurement and Rating of the Noise Output of Consumer Appliances	20.00

*Available in French

Copies of these standards may be purchased from CSA in Toronto, Vancouver, Edmonton, Winnipeg or Montreal. The Toronto address is: Sales Group, Canadian Standards Association, 178 Rexdale Blvd., Rexdale (Toronto), Ontario M9W 1R3.

There is a postage and handling charge of $1.00 in Canada, $2.50 U.S.A and foreign, per order. A cheque is required with all orders under $75.00. VISA or MASTERCARD orders will be accepted by phone at (416) 744-4044.

CSA Endorsed Standards

The following standards, non-Canadian in origin, have been reviewed by the appropriate Technical and Standards Steering Committees and are deemed acceptable for use in Canada without modification.

ANSI/ASTM C384-1977	Impedance and Absorption of Acoustical Materials by the Impedance Tube Method
ANSI/ASTM C423-1977	Sound Absorption and Sound Absorption Coefficients by the Reverberation Room Method
ANSI/ASTM E336-1977	Measurement of Airborne Sound Insulation in Buildings
ANSI/ASTM E492-1977	Laboratory Measurement of Impact Sound Transmission Through Floor Ceiling Assemblies Using the Tapping Machine

Coming Soon....

CSA Standards Near Completion

Z107.0	Definitions of Common Acoustical Terms
Z107.24	Procedure for the Measurement of the Exterior Sound Level of Railbound Vehicles
Z107.25	Procedure for Measurement of the Exhaust Sound Level of Stationary Motorcycles
Z107.31	Test Procedure for the Measurement of Sound Levels from Agricultural Machines
Z107.32	Test Procedures for the Measurement of Exterior and Operator Sound Emitted from Construction, Foresty and Mining Machines

Standards Under Development

- Procedure for Measuring Sound Levels at a Distance
- Noise Monitoring Around Industrial Complexes
- Procedure for Measuring Blasting Noise
- Recommended Practice for the Calculation of Noise Dose
- Procedure for Measuring Impulsive Noise
- Procedure for Using Dosimeter to Determine the Noise Exposure of Industrial Workers

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