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PROCEEDINGS ISSUE

ACOUSTICS WEEK IN CANADA
SEMAINE CANADIENNE D'ACOUSTIQUE
1994

CAHIER DES ACTES

canadian acoustics

acoustique canadienne

THE CANADIAN ACOUSTICAL ASSOCIATION
P.O. BOX 1351, STATION "F"
TORONTO, ONTARIO M4Y 2V9

L'ASSOCIATION CANADIENNE D'ACOUSTIQUE
C.P. 1351, SUCCURSALE "F"
TORONTO, ONTARIO M4Y 2V9

CANADIAN ACOUSTICS publishes refereed articles and news items on all aspects of acoustics and vibration. Articles reporting new research or applications, as well as review or tutorial papers and shorter technical notes are welcomed, in English or in French. Submissions should be sent directly to the Editor-in-Chief. Complete instructions to authors concerning the required camera-ready copy are presented at the end of this issue.

ACOUSTIQUE CANADIENNE publie des articles arbitrés et des informations sur tous les domaines de l'acoustique et des vibrations. On invite les auteurs à proposer des manuscrits rédigés en français ou en anglais concernant des travaux inédits, des états de question ou des notes techniques. Les soumissions doivent être envoyées au rédacteur en chef. Les instructions pour la présentation des textes sont exposées à la fin de cette publication.

CANADIAN ACOUSTICS is published four times a year - in March, June, September and December. Publications Mail Registration No. 4692. Return postage guaranteed. Annual subscription: $10 (student); $35 (individual, corporation); $150 (sustaining - see back cover). Back issues (when available) may be obtained from the Associate Editor (Advertising) - price $10 including postage. Advertisement prices: $350 (centre spread); $175 (full page); $100 (half page); $70 (quarter page). Contact the Associate Editor (advertising) to place advertisements.

ACOUSTIQUE CANADIENNE est publiée quatre fois par année - en mars, juin, septembre et decembre. Poste publications - enregistrement n°. 4692. Port de retour garanti. Abonnement annuel: $10 (étudiant); $35 (individuel, société); $150 (soutien - voir la couverture arrière). D'anciens numéros (non-épuisés) peuvent être obtenus du rédacteur associé (publicité) - prix: $10 (affranchissement inclus). Prix d'annonces publicitaires: $350 (page double); $175 (page pleine); $100 (demi page); $70 (quart de page). Contacter le rédacteur associé (publicité) afin de placer des annonces.

EDITORIAL

This issue constitutes the Proceedings of the technical symposium of Acoustics Week in Canada 1994, to be held in Ottawa in October. Those of you attending should bring this issue along with you as a valuable source of reference. So far over ninety papers have been submitted; over sixty two-page summaries are published here. This will be an exciting meeting with interesting sessions, courses, banquet, CAA prize awards and, apparently, a large proportion of the Association membership in attendance.

For those of you still undecided as to whether or not you will attend, and who need a little bit more seduction, I am pleased to announce a special event that will take place during the symposium. Association President Raymond Hétu is organizing a special plenary session on "The Acoustical Environment - a Social Concern". It will involve a press conference and a round-table discussion of acoustical issues by representatives a number of organizations. The aim is to increase public awareness of acoustical issues and increase the visibility of the CAA - that is of you and your activities. You won't want to miss this event! In the words of Marge, my departmental administrator, "Be there or be square".

In my last editorial I mentioned that reader reaction to the Hétu article published in the March issue had been light, but positive. My remarks were clearly premature. Since then I have received considerably heavier reaction, not all of it positive. I plan to publish reader reaction, and the author's response, in the December issue. Anyone wishing to add their comments to the debate should do so to me by early October.

Cette édition du journal est composée des Actes du symposium technique de la Semaine Candienne d'Acoustique 1994, qui se tiendra à Ottawa en octobre. Ceux qui participeront au symposium sont invités à apporter leur copie; elle vous servira de source de références indispensable. Jusqu'à maintenant plus de 90 communications ont été soumises; plus de 60 résumés de deux pages sont publiés dans ce numéro. Le congrès promet d'être excitant avec des sessions intéressantes, des cours, le banquet, les prix de l'ACA et, paraît-il, une bonne participation des membres.

Pour ceux qui sont encore indécis quant à leur participation et qui doivent se faire davantage séduire, je suis heureux d'annoncer qu'un évènement spécial aura lieu au cours du symposium. Le Président de l'Association, M. Raymond Hétu, organise une session plénière spéciale sur "L'environment sonore - une prèoccupation sociale." Il y aura une conférence de presse et une table-ronde réunissant des représentants de plusieurs organisations qui aborderont diverses questions reliées à l'acoustique. L'objectif est de sensibiliser le public aux questions reliées á l'acoustique et d'accroître la visibilité de l'ACA - c'est-à-dire vous et vos activités. Vous ne voudrez pas manquer cet évènement. Comme le dirait si bien Marge, l'administrateure de mon département, "Soyez-y, sinon vous ne serez pas dans le coup".

Lors de mon dernier éditorial, je mentionnais que la réaction des lecteurs à l'article de Hétu paru dans le numéro de mars était limitée mais positive. Mes remarques étaient clairement prématurées. En effet, depuis ce temps, j'ai reçu des réactions beaucoup plus importantes et pas toutes aussi positives. Je prévois publier les réactions des lecteurs et les réponses de l'auteur dans le numéro de décembre. Tous ceux qui désirent participer au débat en me transmettant leurs commentaires peuvent le faire avant le début octobre.

ACOUSTICS WEEK IN CANADA 1994

Citadel Inn Ottawa

October 17-21, 1994

INVITATION TO PARTICIPATE

You are invited to participate in Acoustics Week In Canada 1994 to be held October 17 through 21 at the Citadel Inn Ottawa. Highlights of the week include seminars, a symposium, a laboratory tour, and entertainment. The week will begin on Monday, October 17 with a seminar on *Acoustical Calibration and Traceability.* Two more seminars will be given on Tuesday. Bruel&Kjaer Canada will address S*ound Power: Measurement and Applicable Standards*, while the other seminar, organized by Alberto Behar, will address *Hearing Conservation and Noise control.*

The Symposium will begin Wednesday morning and will consist of two full days of organized sessions on all aspects acoustics. Each day there will be three simultaneous sessions with invited and contributed papers. Specially catered luncheons for the delegates as well as a Wednesday evening Reception and Banquet will be held in the ballroom atop the Citadel Inn which offers a beautiful panoramic view of the city. The annual general meeting and student awards will be held on Thursday after the close of the symposium. Friday, members of the IRC Acoustics Laboratory at the National Research Council Canada will provide a tour of their facilities and a seminar in which details of current work will be given. Also included in the tour is a complimentary luncheon.

The venue for Acoustics Week In Canada will be the newly renovated Citadel Inn Ottawa located in the heart of downtown, only a short walk from Parliament Hill and other attractions. A large exhibition space central to the lecture rooms has been secured in which morning and afternoon coffee will be served. Discount hotel rates of $85 Single, $90 Double which include free breakfast, are available by telephoning the hotel directly at 1-800-567-3600 and identifying yourself as a CAA conference delegate. Members are encouraged to stay at the Citadel Inn as meeting room charges are determined by the number of guest rooms occupied by our delegates.

The cost of the Symposium will be $130 per person for CAA members, $165 for non-members, $40 for student members, $50 for non-member students. This includes both luncheons and the Banquet. Symposium registration will be conducted at the door, while seminar registration will be done in advance (forms in this issue).

Conference Chair
Dr. Trevor R. Nightingale, Tel: (613) 993-0102

A Note on Air Travel

Canadian Airlines International has been appointed as the official airline for our national meeting in Ottawa. Savings of up to 50% on full fare economy are available to delegates, pending availability and restrictions. Reservations should be made by calling Canadian Airlines Conventionair at 1-800-665-5554 and quoting event number "5437 in Ottawa."

SEMAINE CANADIENNE DE L'ACOUSTIQUE 1994

Citadel Inn, Ottawa

17-21 octobre 1994

INVITATION

Vous êtes invités à participer à la Semaine canadienne de l'acoustique, qui aura lieu au Citadel Inn, à Ottawa, du 17 au 21 octobre prochain. Au programme : des séminaires, un symposium, une visite de laboratoire et une réception avec banquet. La première journée, soit le lundi, il y aura un séminaire sur *L'étalonnage et la traçabilité en acoustique*. Deux autres séminaires seront présentés le mardi. Bruel&Kjaer Canada traitera de *La puissance acoustique : mesures et normes applicables*; quant à l'autre séminaire, organisé par Alberto Behar, il portera sur *La protection de l'ouïe et la lutte contre le bruit.*

Le symposium commencera mercredi matin et comportera deux journées complètes de séances organisées sur tous les aspects de l'acoustique. Trois séances simultanées de présentations sollicitées et offertes sont prévues chaque jour. Des repas du midi spécialement préparés par un traiteur et un banquet-réception (mercredi soir) seront servis dans la salle de bal à l'étage supérieur de l'hôtel, qui offre une vue panoramique de la ville. L'assemblée générale annuelle et la remise des prix aux étudiants auront lieu le jeudi après le symposium. Vendredi, des membres du personnel du Laboratoire d'acoustique de l'IRC au Conseil national de recherches du Canada dirigeront une visite de leurs installations et animeront une conférence portant sur lex travaux en cours. Un repas du midi gratuit est prévu à cette occasion.

La Semaine canadienne d'acoustique se déroulera à Ottawa dans l'Hôtel Citadel Inn récemment renové, qui se trouve en plein centre-ville et à quelques pas seulement de la Colline du Parlement et d'autres lieux attrayants. Un grand local d'exposition tout près des salles de conférences a été réservé et c'est là qu'auront leiu les pauses du matin et de l'après-midi. Les participants pourront réserver une chambre d'hôtel à prix réduit (chambre individuelle à 85 $, chambre double à 90 $, petit déjeuner compris) en communiquant directement avec l'hôtel au 1-800-567-3600 et en mentionnant leur participation au congrès de l'Association canadienne de l'acoustique. Nous encourageons les participants à loger à l'hôtel Citadel Inn puisque le tarif des salles de réunion est fonction du nombre de participants hébergés.

Le droit de participation au symposium est de 130 $ par personne pour les membres de l'ACA, de 165 $ pour les non-membres, de 40 $ pour les membres étudiants et de 50 $ pour les étudiants qui ne sont pas membres. Ce prix englobe les repas du midi et le banquet. On pourra s'inscrire au symposium le jour même, tandis que l'inscription aux conférences se fera à l'avance (formulaires inclus dans ce numéro).

Président du congrès

D^r^ Trevor R. Nightingale, téléphone (613) 993-0102

Voyages par avion

Les Lignes aériennes Canadian International sont le transporteur officiel pour notre rencontre nationale à Ottawa. Les délégués pourront épargner jusqu'à 50 % du plein tarif de classe économique, suivant les places disponibles et les restrictions applicables. On pourra réserver une place en communiquant avec Canadian Airlines Conventionair au 1-800-665-5554 et en mentionnant l'événement "5437 à Ottawa".

ACOUSTICS WEEK IN CANADA 1994

Citadel Inn Ottawa

SEMINARS

ACOUSTICAL CALIBRATION AND TRACEABILITY

Date: October 17, 1994.
Presented by: Dr. George Wong.
Location: National Research Council Canada, Building M-36.
Information contact: Elizabeth Lambe (613-993-5976, or FAX at 613-952-5113)
Language: English
Cost: $200 (includes lunch and coffee)

This one day event is designed for engineers, scientists and technologists who are involved with acoustical calibrations, tests and standards. It will follow an informal approach and will include discussions on:

- primary acoustical standards
- calibrators (reference sound sources such as pistonphones and field calibrators
- theory and selection of acoustical measuring instruments: sound level meters, integrating-averaging sound level meters, dosimeters, etc.
- ISO 9000 requirements
- acoustical calibration techniques and philosophy

The course will provide answers on acoustical calibrations, such as recommended calibration intervals and minimum requirements to ensure confidence in measurements with traceability to the primary standards maintained at the National Research Council of Canada.

Seminar Registration

Registration should be made through Elizabeth Lambe, the Information contatct. For registrations received after August 31, 1994, the fee will be $250. Please, note that these courses will only be offered if there is sufficient registration by August 31, 1994.

SEMAINE CANADIENNE DE L'ACOUSTIQUE 1994

Citadel Inn, Ottawa

SÉMINAIRES

L'ÉTALONNAGE ET LA TRAÇABILITÉ EN ACOUSTIQUE

Date : 17 octobre 1994
Présenté par : George Wong, Ph.D.
Endroit: Conseil national de recherches Canada, bâtiment M-36
Pour information : Elizabeth Lambe (613 993-5976, télécopieur 613 952-5113)
Langue : anglais.
Coût : 200 $ (comprend repas et café)

Ce cours d'une journée est destiné aux ingénieurs, scientifiques ou technologues s'occupant d'étalonnage, d'essais ou de normes dans le domaine de l'acoustique. On y traitera de manière informelle :

- des étalons primaires en acoustique
- des systèmes d'étalonnage (sources sonores de référence comme les pistonphones et les systèmes d'étalonnage sur place)
- de la théorie de la mesure acoustique et du choix des appareils : sonomètres, sonomètres intégrateurs-pondérateurs, dosimètres, etc.
- des exigences ISO 9000
- de la philosophie de l'étalonnage en acoustique et des techniques utilisées dans ce domaine

Des informations utiles seront fournies concernant l'étalonnage en acoustique, par exemple en ce qui a trait aux intervalles recommandés et aux exigences minimales à respecter pour obtenir des mesures fiables dont les résultats peuvent être reliés aux étalons primaires conservés au Conseil national de recherches du Canada.

Inscription aux séminaires

L'inscription doit être faite avec Elizabeth Lambe. Les frais d'inscription seront de 250 $ pour les inscriptions reçues après le 1er août 1994. Veuillez noter que les séminaires ne seront présentés que si le nombre d'inscrits, à cette date, est suffisant.

ACOUSTICS WEEK IN CANADA 1994

Citadel Inn Ottawa

SEMINARS

HEARING CONSERVATION AND NOISE CONTROL
Date: Tuesday October 18, 1994.
Presented by: Alberto Behar, Winston Sydenborgh, and Bob Pemberton.
Information contact: Alberto Behar (416-265-1816)
Language: English.
Cost: $100.

This is a practical one day course for plant personnel involved in hearing conservation programmes. This would include members of health and safety committees, safety officers, occupational nurses, and others involved in work place health and safety.

The course content will consider: how to design, implement, and assess hearing conservation programmes, including the selection of hearing protectors, as well as the measurement of noise levels and exposures. The second half of the course will consider engineering noise control issues and the selection of materials to reduce occupational noise levels.

SOUND POWER: MEASUREMENT AND APPLICABLE STANDARDS
Date: October 18, 1994.
Presented by: Bruel&Kjaer Canada.
Information contact: Robert Trepanier (514) 695-8225
Language: English.
Cost: $175.

It is becoming increasingly important to certify the sound power output of products. Equipment sold or exported to Europe must be labeled for sound power in accordance with standardized methods. With the growing importance of the European Community and the global market, the ability to provide a certificate of sound power is key to accessing these markets.

This seminar specifically addresses possible standards that can be used to obtain a recognized measure of sound power. The various accepted methods will be compared for accuracy and ease of implementation. A demonstration of sound power measurement using acoustic intensity is given and shown to be simple and accurate for most applications. Measurement technique, quality control indicators, and result interpretation will be discussed in detail. A basic knowledge and understanding of acoustics are required.

Seminar Registration

Please complete the registration form included in this issue. Note that these courses will only be offered if there is sufficient registration by August 31, 1994 and that space may be limited.

SEMAINE CANADIENNE DE L'ACOUSTIQUE 1994

Citadel Inn, Ottawa

SÉMINAIRES

PROTECTION DE L'OUIE ET LUTTE CONTRE LE BRUIT

Date : mardi 18 octobre 1994
Présenté par : Alberto Behar, Winston Sydenborgh et Bob Pemberton
Pour information : Alberto Behar (416 265-1816)
Langue : anglais
Coût : 100 $

Il s'agit d'un cours pratique d'une journée à l'intention du personnel d'usine faisant partie de programmes de protection de l'ouïe : membres de comités de santé et sécurité, agents de sécurité, infirmières spécialisées en hygiène professionnelle et autres personnes s'occupant de santé et de sécurité au travail.

La première partie du cours portera sur la façon de concevoir les programmes de protection de l'ouïe, de les mettre en oeuvre et de les évaluer, notamment sur le choix des protecteurs d'oreilles appropriés, ainsi que sur la mesure des niveaux de bruit et des expositions à celui-ci. La seconde partie sera consacrée aux aspects techniques de la lutte contre le bruit et au choix des matériaux destinés à abaisser les niveaux de bruit industriel.

PUISSANCE ACOUSTIQUE : MESURE ET NORMES APPLICABLES

Date : 18 octobre 1994
Présenté par: Bruel&Kjaer Canada.
Pour information : Robert Trépanier (514 695-8225)
Langue : anglais
Coût : 175 $

Il est de plus en plus important de certifier la puissance acoustique de sortie des produits. La puissance acoustique des appareils vendus ou exportés en Europe doit être déterminée suivant des méthodes normalisées. Étant donné l'importance croissante du Marché commun européen et la mondialisation des échanges, la capacité de fournir un certificat de puissance acoustique est essentielle aux entreprises qui désirent exporter.

Ce séminaire portera sur les normes que l'on peut utiliser pour en arriver à une mesure de la puissance acoustique qui soit reconnue. Les diverses méthodes acceptées seront comparées aux points de vue fiabilité et facilité de mise en oeuvre. On fera une démonstration de mesure de la puissance acoustique basée sur l'intensité sonore et on montrera que cette technique est simple et fiable dans la plupart de ses applications. La technique de mesure, les indicateurs de contrôle de la qualité et l'interprétation des résultats feront l'objet d'un examen approfondi.

Il faut connaître les grands principes de l'acoustique.

Inscription aux séminaires

S'il-vous-plaît remplir le formulaire d'inscription qui se trouve dans ce numéro. Veuillez noter que les séminaires ne seront présentés que s'il y a inscription suffisante au 31 août 1994, et que l'espace peut être limité.

ACOUSTICS WEEK IN CANADA 1994

Citadel Inn Ottawa

SEMINARS

INSTITUTE FOR RESEARCH IN CONSTRUCTION, ACOUSTICS LABORATORY TECHNOLOGY UPDATE SEMINARS

Date: October 21, 1994.
Presented by: Acoustics Laboratory, Institute for Research in Construction, National Research Council Canada.
Location: National Research Council Canada, Institute for Research in Construction, Acoustics Laboratory.
Information contact: Maria Clancy (613-993-2305)
Language: English.
Cost: $10.

This one day event will include both seminar presentations and laboratory tours conducted by IRC, Acoustics Laboratory researchers. The seminars will be of particular interest to practitioners of building acoustics as well as persons who design and develop and use acoustically engineered building products.

The morning will include presentations giving applied and practical information from our most recent client funded projects:

- Flanking sound transmission in wood frame constructions;
- Sound transmission through gypsum board walls;
- Degradation of sound insulation due to electrical outlets in walls;
- Sound power measurement of HVAC systems, and of low frequencies;
- Aircraft noise issues at Canadian Airports;
- Room acoustics measurement techniques and subjective testings;
- Design and commissioning of a new floor transmission test facility.

Where possible, handouts and report reprints will be made available to participants.

In the afternoon there will be guided tours of the Acoustics Laboratory facilities which include the wall and floor test suites, the anechoic room, the room acoustics test suite, and the flanking transmission test suite. The tour will provide participants with an excellent opportunity to discuss technical matters and plans for future projects with the researchers.

Buses will provide transportation to and from the Citadel Inn.

Seminar Registration

Please complete the registration form included in this issue. Note that these courses will only be offered if there is sufficient registration by August 31, 1994 and that space may be limited.

SEMAINE CANADIENNE D'ACOUSTIQUE 1994

Citadel Inn Ottawa

SÉMINAIRES

INSTITUT DE RECHERCHE EN CONSTRUCTION, SÉMINAIRES SUR L'ACTUALITÉ TECHNOLOGIQUE DU LABORATOIRE D'ACOUSTIQUE

Date: le 21 octobre 1994.
Presentées par: le Laboratoire d'acoustique, Institut de recherche en construction, Conseil national de recherches du Canada.
Endroit: Conseil national de recherches du Canada, Institut de recherche en construction, Laboratoire d'acoustique.
Contact: Maria Clancy (613-993-2305)
Langue: l'anglais.
Coût: 10 $.

Cette rencontre d'une journée englobera à la fois des exposés et des visites de laboratoire dirigées par des chercheurs du Laboratoire d'acoustique de l'IRC. Les séminaires intéresseront tout particulièrement les praticiens de l'acoustique architecturale et les personnes qui conçoivent, mettent au point et utilisent des matériaux de construction insonorisés.

Au cours des exposés du matin, les chercheurs de "IRC présenteront de l'information concrète tirée de leurs plus récents projets financés par la clientèle:

- La transmission latérale du son dans les bâtiment à ossature de bois;
- La transmission du son à travers les murs de plaques de plâtre;
- La dégradation de l'insonorisation sous l'effet des prises électriques dans les murs;
- La mesure de la puissance sonore des systèmes CVC et des faibles fréquences;
- Le bruit des avions dans les aéroports canadiens;
- Les techniques de mesure et les essais subjectifs en acoustique des salles;
- La conception et la mise en service d'une nouvelle installation d'essais (transmission par les planchers

Les participants recevront dans la mesure du possible des documents et des tirés à part.

Dans l'après-midi, on pourra visiter les installations du Laboratoire d'acoustique, y compris les locaux d'essais pour murs et planchers, la chambre anéchoïque, les locaux d'essais en acoustique des salles, ainsi que les locaux d'essais de la transmission latérale du son. Cette visite offrira aux participants une excellente occasion d'aborder des questions techniques et des projets futurs avec les chercheurs.

Des autobus feront la navette entre le CNRC et l'hôtel Citadel Inn

Inscription aux séminaires

S'il-vous-plaît remplir le formulaire d'inscription qui se trouve dans ce numéro. Veuillez noter que les séminaires ne seront présentés que s'il y a inscription suffisante au 31 août 1994, et que l'espace peut être limité.

ACOUSTICS WEEK IN CANADA 1994

Citadel Inn Ottawa, 101 Lyon Street, Ottawa

REGISTRATION FORM AND INFORMATION

SYMPOSIUM

Registration fee is $130 and will be collected at the door; pre-registration is not required.

SEMINARS

Registration for all seminars should be done in advance by completing this form.

ACOUSTICAL CALIBRATION AND TRACEABILITY
Date: October 17, 1994.
Registration contact: Elizabeth Lambe (613-993-5976, or Fax at 613-952-5113)

HEARING CONSERVATION AND NOISE CONTROL
Date: Tuesday October 18, 1994.
Cost: $100 before August, 31 1994, $125 after August, 31, 1994. $______

SOUND POWER: MEASUREMENT AND APPLICABLE STANDARDS
Date: October 18, 1994.
Cost: $175 before August, 31 1994, $200 after August, 31 1994. $______

INSTITUTE FOR RESEARCH IN CONSTRUCTION, ACOUSTICS LABORATORY TECHNOLOGY UPDATE SEMINARS
Date: October 21, 1994.
Cost: $10 before August, 31 1994, $15 after August, 31 1994. $______

Please, note that seminars will only be offered if there is sufficient registration by August 31, 1994 and that space may be limited.

Completed forms should be returned along with a cheque or money order made payable in Canadian funds to **CAA '94 Ottawa** at the address below:

Dr. Trevor Nightingale
CAA '94 Conference Chair
Post Office Box 74068
Ottawa, Ontario K1M-2H9

CONFERENCE HOTEL: CITADEL INN

The citadel Inn, as the conference hotel, has promised to offer our delegates the best available room rate for the days of our conference: $85 single, $90 double per night, including breakfast. You are urged to stay at the Citadel Inn since meeting room costs depend on the number of CAA delegates staying at the hotel. Please make reservations by calling: 1-800-567-3600 and identify yourself as a CAA delegate.

CONFERENCE AIRLINE: CANADIAN AIRLINES INTERNATIONAL

Canadian Airlines International, as the conference airline, will offer savings of up to 50% on full fare economy, pending availability and some restrictions. Reservations should be made by calling Canadian Conventionair at 1-800-665-5554 and quote event number "5437 in Ottawa."

SEMAINE CANADIENNE DE L'ACOUSTIQUE 1994

Citadel Inn, 101, rue Lyon, Ottawa

RENSEIGNEMENTS ET FORMULAIRE D'INSCRIPTION

SYMPOSIUM

Les droits d'inscription de 130 $ seront perçus à l'entrée; il n'est pas nécessaire de s'inscrire à l'avance.

SÉMINAIRES

Pour tous les séminaires, il faut s'inscrire à l'avance en remplissant ce formulaire.

L'ÉTALONNAGE ET LA TRAÇABILITÉ EN ACOUSTIQUE
Date : le 17 octobre 1994
Inscription: Elizabeth Lambe (613-993-5976, ou télécopieur au 613-952-5113)

LA PROTECTION DE L'OUÏE ET LA LUTTE CONTRE LE BRUIT
Date : le 18 octobre 1994
Coût : 100 $ avant le 31 août 1994, 125 $ après cette date $______

LA PUISSANCE ACOUSTIQUE : MESURES ET NORMES APPLICABLES
Date : le 18 octobre 1994
Coût : 175 $ avant le 31 août 1994, 200 $ après cette date $______

INSTITUT DE RECHERCHE EN CONSTRUCTION, LABORATOIRE D'ACOUSTIQUE, SÉMINAIRES « LE POINT SUR LA TECHNOLOGIE »
Date : le 21 octobre 1994
Coût : 10 $ avant le 31 août 1994, 15 $ après cette date $______

Veuillez noter que les séminaires ne seront présentés que si le nombre d'inscription au 31 août 1994 est suffisant, et que l'espace peut être limité.

Les formulaires dûment remplis, accompagnés d'un chèque ou mandat (en argent canadien) à l'ordre de l'**ACA 94 Ottawa,** doivent être envoyés à l'adresse suivante :

M. Trevor Nightingale, Ph.D.
Président du congrès ACA 94
Case postale 74068
Ottawa (Ontario) K1M 2H9

HÔTE DU CONGRÈS : LE CITADEL INN

Le Citadel Inn s'est engagé à offrir à nos délégués des chambres au meilleur tarif possible : 85 $ la nuit pour une personne et 90 $ pour deux, petit déjeuner compris. Nous vous conseillons vivement de loger au Citadel Inn, car le coût des salles de réunion dépendra du nombre de délégués de l'ACA qui y descendront. Pour réserver, veuillez composer le 1-800-567-3600 et dire que vous êtes un délégué de l'ACA.

TRANSPORTEUR : LES LIGNES AÉRIENNES CANADIEN INTERNATIONAL

Les Lignes aériennes Canadien international offriront un rabais pouvant atteindre 50 % du plein tarif en classe touriste, selon la disponibilité des places et à certaines conditions. Pour réserver, il faut appeler le service Conventionair de Canadien international (1-800-665-5554) et donner la référence «5437 à Ottawa».

15th International Congress on Acoustics

Trondheim, Norway, 26-30 June 1995

hosted by the Acoustical Society of Norway

GENERAL INFORMATION

Trondheim: the third largest city in Norway with a population of about 150,000. Located on the coast it enjoys a temperate climate thanks to the Gulf Stream. Trondheim can be reached by plane, train, and boat. In June you will experience the magnificent summer nights when it is never really dark, and the days are only separated by a couple of hours of twilight.

The Congress and Exhibition will be held at the campus of the Norwegian Institute of Technology (NTH), an integrated part of the University of Trondheim, from Monday 26 June to Friday 30 June 1995. The official opening will take place on the morning of 26 June in the nearby Olavs Hall.

Accommodation: the Congress organizers have made reservations at hotels of high standard within various price categories. Most hotels will be within walking distance of the Congress.

Lunches: As there are few restaurants open for regular lunch service in the vicinity of the University Campus, therefore lunch is provided for all participants in the nearby Student Union Building. It is absolutely necessary for us to make reservations in advance. Tickets for the lunches and also for the Congress Banquet will therefore be included in the total Congress fee.

An international **exhibition** of equipment, instrumentation, materials, and literature will be held in connection with the congress at NTH. Those wishing to participate in this exhibition should contact the ICA'95 Secretariat immediately for further details and to reserve space.

TECHNICAL PROGRAM

The congress shall cover all aspects within the subject of acoustics and associated subjects. The working languages for the congress are English, French and German. There will be no simultaneous translation services, and the papers will be presented in the language of the author's choice. The technical program of the congress will consist of four types of sessions: plenary sessions, special structured sessions, sessions for contributed papers and poster sessions.

Plenary Sessions: a number of lectures will be presented by invited distinguished speakers.

Structured Sessions: will focus on well-defined themes, lasting half a day each. They will consist of invited review or tutorial papers, workshops, panel discussions, or a combination of these formats. They will be organized and lead by an expert in that particular field.

Sessions of Contributed papers: will be arranged in connection with Structured Sessions on the same subjects. The papers will be presented orally and followed by a brief discussion, for a total period of 20 minutes per paper. These will take place in multiple parallel sessions.

Poster sessions: will take place in the Exhibition Area. The area set aside for posters and the exhibition is particularly well suited, being a glass covered area between the lecture theaters and other buildings. This is also the area where refreshments are served during the coffee breaks.

Proceedings: all papers (invited, contributed and poster presentations) will be included in the Congress Proceedings. Detailed information on how to prepare manuscripts and posters for the Proceedings will be provided directly to each author. The Proceedings are included in the registration fee, they will be issued to delegates at congress registration.

GUIDELINES FOR PROSPECTIVE AUTHORS

Initial submission
All those wishing to submit a paper should already have sent a tentative title and an abstract to the ICA 95 Secretariat. This preliminary abstract is for the use of the Scientific Program Committee only, to give a basis for the selection of papers. Abstracts for the Congress publications will be requested at a later stage.

Final submission
Authors whose papers have been accepted for presentation, will be notified in October 1994. They will also be given detailed instructions for the presentation of camera-ready final abstracts and papers for the Congress Proceedings. Deadline for submission of final abstracts and papers will be 15 February 1995.

Each paper submitted must be accompanied by a registration fee. This fee, which must be received before 15 February 1995, will be credited to the first-named author only. If a paper is withdrawn after submission, the discounted registration fee minus the NOK 500 administration charge will be refunded only if written notification is received by 1 May 1995 and you do not wish to participate.

SOCIAL PROGRAM

During the congress a number of concerts will be arranged, including a symphonic concert with works by renowned Norwegian composers and a concert in the medieval Nidaros Cathedral. Mini concerts will be held every day during the lunch break in the Student Union Building, the ICA Club will be open every night. The Congress Banquet will be held on the evening of Friday 30 June.

TIMETABLE FOR AUTHORS

15 September 1994	Deadline for receipt of preliminary abstract.
15 October 1994	Acceptance of papers. Authors will receive instruction.
15 February 1995	Deadline for receipt of final paper for proceedings.

FURTHER INFORMATION

Further information about ICA 95 is available from the congress secretariat:

ICA'95
SEVU, Congress Department
N-7034 Trondheim
Norway

Telephone: +47 7359 5251/7359 5254
Fax: +47 7359 5150
Electronic post: ica95@sevu.unit.no

Acoustics Week in Canada 1994 / Semaine Canadienne d'Acoustique 1994

Wednesday, October 19, morning -- PROGRAMME 1 -- mercredi, 19 octobre, matinée

Date and Time	**Joliet Room**	**La Chaudiere Room**	**Frontenac Room**
Wednesday	**Hearing in the Workplace**	**Outdoor Propagation**	**Architectural Acoustics - I**
October 19	**Chair: C. Laroche**	**Chair: M. Stinson**	**Chair: J. Swallow**
8:55	Opening Remarks	Opening Remarks	Opening Remarks
9:00	The Effects of Conventional and Active Hearing Protectors on Sound Localization by Normal and Hearing-Impaired Listeners *S.M. Abel, V.H. Hay*	Review of Physical Mechanism and Computational Models *G.A. Daigle*	Shear Membranes in Wood Frame Party Walls *D.B. Larson, R.A. Strachan*
9:20	A Case Report of Noise-Induced Tinnitus and Stuttering *T. Leroux*	Effects of Excess Ground Attenuation on Aircraft Noise Contours *J.S. Bradley*	Sound Transmission Between Columns and Plates *J.A. Steel*
9:40	Recognition of Reverberated Pulsed Signals *R. Larocque, R. Hétu, H.T. Quoc*	Science and Airport Noise Contour Prediction Programs *G. Bourgeois*	Flanking Transmission Caused by Fire Stops in Wood Frame Constructions *T.R.T. Nightingale*
10:00	Characterization of Occupational Sound Exposure of Professionals Involved in Highly Amplified Music Reproduction *M. Fortin, R. Hétu*	Highway Noise Prediction Improvements: Ground Effects and Multiple Barrier Diffraction *C.T. Blaney*	Numerical Modelling of Wood Frame Joints Having Fire Stops *T.R.T. Nightingale, R.J.M. Craik*
10:20	**Break**	**Break**	**Break**
10:40	Cases of Possible Job Discrimination Based on Hearing Loss *C. Laroche*	Insertion Loss Characteristics of Barriers and Berms *CC.. Harrison, K.R. Fyfe, L. Cremers*	Practical Aspects of Operable Wall Sound Isolation *A.D. Lightstone*
11:00	Finite-Element Modelling of the Normal and Surgically Repaired Cat Middle Ear *H.M. Ladak, W.R.J. Funnell*	The Draft International Standard Method (ISO/DIS 9613-2) for Calculating the Attenuation of Sound During Propagation Outdoors *J.E. Piercy*	Assessment of Errors in Sound Pressure Measurement in a Large Anechoic Chamber *S.E. Keith, M.G. Davidson, S.H.P. Bly*
11:20	Middle Ear Stapedius Muscle Acoustic Reflex and Laryngeal Amplitude Response *B. Orser*	The Estimation of the Linear Sound Speed Profiles Under General Meteorological Conditions *A. L'Espérance*	Sound Absorption Properties of a Double Layer Fibrous Panels - Application to the New Roof of the Olympic Stadium in Montreal *J.G. Migneron*
11:40	Temporal Acuity and Frequency Selectivity Among Normal Hearing Elderly Listeners *J.F. MacNeil, E.B. Slawinski*	Measuring the Effects of Turbulence *D.I. Havelock*	Occupational Noise Exposure in the High School Music Practice Room *W. Gastmeier, D. Pernu, M. Chasin*
12:00	**Lunch - TOP OF THE HILL - (Penthouse)**		

Acoustics Week in Canada 1994 / Semaine Canadienne d'Acoustique 1994

Wednesday, October 19, afternoon -- PROGRAMME 2 -- mercredi, 19 octobre, après-midi

Date and Time	Joliet Room	La Chaudiere Room	Frontenac Room
Wednesday	**Speech Perception and Production - I**	**Vibration**	**Architectural Acoustics - II**
October 19	**Chair: I. MacKay**	**Chair: M. Bracken**	**Chair: E.A.G. Shaw**
1:30	DSA (Digital Speech Aid) for Stuttering People *M. Roland-Mieszkowski, A. Czyzewski, B. Kostek*	Tuned Mass Dampers for High Rise Buildings - A Case Study *M.P. Sacks, J.S. Swallow*	Wave Versus Ray-Based Room Noise Models *C. Tang, K.R. Fyfe*
1:50	Rates of Normal Disfluency in the Speech of Stutterers and non-Stutterers *J. MacKay, A. Meltzer*	Multi-Celled Liquid Dampers to Eliminate Annoying Floor Vibrations *R.L. Shope, T.M. Murray*	When is Diffuse-Field Theory Accurate? *M. Hodgson*
2:10	Infant Dependence on Acoustic Cue Redundancy: Discrimination of Word-Final Voicing Contrast, /t/ - /d/ *M.A. Orme, L. Polka*	Exercise-Induced Building Vibrations: a Modern-Day Happening *G. Pernica*	A Review of Objective Descriptors for Sound Diffuseness *A. Abdou, R.W. Guy*
2:30	Voice Pitch as an Aid to Speechreading in Young Children *J.D. Fagg*	New Design Criterion for Walking Vibrations *D.E. Allen*	On Response Measurements Using Deterministic Types of Periodic Signal and FFT *W.T. Chu*
2:50	**Break**	**Break**	**Break**
3:10	Building Voice Lineups: One Method's Bias Problems *A.S. Laubstein*	Forced Vibrations of a Steel Cantilever Beam with Thick Viscoelastic Damping Layer *D.C. Stredulinsky, J.P. Szabo*	Comparison of Room Impulse Response Measurement Methods *S. Norcross, J.S. Bradley*
3:30	Effects of Noise on Identification of Topic Changes in Discourse *G.A. Tidball, M.K. Pichora-Fuller, J.H.V. Gilbert, N. Lamb*	Détectiopn Vibroacoustique des Fissures de Fatigue dans les Poutres *K. El Bikri, A. Berry, R. Gauvin*	Acoustical Analysis of the Orpheum Theatre *J.S. Bradley, G.A. Soulodre*
3:50	Is the Voiced-Voiceless Phonemic Boundary Influenced by an Intensity Level of the Presentation? *E.B. Slawinsky, J.F. MacNeil*	Determination of Dynamic Properties of Non-Linear Structures Using the Pull-Release Test *J.H. Rainer, G. Pernica*	The Components of Spatial Impression in Concert Halls *G.A. Soulodre, J.S. Bradley*
4L10	Perception, Production and Acquisition of English /r/ and /l/ Speech Contrasts by Korean Listeners *K. Yu, D.G. Jamieson*	P-C Based Measurement and Analysis of Traffic-Induced Vibrations *M.O. Al-Hunaidi, J.H. Rainer, G. Pernica*	Testing the Sensitivity of Stage Acoustics Measurement Techniques *J. O'Keefe*
4:30	Acoustic and Phonological Factors in the Perception of English /r/ and /l/ by Japanese Listeners *J.S. Logan, K. BharrathSingh*	Evaluation of Appropriate Sample Size for Measurement of Vibration Levels Induced by Rail Transit Vehicles *M.O. Al-Hunaidi, M. Bracken*	Acoustics Conditions in and Propagating from Orchestra Pits *J. O'Keefe*
7:00	**RECEPTION - TOP OF THE HILL - (Penthouse)**		
8:00	**BANQUET - TOP OF THE HILL - (Penthouse)**		

Acoustics Week in Canada 1994 / Semaine Canadienne d'Acoustique 1994

Thursday, October 20, morning -- PROGRAMME 3 -- jeudi, 20 octobre, matinée

Date and Time	**Joliet Room**	**La Chaudiere Room**	**Frontenac Room**
Thursday	**Hearing Accessibility**	**Underwater Acoustics**	**Noise Control - 1**
October 20	**Chair: K. Pichora-Fuller**	**Chair: D. Chapman**	**Chair: J. Nicolas**
8:25	Opening Remarks	Opening Remarks	Opening Remarks
8:30	The Effects of Acoustical Enviornments on People *M. Hodgson*	An Experimental Approach to Evaluation of Acoustic Masking of Beluga Communication by Ship Noise *C. Erbe, D.M. Farmer*	Controlling Control Valve Noise: A Case Study *R. Ramakrishnan, R. Gaspar*
8:50	Perceptual and Cognitive Factors Affecting Speech Understanding *B. Schneider*	Relationship of Underwater Acoustic Intensity Measurements to Beamforming *G.L. D'Spain*	Piezofilm Sensors for the Detection of Propagating Acoustic Pressure in Pipes *B.V. Chapnik, I.G. Currie*
9:10	Engineering Aspects of Assistive Device Technologies for Hard of Hearing and Deaf People *C.A. Laszlo*	Acoustic Intensity Measurements with Swallow Floats *F. Desharnais, G.L. D'Spain*	Reducing the Noise of Pressure Pulp Screens *R. Oddo, J. Nicolas, R. Panneton*
9:30	Electromagnetic Interference in Hearing Aid T-Coil Applications *B. McKinnon*	Minimizing Instrument Effects in an Ocean Bottom Seismometer *D.J. Dodds, D.M.F. Chapman, J.S. Osler, W.C. Risley*	Design of Circular Saw Blade for Quiet Operation *J-L. Wojtowicki, O. Beslin, J. Nicolas*
9:50	Building Performance for Hearing Impaired People *J.R. Champagne*	**Break**	**Break**
10:10	Validation of Masked Threshold Predictions Among People with Sensorineural Hearing Loss *R. Hétu, H.T. Quoc*	Seismo-Acoustic Measurements Using an Ocean Bottom Seismometer in the High Arctic *S.E. Dosso, G.H. Brooke, D.G. Baade*	Characterization of the Noise Generation Mechanism of Percussion Drill Steel Rods in Real Operating Conditions *C. Lesage, R. Oddo, Y. Champoux*
10:30	Hearing Accessibility in a Home for the Aged *M.K. Pichora-Fuller*	Panel Discussion: Use of Geophones in Under Water Acoustics Research	Response of a Non-Baffled Sandwich Panel Submitted to a Reverberant Chamber Acoustic Environment *H. Nélisse, O. Beslin, J. Nicolas*
10:50	Acoustics and Technology: A Hard of Hearing Perspective *R. Warrick*		An Analytical Approach for the Frequency Response of a Multilayer Disc *B. Cournoyer, N. Atalla, J. Nicolas*
11:10	Multi-Disciplinary Government Policy and Accessibility *J. Harvey*		Wheel Squeal in Rail Transit Systems *D. Chin-Quee*
11:30	**Plenary Panel Discussion:** L'Environnement sonore: une préoccupation sociale/ The Acoustical Environment: A Social Concern		
12:30	**Lunch - TOP OF THE HILL - (Penthouse)**		

Acoustics Week in Canada 1994 / Semaine Canadienne d'Acoustique 1994

Thursday, October 20, afternoon -- PROGRAMME 4 -- jeudi, 20 octobre, après-midi

Date and Time	Joliet Room	La Chaudiere Room	Frontenac Room
Thursday	**Speech Perception and Production - II**	**Audio Engineering**	**Noise Control - II**
October 20	**Chair: V. Bouché**	**Chair: G. Soulodre**	**Chair: C. Sherry**
2:00	Etudes Acoustiques Comparatives des Tons "hoi" - "nga" en Vietnamien du Nord et du Sud *Nguyen Ngoc-Quang*	Subjective Evaluation of High Quality Audio Systems *T. Grusec*	Broad-Band Active Noise Reduction in Communication Headsets by Digital Feedforward Control *G.J. Pan, A.J. Brammer, R.A. Goubran, J.G. Ryan, J. Zera*
2:20	Relâchement et Abrègement des Voyelles: Phonétique ou Phonologie? *F. Poiré*	Digital Signal Processing Applied to the Equalization of the Loudspeaker Room Interaction *C.R. Fortier, P. Côté*	Audiometric Services for Medium Size Companies *A. Behar*
2:40	Le Focus et l'Intonation en français parlé à Montréal *L. Levac*	Reducing the Variability of Loudspeaker Preference Ratings Through Digital Equalization *J.G. Ryan*	The Acoustical Challenge of Quarry Design *J. Emeljanow, A.D. Lightstone*
3:00	Pharyngeals in Tigrinya *J. Anderson, H. Rogers*	Ambisonic Sound for the Masses *J.S. Bamford*	The New CSA Standard Z107.56 - Occupational Noise Measurement *T. Kelsall, A. Behar*
3:20	**Break**	An Overview of Audio Technologies in Teleconferencing: From Source to Receiver and Back *B.P. Basnett*	**Break**
3:40	An Acoustic Study of Nasalization in Inor *A. Indrissi*	Building a Recording Studio in a Bank! A Practical Examination of Some of the Acoustical and Audio Design Considerations for a New Multi-Track Digital Facility for Montreal's Les Disques Star *T. Hewlings, J-L. Louradour*	Venturis as Silencers - a Case History *T. Kelsall, T. Gerritsen*
4:00	Digital Audio Laboratory Station *M. Roland-Mieszkowski, D. Roland-Mieszkowski, A. Czyzewski, B. Kostek*	Sound System Modelling Software *P. Giddings*	
4:20	Accounting for Musicians' Superior Auditory Serial-Order Identification: Audition or Notation? *J.A. Lymburner, A.J. Cohen*	Room Acoustics Considerations for Loudspeaker Systems in a Large Reverberant Space *J. O'Keefe*	
4:40	Rhythm and Compression in Venezuelan Caribbean Spanish *G.A. Toledo*	Sound for a Million - An Overview of Design Guidelines for Outdoor Sound Systems for Very Large Group Events *T. Paige*	
5:00			**CAA Business Meeting**

PROCEEDINGS OF ACOUSTICS WEEK IN CANADA 1994
ACTES DE LA SEMAINE CANADIENNE D'ACOUSTIQUE 1994

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SHEAR MEMBRANES IN WOOD FRAME PARTY WALLS

D. B. Larson and R. A. Strachan
Brown Strachan Associates
Two Yaletown Square
1290 Homer Street
Vancouver, BC, Canada. V6B 2Y5

1. Introduction

On a recent project, Brown Strachan had recommended a standard party wall construction consisting of 2 x 6 wood studs @ 16" o.c., separated by a 1" air space, with fibreglass batt insulation in the stud cavities, and one layer of 5/8" gypsum wall board (gwb) on each outside face. This wall has a typical sound transmission class rating of STC 57.

Late in the design stage, however, the structural consultant added a requirement for 1/2" plywood on both inside faces of some walls. The plate spacing was increased to maintain a 1" clear space between the two inside plywood faces. A Gypsum Association test shows STC 57 with 1/2" Type X gwb on the inside face of studs @ 24" o.c. On the basis of this test, the architects concluded that the change was acceptable, as the Code required a STC 50 rating.

Geiger & Hamme had tested a similar wall consisting of 5/8" drywall on the inside of studs spaced 16" o.c., and obtained a rating of STC 45. Obviously, the bending stiffness of the inside face had a major effect on the net stiffness of the air trapped between the two inside layers, spaced 1" apart. Plywood is about eight times as stiff as gwb, and since the stiffness of the inside plate is the key factor in determining the stiffness of the air, a problem was clearly indicated.

Potential treatment options included resilient channel, extra layers of drywall, and holes in the plywood membrane which would act to vent air in the 1" airspace to the fibreglass filled stud cavity. A mathematical model was used to study the effects of the various treatment options, and field measurements were made to determine the treatment results.

2. Modelling

The frequency region of concern lies between 80 Hz and 250 Hz, well below the coincidence frequencies of the plywood and drywall surfaces. For this reason, it was felt that an analysis of the forced wave transmission loss, considering the net volume displacements of a representative stud section, should provide a useful indication of the wall motions.

Initially, the assumption was made that the studs would remain effectively motionless, relative to the drywall and plywood motion between the studs. Although stud motion had been included to allow coupling, the effect of the stud motion coupling to the other motions via the stiffness of the air space had not been considered. This led to poor agreement between predictions and measurements, to be discussed below.

Eventually, ten degrees-of-freedom were defined: four describing the motion of each of the four plates at a point centred between the studs, four describing the plate motion at the studs, and two describing the motion of air in holes drilled in the inner shear membranes. The plate motion was considered to be the sum of a sine term, with zero motion at the studs and maximum motion at the plate centre, plus a cosine term, with maximum motion at the studs and zero motion at mid-plate.

An evanescent term could have been introduced to match a clamped-clamped edge condition, but a simply supported mode shape was assumed. Appropriate adjustments to the plate bending stiffness were made in consideration of a clamped-clamped condition. For forced radiation, the effective volume displacements may be out by about 20%, or less than two decibels.

If a high plate stiffness is assumed, the model degenerates to mass law transmission, and we obtain standard mass law transmission loss below the mass-air-mass resonance. There is a five decibel reduction in reverberant transmission loss, relative to normal incidence transmission loss. For simplicity, this has been approximated by the transmission loss at 60°, with a six decibel reduction relative to normal incidence. As the model operates well below coincidence, this is considered a reasonable approximation.

To summarize, the model has a diagonal mass matrix and a fairly full stiffness matrix. This stiffness matrix includes both bending stiffnesses and air stiffnesses, expressed as equivalent volume stiffnesses per square meter. Damping terms are included as complex stiffness moduli.

At low frequencies, as discussed earlier, only a forced wave was considered, with radiation proportional to the square of the volume displacement. Thus, we have the condition that sine squared plus cosine squared equals one, and we have uniform radiation for a homogeneous plate.

3. Recommendations and Results

Early discussions regarding potential wall modifications to meet the STC 50 Code requirement focused primarily on venting the air in the 1" airspace by means of drilling holes in one of the two shear membranes. To reduce the stiffness of the airspace in the critical 80 Hz to 250 Hz region, a hole with a Helmholtz resonance of approximately 300 Hz was recommended. This corresponded to 4" holes @ 16" o.c. This should have allowed air to vent into the stud cavity and thus provide an improvement of about 10 decibels at 125 Hz, as the stiffness would have been reduced to one third of its original value. Figure 1 shows transmission loss results predicted using the mathematical model. Notably, transmission loss at 160 Hz increases with 4" holes drilled in one shear membrane.

The predicted improvement in transmission loss was not observed in field measurements. Figure 2 shows field measurement results for the tests described. As indicated in Figure 2, this wall construction received a STC 39 rating, limited in the 125 Hz and 160 Hz third octave bands. It was found that the flow resistance of the fibreglass at the hole was significant, relative to the inertia of the air in the hole. Once this was recognized, the model was modified to consider fibreglass resistance.

To reduce flow resistance at the hole, still larger diameter holes were recommended. As 8" diameter holes were unacceptable to the structural consultant, a wall with 4" holes in one shear membrane and 6" holes in the second membrane was tested. This test obtained a STC 50 rating, but indicated that a substantial open area would be required. As the costs associated with such an extensive amount of drilling proved prohibitive, alternate treatments were studied.

Evaluation of a wall system with no coupling between the studs and drywall indicated that significant improvement should be obtained by using very flexible resilient channel. It was therefore recommended that a wall using the most flexible resilient channel available be tested. A wall with resilient channel on one side and one layer of drywall on each side obtained a STC 48 rating.

A subsequent test of a wall with two layers of gwb on resilient channel received a STC 50 rating. The use of two layers of gwb was thought to raise the "beam" resonant frequency of the drywall on a 14.4" "span", to the extent that the drywall and stud motions remain in phase, thus adding to the effective net inertia at the stud. This effect would lower the low frequency mass-air-mass resonant frequency, a resonance which was considered the major contributor to the poor performance in the 125 Hz and 160 Hz third octave bands. The tested wall did have 4" holes drilled in one side of the shear membrane but as discussed above, the 4" holes proved to be ineffective because of high flow resistance. In fact, tests of wall systems that only differ in that one wall has 4" holes drilled in a shear membrane and one does not have received STC ratings differing by only one point.

One final assembly tested consisted of a wall with a single layer of drywall and resilient channel on each side. This wall received a STC 48 rating. This rating was partially attributed to the inadvertent use of "standard" resilient channel, rather than the more flexible channel recommended. In fact, site coordination problems such as this highlighted the need for the simplest possible remedial action.

4. Conclusions

The behaviour of a standard double stud party wall below 300 Hz may be adequately modelled as a ten degrees-of-freedom system. Consideration of the net volume displacements of a typical stud section provided useful indications of wall motions.

There are treatments available to minimize the degradation in noise isolation that occurs in standard double stud party walls when shear membranes are applied to the interior of the studs. Venting of the encapsulated air space by means of holes in the shear membrane can be effective providing that the holes are large enough to overcome the flow resistance of insulation adjacent to the holes. Alternately, resilient channel and extra layers of drywall may be added to the wall to bring STC ratings up to minimum Code requirements.

5. References

Beranek, L. L. and Vér, I. L. (1992), *Noise and Vibration Control Engineering*, John Wiley & Sons, Inc., New York.

DuPree, R. B. (1981), *Catalog of STC and IIC Ratings for Wall and Floor/ Ceiling Assemblies With TL and ISPL Data Plots*, Office of Noise Control, California Department of Health Services, California

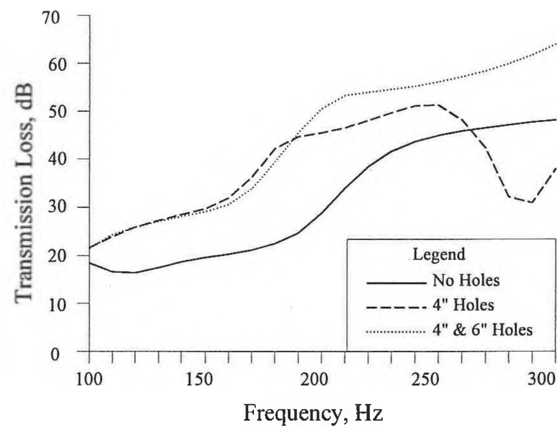

Figure 1: Predicted Transmission Loss of Various Wall Constructions

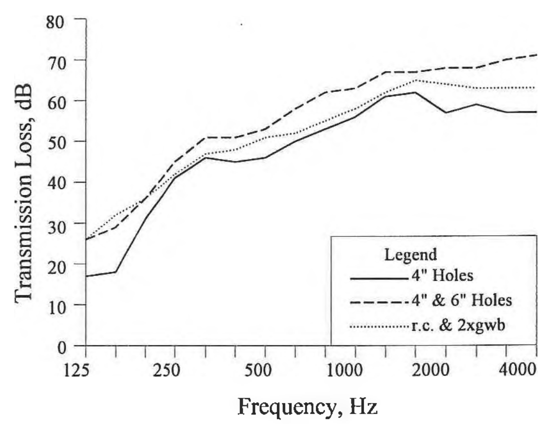

Figure 2: Field Transmission Loss Test Results for Various Wall Constructions

SOUND TRANSMISSION BETWEEN COLUMNS AND PLATES

J. A. Steel

Department of Mechanical Engineering, Heriot-Watt University, Riccarton, Edinburgh EH14 4AS

1. INTRODUCTION

While there have been a large number of studies of sound transmission in traditional buildings consisting mainly of walls and floors the same cannot be said for framed buildings. As framed structures become more common in the design of industrial and domestic buildings so understanding the transmission mechanisms between columns, beams and plates becomes more important. The types of joint that can be found include connected columns and floors and joists and walls. Normally a column /floor joint is complicated by beams which support the floor at the top of columns.

In this work the transmission mechanism at such joints is studied both theoretically and experimentally using Statistical Energy Analysis (SEA). The effects of beams at the joint are investigated.

2. COUPLING LOSS FACTORS

The coupling loss factor (CLF), η_{ij}, is defined as the power flow in one cycle of vibration from subsystem i to subsystem j. For a bending wave on a column which is incident on a joint with another structure the CLF is given by Steel [1] in terms of the transmission coefficient, τ_{ij}, as,

$$\eta_{ij} = \frac{2C_i\tau_{ij}}{\omega L_i} \qquad (1)$$

where C_i is the bending wave speed and L_i is the length of the column. The transmission coefficient can be calculated using expressions for the impedance of the plate structure at the joint [1,2,3].

Predicted CLF's were calculated using equation (1). Measured CLF's were calculated from the product of the measured acceleration level differences and damping.

3. TEST STRUCTURES

Measurements were made on two laboratory models. A steel bar (2.5x0.025x0.006 m) was welded to a steel plate (3x1.5x0.006 m) so that the model investigated by Lyon and Eichler [4] could be studied. Bars were then welded along an edge to the plate and the end of the column, through the joint. This allowed investigation of vibration transmission for symmetric and asymmetric excitation of the plate/beam structure. Another beam was added to form cross beams at the joint. Impedance's derived by Goyder and White [2,3] were used when calculating transmission coefficients and coupling loss factors for the joint for various axis of vibration. A similar process was carried out for a concrete model (column 2x0.1x0.1 m; plate 2.4x2x0.1 m; beams 2.4x0.2x0.2 m). The steel column and plate always had at least one mode of vibration in each 1/3 octave band for the frequency range considered. The concrete column had a number of discrete modes in the frequency range considered. This allowed the study of coupling where there are few modes.

The test structures were excited by tapping a plastic headed hammer over the surface of a subsystem for 15 seconds and measurements were repeated until confidence limits were less than ±3dB.

4. RESULTS

The measured and predicted CLF's for coupling from the concrete column are shown in Figure 1. Two measured results are shown for symmetric and asymmetric excitation of the plate/beam structure. When the column is excited to twist the plate/beam structure, symmetrically, the predicted CLF (solid line) shows a peak at 2000 Hz and the measured results fluctuate about the prediction showing good agreement. When the column is excited to bend the plate/beam structure, asymmetrically, the CLF falls with increasing frequency. Similar results are found with the steel model.

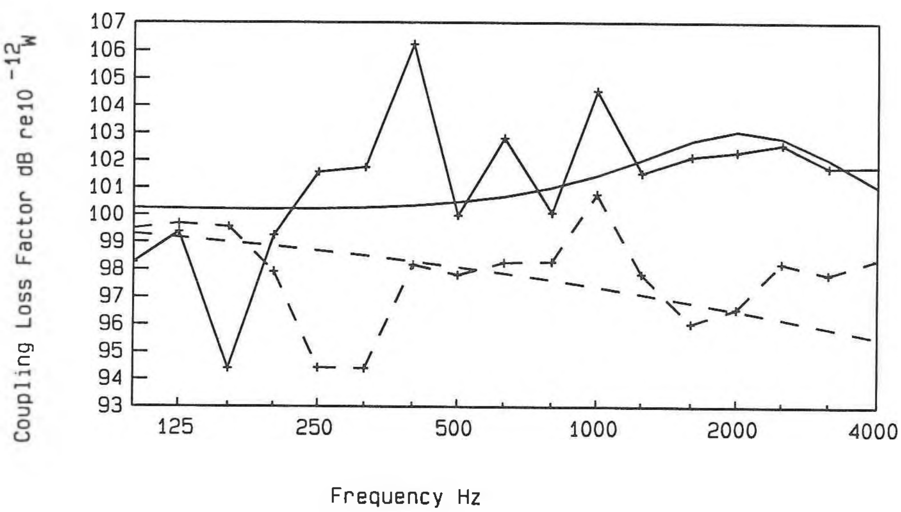

Figure 1. Measured and predicted coupling loss factors for transmission from a concrete column to a plate/beam.— predicted (symmetric); —+— measured (symmetric); – – predicted (asymmetric); – + – measured (asymmetric).

The case where cross beams are found at the joint is also investigated and coupling is found to be similar to the results for asymmetric excitation of a single beam.

When the plate is excited the coupling to the column is dominated by the modal response of the column.

5. CONCLUSIONS

Measured and predicted results show good agreement. Coupling between columns and plate/beam structures has been investigated. The beams can influence the transmission characteristics at these joints. The orientation of the beams effects the coupling at the joint.

6. REFERENCES

1. J. A. Steel and R.J.M. Craik, Sound transmission between columns and floors in framed buildings. *Applied Acoustics.* **39** (1993) 191-208.

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FLANKING TRANSMISSION CAUSED BY FIRE STOPS IN WOOD FRAME CONSTRUCTIONS

T.R.T. Nightingale
Acoustics Laboratory, Institute for Research in Construction,
National Research Council Canada,
Ottawa Ontario K1A 0R6

Introduction

In multi-family dwellings, the nominal party wall or floor between units is both fire and sound rated. At the joint between two fire rated assemblies, there must exist a fire stop to control smoke and flame spread in the event of a fire. Typically, the fire stop provides a physical connection between building elements thereby causing a flanking path. The degradation of the net sound isolation of a double leaf construction is examined for two types of fire stop materials listed in the National Building Code of Canada (NBCC). In this paper, fire stops formed from continued room surfaces are examined. A section through the specimen without any fire stopping is shown in Figure 1.

Measured Results

Figure 2, a plan section through the party and end walls, shows the installed fire stop; both layers of the 13 mm gypsum board were run across the end of the party wall separating rooms A and B. This represents an acceptable fire stop construction as gypsum board having a thickness of greater than 12 mm satisfies the NBCC criterion[1].

Figure 3 shows the measured net sound isolation for the assembly with and without the fire stop at the end of the party wall. The presence of the fire stop between rooms A and B has degraded the net sound isolation by 9 STC points. The outstanding sound isolation of STC 63 has been reduced to STC 54. Other room pairs were not impacted.

Figure 4 shows the measured transmission loss data between rooms A and B (ASTM E336) for the construction with and without the fire stop at the end of the partition wall. From the figure it is clearly evident that the presence of the fire stop has degraded the sound isolation over the frequency range 500 to 4000 Hz. However, the sound isolation for frequencies greater than 2000 Hz was severely degraded; by 10 dB or more.

The second fire stop to be examined is at the base of the party wall separating Rooms A and B. Figure 5 shows the fire stop formed by running the 16 mm thick plywood floor decking under the partition wall. Figure 6 shows the measured net sound isolation between the various room pairs expressed in terms of STC. From the figure it can be seen that the sound isolation between rooms A and B was severely degraded by the fire stop. The system, with the fire stop, achieved an STC 45, a degradation of 17 STC points. As installed, it would fail to meet the NBCC criterion of STC 50. Rooms on the diagonals were also effected but to a much lesser extent. From Figure 4, it can be seen that the flanking path caused by the fire stop was the dominant transmission path for the frequency range 160-4000 Hz. The affected range of

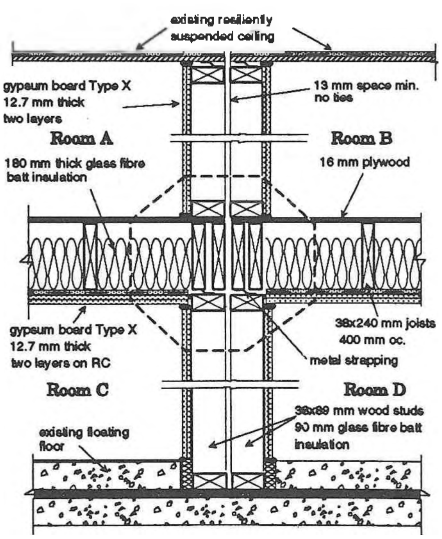

Figure 1: Section through the specimen without any fire stopping.

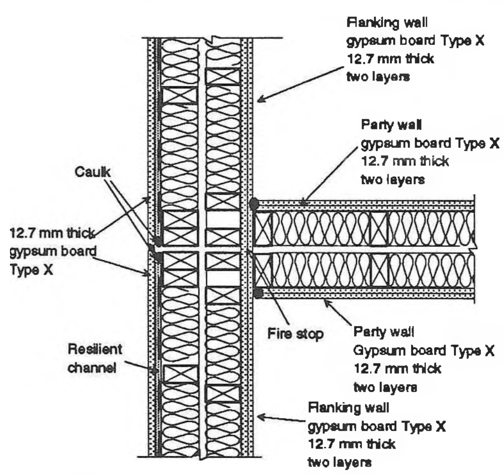

Figure 2: Plan section through the party and end walls showing the fire stops.

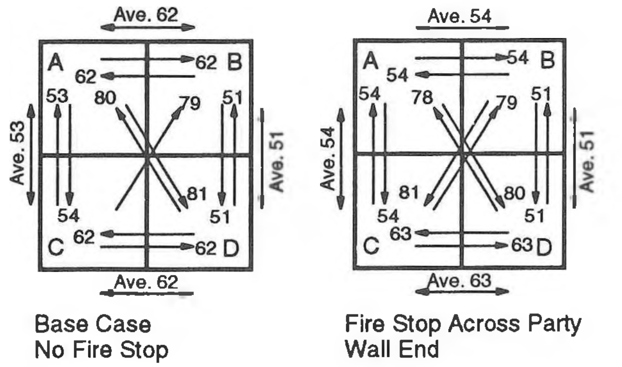

Figure 3: Measured net sound isolation(STC) for room pairs with and with out the fire stop across the end of the party wall.

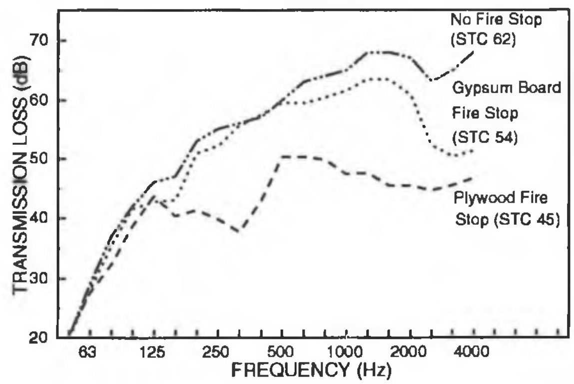

Figure 4: Measured net transmission loss (ASTM E336) between rooms A and B for the three cases.

frequencies represents nearly all of the building acoustics range which makes the impact so severe.

Discussion

The two fire stops considered – which are not typical of all fire stops – were constructed by continuing a room surface across or under the partition. This provided a very efficient flanking path for vibratory energy to propagate from one room to the other without having to pass through the partition wall. The magnitude of this flanking transmission and hence the degradation of the net sound isolation is a function of three factors. They are:

- Ability of the surface to accept airborne acoustic energy in the source room and convert it to vibration energy;
- Effectiveness of the continued surface (i.e. fire stop) for transmitting the vibration energy;
- Ability of the receiving room surface to convert the vibratory energy to airborne acoustic energy.

The effectiveness of the continued surface to transmit the vibratory energy through the joint at the two walls is discussed in the companion paper[2].

The ability of the source surface to accept and the receive surface to radiate sound energy is directly related to the radiation efficiency of the surface. The radiation

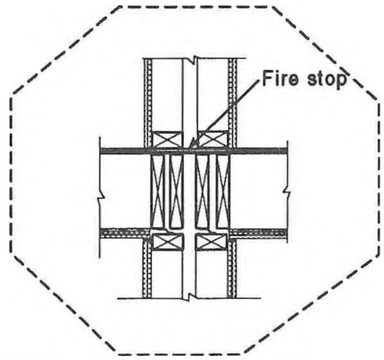

Figure 5: Section through the specimen showing the fire stop under the party wall separating rooms A and B.

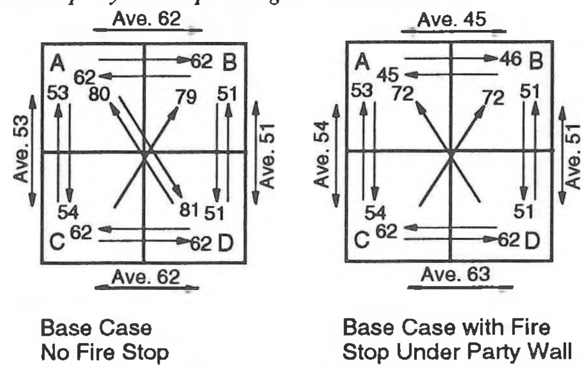

Figure 6: Measured net sound isilation (STC) for room pairs with and without the fire stop under the party wall.

efficiency of a surface is a maximum at the critical frequency and is only slightly less for frequencies above the critical frequency Thus, for a system having a joint whose transmission characteristics are reasonably independent of frequency, it is the radiation efficiency of the flanking surface that defines the frequency at which the flanking transmission will be the greatest. This was shown in Figure 4 where the frequency at which severe degradation begins (~2000 Hz) is very close to the critical frequency for the double layer of the 13 mm gypsum board (~2500 Hz) that forms the fire stop and the continued room surfaces. With the floor decking under the party wall, the wide range of affected frequencies may be explained by viewing the floor as having two distinctly different stiffnesses. The floor stiffness in the direction perpendicular to the joists will be similar to that of the plywood, while in the direction parallel to the joists the apparent stiffness will be very much higher. This causes there to be two critical frequencies – one for each of the orthogonal directions.

Fire stops formed by continuing a room surface across or under the nominally separating element should be avoided as they form flanking paths.

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[1]National Building Code of Canada, Fire Stopping 9.10.15.

[2]Nightingale,T.R.T, Craik,R.J.M., Numerical Modelling of Wood Frame Joints Having Fire Stops, Canadian Acoustics Proceedings 1994.

NUMERICAL MODELLING OF WOOD FRAME JOINTS HAVING FIRE STOPS

T.R.T. Nightingale
Acoustics Laboratory, Institute for Research in Construction,
National Research Council Canada, Ottawa Ontario K1A 0R6
Robert J.M. Craik
Department of Building Engineering and Surveying,
Heriot Watt University, Riccarton, Edinburgh UK EH14 4AS

Introduction

In a companion paper[1] the degradation of the net air borne sound isolation resulting from the flanking path caused by a special class of fire stops was investigated. This paper will examine a simple model for the propagation of bending waves through the joint involving a fire stop constructed by continuing one of the room's surfaces across or under the nominally separating element. Figure 1 shows a simplified section through the partition wall and the end wall that will be modelled. Due to the symmetry of the joint it can be viewed as a pair of corner joints coupled by an element (i.e., the fire stop) having a unique bending stiffness. In the simplified model presented here, only the transverse component of the surface velocity will be considered as it is this motion (i.e., that caused by bending waves) that will be, by far, the most effective at converting structural vibration to airborne acoustic energy.

Joint Equations

In deriving the equations for the joint it is assumed that each corner is pinned (i.e., there is no translational motion) and the fire stop that couples the corner joints has only stiffness. This is not the case in reality as the fire stop usually has a width equal to the 25 mm nominal gap left in double stud walls. The effects of the studs at the joint will not be considered as their treatment will introduce wave types other than bending waves. A mechanical representation of the simple system is shown in Figure 2. If the corner joints are assumed to be pinned rigid then the following equations of motion can be written,

$$\phi_1 = -\phi_2 \text{ and } \phi_3 = -\phi_4, \qquad [1a]$$

$$M_1 - M_2 + \hat{M} = 0 \text{ and } M_3 - M_4 - \hat{M} = 0, \qquad [1b]$$

where φ is the angular rotation, and M is the moment. The subscript indicates the arm of the joint (as per Figure 1) while the circumflex refers to the fire stop material that connects the two corner joints. If the bending wave enters the joint from arm 1 then the following equations define the transverse displacement, ξ, in each arm,

$$\xi_1 = \left(e^{-ik_1 x\cos\theta_1} + T_1 e^{+ik_1 x\cos\theta_1} + T_{n1} e^{+k_{n1}x}\right) e^{-ik_1 y\sin\theta_1}, \qquad [2a]$$

$$\xi_2 = T_2 e^{-ik_2 z\cos\theta_2} + T_{n2} e^{-k_{n2}z}, \qquad [2b]$$

$$\xi_3 = T_3 e^{-ik_3 x\cos\theta_3} + T_{n3} e^{-k_{n3}x}, \qquad [2c]$$

$$\xi_4 = T_4 e^{ik_4 z\cos\theta_4} + T_{n4} e^{k_{n4}z}, \qquad [2d]$$

where T is the amplitude of the bending wave on the arm indicated by the subscript, the 'n' of the subscript indicates the near field or evanescent wave, k is the wavenumber for travelling waves, k_n is the wavenumber for the evanescent waves and θ is the angle of propagation. The displacements given in Equation[2] are related to Equation[1] by,

$$\phi_1 = \frac{d\xi_1}{dx} \text{ and } M_1 = -B_1\left[\frac{d^2\xi_1}{dx^2} + \mu\frac{d^2\xi}{dy^2}\right], \qquad [3]$$

where B is the bending stiffness of the arm and μ is Poisson's ratio. Equations[1], [2] and [3] represent a set of four simultaneous equations involving the eight unknown coefficients. The additional four equations required for a unique solution can be realized by considering the constraint placed on the motion at the joint. This motion must be purely rotational (i.e., the displacement is zero at x=z=0). This gives,

$$T_1 + T_{n1} = -1,\ T_2 + T_{n2} = 0,\ T_3 + T_{n3} = 0,$$

and $$T_4 + T_{n4} = 0. \qquad [4]$$

Now a set of eight simultaneous equations can be created to describe the amplitude of the bending waves in each of the four arms. The resulting matrix for the case of normal incidence is Equation[5] (shown on next page).

Fire Stop Modelling

Equation[5] was used to calculate the velocity level difference or velocity transmission loss across the joint shown in Figure 1. The effective bending stiffness of the fire stop material is given by,

$$\hat{B} = \frac{Et^3}{(1-\mu^2)}\frac{1}{d} \qquad [6]$$

where E is Young's Modulus, μ is Poisson's ratio, and *d* is the span of the fire stop. (Typically this is about 25 mm since this is the distance between plates in a double stud wall.) The predicted surface velocity transmission loss at normal incidence between arms 1 and 3 (TL_{13}) is shown in Figure [3] for a few possible fire stops. Due to symmetry in constructions of arms 3 and 4, $TL_{13} = TL_{14}$. (Each arm had a bending stiffness per unit width of 340 Nm.) From the figure it is evident that the TL across the joint is a function of both stiffness and frequency. The 0.38 mm sheet steel fire stop, having very little bending stiffness, offers the best vibration isolation with the TL increasing 6 dB per doubling of frequency. Comparing the TL for 13 mm gypsum board with spans of 8 mm and 25 mm one can see that reducing the span of the fire stop reduces the TL. A fire stop made from two layers of 13 mm gypsum board only provides slightly better TL than a single layer of 16 mm plywood. For the frequency range 100 to 100 Hz the plywood fire stop is only marginally better than if the joint were perfectly rigid (i.e., a cross joint). It should be noted that the surface velocity transmission losses given here can not be compared directly to standard measures of airborne sound reduction (i.e., ASTM E90 or E336 which are measures of sound power reduction). This is due to the fact that sound power accepted/radiated by a surface is related to the surface velocity through the radiation factor which has not been considered in this discussion.

$$\begin{bmatrix} ik_1 & k_{n1} & ik_2 & k_{n2} & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & -ik_3 & -k_{n3} & -ik_4 & -k_{n4} \\ B_1k_1^2 - i\hat{B}k_1 & -B_1k_{n1}^2 - \hat{B}k_{n1} & -B_2k_2^2 & B_2k_{n2}^2 & -i\hat{B}k_3 & -\hat{B}k_{n3} & 0 & 0 \\ i\hat{B}k_1 & \hat{B}k_{n1} & 0 & 0 & -B_3k_3^2 + i\hat{B}k_3 & B_3k_{n3}^2 + \hat{B}k_{n3} & B_4k_4^2 & -B_4k_{n4}^2 \\ 1 & 1 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 1 & 1 \end{bmatrix} X \begin{bmatrix} T_1 \\ T_{n1} \\ T_2 \\ T_{n2} \\ T_3 \\ T_{n3} \\ T_4 \\ T_{n4} \end{bmatrix} = \begin{bmatrix} ik_1 \\ 0 \\ -B_1k_1 - i\hat{B}k_1 \\ i\hat{B}k \\ -1 \\ 0 \\ 0 \\ 0 \end{bmatrix} \quad [5]$$

Conclusions

To optimize the sound isolation of partition assemblies having fire stops, it is critical to select the correct fire stop material. This is especially true when high degrees of sound isolation are required. The structural decoupling (hence superior sound isolation) gained by using a double stud wall may be completely defeated if a very stiff fire stop is used. The only way to maximize the airborne transmission loss of a partition is to use a fire stop whose effective bending stiffness is very much lower than the bending stiffness of the surfaces to which it is connected. Thus, of all the materials listed in the NBCC, the 0.38 mm sheet steel is most likely to satisfy this criterion and offer the greatest sound isolation. The span of the fire stop should be maximized where possible.

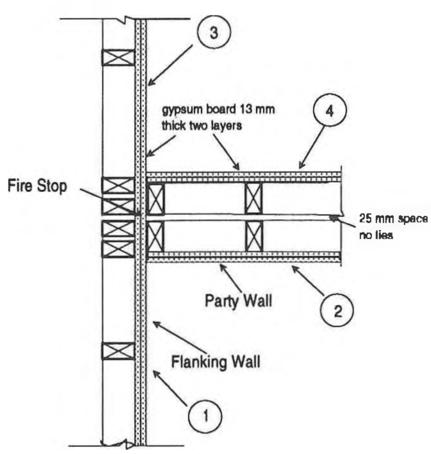

Figure 1: Plan section through the joint.

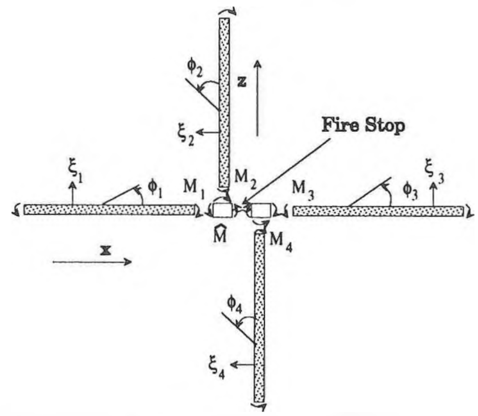

Figure 2: Mechanical representation of the joint.

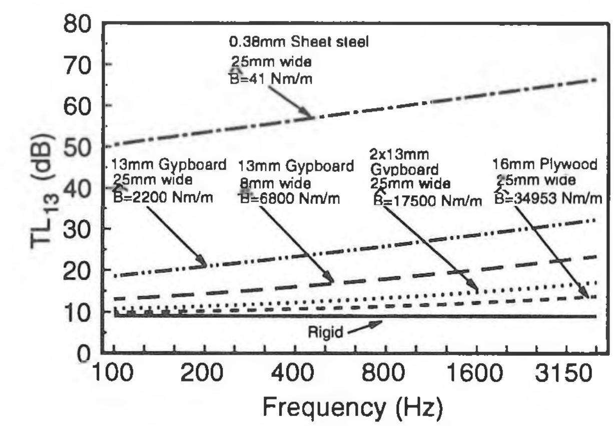

Figure 3: Predicted normal incidence velocity transmission loss for the paths 13 and 14 obtained from Equation 5. Effective bending stiffnesses of the fire stops are given in N-m/m.

[1] Flanking Transmission Caused by Fire Stops in Wood Frame Constructions, Proceedings Issue of the 1994 Canadian Acoustical Association Conference, Ottawa, 1994.

PRACTICAL ASPECTS OF OPERABLE WALL SOUND ISOLATION

A.D. Lightstone, Ph.D., P.Eng.
Valcoustics Canada Ltd., 30 Wertheim Court, Unit 25
Richmond Hill, Ontario L4B 1B9 (905)764-5223

INTRODUCTION

Operable walls are typically integral parts of school, major meeting room, conference/convention centre, and hotel facilities. Where the purpose is for simultaneous uses on opposite sides, high sound isolation, often difficult to achieve, is required.

SPECIFICATIONS

Great care is required interpreting manufacturers' specifications. Typically, Sound Transmission Class (STC) ratings according to the laboratory test procedure ASTM E90 are available. However, this may be for the panels alone, not for an actual operable wall, including top, bottom, vertical and edge joints. To be certain, reference must be made to the original *independent* test report, not the sales brochure. Clearly, significantly higher STC ratings can be expected for the panel alone, without all of the joints or operating mechanisms, and this type of specification can be totally misleading.

A test and certification procedure exists through the Operable Partition Manufacturers Association Section/National School Supply and Equipment Association (OPMA/NSSEA). This calls for a fully operable installation nominally 14 ft wide by 9 ft high, including head, jamb, etc., as normally used in the field, installed in the laboratory test opening, for testing according to ASTM E90. OPMA/NSSEA will certify supervised tests in an approved laboratory. There are several problems with this approach; not all manufacturers are members; few partitions have certified tests; the test partition size is much smaller than is typical in actual use. Panel heights can range from two to three times taller and many times longer, exacerbating any difficulties with field installation aspects that affect sound isolation, such as construction tolerances.

Some manufacturers also report independent field tests of operable installations. These are in the form of field transmission loss (FTL) or noise reduction (NR), and Field STC (FSTC) or Noise Insulation Class (NIC). FSTC and NIC are single number indices computed according to ASTM 413 from the spectrum of FTL or NR, as is STC. Field performance ratings are usually best for specification use, because of a more realistic indication of results to be expected. Generally, the FSTC/NIC ratings are about 10 points lower than the laboratory performance. The practical upper limit for well installed, readily available, operable walls is about NIC 42 to 44; while lab STC ratings as high as 58 may be reported. Field results are poor to marginally acceptable for many situations, particularly when sound systems are used. For many uses, this level of sound isolation is inadequate.

PARTITION SYSTEMS

There are many different models of operable walls available, with different operational and installation implications that are relevant to associated sound isolation details. Panels may be individual, hinged in pairs or more, or in a train, all hinged together. Individual panels can negotiate more complex turns and with track switches can be used in different spaces or along different boundaries. When open, panels may nest in set-back pockets out of the space, or simply against a wall in the space.

SOUND ISOLATION FACTORS

There are many relevant factors; some directly related to the partition and its hardware; some related to the base building.

Partition System

Clearly the partition construction itself is the starting point. Various types, with different thicknesses and weights incorporating various materials such as steel, particle board, and gypsum board are available. Typical weights range from 6 to 12 lbs/ft^2. There is a height limit associated with each type, with more substantial construction required for tall panels to avoid unacceptable twisting or other deformations. The vertical joints between panels must nest and seal tightly. The better systems have fairly sophisticated interlocking shapes. Most operable walls are suspended, requiring a track and roller system to carry the often substantial weight. A proper sound seal across the top offers a design challenge.

In some cases the top can be contained within a bulkhead extending below the ceiling, to achieve a sound seal. However, in most cases, the partition must be below the ceiling, (with the support track flush mounted) to allow turning the panels. In these cases, either movable mechanical seals, or fixed flexible sweep seals are used to close the clearance gap between the panels and the ceiling. These clearance gaps are typically about 25 mm and ensure that the partition can be operated. The mechanical seals are superior acoustically because when activated on panel placement, a positive force is created against the track; but they are more complex and costly. The bottom seal situation is similar to the top except that a larger closure force can be used.

Alternative techniques are used to seal the end vertical panels, termed the jamb, including hinged closure panels, and (superior) expanding jamb panels which are analogous to the top and bottom mechanical seals.

Base Building

There are several potential flanking paths that must be addressed in design. Usually, a bulkhead is needed above the track to close off the ceiling space. This precludes a common return air plenum in the ceiling spaces. Separate ducting or silenced transfer ducts must be used. The bulkhead system must allow access to the track system for initial adjustment and for service. In gymnasia, rehearsal halls, or computer rooms there may be floor cavities that should be closed off (along the partition line). Where carpet is the floor finish, bottom seals are compromised. Solid thresholds are preferred.

Potentially, the most significant base building factors in achieving high sound isolation from operable partition relates to construction tolerances. The jambs must be plumb and true (±3.2 mm/ every 3 m). The structure carrying the head track (and the track itself) must be very rigid and designed to include the partition load, so that non-cumulative deflection does not exceed 3.2 mm every 3.7 m. The partition track would normally be installed with the structure, early in construction. As the building progresses, the loading will increase and so will deflections. With concrete construction, the deflections may continue over many months. Thus, the track system must have ready means to adjust its level before and after the partition is hung (with bulkhead access not compromising sound isolation). In some cases, the structural engineer can use reverse camber beams and trusses so that when loaded, they take a straight horizontal shape across the bottom.

Similarly, the floor must have adequate flatness and be smooth (±1.6 mm from flat). To the extent that base building tolerances are exceeded, the partition may not be able to seal properly.

CONCLUSIONS

Design/use decisions should not be based on manufacturer's laboratory STC ratings. Rely only on field test results from independent authorities. Aside from treatment of potential flanking paths, control of construction tolerances for floor flatness, jamb plumbness, head track level and structural deflections are important for proper operation and sealing. Achieving greater than about NIC 42 with readily available systems in practice is not possible (except by use of two partitions separated by a cavity - usually not done because of cost and space.)

Assessment of Errors in Sound Pressure Measurement in a Large Anechoic Chamber

S.E. Keith, M.G. Davidson, S.H.P. Bly
Health Canada, Radiation Protection Bureau, 775 Brookfield Rd., Ottawa, Ontario, K1A 1C1

Introduction
A large acoustical anechoic chamber has recently been put into operation at the Canadian Radiation Protection Bureau. The walls, floor and ceiling of the anechoic chamber are lined with flat-tipped fibreglass wedges designed for a cut-off frequency of 50 Hz. The interior (wedge tip to wedge tip) is 13 m long, 9 m wide and 8 m high.

In order to create well-defined noise exposures and measurements of sound power and sound pressure in the chamber, its free-field performance must be quantified. This provides baseline estimates for errors in sound pressure measurements and a technique for error estimation as measurement configurations change. This paper describes characterization of the free field performance from 50 to 5000 Hz along the chamber axis. Implications for measurement uncertainties in the anechoic chamber are discussed.

Method and Apparatus
The technique used was similar to that recommended in ISO[1] and ANSI[2] standards and used to characterize other large anechoic chambers[3, 4].

Free-field performance was characterized by measured deviations from the $1/r$ dependence of pressure, p, on distance, r, for a point source in a free field (inverse square law). This can be expressed as:

$$1/p = r/A \qquad (1)$$

where A is proportional to the source strength.

The deviation, $\epsilon(r)$, from the inverse square law was obtained from:

$$\epsilon(r) = 20\log\frac{p_{meas}(r)}{p_{fit}(r)} \qquad (2)$$

where p_{meas} was the measured pressure at position r. The quantity p_{fit}, at any value of r, was the pressure given by a linear least squares fit to equation (1) of the pressures measured over a limited range of r values.

The data range to be fitted was arrived at as a compromise between several competing criteria. Ideally, to reduce the effects of echoes, the least squares fit should be determined from data measured as close to the source as possible. However, the measurement also has to be far enough away that the field is omnidirectional within ±1 dB[1, 2]. This limit on source directivity means that estimation of the maximum measurement error (ϵ_{max}) can be determined experimentally within ±0.2 dB, for ϵ_{max} values less than 2 dB[3]. In addition, the measurement has to be far enough away that, to a good approximation, the free-field pressure field due to the source varies with 1/r. This was diagnosed by measurements of sound source directivity as a function of distance as described below. Finally the range must be large enough to sufficiently reduce statistical uncertainties in the linear least squares fit.

Generation of the acoustic pressures to be measured was done with two novel sound sources; a low frequency dodecahedral array of speakers for 50 to 500 Hz[5] and a higher frequency piezoceramic sphere for 500 to 5000 Hz. Both sources were symmetric on three axes, about a reference point. The low frequency source was constructed from twelve, 6 inch, 100 watt loudspeakers set in the faces of a dodecahedron with an average spherical radius of 19 cm. The higher frequency source was a lead zirconate sphere with a 96.52 mm O.D. and a 4.92 mm wall thickness (Channel Industries, Santa Barbara, CA).

Each sound source was supplied with a simultaneous mix of 11 computer generated sinusoids at 1/3 octave band centre frequencies spanning the range appropriate to each source. For the low frequency measurements, the signal was supplied from a Nagra IV-SJ tape recorder through a Brüel & Kjær (B&K) 2706 power amplifier. The driving signal for the higher frequency source was output by computer through a National Instruments AT-DSP2200 digital signal processor board and amplified using a Yamaha PC 5002 M 1000 W power amplifier.

For directivity measurements, the low frequency source was rotated by mounting it on a tripod centred on a B&K 3922 turntable. The higher frequency source was rotated by supporting it in the end of a nylon stocking, hung 2 m below the spindle of a B&K 3923 rotating microphone boom. The sources rotated about their geometric centres, and measurements were taken every 4 degrees. Repeated revolutions gave a total averaging time of 4 seconds per 4 degree sector. Nine directivity measurements were made at distances between 31.5 and 200 cm from the centre of each source.

Deviations from the inverse square law were obtained with the sources hung from the ceiling and centred vertically in the chamber. Measurements were made along a horizontal line parallel to the long axis of the chamber. The mid point of this measurement traverse was near the centre of the chamber. Low frequency measurements were taken from 50 cm to 560 cm in 2 cm to 10 cm steps using 64 second linear time averaging. Measurements of the higher frequency source were averaged for 8 seconds at each position. At frequencies above 2500 Hz the microphone was positioned from 50 to 530 cm in 1 cm steps. For frequencies from 500 Hz to 2500 Hz, the microphone was positioned from 50 to 545 cm in 5 cm steps.

All measurements of the dodecahedron were made using a B&K 3545 intensity probe, with 12 mm 4181 microphones and a 50 mm spacer. Microphone calibrations were made before and after measurements using a B&K 3541 system (with pistonphone). For the piezoceramic sphere, all directivity measurements and deviation measurements above 2500 Hz were made with a B&K 4165 1/2" microphone, attached to a nosecone, and powered by a B&K 2807 microphone power supply. A B&K 4228 pistonphone was used for microphone calibration for this latter system. To obtain an adequate signal to noise ratio at frequencies from 500 Hz to 2500 Hz with the piezoceramic sphere, deviation measurements were made with a 1" diameter B&K 4179 low noise microphone with nose cone. All pressure measurements were made with a signal to noise ratio of at least 30 dB.

All positioning of the measuring microphone was controlled using a B&K 9654 robot. A B&K type 2133 1/3 octave band frequency analyzer was used for data analysis.

For both directivity and deviation measurements, all ancillary structures within the anechoic chamber that could act as acoustical reflectors, were carefully wrapped with fibreglass batts.

Results and Conclusions

The least squares fit to equation (1) was made over the range r=0.5 to 1 m, where r was taken as the separation between the microphone and the source geometric centre. This was justified by the spatial symmetry of the sources described above.

Measurements suggested that the minimum source receiver separation for the least squares fit should be 50 cm. At distances greater than 50 cm, the change in directivity with distance was 0.1 to 0.4 dB for the dodecahedron and 0.2 to 0.5 dB for the piezoceramic sphere. In most frequency bands, the inverse square law was validated to within ±0.4 dB for the sources used in this work. Furthermore, at distances greater than 50 cm, the maximum deviation from omnidirectionality was less than 0.8 dB.

The results shown in Fig. 1 are of maximum absolute deviations versus frequency for four source-receiver separation distances up to 1, 2, 4 and 5.5 m. In the frequency range from 100 to 5000 Hz, the maximum deviations from free field ranged from 0.4 dB at 100 Hz to 1.0 dB at 5 kHz. At 50 Hz the maximum deviation rose to 3.6 dB. Below 100 Hz, the source receiver separation must be reduced to about 2 metres in order to reduce errors to 1 dB.

The maximum deviations described above, gave a "worst case" error estimate for pressure measurements in the anechoic chamber. These errors could only occur for a source with strong tonal components. The maximum deviation, or error (ϵ_{max}) can be calculated by the summation of pressure amplitudes of direct and reflected sound[6]:

$$\epsilon_{max} = \left| 20\log\left(1 \pm Rr\sum_{i=1}^{6}\frac{1}{d_i}\right)\right| \qquad (3)$$

where R is the chamber wall reflection coefficient (<0.1 above the cutoff frequency for an anechoic chamber), r is the distance between source acoustic centre and microphone, d_i is the distance travelled from source to microphone by a wave reflected from the ith wall. The deviations presented here were consistent with the chamber being anechoic down to 50 Hz. Measured deviations only reach the deviations calculated using equation (3) (with R=0.1) at the chamber cutoff frequency of 50 Hz.

The error estimate in equation (3) is conservative for spatially averaged measurements, a broadband source using a large analysis bandwidth (i.e., A-weighted totals), or a large device containing multiple incoherent sources. In these cases the cross terms in equation (3) tend to cancel and in the limit the deviations are reduced to an energy type summation of the direct and reflected sound.

$$\epsilon_{max} = 10\log\left(1 + R^2 r^2\sum_{i=1}^{6}\frac{1}{d_i^2}\right) \qquad (4)$$

The difference between the two measurement situations is seen, for example, when the deviation from equation (3) is 1 dB. Then the estimated error in equation (4) will be in the range 0.02 to 0.1 dB (the latter value is associated with a source or receiver position close to a single wall).

For an omnidirectional source, initial estimates of expected errors can be made using the data from figure 1. For example, using equation (3), R can be estimated as 0.1 at 50 Hz, and ranges from 0.02 to 0.06 above 100 Hz. Substituting values for d_i's in equations (3) and (4) can give upper and lower bounds on pressure measurement errors in any position in the chamber. If the source is not omnidirectional, the d_i's in equations (3) and (4) must be weighted to obtain an error estimate.

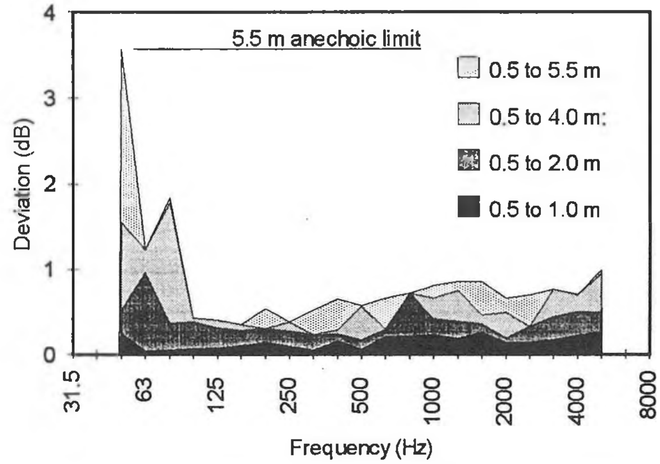

Figure 1: Measured maximum deviations from inverse square law for four source-receiver separation distances.

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PROPRIÉTÉS ABSORBANTES DE PANNEAUX À DOUBLE MEMBRANE APPLICATION AU NOUVEAU TOIT DU STADE OLYMPIQUE

MIGNERON, Jean-Gabriel, *Laboratoire d'acoustique, Faculté d'aménagement et d'architecture, Université Laval, 1 Côte de la Fabrique, Québec, Qué., G1K 7P4*

ABSTRACT:
In order to develop a special absorbent material to control the reflections on the concave surface of the new roof of the Olympic Stadium of Montréal, a study has been conducted on the acoustical properties of a double layer absorbent panels, with a special interest concerning the low frequency comportment of this composite material and the effects of the thickness and the density of the fibrous material, the distance between the two parallel panels, the external covering and the finishing paint of the exposed panels. Different samples have been tested in a reverberation chamber, both in vertical and horizontal position, in relation with the opening between two juxtaposed panels and the distance between the acoustical treatment and the new steel roof. The conception of the acoustical material takes into account the predicted reverberation time, the durability of the panels, the mechanical and fire resistance and, finally, the cost of the total treatment.

Suite aux différentes discussions avec l'équipe des architectes de la nouvelle toiture du Stade Olympique de Montréal, une décision a été prise pour placer les panneaux acoustiques horizontalement, suivant la courbure de la charpente du nouveau toit suspendu, soit à 1.3 mètre du platelage d'acier. Le type de traitement acoustique recherché pour la nouvelle toiture devait offrir une bonne absorption des basses fréquences, du fait des très longs temps de réverbération en présence (maximum de 19.4 sec. vers 315Hz), du comportement vibratoire particulier de la toile de kevlar actuelle (qui dissipe en partie les basses fréquences) et puisque, dans une très grande enceinte, les hautes fréquences sont atténuées par l'absorption moléculaire de l'air. Un traitement acoustique à double membrane permettait d'obtenir l'absorption souhaitée, tout en respectant les autres contraintes. À partir de ces paramètres de base, des panneaux ont été optimisés en fonction des détails de construction, du poids et de la durabilité du traitement et bien sûr de l'absorption acoustique requise. La présentation fait état des tests acoustiques effectués sur différents échantillons et des principales considérations examinées lors de la conception des échantillons.

ÉCHANTILLONS TESTÉS EN LABORATOIRE

Des tests ont été effectués en laboratoire sur sept échantillons, afin d'optimiser l'absorption dans les basses fréquences des panneaux à double membrane. Les paramètres étudiés ont été la densité de la laine minérale employée, son épaisseur, l'espace d'air requis entre les deux laines, le revêtement de finition nécessaire pour assurer une bonne durabilité de l'ensemble du traitement, ainsi que les dispositifs d'accrochage à la charpente d'acier. Le type de traitement retenu est constitué de deux panneaux doubles de laines minérales de 1.22x1.22m espacés entre eux par un cadre métallique qui constitue la structure même du panneau.

Les sept échantillons testés en laboratoire ont été constitués de six panneaux (8.92 m^2) et sont décrits dans le tableau nº 1. Pour les échantillons nº1 et 2, les cadres des panneaux ont été fabriqués avec des colombages d'aluminium de 50 mm de largeur. Les cadres de tous les autres échantillons ont été fabriqués avec des montants métalliques de 92 mm d'âme et de jauge 26 (0.45mm d'épaisseur).

Nº de l'échantillon	Densité de la laine minérale (kg/m^3)	Épaisseur de la laine minérale (mm)	Espace d'air (mm)	Revêtement employé	Position horizontale ou verticale	Espace entre les panneaux
1	48.1	2x50	50	néoprène	H	-
2	48.1	2x50	50	néoprène	H	200mm
3	39.7	2x25	92	néoprène	H	-
4	48.1	2x50	92	néoprène	H	-
5	48.1	2x50	92	néoprène avec latex	H	-
6	48.1	2x50	92	néoprène avec latex	V	-
7	48.1	2x50	92	néoprène et peinture granitée	H	-

Tableau Nº1: *Description des différents échantillons de panneaux testés en laboratoire.*

Les panneaux ont été fixés aux montants métalliques mécaniquement et par collage. Le dernier type de cadre offre une meilleure rigidité que les colombages d'aluminium, mais il sera préférable d'ajouter des équerres de renfort dans les angles (en plus de la quincaillerie de suspension et de clips de sécurité). Enfin, les matériaux ayant servi aux tests d'absorption acoustique en laboratoire (laine minérale, cadre, quincaillerie) ont été pesés, le critère de poids étant très important pour la nouvelle toiture (limite de 5.9 kg/m^2).

Dans la chambre réverbérante de 200m^3, tous les panneaux de laine minérale ont été installés horizontalement à 1.3 mètre du plancher de la chambre réverbérante (afin de permettre les réflexions arrières), sauf l'échantillon nº6 (composé de deux laines de 50mm espacées de 92mm entre elles et peintes au latex) qui a été installé verticalement, directement sur le plancher.

RÉSULTATS DES TESTS D'ABSORPTION

Les coefficients d'absorption obtenus pour les différents échantillons de panneaux placés horizontalement sont reproduits sur le graphique nº1. Comme on peut le voir sur ce graphique , l'échantillon composé de deux laines minérales de 25mm d'épaisseur, recouvertes de néoprène et de plus faible densité (39.7kg/m^3 au lieu de 48.1kg/m^3), bien qu'espacées de 92mm, est moins performant dans les basses fréquences. Ensuite on retrouve les échantillons composés de deux laines minérales de 50mm d'épaisseur, recouvertes de néoprène et espacées de 50mm, avec ou sans espacement latéraux de 200mm entre les six panneaux.

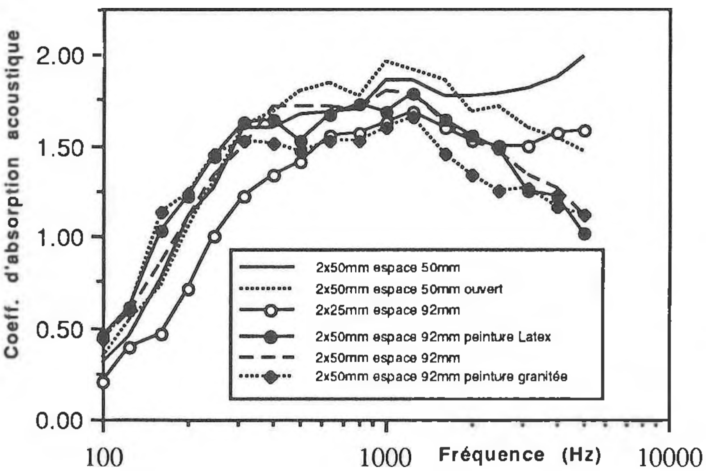

Figure N°1: Absorption des différents échantillons horizontaux.

On remarque que le fait d'espacer ou non les panneaux les uns des autres, n'a pas une grande influence sur les coefficients d'absorption des basses fréquences. Le gain d'absorption se trouve plutôt dans les fréquences moyennes, lorsqu'on permet aux ondes acoustiques de pénétrer vers la seconde face des panneaux.

Par la suite, trois échantillons composés de deux laines minérales recouvertes de néoprène de 50mm d'épaisseur, espacées de 92mm, ont été testés soit avec le néoprène seul, soit ensuite peinturés au latex ou à la peinture granitée. On remarque que ce sont les échantillons peints au latex ou à la peinture granitée qui permettent une plus grande absorption des basses fréquences. De plus, la peinture au latex offre une meilleure absorption des hautes fréquences que la peinture granitée, cette dernière est en effet trop épaisse et affecte la porosité des panneaux absorbants.

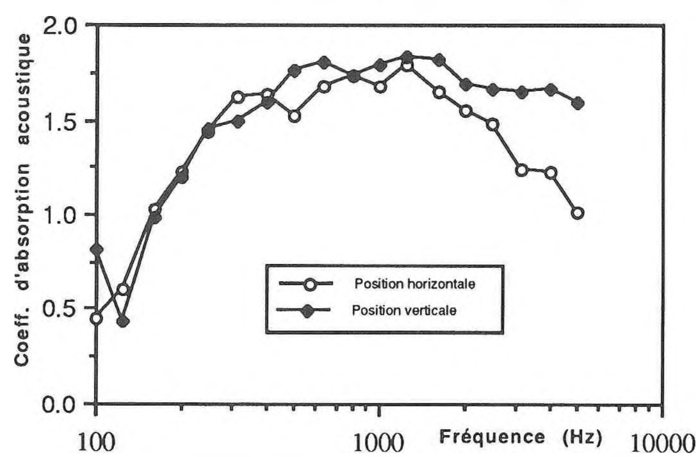

Figure N°2: Comparaison entre les montages horizontal et vertical.

Par ailleurs, le graphique n°2 présente une comparaison entre le panneau constitué de deux laines minérales de 50mm d'épaisseur, espacées de 92mm, et peint au latex, installé horizontalement et ensuite verticalement.

On remarque, dans la position verticale, une baisse d'absorption dans la bande de fréquence de 125 Hz, baisse qui ne se retrouve pas dans la position horizontale. Ce comportement peut s'expliquer par le fait que l'effet de membrane est affecté ou non par le poids du matériau. Par ailleurs, l'échantillon installé verticalement permet une meilleure absorption des hautes fréquences, étant donné que dans cette position, les deux surfaces du panneau sont plus exposées aux ondes acoustiques directes qu'en position horizontale (une des faces de l'échantillon n'est exposée qu'à une partie des ondes réfléchies).

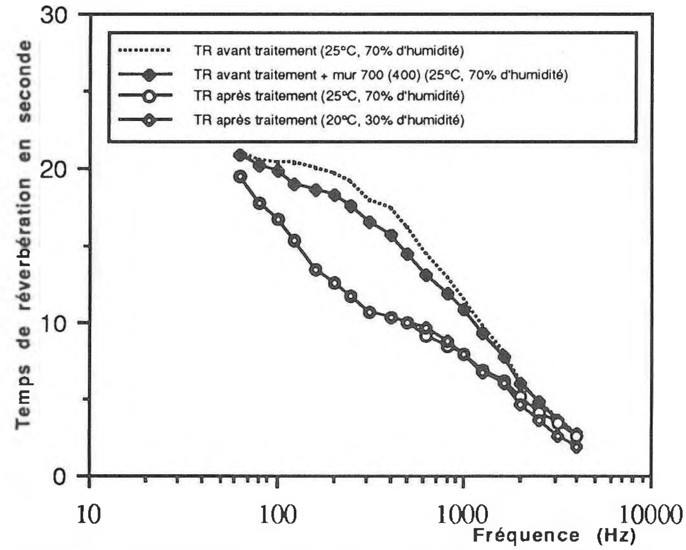

Figure N°3: Temps de réverbération prévus après construction.

CONCLUSION

Les temps de réverbération prévisibles ont été calculés (pour différentes températures et humidités) à partir des coefficients d'absorption obtenus avec l'échantillon de deux laines minérales de 50mm recouvertes de néoprène et peintes, distantes entre elles de 92mm. Le graphique n°3 présente les temps de réverbération escomptés, avant et après traitement acoustique (la première étape prévue concerne le mur arrière du niveau 700). Ainsi, le traitement acoustique de la nouvelle toiture, en plus de contrôler les réflexions, devrait permettre une diminution du temps de réverbération dans l'enceinte du Stade d'environ 2 à 3 sec. dans les très basses fréquences et d'approximativement 5 à 7 sec. dans les basses et moyennes fréquences, comparativement aux temps de réverbération actuels du Stade. Bien que le temps de réverbération moyen pourrait être sensiblement diminué, l'enceinte devrait conserver une atmosphère satisfaisante pour les effets de foule, avec une réverbération moyenne aux alentours de 10 secondes. Il faut également mentionner que l'intelligibilité de la sonorisation devrait se trouver sensiblement améliorée dans tous les gradins.

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Occupational Noise Exposure in the High School Music Practice Room

Bill Gastmeier, HGC Engineering, 2000 Argentia Rd, Suite 203, Plaza 1, Mississauga, Ontario, L5N 1P7

David Pernu, Noront Audiometric Testing Inc., P.O. Box 844, Sudbury, Ontario, P3E 4S1

Marshall Chasin, Director of Research, Canadian Hearing Society, 271 Spadina Rd., Toronto, Ontario, M5R 2V3

Introduction

Occupational noise is most often considered potentially hazardous in industrial workplaces or on construction sites. Another workplace where high sound levels commonly exist is the music practice room of an educational facility.

Measurements were performed of the sound levels produced during two instruction periods in a high school music room. Both personal dosimetry and engineering measurements were performed. Through analysis, the typical daily noise exposure of the instructor was determined and was found to exceed the guidelines provided in the Occupational Health and Safety Act.

The implications of this excess in terms of hearing risk criteria and architectural acoustic design are discussed.

Sound Level Measurements

Sound levels were measured during the 09:45 and 11:40 instruction periods approximately 2 m from the instructor at ear level. The musical instruments included electronically amplified bass and guitar, a full drum set, and various wind and brass instruments including saxophones, trombones and trumpets.

Figures 1 & 2 show the results in terms of the average sound level in each 10 second interval. Warm up and active rehearsal occupied from 40 to 50 minutes of each period during which the sound levels varied from as low as 60 dBA during lulls to as high as **103 dBA** during fortissimo brass passages. The one hour average sound level was 93.4 dBA and 91.1 dBA for the two classes respectively. The brass instruments were observed to be the major contributors to the measured levels and were pointed directly at the instructor.

Calculation of Exposure

The following schedule was provided by the instructor for the measurement day, which is understood to be a typical day of instruction, occurring 5 days a week.

Time	Class	# of Students
08:50 - 09:40	Grade 9	19
09:45 - 10:38	Senior Class	21
10:38 - 11:45	Spare	0
11:40 - 11:50	Recording	3
11:50 - 12:25	Band Practice	20
12:35 - 13:35	Grade 9	16
13:35 - 15:20	Spares	0
15:30 - 17:30	Band Practice	20

This schedule indicates that activities similar to those measured occur for 5.47 hours per day. Calculations indicate that the instructor is exposed to 91 dBA on a daily (8 hour) basis. This estimate is considered to be conservative in that it does not take into account evening activities and was measured 2 m from the instructor. Personal dosimetry verified these results.

Occupational Health and Safety Considerations

The sound levels to which the instructor is exposed exceed existing guidelines (90 dBA - 8 hour Leq) found in the Occupational Health and Safety Act[1] for industrial workplaces. They significantly exceed proposed guidelines (85 dBA).

In an industrial workplace the Act recommends that consideration be given to the reduction of noise levels through any control measures which may be feasible. The use of hearing protective devices is required. Music education facilities are not considered industrial workplaces. **We are concerned that significant (but preventable) hearing damage is occurring in this group of dedicated professionals.**

Damage Risk Criteria

A spectrum of the sound during one of the loudest passages is shown in Figure 3. It can be seen that significant sound is present in the 250 to 4000 Hz Octave bands, characteristic of trumpets and cymbals.

This is of particular concern to music instructors since it is well documented that "a threshold shift occurs most readily at 4000 Hz. and that it is most readily caused by noise energy in the frequency region about one octave lower (1000 to 2000 Hz or so)"[2]. Considering the high frequency nature of the sound, it may be that music teachers are even more at risk than persons exposed to the same dBA levels of noise in industry where the spectrum may be more low frequency in nature. **This is an area where the authors feel further research is required.**

Noise Control Measures

Much can be done in the music rehearsal room to moderate sound exposure. The following points are offered as helpful suggestions in that regard, but should not be interpreted as design criteria to eliminate the problem. Each practice room is unique and each situation must be considered individually.

a) Risers (1 - 1.5 feet in height) can be provided. Trumpets should be elevated so that they do not point at the heads of the musicians in front of them and the risers arranged so that they also do not point directly at the instructor. An offset of at least 30 degrees would be beneficial in this regard.

b) Acoustical wall and ceiling panels, a suspended acoustical ceiling and other treatments can be appropriate to control reverberation and overall noise levels. A good implementation should be effective through the entire frequency range (both bass and treble frequencies) and requires careful design. The thickness, amount and precise location of the treatment will depend on the room volume, layout and activities. In a retrofit situation, acoustical measurements are very helpful. The services of a consulting acoustical engineer may be useful in this regard.

c) It is also appropriate to incorporate some changeable absorption within the space. Pull down baffles or curtains can be temporarily placed over blackboards to attenuate reflections during rehearsals or to assist in recording. Similar treatments can be useful on walls near the drum kit.

d) It may be appropriate to use carpet for additional high frequency absorbtion near the drum kit and brass instruments. It does not pose a sanitary problem as any debris from wind instruments is primarily the result of condensation. Generally, however, musicians will hear themselves and others more clearly if they are not standing on carpeted areas.

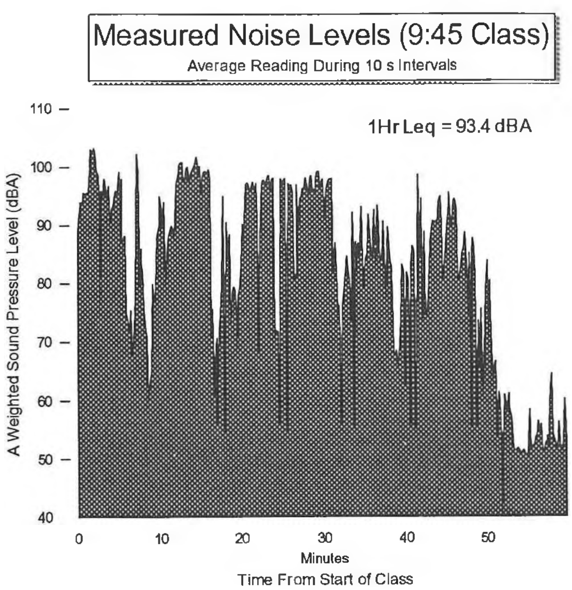

Figure 1

e) Hearing protectors are available with a uniform attenuation at all frequencies for a more natural sound. Instructors should strongly consider the use of such devices since the risk of damage is real.

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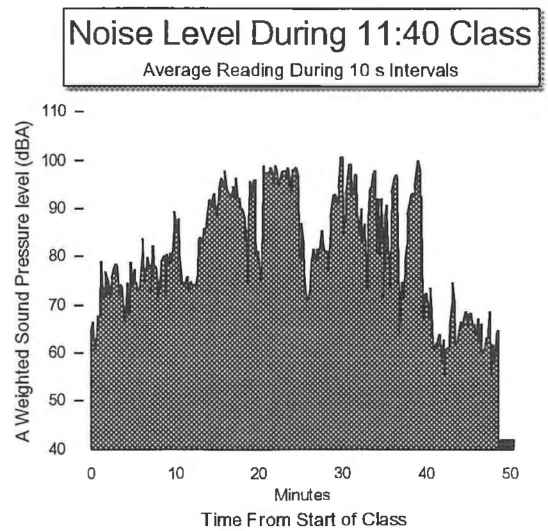

Figure 2

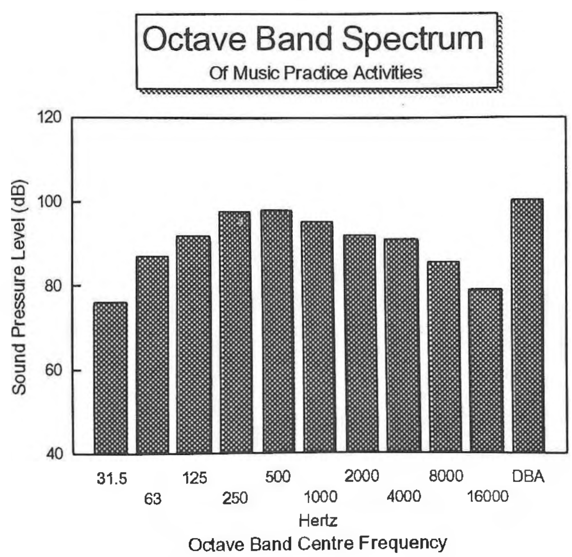

Figure 3

Wave versus Ray-Based Room Noise Models

C. Tang and K.R. Fyfe
Department of Mechanical Engineering,
University of Alberta, Edmonton, Alberta, Canada

INTRODUCTION

Traditionally, room acoustics prediction models have been ray-based (geometrical) and thus ignore phase. Wave-based models on the other hand, make use of phase information to properly determine interference phenomena. This paper outlines some preliminary studies performed on two-dimensional geometries to compare these two classes of methods.

PROCEDURE

In order to compare predictions garnered from wave-based methods with those from geometrical techniques, a procedure was devised to compare harmonic signals with sources that emit energies within a limited frequency band. The reason for this is that mono-frequency responses are highly dominated by modal effects. These effects cannot be predicted by geometrical approaches due to its inherent assumption of a diffuse sound field. Nevertheless, comparisons can still be made if banded signals are used. By selecting a sufficiently wide frequency spectrum, modal effects caused by any single frequency are damped out when all the responses within the bandwidth are superimposed. This can be done by assuming that a banded source behaves like a combination of N of elemental point sources, each of which radiates at different frequencies. With this in mind, the total bandwidth can be subdivided into small intervals, within which each sound power remain relatively constant. The total power, W, radiated by the banded source then equals the summation of all the elemental powers, W_i, at discrete frequency intervals. The total pressure response can then be found by treating these elemental sources as un-correlated noise signals and adding their mean square values. Thus,

$$P_t^2 = \sum_{i=1}^{N} P_i^2$$

from which the sound pressure level (SPL) can be found. The sound pressure level is one of the most fundamental quantities easily produced by either class of methods and is directly measurable by standard acoustical equipment.

This study will make use of a two-dimensional empty rectangular cavity with a single point source. A 7m x 4m room is modelled with a point source located at (1.0m, 1.0m). Uniform damping on all four walls will be used for the subsequent analysis of the room. Air absorption is assumed small and therefore will be neglected.

In a wave-oriented approach, most boundary interactions are described by a complex quantity known as the acoustic impedance, **Z**, or admittance, β, of the surface. A locally reacting surface is assumed. The specific admittance is defined by

$$\beta = \frac{\rho c}{Z} = \xi - i\sigma$$

where ρc is the characteristic impedance of air. In geometrical acoustics, phase relationships between acoustic waves and the boundary are neglected. Instead, only the rate of sound energy being absorbed by the boundary surface is modelled. The rate of absorption is usually given in terms of a constant known as the Sabine absorption coefficient, α. This absorption coefficient can be related to the acoustic admittance as follows [1],

$$\alpha = 8\xi\left[1 + \frac{\xi^2 - \sigma^2}{\sigma}\tan^{-1}\left(\frac{\sigma}{\sigma^2 + \xi^2 + \xi}\right) - \xi \ln\left(\frac{(\xi + 1)^2 + \sigma^2}{\xi^2 + \sigma^2}\right)\right]$$

Iterative calculations using an absorption coefficient, α, equal to 0.1 yields a specific admittance β of 0.0141. It should be noted that this is not a unique solution to the equation.

RESULTS

Using the defined room, source and wall properties, the acoustic response over the area of this room was evaluated by the finite element method (FEM) [2]. The results at 50 Hz and 1000 Hz are shown in Figures 1 and 2 respectively.

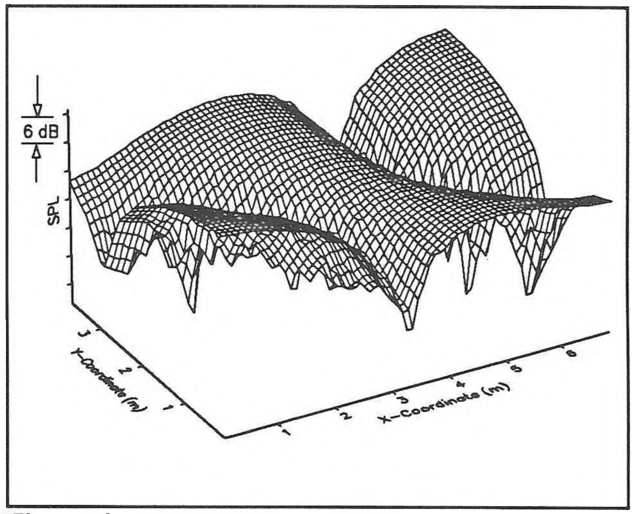

Figure 2- SPL at 50 Hz - 10% Absorption

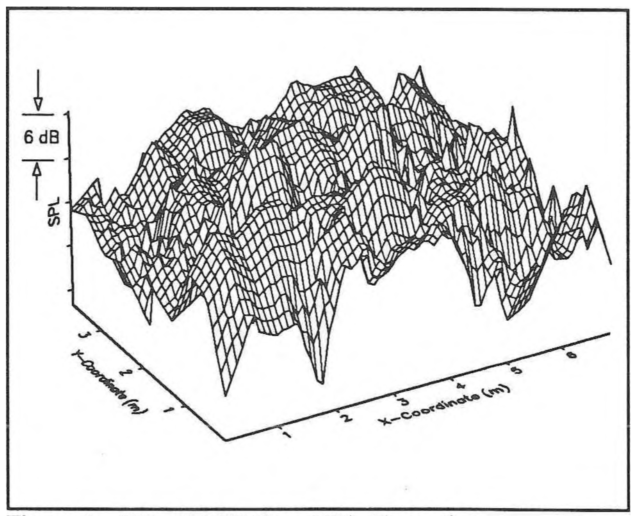

Figure 2 - SPL at 1000 Hz - 10% Absorption

From these figures, it is seen that a low frequency response is very modal dominated while the higher frequency test exhibits a more uniform and almost *random* response. This can be explained by knowing something about the *modal density* in the room. At low frequencies, where there are few room modes governing the pressure distribution in the enclosed space, the sound energy propagation is dominated by the mode nearest the excitation frequency. However, as the modal density increases, the effects of each individual room resonance are summed up so that the effects of any single room mode are averaged and reduced. Eventually, there are enough modes present so that the sound energy propagation are nearly uniform in all directions. This is called a diffuse sound field and is the fundamental assumption used by geometrical approaches.

At a low frequency, the SPL fluctuates dramatically with frequency. As the frequency is increased, the SPL remain relatively constant, indicating a more uniform energy level.

Banded signals can be used to further reduce the effect of any single room mode so that direct comparisons can be made with geometrical techniques. The advantage of using banded signals is that individual frequency response are summed and averaged so that the effects of any single room resonance are reduced.

To determine a suitable bandwidth for use in comparison, a frequency response was plotted for a point located at (6.0m, 1.5m) as shown in Figure 3.

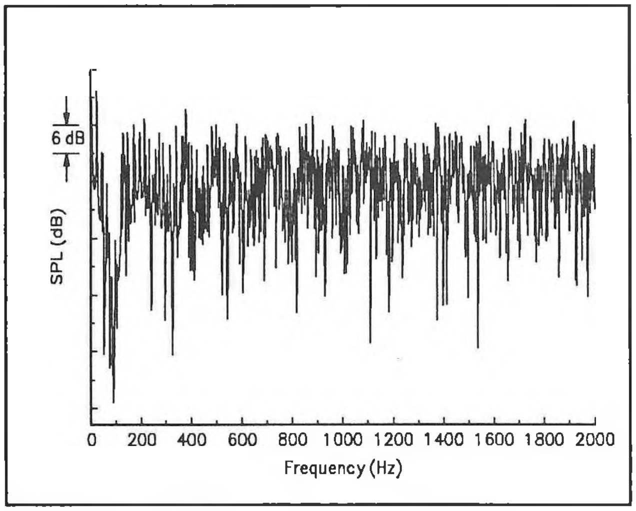

Figure 3 - Frequency Response at a Single Point

A statistical analysis was performed on this curve using the statistical sampling equation,

$$N = \left(\frac{z_{\alpha/2}\,\sigma}{E} \right)^2$$

where N is the number of sampling points required to calculate the sample mean with a 90% chance (α) that the error (E) will not exceed 1.5 dB. By assuming a normal distribution, the minimum number of calculation points required is 85. Using a 56 x 32 finite element mesh and using the 5 node per wavelength requirement, this limits the maximum frequency for accurate modelling to about 700 Hz. Using these parameters, a frequency spectrum from 500 Hz to 680 Hz discretized at a 2 Hz interval, was selected for simulation of the limited band signal.

A comparison between the banded signal generated by the FEM and the ray-tracing method is shown in Figure 4. This plot shows the SPL predictions along Y = 1.0m at ten, thirty and fifty percent boundary absorptions. The overall sound propagation predicted by the FEM shows good agreement with those predicted by the ray-tracing technique. Figure 5 shows the SPL distribution with a 3m high barrier located at x = 3.5m. Again, good agreement is achieved between the two approaches. However, the banded signal simulated by the FEM still obviously contains wave characteristics.

One particular area of interest in comparison is at the boundary surface. In Figures 4 and 5, the FEM consistently predicts a higher SPL than that using the ray-tracing technique. This can be explained by the fact that individual standing waves of a rectangular enclosure can only be excited to its fullest extent by a sound source located in regions where the particular standing wave pattern has a pressure antinode. In order to excite every mode to its fullest extent, the source must be located at the corner of a room. Similarly, a point on the boundary surface will excite more modes to its fullest than any points not on the boundary. This modal effect cannot be accounted for using geometrical methods.

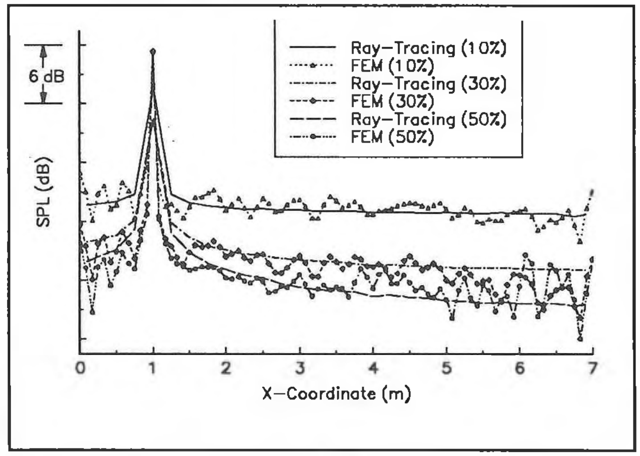

Figure 4 - Response in Room at Different Absorption Levels

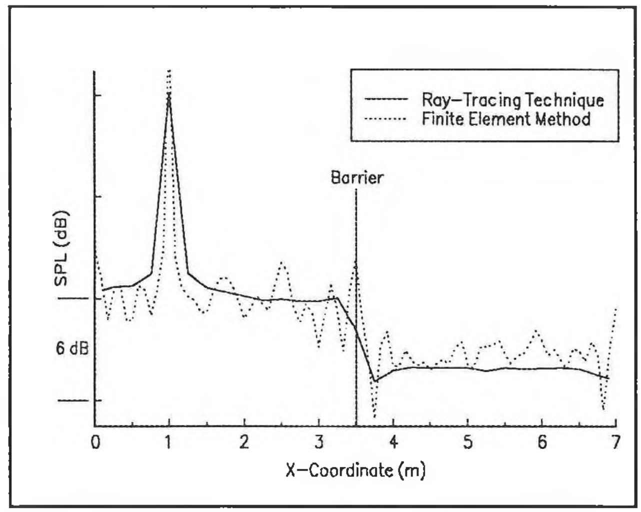

Figure 5 - Sound Levels due to Presence of Barrier

CONCLUSIONS

In general, good agreement between the two methods is observed except at the boundaries. There are a several factors that contribute to deviations between the two approaches. First, the bandwidth of the signal may not be wide enough to eliminate biasing the room resonance within the frequency band chosen. Secondly, the number of calculation points used is insufficient to statistically average out the modal effects of the signal. Finally, the frequency of the banded signal may be too low to exhibit geometrical behaviour.

ACKNOWLEDGEMENTS

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WHEN IS DIFFUSE-FIELD THEORY ACCURATE?

Murray Hodgson

Occupational Hygiene Programme and Department of Mechanical Engineering, University of British Columbia, 2206 East Mall, Vancouver, BC, Canada V6T 1Z3

Diffuse-field theory is used by practitioners to predict sound fields in rooms of every type. Often forgotten is the fact that the theory is based on assumptions which may limit its applicability. If the theoretical assumptions do not hold in the case of a particular room for which predictions are to be done, the predictions may not be accurate.

The objective of this paper is to review what is known about the applicability of diffuse-field theory. This is mainly based on extensive work by Kuttruff ([1] and references therein) and by the author [2, 3, 4, 5, 6], comparing predictions by diffuse-field theory and a ray-tracing model. It will consider two versions of diffuse-field theory - the Eyring and Sabine versions - and the prediction of both sound decay / reverberation time, and steady-state sound pressure level. Note that both diffuse-field theory and ray-tracing are energy-based models which ignore wave effects and thus may be inherently inaccurate at lower frequencies.

I. DIFFUSE FIELDS AND PARAMETERS STUDIED

The discussion will consider the accuracy of diffuse-field theory with respect to the following room-acoustical parameters:

- room shape - as described by the aspect ratio (length:width:height);
- surface absorption - its distribution, and its magnitude as described by the average surface-absorption coefficient;
- surface reflection - as described by the diffuse-reflection coefficient, equal to the proportion of reflected energy which is reflected diffusely [4]. That energy not reflected diffusely is reflected specularly;
- fittings (or volume scatterers) - these are obstacles in the room volume that scatter sound randomly, as described by their volume density.

The basic assumption of diffuse-field theory is that the sound field in the room is diffuse. The sound field is diffuse if the following two conditions apply:

1. At any position in the room the reverberant sound waves are incident from all directions with equal intensity (and random phase relations);
2. The reverberant sound field is the same at every position in the room.

The question that is effectively being asked here is: for what values of the above room-acoustical parameters do the above two conditions hold? Note that it has recently been argued that the first condition is sufficient with respect to sound decay / reverberation time prediction. However, both conditions are relevant to steady-state sound pressure level prediction [6].

The various effects will be discussed qualitatively; space does not permit the use of figures to illustrate each point.

II. ROOM SHAPE

Sound Decay / Reverberation Time - In the case of specular reflection, Eyring prediction is accurate in regularly-shaped (ie quasi-cubic) rooms. As room aspect ratio increases it becomes less accurate. Accuracy increases with diffuse surface reflection and, up to some limit, fitting density.

Steady-State Sound Pressure Level - Eyring prediction is accurate in empty, regularly-shaped rooms with specularly reflecting surfaces. It becomes increasingly inaccurate as the fitting density, room aspect ratio and diffuse surface reflections increase. In particular, levels near a source are increasingly underestimated; those far from sources are increasingly overestimated.

III. SURFACE REFLECTION

Sound Decay / Reverberation Time - Eyring prediction is accurate in regularly-shaped rooms with specularly reflecting surfaces. Its accuracy decreases with increasing aspect ratio. Increasing the diffuse reflection coefficient improves accuracy.

Steady-State Sound Pressure Level - Eyring prediction becomes increasingly inaccurate as the diffuse-reflection coefficient increases from zero (specular reflection). In particular, levels near a source are increasingly underestimated; those far from sources are increasingly overestimated.

IV. FITTING DENSITY

Sound Decay / Reverberation Time - Eyring prediction is accurate only if the fitting density takes an optimum value. This value depends on the room shape and surface absorption distribution. It is low in regularly-shaped rooms with uniformly distributed absorption. It increases with room aspect ratio and the non-uniformity of the absorption. If the fitting density is either lower or higher than the optimum value, the rate of sound decay is lower than that predicted by Eyring;

Steady-State Sound Pressure Level - Eyring prediction becomes increasingly inaccurate as the fitting density increases from zero (empty room).

V. SURFACE-ABSORPTION DISTRIBUTION

Sound Decay / Reverberation Time - In the case of low diffuse-reflection coefficient Eyring prediction is accurate only if the surface absorption is uniformly distributed. Its accuracy decreases with non-uniformity;

Steady-State Sound Pressure Level - Generally speaking, surface-absorption distribution has only a small effect on room steady-state sound pressure levels.

VI. SURFACE-ABSORPTION MAGNITUDE [5]

Sound Decay / Reverberation Time - Eyring prediction can be accurate for any average surface-absorption coefficient. Sabine prediction is only accurate if the average surface absorption coefficient is sufficiently low;

Steady-State Sound Pressure Level - Eyring prediction can be accurate for any average surface-absorption coefficient. Sabine prediction is only accurate if the average surface absorption coefficient is sufficiently low.

VII. SUMMARY

Following is a summary of the conditions under which diffuse-field theory would be expected to be accurate:

Sound Decay / Reverberation Time:	Eyring accurate if	diffuse surface reflection OR optimum fitting density OR specular reflection AND cubic shape AND uniform absorption
Steady-State Sound Pressure Level:	Eyring accurate if	specular reflection AND cubic shape AND uniform absorption AND no fittings (empty)
Sabine cf Eyring:	Eyring accurate Sabine accurate	for any average surface absorption coefficient for low average surface absorption coefficient

VIII. REAL ROOMS

Of fundamental practical interest is the question of how do the above results apply to real rooms. This question is not easy to answer. This is because the applicable values of certain key room-acoustical parameters are not well known for particular rooms. In particular, applicable values for surface-absorption and diffuse-reflection coefficients, and for fitting densities, are not well known.

Here is my personal experience, from having measured sound fields in hundreds of rooms of many types. Note that I am only referring to rooms which consist of a single volume - not coupled spaces. Generally, sound-decay curves are quite linear, and diffuse-field reverberation-time prediction is quite accurate in most real rooms. However, diffuse-field steady-state sound pressure level prediction is seldom accurate in real rooms and can, in fact, be highly inaccurate. These conclusions are consistent with the above conclusions regarding the applicability of diffuse-field theory and the existence in real rooms of many sound diffusing mechanisms (diffusely reflecting surfaces and/or fittings). This is supported by results published elsewhere [4].

It is easy to illustrate how inaccurate diffuse-field theory can be in predicting steady-state sound pressure levels. The figure shows the 1000-Hz sound-propagation curves (the variation with distance from a single omnidirectional point source of the sound pressure level minus the source sound power level) measured in three different rooms - a squash court (9.7 x 6.4 x 5.5 m), an open-plan office (40 x 25 x 2.7 m), and a machine shop (23 x 9 x 4.6 m). Clearly only in the case of the squash court (a good approximation to a reverberation room) does the sound propagation curve level off at large distances as predicted by diffuse-field theory. In all other cases levels decrease monotonically with increasing distance. Eyring theory underestimates levels near the source and overestimates levels far from the source.

IX. CONCLUSIONS

The results of research by Kuttruff and the author, amongst others, have established for what values of certain room-acoustical parameters the sound field in a room would be expected to be diffuse and, thus, predictions by diffuse-field theory accurate. Further research is required to determine exactly how these results apply to real rooms. However, practitioners using diffuse-field theory should be aware that the assumption of a diffuse sound field may seriously limit the accuracy of prediction - particularly of steady-state sound pressure level. Models - such as the method of images and ray tracing - which are accurate in the case of non-diffuse sound fields, are available.

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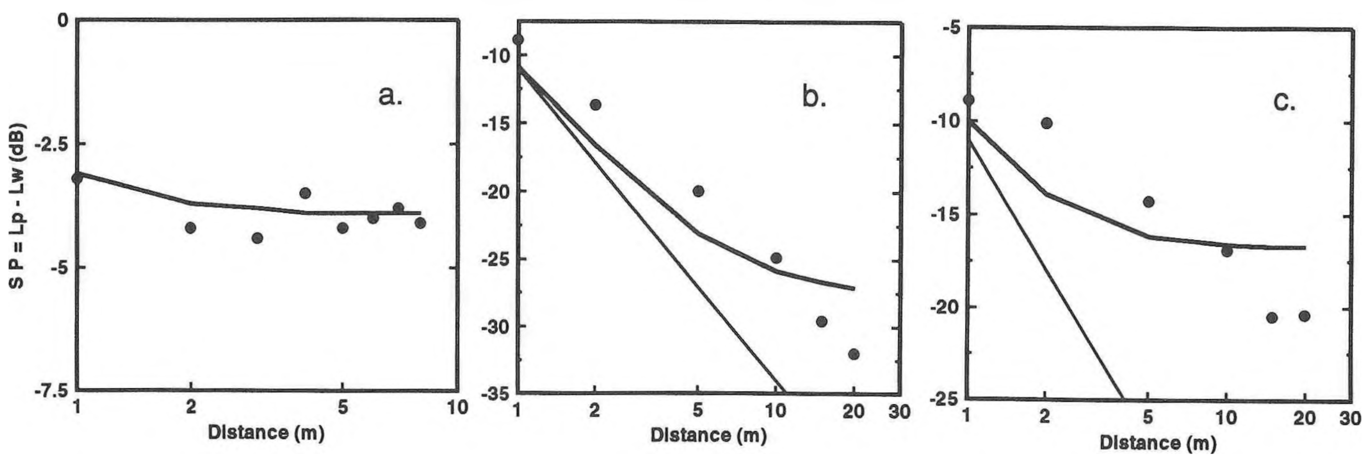
FIGURE

1000-Hz octave-band sound-propagation curves as (•) measured and (▬) predicted by Eyring theory for: a) a squash court (9.7 x 6.4 x 5.5 m); b) an open-plan office (40 x 25 x 2.7 m); c) a machine shop (23 x 9 x 4.6 m). (——) free-field sound-propagation curve.



A REVIEW OF OBJECTIVE DESCRIPTORS FOR SOUND DIFFUSENESS

ABDOU A. and *GUY R. W.*

Centre for Building Studies (CBS),
Concordia University,
Montreal, Quebec, H3G 1M8, CANADA.

INTRODUCTION

Recently, Barron [1] commented that the largest gap in the objective description of concert hall acoustics appears to be the lack of a measure relating to subjective diffuseness or spatial distribution of the reverberant sound. ***The objective of the present study is to review known measurement methods and existing objective descriptors for quantifying sound field diffuseness in an enclosure.*** The *presentation* will address their concepts, merits, shortcomings and potential use. In addition, the concern of where and when "*diffuse*" sound occurs will be explored. Measurements of directional information and their applications for diffuseness quantification employing new objective descriptors proposed and reported by the authors [2,3] will be discussed.

"DIFFUSE" SOUND DEFINITION AND PROPERTIES

A widely accepted definition for sound diffuseness [4,5,6,7] states that a sound field is referred to as "diffuse" when the amplitude of the incident waves are uniformly distributed over all possible directions of incidence i.e. they have equal probability of distribution in all directions and subsequently an equal probability of impinging on the boundary surfaces of the enclosure at any angle. In addition the phases of these arriving waves should be randomly distributed and therefore, their energies can be added. In decaying sound fields these conditions should be fulfilled at each moment of the decay process or over short time intervals compared with the decay process duration.

IMPORTANCE OF "DIFFUSE" SOUND QUANTIFICATION

Sound field diffuseness is considered a crucial condition for the validity of the decay process in an enclosure and subsequently, most of the contemporary room-acoustic indicators are derived on this assumption. In room-acoustics, adequately diffuse sound is desired for acoustical quality. Lack of diffusion may occur due to either simplicity of enclosure geometry or non-uniform distribution of boundary surfaces absorption. Traditional theory for a diffuse space divides the total sound into two components, the direct and reflected sound; the reflected sound is usually subdivided into early and late parts or regimes, the later is then assumed diffuse. Simple diffuse-field theory predicts relatively constant sound pressure levels with increasing source-receiver distance under steady-state conditions and the decay of energy will follow an exponential law. Measurements in existing halls show that the field is unlikely to be diffuse particularly in the late time period. Decay curves are neither exponential nor independent of the receiver position; for large distances from the source however they do approach classical decay theory. Barron [8] developed a revised theory to better explain the variations of sound levels in concert halls, and Hodgson [9] reviewed knowledge about the accuracy and applicability of diffuse-field theory with respect to some acoustical parameters.

Recently, Souldore and Bradley [10] undertook a subjective study on the influence of late arriving energy on spatial impression in a concert hall and found that listener envelopment is produced by late arriving energy as well as early reflections and it is affected by the level and arrival time; surprisingly, since late reverberation is usually considered detrimental to other subjective impressions such as music clarity and definition. The temporal and directional characteristics of the late reverberant reflections required to achieve a certain degree of envelopment however are still unclear.

Since late reflections associated with reverberation affect the subjective judgment of envelopment and diffusion is closely related to reverberation, many questions are raised. For example to what extent should the sound be diffuse and how could this be judged or quantified. Moreover, since diffuse sound conditions are also necessary for many acoustical tests such as absorption measurements in reverberation rooms and transmission loss tests, should qualifying indicators be reconsidered. Methods of enforcing sufficient diffusion in a particular room can neither be decided upon nor their success or failure judged without an objective method and description of diffuseness degree.

KNOWN MEASUREMENT METHODS AND OBJECTIVE DESCRIPTORS

Temporal Diffusion: Sound diffusion has been quantified from the pressure impulse response with respect to time by "Temporal Diffusion", Δ proposed by Kuttruff [4].

$$\Delta = \frac{\Phi_{(0)}}{\Phi_{max(\tau\neq0)}} \tag{1}$$

where,

$$\Phi_{(\tau)} = \int_0^{\infty} P(t)\,.\,P(t+\tau)\,dt$$

It characterizes the randomness of the impulse response by the ratio of the maximum value of the auto-correlation function of the impulse response, $P(t)$ at zero lag to the maximum value outside the origin.

Spatial Diffusion: A very crude measure based on uniform pressure at all locations in diffuse field is to check the spatial variance of the steady-state sound pressure level at different position in the room excited by random noise.

Another objective descriptor, the directional diffusion index, depends on a knowledge of the sound field directional distribution. Sound directional distribution is characterized by the angles describing incidence (θ and ϕ) at an instant of time. By exciting the room with a stationary sound these parameters can be measured by scanning all directions with a directional microphone of high resolution, then the degree of diffuseness can be quantified by the diffusion index proposed by Thiele **[4,5,6]**:

$$\Theta = 1 - (\mu / \mu_o) \tag{2}$$

where,

$$\mu = \frac{1}{\Omega \langle f \rangle} \int | f - \langle f \rangle | \, d\Omega$$

$\langle \hat{I} \rangle$ = average intensity, w/m^2, $\hat{I}$ = incoming intensity, w/m^2, Ω = solid angle of interest, and $\mu_o = \mu$ measured in free field with the same microphone

An indirect but reliable measure is to calculate the correlation coefficient between the steady-state sound pressure signals at different locations in the room expressed by **[4,5]**:

$$\psi = \overline{P_1 P_2} / (\overline{P_1^2} . \overline{P_2^2})^{1/2} \tag{3}$$

Coherence between sound pressure and particle-velocity has been shown to reflect the nature of the sound field **[11]**. Theoretical diffuse field quantification by the two microphone intensity technique has been proposed by Gerges **[12]**, where the coherence function between the acoustic pressure and particle velocity is employed as a quantitative indicator of the sound field diffusion in a reverberant field and defined by:

$$\gamma^2(f) = |G_{pu}|^2 / (G_{pp} . G_{uu}) \tag{4}$$

where, G_{pu} is the cross spectrum of the pressure, $p(t)$ and particle velocity, $v(t)$ signals and G_{pp} and G_{uu} are the auto-spectrum of both respectively.

NEW QUANTIFIERS OF SOUND DIRECTIONAL DISTRIBUTION AND DIFFUSENESS

An ideal diffuse sound field exists when the energy flow at a given position is the same in all directions for all arrival times, hence there is no acoustic net energy flow and the instantaneous sound intensity is zero. Diffusion of the sound field can also be viewed as the sound being isotropic in all direction. In this case the sound decay in all directions should exhibit the same decay rate. Deviation of a directional decay component from the others indicates a lack of spatial homogeneity. Examples of quantifying diffuse sound from the point view of energy flow and directional decay curves will be shown.

CONCLUSION

Objective descriptors of sound field diffuseness exist but both objective and subjective quantifiers require further study. There is also a need for parallel subjective studies to determine the subjectively perceived onset time and required adequate degree of diffuseness that characterizes the late arriving reverberant energy at a listener position.

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ON RESPONSE MEASUREMENTS USING DETERMINISTIC TYPES OF PERIODIC SIGNAL AND FFT

Wing T. Chu
Acoustics Laboratory, Institute for Research in Construction
National Research Council Canada, Ottawa, Ontario, K1A 0R6

1. Introduction

System response measurements appear in many areas of scientific and technical investigations. The speed of these measurements has been greatly improved with the advent of FFT analyzers. Conventional FFT analyzers require two channels to measure the system response using white noise as a stimulus. Bias errors such as aliasing, leakage, and picket fence effects have to be properly addressed and random errors have to be minimized by averaging. Uses of deterministic types of periodic signal as stimulus are not common although most of these errors would disappear. The inflexibility of the commercial FFT analyzers may be an important reason. To analyze periodic signals, it is necessary to use an 'exact' multiple of the period of the signal to avoid leakage effect. It is also necessary to ensure that the signal frequencies coincide 'exactly' with the analysis frequencies to avoid picket fence effect. This will require some tailoring of the sampling frequency and the number of data points. In this paper, we will explore the application of deterministic types of periodic signal for system response measurements using a dual channel A/D converter and a special FFT routine. Two types of periodic signal will be discussed: (1) pure tones for single frequency measurements and (2) m-sequence or maximum-length sequence (MLS) for broad-band measurements.

2. Pure-tone Tests

This is a simple signal to use for system response measurements. Usually, the magnitude is measured with a ratio meter and the phase with a phase meter. More accurate results can be obtained using a lock-in amplifier. Chu[1] has used such a technique for impedance tube measurements based on the two point transfer-function technique. Since the lock-in amplifier is not a common laboratory instrument, an alternative way of measurement using the FFT has been considered. Figure 1 shows the arrangement of equipment for checking the accuracy of the proposed method. Pure tones of different frequencies (125, 250, 500, 1000, 2000, 4000 Hz) were generated by a stable function generator (HP 3325A). A 12 bit dual-channel A/D converter was used to sample simultaneously both the input and output signals of the linear system under test. The sampling frequency used was 10 kHz and the number of data points chosen was 2000. This combination satisfied the two conditions mentioned in Section 1. To perform the discrete Fourier transform on 2000 points, a special non-power of 2 FFT routine is required. Two routines are available; one is the Chirp Z transform[2] and the other is the Glassman's general n points FFT[3]. The latter was used in this study. Table 1 shows the phase response of a digital delay system (Klark-Teknik DN700) at different delay settings. The average and sigma were obtained from 20 repeated measurements. The measured results compare very well with the calculated results from the delay settings.

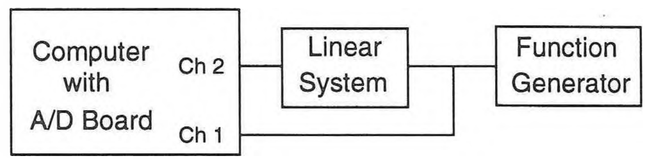

Figure 1. Diagram of equipment used in pure-tone tests.

3. m-Sequence Tests

An m-sequence is a periodic binary sequence generated by digital shift registers with appropriate feedback[4]. The most important property of any m-sequence is that its circular autocorrelation function is essentially an

Frequency	125	250	500	1000	2000	4000
0.0265ms delay	-1.19	-2.38	-4.77	-9.54	-19.08	-38.16
Measured<Phase>°	-1.19	-2.39	-4.77	-9.55	-19.11	-38.18
Sigma (deg)	0.006	0.007	0.009	0.011	0.021	0.018
0.053ms delay	-2.38	-4.77	-9.54	-19.08	-38.16	-76.32
Measured<Phase>°	-2.38	-4.77	-9.54	-19.09	-38.22	-76.33
Sigma (deg)	0.005	0.008	0.008	0.012	0.017	0.014
0.0795ms delay	-3.58	-7.16	-14.31	-28.62	-57.24	-114.48
Measured<Phase>°	-3.58	-7.16	-14.31	-28.64	-57.32	-114.46
Sigma (deg)	0.006	0.007	0.008	0.017	0.011	0.015

Table I. Comparison of the expected phase response of a digital delay system with measurements using pure-tones. Averages and sigma were computed from 20 repeated measurements.

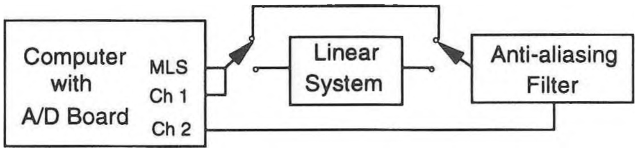

Figure 2. Diagram of equipment used in m-sequence tests.

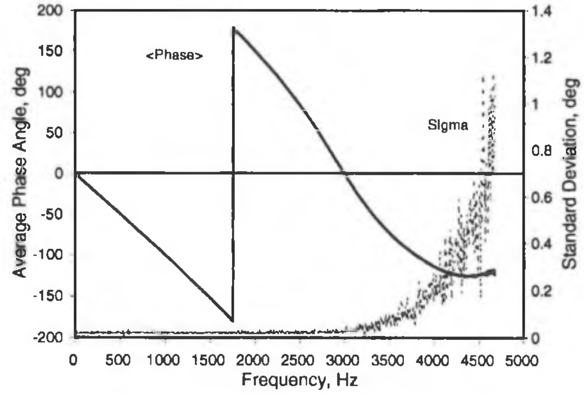

Figure 3. Average and standard deviation of the phase response of a 3 kHz low-pass filter from 10 measurements.

impulse. This property immediately suggests that the impulse response of linear systems can be measured by cross-correlating the system output with the m-sequence excitation signal[5]. The periodic length, L, of the m-sequence is always one less than an integer power of 2. Special technique is available to calculate the impulse response[6,7]. Frequency response can also be determined directly using the general n points FFT routines mentioned in the previous section. Since the m-sequence is periodic, one can use an exact multiple number of periods to eliminate the leakage effect. Using the same clock frequency for both the m-sequence generation and the sampling frequency of the A/D converter will ensure that the analysis frequencies coincide exactly with those of the m-sequence. An anti-aliasing filter is required for accurate measurements, whose response has to be determined and corrected for in the computation of the unknown system response. Figure 2 shows a schematic diagram of the equipment set-up. An m-sequence of 4095 points was used with a clock frequency of 10 kHz. The impulse response of the system was determined first and the frequency response was obtained by Fourier transforming the impulse response. Figure 3 shows the average phase response of a 3 kHz low-pass filter and the standard deviation obtained from 10 repeated runs as a function of frequency. Values at 125, 250, 500, 1000, 2000, 4000 Hz were picked and compared with those obtained using pure-tones. Table II shows good comparison for both magnitude and phase except for the phase result at 4 kHz obtained by the m-sequence method. Better agreement was obtained for this case when the sampling frequency was increased to 16 kHz.

4. Conclusion

We have shown that accurate response measurements can be made using periodic signals and FFT. With proper choice of the sampling frequency and the number of data points, bias errors associated with FFT can be eliminated. Random errors become insignificant because the periodic signals used are deterministic.

Frequency	125	250	500	1000	2000	4000
			<Magnitude >dB			
Pure-tone						
Average	0.07	0.05	0.03	-0.01	-0.12	-20.36
Sigma	0.001	0.000	0.001	0.001	0.001	0.002
MLS:10k SF						
Average	0.07	0.06	0.04	-0.00	-0.12	-20.33
Sigma	0.003	0.002	0.004	0.004	0.002	0.026
MLS:16k SF						
Average	0.07	0.06	0.05	0.00	-0.13	-20.21
Sigma	0.004	0.003	0.004	0.004	0.003	0.018
			<Phase>deg			
Pure-tone						
Average	-12.44	-24.73	-49.40	-99.60	151.52	-118.46
Sigma	0.002	0.003	0.003	0.005	0.010	0.013
MLS:10k SF						
Average	-12.45	-24.76	-49.42	-99.63	151.59	***-113.59***
Sigma	0.024	0.019	0.023	0.020	0.017	0.178
MLS:16k SF						
Average	-12.44	-24.79	-49.45	-99.66	151.54	-118.66
Sigma	0.028	0.015	0.031	0.023	0.021	0.103

Table II. Comparison of the frequency responses of a 3 kHz low-pass filter measured with pure-tones and with the m-sequence using different sampling frequencies (SF).

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COMPARISON OF ROOM IMPULSE RESPONSE MEASUREMENT METHODS

S.Norcross
Audio Research Group, University of Waterloo, Waterloo, Ontario, N2L 3G1.
J.S. Bradley
Institute for Research in Construction, National Research Council, Ottawa, Ontario, K1A 0R6.

INTRODUCTION

One of the most important measures of a system such as a room is is impulse response. Almost all characteristics of that system can be calculated directly from the impulse response, i.e. in room acoustics numerous objective parameters such as reverberation time, early/late ratios and RASTI can all be obtained from it. In room acoustics, due to large reverberation times, longer impulse responses are needed and a large dynamic is desired.

The impulse response of a system is defined as the output when a perfect pulse, or delta function, is applied to it. Hence, the simplest of all techniques is just to apply a short duration pulse to the room and then measure its response. Other broadband signals can be used, with various processing, to calculate the impulse response. One example is the use of a chirp, a short duration sine sweep. This approach has more energy output than a pulse, but requires post-processing. If one stretches out the chirp so that it is a continuous repeating sweep, even more energy is output, and this requires similar post-processing. Finally, a pseudorandom noise, or a maximum-length sequence signal, with a Fast Hadamard transform can be used, to calculate the impulse response. For all of these techniques to be valid, the system under study must be linear and time invariant (except with the pulse method where time invariancy is not a problem.)

This paper will compare each of these approaches, and describe the strengths and weaknesses of them when applied to room acoustics.

THEORY

As mentioned above, the simplest approach is to excite the room with a short pulse and then measure the response. This approach does not produce much energy in the room. Therefore, to achieve an adequate signal-to-noise ratio, numerous averages have to be made. This requires more time and is not always practical.

A chirp, which is generated simply by creating a sine wave that changes frequency in time, can have a duration of about 10-100 ms and can be repeated every 2 to 3 seconds. One measures the response of the room due to that chirp and later calculates the impulse response. The processing is a correlation of the input and output to obtain the impulse response. This can be accomplished by using FFTs, dividing the complex spectra of the source and received signals followed by an inverse FFT to get the impulse response. One problem is that the correlation with FFTs is actually a circular one and a wrap-around effect is present. This can easily be fixed by padding with zeros or subtracting the length of the chirp from the end of the calculated impulse response[1]. Due to the chirp frequency being dependent on time, one can contour the frequency spectrum of the chirp by varying the amplitude of individual frequencies. Thus, one can produce a white spectrum, or taper the frequencies to what is desired.

A sine sweep is similar to a chirp but it is continuously repeated so that there are no gaps in the signal. The processing is similar, but since the signal is continuous and is periodic one does not have to worry about the circular correlation problem. Therefore, to obtain the impulse response, simple FFT methods can be used. The one advantage over the chirp is the amount of energy that is output with the continuous sweep is greater.

Finally, a maximum-length-sequence signal and a Fast Hadamard transform can be used to calculate the impulse response[2,3]. The basic idea here is to excite the system with the MLS, acquire the response of the room, then cross-correlate that response with the MLS source signal. The result is the system impulse response.

EXPERIMENTAL PROCEDURE AND DATA

Each of the above mentioned techniques was implemented using a PC and a 16-bit A/D board, and data was sampled at 12780 Hz. For the pulse method, the pulse width was 0.23 ms and the peak was approximately 4.5 volts. The chirp duration was about 40 ms, while the sweep and MLS signal were about 2.5 seconds in duration.

Figures 1 through 4 show the first 60 points of the impulse response of a low-pass filter with a cutoff frequency of 5.22 KHz cutoff. Figure 1, the pulse method, shows little energy output, while Figures 2 and 3 (chirp and sweep) show about the same energy output. This is due to correlation process normalizing the impulse response with the source signal. The most energy output is by the MLS. All the impulse responses look very similar and further tests in rooms will show how the strengths of each method when dealing with longer impulse response and background noise.

CONCLUSIONS

In a simple test, each of the methods produced similar results, except for the pulse with little energy which was expected. Therefore, tests in rooms need to be performed to further investigate the four methods.

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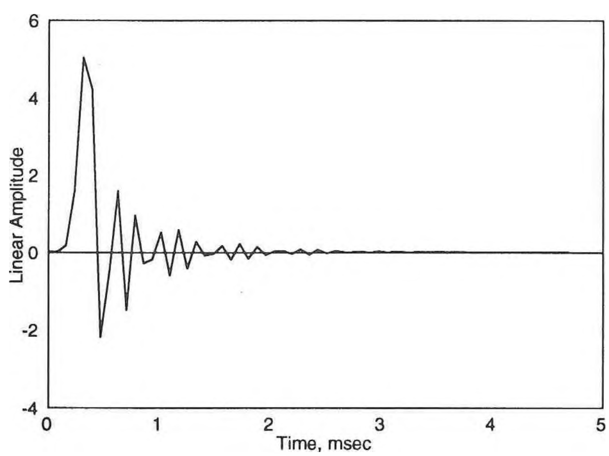

Figure 1. Impulse response from pulse method.

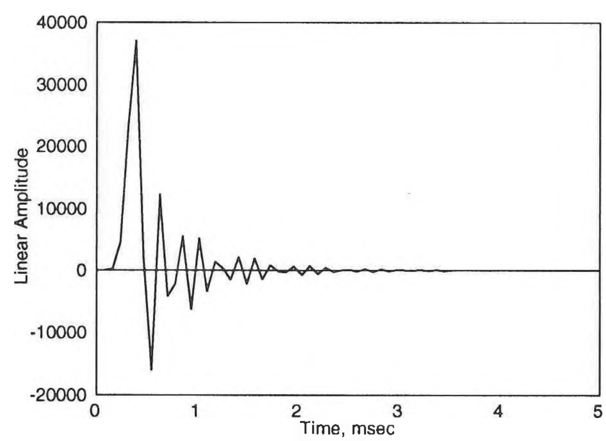

Figure 3. Impulse response from sweep method.

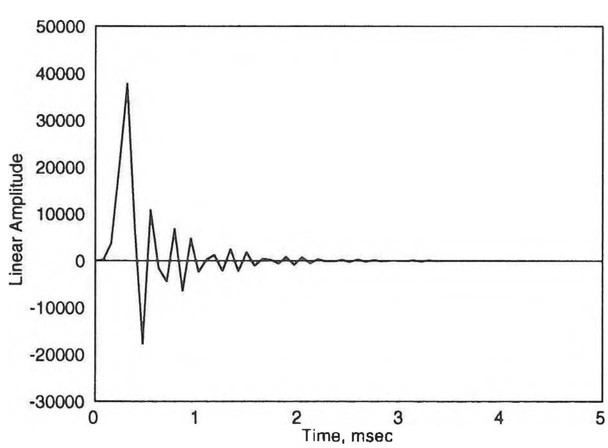

Figure 2. Impulse response from chirp method.

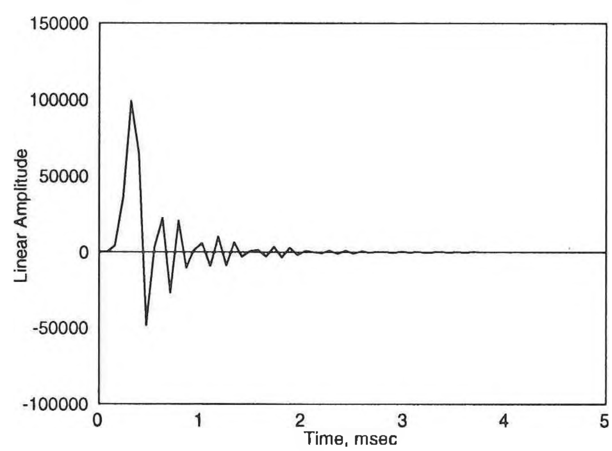

Figure 4. Impulse response from MLF signal methd.

ACOUSTICAL ANALYSIS OF THE ORPHEUM THEATRE

J.S. Bradley
Acoustics Laboratory, Institute for Research in Construction, National Research Council, Ottawa, Ontario K1A OR6
G.A. Soulodre
McGill University, Montreal, Quebec H3A 1E3.

Introduction

Extensive acoustical measurements and analyses were carried out in the Orpheum Theatre (Vancouver) for Aercoustics Engineering Ltd. as part of studies to develop plans for renovating the theatre. Measurements were made both on stage and for a number of locations in the audience seating area. Complete sets of measurements were repeated to evaluate the effects of the existing over-stage reflectors and an under-balcony electro-acoustic enhancement system. The measurements quantified problems associated with focusing effects and the different conditions found under a large balcony overhang. The measurements are compared with similar results found in other large halls.

Measurements were made with our RAMSoft-II computer-based measurement system that uses an MLS signal and Fast Hadamard Transform techniques to calculate impulse responses. The impulse responses were filtered into octave bands and a dozen different acoustical quantities were calculated while in situ at each measurement position in the hall. For these measurements, three different source positions on stage and 15 receiver positions in the audience area of the hall were used.

Comparison of Hall-Average Measurements

Initial analyses compared mean values of the principal acoustical measures with average values from other large halls. Figure 1 compares the mean RT values from the Orpheum Theatre with the range of Mean RT values from 11 other large halls. Also shown are the hall-average EDT values versus frequency from the Orpheum Theatre. While mid-frequency RT values are intermediate to those of other halls, at the lowest octave band the Orpheum RT value is longer than in the other halls. The mean EDT value is a little less than the RT value, indicating decays are not completely linear and the hall will sound somewhat less reverberant than indicated by the measured RT values.

Sound levels as measured by the relative level or strength (G) are shown in Figure 2. The mean measured G values are comparable to the lowest of the other 11 halls at mid- and high-frequencies.

Mean values of C80, a measure of the clarity or the balance between clarity and reverberance, were intermediate to those of other halls. Similarly, mean values of measurements of the lateral energy fraction, LF, suggested that spatial impression in the Orpheum Theatre would be in the middle of the range of values found in other halls.

Variations with Source-Receiver Distance

The acoustical quality of a hall can also be assessed by the spatial homogeneity of various measures. Figure 3 compares the variation of 1000 Hz RT and EDT values with source-receiver distance. While the RT values are reasonably constant with distance, EDT values decrease quite dramatically towards the rear of the hall. EDT values from seats under the balcony were consistently lower than

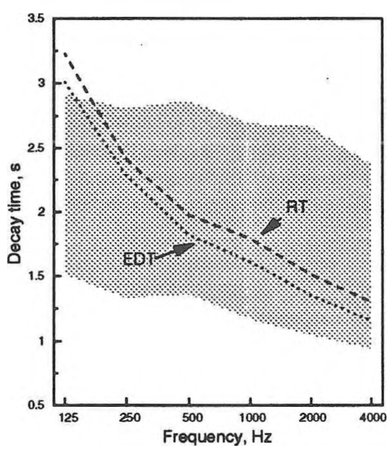

Figure 1. Mean RT and EDT in the Orpheum Theatre compared to the range of hall-average RT values from 11 halls.

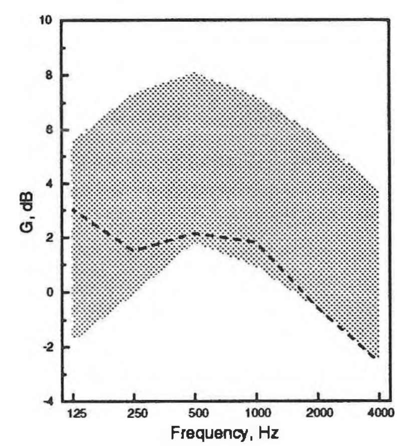

Figure 2. Comparison of mean G values in the Orpheum Theatre and the range of mean G values from 11 halls.

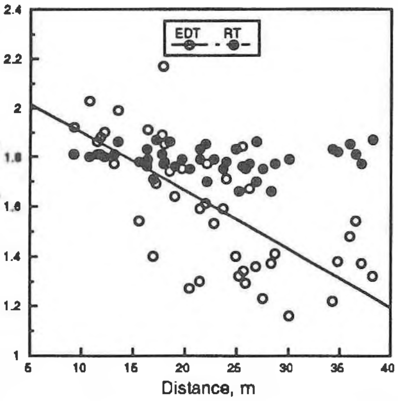

Figure 3. RT and EDT decay times versus distance at 1000 Hz.

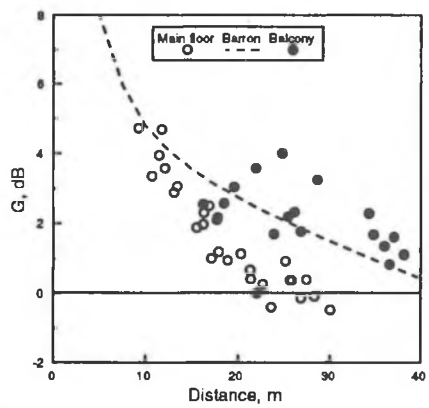

Figure 4. G values versus distance at 1000 Hz.

EDT values measured at seats in the balcony.

Figure 4 plots 1000 Hz G values versus source-receiver distance. Main floor seats under the balcony are about 2 dB lower than predicted by Barron's revised theory[1]. Some seats in the balcony have G values in close agreement with Barrons's theory while others exceed the theoretical prediction. This is illustrated by the plot of G80 values (the relative level of the first 80 ms of the impulse responses) versus distance in Figure 5. Here most measurements both in and under the balcony agree quite well with Barron's theory. Eight measurements at Balcony seats are approximately 2 dB greater than the theoretical prediction. These results are due to focussed reflections from various curved areas of the ceiling. Further unusual results are seen for the G(late) values shown in Figure 6. Here the relative level of the late arriving sound is significantly lower and decreases more rapidly with distance under the balcony.

Particular Effects

At some locations, the focusing effects of the ceiling (that influenced the G80 results in Figure 5) were even more significant. Impulse responses showed very strong reflections and G80 values exceeded the theoretical prediction by over 5 dB at 1000 Hz. At these locations,

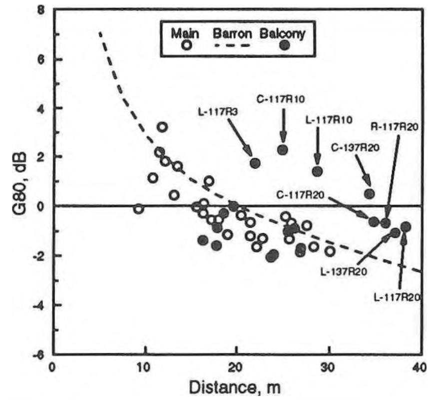

Figure 5. G80 values versus distance at 1000 Hz.

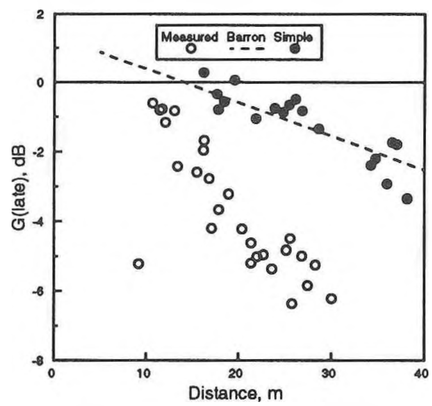

Figure 6. G(late) values versus distance at 1000 Hz.

image shifts were so audible that the soloist appeared to be located in the ceiling of the hall.

Particular efforts were made to identify the effects of an array of over-stage reflectors. Complete sets of measurements (3 source positions by 15 receiver positions) were made both with and without the reflectors in place. Plots of G80 values versus frequency were compared for each measurement location. Some main floor seats in front of the balcony showed small increases in 4000 Hz early sound levels. On-stage measurements at some positions also showed effects of these reflectors.

The hall has an electro-acoustic enhancement system to increase sound levels under the balcony. In its present state, this system was found to be quite ineffective. The under balcony G(late) values shown in Figure 6 suggest that an ideal enhancement system should increase the late arriving sound levels so that they are more equivalent to those in other areas of the hall.

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THE COMPONENTS OF SPATIAL IMPRESSION IN CONCERT HALLS

Gilbert A. Soulodre
McGill University
Montreal, Quebec, H3A 1E3.

John S. Bradley
National Research Council
Ottawa, Ontario, K1A 0R6.

Introduction:

Spatial impression is usually loosely described as the sense of being enveloped by the sound or as an increase in the apparent width of the source. In a previous study by the authors [1], the findings of Barron and Keet [2,3] were verified in experiments using sound fields consisting of a direct sound and a few early reflections. The results confirmed that the apparent width of the source increases with increasing early lateral energy. It was also found however that although a broadening of the source was evident for these simple sound fields, listeners never felt enveloped (surrounded) by the sound. In a series of informal listening tests, it was found that a sense of *listener envelopment* could be obtained by including late arriving or reverberant energy in the sound fields. Furthermore, the addition of reverberant energy was found to reduce the listener's ability to discriminate the effects of early lateral energy. Several subjective experiments were conducted to explore these issues [4,5]. The results indicate that spatial impression has two distinct components: apparent source width (ASW) and listener envelopment (LE).

Method:

All tests were in the form of double-blind paired comparisons using music as the stimulus. Subjects could switch between the two sound fields in a pair for as long as they required in order to make their decision. Two methods were employed to simulate the sound fields used in the experiments: multiple loudspeakers in an anechoic chamber, and reproduction of anechoic music convolved with binaural impulse responses of real concert halls.

Effect of reverberation on ASW:

The first experiment was designed to show how late arriving sound affects a listener's ability to detect changes in the early lateral energy. Subject's listened to pairs of sound fields and rated the difference in ASW on a five-point scale. The sound fields consisted of four distributions of early lateral energy and three levels of reverberant energy, thus giving a total of 12 sound fields. The results, shown in Figure 1, demonstrate that differences in early lateral energy (as measured by ΔLF or ΔIACC) become increasingly difficult to detect as the level of reverberation (as measured by C80) is increased.

This finding was confirmed in a test using the binaural simulator. Ten sound fields of varying early lateral energy were subjectively rated in terms of ASW. The impulse responses used to create the sound fields were then truncated to eliminate all energy after 80 ms, and the rating was repeated. It was found that changes in ASW were more easily and consistently detected for the truncated sound fields.

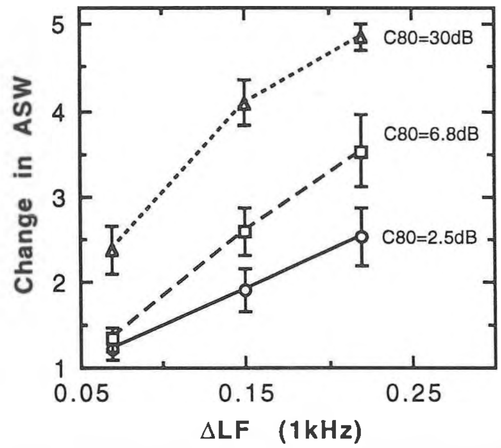

Figure 1. Perceived change in ASW vs. ΔLF with 90% confidence limits.

The degree to which reverberation affects the audibility of changes in ASW can be appreciated when one notices in Figure 1 that a difference in LF of 0.22 with C80=2.5 dB is perceived to be the same as a difference in LF of 0.07 with C80=30 dB. A ΔLF=0.22 approaches the maximum measured in real halls, while a ΔLF=0.07 approaches the limit of detectability under free field conditions.

Listener Envelopment (LE):

Informal listening tests indicated that some late arriving energy was required to produce a sense of listener envelopment. Therefore, a series of subjective experiments were conducted in an anechoic chamber to examine how the temporal distribution, level, and spatial distribution of the late energy affected the perception of listener envelopment.

The temporal distribution of the late energy was found to be important to listener envelopment. Results indicate that some energy must arrive after about 80 ms in order to obtain a sense of LE. LE was also found to be related to reverberation time. Longer RT's provided a greater sense of listener envelopment.

The level of the late energy was found to have a strong effect on LE. Specifically, higher levels of reverberant energy resulted in increased listener envelopment.

The effect of the spatial distribution of the late energy was analogous to the effect found by Barron [2] for early lateral reflections. Listener Envelopment increased as the late energy from lateral angles increased. The highest degree of listener envelopment was obtained when the late sound included energy from ± 90 degrees.

Predicting Listener Envelopment:

Using the binaural simulation method, a subjective test was conducted in which ten subjects rated ten sound fields according to LE. The findings from the anechoic chamber tests were then used in an attempt to derive a suitable predictor of these subjective results.

A measure of LE must include terms that account for the level and spatial distribution of the energy arriving after 80 ms. As such, it was found that LE was very strongly related to the A-weighted level of the late lateral energy as defined in Equation 1. An A-weighting was used since it was found to correlate best with the perception of loudness in concert halls [6].

$$LG_{80}^{\infty} = \int_{80}^{\infty} p^2(t) \cdot \cos(\alpha) dt \Big/ \int_{0}^{\infty} p_A^2(t) dt \qquad (1)$$

where p(t) in the impulse response of the room and α is the angle between the direction of arrival of the reflection and a line through the two ears. The symbol $p_A(t)$ represents the response of the source measured at a distance of 10 m in a free field. The results of the subjective test are plotted against LG_{80}^{∞} in Figure 2.

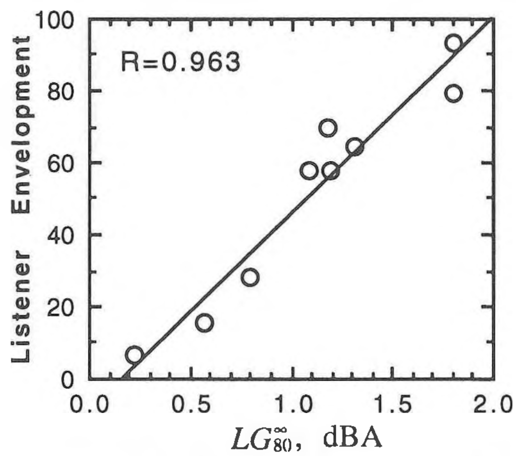

Figure 2. Listener Envelopment vs. LG_{80}^{∞}.

A final experiment was conducted to verify that ASW and LE are indeed separate perceptual phenomena. Four sound fields consisting of two levels of ASW and two levels of LE, were presented to subjects in randomly ordered pairs. The subjects were asked to identify differences in the sound field pairs as either a change in ASW or a change in LE. The results showed that there are two distinctly different aspects to spatial impression, and that subjects could correctly identify them.

The difference between ASW and LE can be explained in terms of the precedence effect. Early reflections will tend to be integrated with the direct sound and will therefore tend to be temporally and spatially fused with the direct sound. This will increase the apparent level of the direct sound and cause some ambiguity as to its perceived location. The result is an increase in ASW. Later arriving sound however, is not integrated with the direct sound and thus leads to more spatially distributed effects that appear to envelop the listener.

Conclusions:

Spatial impression is composed of two components: apparent source width and listener envelopment. Apparent source width is related to early lateral reflections, while listener envelopment is related to the relative level of the late lateral reflections. It has further been shown that the effects of early lateral reflections are less audible in the presence of reverberation. This suggests that listener envelopment is the more important component of spatial impression.

The new results have considerable significance for the design of concert halls. Prior to this study, increased spaciousness was assumed to require only strong early lateral energy. As a result, some newer halls have introduced large reflector panels specifically to add strong early lateral reflections. This design approach however, does not guarantee that sufficient late lateral energy will result. Conversely, designs intended to maximize later arriving lateral energy will usually tend to also increase early lateral reflections and thus ASW. For example, a shoe-box shaped hall will tend to provide both early and late lateral energy while a fan-shaped hall might include reflector panels to provide early lateral reflections without producing significant late lateral energy. The importance of late lateral energy may be another reason for the success of shoe-box shaped concert halls.

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Testing The Sensitivity Of Stage Acoustics Measurement Techniques

John O'Keefe

Aercoustics Engineering Ltd.
50 Ronson Drive, Suite 127
Toronto, Canada M9W 1B3

INTRODUCTION

The stage acoustics parameters proposed by Gade[1] and Naylor[2] in the 1980s have gained general acceptance but been have applied in various and sometimes incompatible ways. Some measure Support at 0.5 m while others measure it at 1.0 m. Some measurements have been performed on empty stages, others on fully equipped stages with chairs and music stands. These new stage measurements offer the promise that the early energy audience related measurements did in the 1960s and '70s. If a consistent database of stage measurements is to be established, it would seem prudent to test the sensitivity of these measurements and the effects of the various measurement techniques.

MEASUREMENT SYSTEM

The acoustic source used in the measurements was a dodecahedron with 75 mm Altec ALS 35 loudspeakers. Source height was typically 1.1 m above the stage and, unless mentioned otherwise, Support ratios were measured at a source receiver distance of 0.5 m. Similar to the author's previous studies[3], measurements were performed at and between five locations on each stage corresponding to Soloist, Violin, Viola, Horn and Bass. For the occupied seating measurements, responses were recorded on Digital Audio Tape (DAT) recorders for subsequent processing in the laboratory. For all other measurements, processing was done on site.

ABSORBENT LAYER

In their initial studies, both Gade and Naylor placed a 50 mm thick glass fibre blanket beneath the source, reasoning that it represented a crude estimate of the absorption of the musician seated in his or her chair. Gade and others no longer use a blanket, O'Keefe & Bracken[3] still do.

Measurements have been performed with and without a blanket in two rooms and at two different source receiver distances. The first room is Roy Thomson Hall, Toronto which at the time was equipped with seating and music stands for approximately fifty musicians. The second room was an empty lobby that had a reverberation time of 1.7 seconds and was devoid of furniture or acoustically absorbent materials.

In Roy Thomson Hall, the effect of the blanket was less than 0.5 dB. In the lobby there was a slightly more pronounced effect, notably at higher frequencies. The effect of the absorbent layer was marginally higher for measurements at 0.5 m than it is at 1.0 m.

SOURCE-RECEIVER DISTANCE

Gade and others measure Support ratios at a distance of 1.0 m from the centre of the dodecahedron. Naylor has pointed out that a 0.5 m source receiver distance is a bit closer to reality, i.e. it is difficult to play an instrument held out 1.0 m from the performer's head! Naylor used a 6 dB adjustment for a modified version of Gade's Early Ensemble Level and some of the Support ratios. The adjustment was to account for hemispherical divergence of the direct sound and allow for direct comparison with Gade's measurements. O'Keefe and Bracken[3] applied a similar adjustment.

Support ratios have been measured on a number of concert platforms and proscenium arch stages using source receiver distances of 0.5 and 1.0 m. The data makes two things immediately clear. The average difference between 0.5 and 1.0 m measurements is less than 6 dB and it varies considerably, both in frequency and from stage to stage. Floor reflections explain part of this discrepancy but cannot explain all of it. The "direct" sound component of a Support ratio is measured over a temporal window of 10 ms. It would seem therefore that any reflecting surface within approximately three metres of the microphone will affect the adjustment factor. Of the six halls that were measured only the Queen Elizabeth Theatre had a completely bare stage. It was also to closest to 6 dB. To conclude, it does not seem practical or prudent to compare 0.5 and 1.0 m Support ratios through the use of a correction factor.

CHAIRS & MUSIC STANDS

Measurements were performed on two stages with and without chairs and music stands. In both cases measurements were performed with a 50 mm layer of glass fibre blanket underneath the source. There were chairs and stands for approximately 24 musicians on both stages. Care was taken to move the music stands away from the source and receiver.

Both Support and Modulation Transfer Functions are remarkably insensitive to the presence of chairs and music stands. The Support ratios changed by less than 1 dB at high frequencies. At 500 and 1000 Hz the difference was less than 0.5 dB. Mean MTFs were hardly changed at all. The only noticeable difference in MTFs was at the higher modulation frequencies (12 to 20 Hz) and here the change was in the order of 0.1. Naylor suggests that this is subjectively insignificant[4].

OCCUPIED CHAIRS

Following up on this finding, it made sense to see if the measurements were sensitive to the difference between empty and occupied chairs. The measurements were performed in a shoe box shaped gymnasium with a mid-frequency reverberation time of 2.3 seconds, unoccupied. Chairs and music stands were set up for 25 musicians at one end of the room and the first set of measurements were performed. The glass fibre blanket underneath the source was omitted in a effort to maximise the difference between the empty and occupied configurations. The musicians were then asked to enter the room and take their seats. Once seated, the measurements were quickly repeated. Again, the Support and MTF changed very little. Mean MTFs changed approximately 2% at low and high (audio) frequencies and about 0.5% at middle frequencies. Support changes were of a similar magnitude. The only significant differences in the impulse response functions were quantified by the Early Decay Time and Reverberation Times which were reduced by about 10% when the musicians entered the room. These results suggest that the presence of musicians on the stage affects the later part of the decay more than the first few reflections.

REGRESSION ANALYSIS

If one assumes for the moment that sound levels on a stage can be quantified with a simple linear regression, it is possible to extract some useful information. Figure 1 shows a typical result at 1000 Hz. The room radius in most concert halls is approximately 5.0 m. In Figure 1 therefore, the classical definition of sound in a reverberant field would suggest a curve. The data in fact has a good fit to the straight line. Figure 2 shows that on a stage, sound levels decrease at a rate of approximately 1 dB/m depending on the octave band frequency. The notable exception is the 250 Hz octave which consistently shows a smaller slope. Both figures demonstrate interesting results and suggest new avenues of research.

ACKNOWLEDGEMENTS

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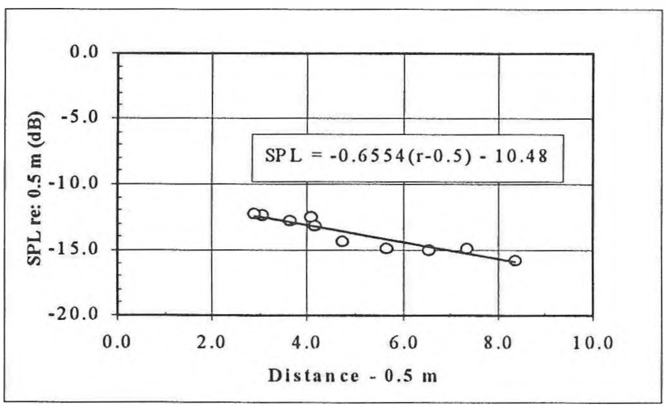

Figure 1 Sound Level Regression, 1000 Hz, duMaurier Theatre, Hamilton, Ont.

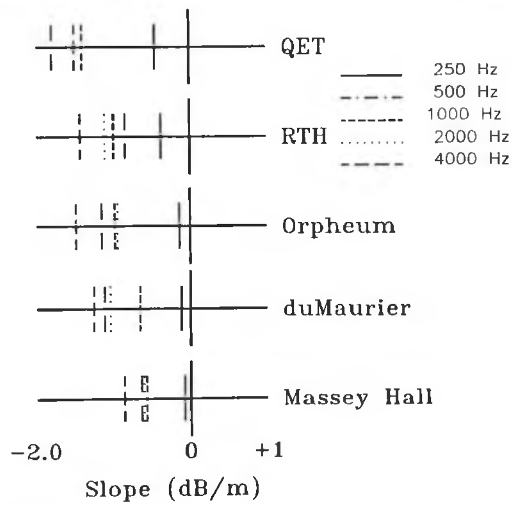

Figure 2 Slopes of linear regressions in dB/m

Acoustic Conditions in and Propagating from Orchestra Pits

John O'Keefe

Aercoustics Engineering Ltd.
50 Ronson Drive, Suite 127
Toronto, Canada, M9W 1B3

Introduction

Orchestra pits are not always easy environments to perform in. Noise levels in most pits often exceed safe levels. In fact, most acoustical research in this area has concentrated on hearing loss. With the emerging understanding of performer related acoustics, it seems timely to expand the investigation of the acoustical conditions in pits.

Measurement Procedure

The measurement system has been described in the companion paper[1]. Measurements were performed at and between three locations inside the pit and between those three locations and the five standard locations measured on stage (Soloist, Violin, Viola, Horn and Bass). Measurements were also performed between the three pit locations and two audience seats and between three stage sources and the same seats. Unless stated otherwise, Support measurements have been measured at 0.5 m. Three rooms were measured including the Queen Elizabeth Theatre (QET) in Vancouver, the Princess of Wales Theatre (POW) and MacMillan Theatre, both in Toronto. Of the three, only the latter pit was empty. The Princess of Wales pit is partially covered while the other two are open, at least in the configurations measured here. At the Princess of Wales, extended coverage of the base building pit was provided by the temporary stage for the musical *Miss Saigon*.

Hearing of Self

Average ST_{total} measurements are higher in the pits than on most stages, as expected. A summary of the results is shown in Table 1.

Table 1
1kHz Support (@0.5 m) and MTF

	STtotal	STearly	MTF
	Self	Hearing of Other	
QET	-14.0	-14.4	0.84
POW	-10.7	-10.8	0.80
MacM.	-16.2*	-17.1*	0.74*

* Erratum - supersedes O'Keefe[2]

Hearing of Other

Gade has found that STearly, measured at 1.0 m, correlates better with Ensemble or Hearing of Other than his Early Ensemble Levels measured across the length and width of the stage[3]. Intuitively this seems a bit odd but when one considers the stage *average* measurements correlated to a group of musicians' *average* response, the findings are perhaps not all that surprising. The assumption, of course, is that all the musicians are in essentially the same acoustic environment. The reasoning breaks down when one considers the communication between a stage and the orchestra pit. In this situation, the two groups of musicians are located in significantly dissimilar acoustical environments. In the measurements presented here, STearly was, on average, 8 dB higher in pits than on stages.

Naylor found the Modulation Transfer Function (MTF) to be a good descriptor of Hearing of Other[4]. Unlike STearly, the MTF is measured between distant locations. In the pit to stage scenario, the MTF seems the more likely alternative to quantify Hearing of Other. At both the Princess of Wales and Queen Elizabeth Theatres, the Mean MTFs are 0.56 between the pit and the stage. There is however a broad range in measured MTF with poor communication between the pit and the back of the stage.

Stage to Pit Balance

In proscenium arch theatres presenting opera or musicals, one of the most important acoustical characteristics is the balance between singer and orchestra. To date, this has received little attention, with the exception of Barron[5]. In the present study, two sets of impulse response functions were measured for a given seat, one with the sources in the pit and the other with the sources on the stage. Barron used a directional source on the stage and an omni-directional source in the pit. This study used an omni-directional source in both locations. The measurements were performed at a single seat on the orchestra level, a few rows in front of the balcony overhang and a single seat on the first balcony. The balance between stage and pit sources was quantified as follows:

$$SPB = \frac{\int_{t1}^{t2} p^2{}_{stage}(t)dt}{\int_{t1}^{t2} p^2{}_{pit}(t)dt}$$

Three variations of the Stage to Pit Balance (SPB) have been considered:

	t1 (ms)	t2 (ms)
SPB_{early}	0	50
SPB_{late}	50	∞
SPB_{total}	0	∞

A 50 ms early energy temporal threshold was chosen rather than the 80 ms that has been used by many for musical clarity. This was done in light of the recent work by Julien et al.[6] suggesting a shift in clarity thresholds. It should also be remembered that in a performance that makes use of an orchestra pit, speech intelligibility or diction is, by definition, more important than orchestral reverberance.

It is not clear that an SPB of 0 dB represents an optimum condition. Given the acoustic energy generated by the orchestra, compared to a singer, it most likely greater than 0 dB. Recognising the importance of singers' formants, the optimum SPB may vary with frequency.

Barron's measurements correspond roughly to SPB_{total}. He found a limited range of about 4 dB inside individual opera houses. The measurements performed here find a similar range for SPB_{total} for the two seating locations that were measured, perhaps even narrower. The SPB_{early} curves in Figure 1 however show a broader range and demonstrate more obvious frequency characteristics. In Figure 1 the QET measurements on the balcony indicate a flat spectrum. The subjective experience on the balcony is one of good balance.

On the QET orchestra level it is difficult to hear the singers. The corresponding SPB_{early} curve shows a noticeable dip at 4000 Hz. This octave contains the important "singer's formant" which allows a soloist to be heard over the much stronger forces of an orchestra in the pit. The objective SPB_{early} measurement appears to agree with subjective experience.

Audience related measurements were performed by John Bradley and Gilbert Soulodre in the QET at the same time as the stage measurements reported here. SPB ratios have been extracted from the data and are shown in Figure 2. The data demonstrates lower overall ratios but similar spectral behaviour, notably at 4000 Hz. Bradley's data was measured over more seats but fewer sources than ours.

These findings are interesting but, for now, anecdotal. Clearly there is room for more work, notably in determining an optimum range for SPB, developing a database from existing rooms and, as demonstrated by comparing Figures 1 and 2, establishing a consistent measurement procedure.

Acknowledgements

In this and the companion paper[1] I would like to thank the management and staff of the various venues who kindly donated their time and space free of charge. This work was supported by a grant from the Ontario Arts Council.

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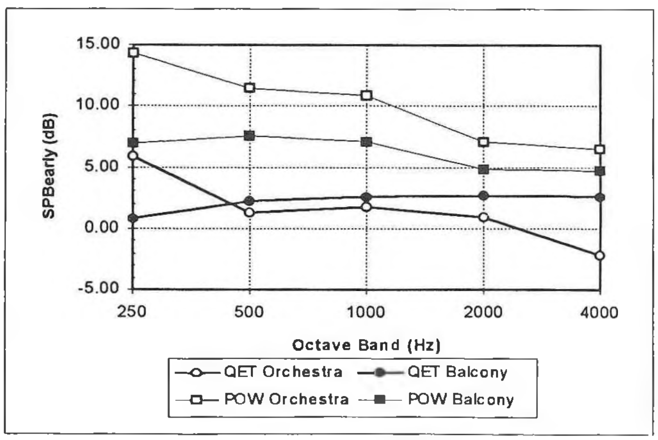

***Figure 1** Stage to Pit Balance (SPB) measured by O'Keefe.*

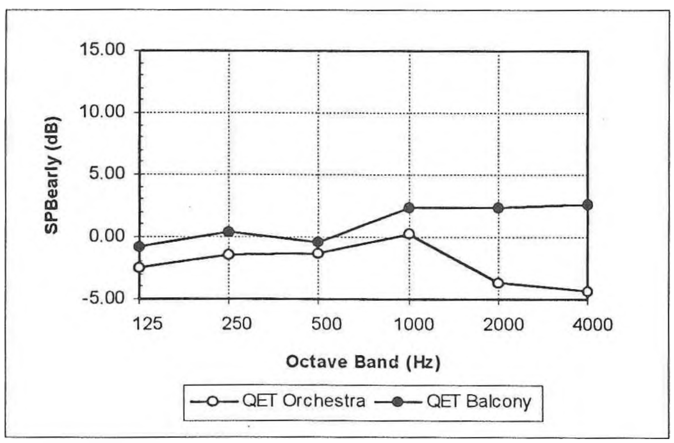

***Figure 2** Stage to Pit Balance (SPB) extracted from Bradley's measurements.*

SUBJECTIVE EVALUATION OF HIGH QUALITY AUDIO SYSTEMS

T. Grusec, Ph.D., Communications Research Centre, 3701 Carling Ave. Ottawa, Ont., K2H 8S2

Radio-frequency spectrum restrictions in radio broadcasting, and space limitations in other applications (such as audio storage on high cost media) demand bit-rate reduction. At the Communications Research Centre, we have been evaluating the subjective quality of audio codecs that reduce digital audio bit-rate by factors of 4 or more, depending on the application. The algorithms used to achieve such reductions are based on psychoacoustic models of hearing so that, theoretically, codecs should be able to operate transparently.

With the best of these codecs, operating at their higher bit-rates, such transparency is likely to be true for average listeners. However, very subtle differences among these codecs can become magnified, for example, by operating several of these codecs in tandem, or by post-processing applied after bit-rate reduction (typical in broadcasting). Differences among the codecs may, thus, become more obvious at the end of complex broadcast chains. Thus, the codecs are not necessarily fully equivalent, and it is essential to make fine-grained comparisons before choosing among them for broadcast, and other critical applications.

For achieving such comparisons, we use special conditions and procedures. We believe these special features are adaptable for the subjective evaluation of any high quality audio devices.

First, our listening room for subjective testing minimizes the effect of room reflections. From 160 Hz upward, our room yields a reverberation time of about 0.20 seconds. This rises slightly at lower frequencies to about 0.35 seconds below 125 Hz. These low values help to ensure that artifacts are not masked while still retaining enough reverberation to make the listening enjoyable. The other important room characteristic is background noise. We have attained an NR rating of 15.

We run listeners one at a time under blind conditions in our evaluation experiments. An innovative disk-based playback system permits seamless switching among three alternative versions of audio materials that each listener compares on each of the trials in a typical experiment. One alternative on each trial ("A") is (usually) an unprocessed reference material. The listener knows that "A" is the standard against which he or she must compare each of the other two versions heard on that trial. One of these other two versions ("B" or "C") is the same audio selection as the reference but processed through one of the codecs under evaluation. The second alternative ("C" or "B") is an "hidden reference," fully identical to the reference "A". The specific assignment of hidden reference and processed versions to "B" or "C" on any trial is not known to the subject. This assignment varies across the different trials in an experiment so that it is unpredictable to the listener.

Each subject must evaluate both "B" and "C" by comparing it to "A" on each trial of the experiment, using a 5-grade rating scale. We instruct subjects to treat this as a continuous scale to single decimal place resolution (a 41-point scale, in effect). Subjects do the switching with a mouse by operating three buttons "A", "B" and "C" seen on a computer screen and corresponding to the audio material version. We design the sessions to take no more than one-half hour for completion by each subject. The actual session length is under the control of the listener. He or she can switch freely among the three versions for as long as needed to decide the scores for each trial. Each 10 to 15 trial session contains all the codecs in the experiment. Each codec is usually presented several times within a session, intermixed unpredictably in the trial to-trial sequence with the other codecs.

Up to three listeners can be run in one afternoon. Two or three sessions of 10 to 15 trials each are usual, with each listener resting while the other two complete a session. Before these individual rating sessions, a group training phase takes place in the morning. In training, all the listeners for a given day can work together, along with a resource person, to become thoroughly familiar with the materials they will be rating in the afternoon. They can interact freely with each other and with the resource person, accessing and discussing all the materials they will be rating later. During training, in contrast to the blind conditions of the afternoon rating sessions, all listeners explicitly know which items are reference and which are processed versions. Thus, they can maximize their sensitivity to the often subtle differences between the versions, learning from each other and from the resource person. Training usually takes up the entire morning.

A crucial process precedes the actual beginning of a subjective evaluation experiment. This is the choosing of "critical" audio materials for use in the experiment. There are no *a priori* methods of making these choices. While work is underway to develop more efficient ways, it is now a tedious, empirical search among standard, commercial CDs and reference or test recordings. The materials must be "fair" ones, so that one should not use artificial materials explicitly designed to "break" a codec. On the other hand, the materials must stress each codec since most "statistically representative" materials will fail to reveal anything due to the high quality of the present generation of codecs.

The method used is simply one of bringing together a number of highly knowledgeable expert listeners and a large library of CD and other materials. Included are versions of these materials processed through the codecs under test. These experts then audition the various materials to find ones that stress each codec to reveal coding artifacts. They try to find a minimum of two stressful materials per codec. Since some materials stress more than one codec, the experts usually find as many materials in total as there are codecs for testing in an experiment rather than twice that number. In the ensuing subjective tests themselves, subjects evaluate each codec against all the materials found for all the codecs. The critical materials search can take up to a month, or more, for 5 or 6 codecs. This search time is usually longer than the time it will take to run the following evaluation experiment.

For the experiments themselves, it is essential that all the subjects are sufficiently sensitive to make the fine discriminations needed to evaluate the codecs reliably. A traditional approach for this purpose is to use pre-screening methods, such as audiometric testing, to choose subjects.

While pre-screening is useful, there are limitations if one uses this approach exclusively. For one thing, one does not usually know whether any given cut-off criterion used for inclusion and exclusion of subjects in conjunction with pre-screening is the most suitable one for a given experiment. If you set the criterion too rigidly, then you may exclude subjects who might have been entirely satisfactory for a given test. On the other hand, a criterion set too loosely may lead to the inclusion of too many deficient listeners. The second of these possibilities is the more serious error for sensitive experiments. From pre-screening alone, there is no way of checking on whether one or the other of these selection errors has occurred, and if so, what its magnitude was.

Also, even though pre-screening suggests that a given subject will do well in an experiment, his or her performance *at the actual time of the experiment* may not be up to that subject's usual capability.

A practical consideration is that formal pre-screening, such as audiometric testing, is costly and time-consuming.

To deal with factors like these, we use a two-fold set of criteria for listener selection. First, we pre-screen in a very loose way by choosing subjects mostly from occupational and interest groups that ought to contain many good listeners. These include audio professionals of various kinds, audiophiles and musicians.

The second step is to measure the actual performance of listeners as shown during an experiment. As mentioned, listeners give ratings to both of the two versions presented on each trial - i.e., to the item they believe to be the hidden reference, and to the one they have concluded is the processed or coded version. Which versions were the true hidden references and which were the coded ones is, of course, known to us as the designers of the experiment. Thus, for all the trials of each subject, we can compare the distribution of scores for the true references with the distribution for the coded items.

We may compare the means or averages of these two distributions with each other statistically by use of a *t*-test that takes into account the correlation between trials, and the variability of the distributions. If the mean for the coded version distribution is significantly different from the one for the reference, then one can infer that the subject was truly discriminating between these two versions. In that case, then, one can conclude that the subject's sensitivity was adequate for the task of the experiment. One can include his or her data along with those of other similarly sensitive subjects in the final analysis of experimental outcomes. On the other hand, if those two means are statistically identical, then one cannot reject the hypothesis that the subject was guessing, overall, rather than properly discriminating between the coded and the hidden reference versions. In this case, then, one can omit that subject's data from the experiment on the grounds of insufficient sensitivity to the experimental task.

Rather than working with the two distributions of scores, there is a fully identical alternative process. One can subtract one of the scores on a trial (reference or coded) from the other one (coded or reference) and work with the resulting single distribution across all trials for each listener. This subtraction procedure automatically takes the correlation between trials into account. A *t*-test would show if the mean of this distribution is statistically zero (indicating guessing) or different from zero (indicating true discrimination).

Over the years, we have built up a pool of listeners and have been able to track the performance over time of those listeners who have been in more than one experiment. Regarding our two stage process of listener selection, we can report that some listeners who were quite adequately sensitive in one experiment are occasionally deficient in another one. This argues that any pre-screening criterion used alone is insufficient to ensure that only good listeners contribute data. A given listener may be good or not, as seen in our *t*-test, depending on factors such as the relative difficulty of detection of the coding artifacts in a specific study.

Although the scores used to measure listener sensitivity are the same ones that we use to draw conclusions about the codecs, we use these scores in independent ways for these two purposes. The listener sensitivity measure (*t*-test) basis is only whether a subject correctly judged items as hidden reference or coded, and not on how he or she evaluated the quality of specific codecs. To illustrate this, we could take any set of data from one of our completed experiments and alter the assignment of rating numbers to specific codecs without disturbing the sensitivity values at all. We usually achieve statistically significant final evaluations of codecs. This means that sensitive listeners tend to be highly consistent with each other in these evaluations, as one would expect.

Our experiments need very few listeners (sometimes as few as half a dozen) to produce conclusive results. Contributing to this reliability is our exclusive use of within-subject (repeated measures) experimental designs. Such designs eliminate individual difference effects.

CONCLUSIONS

The high sensitivity of our experiments is due to many factors. These include: the listening environment and equipment; the use of critical materials; a carefully conducted training phase; rigorous double blind testing conditions; the use of a listener-controlled, seamless switching playback system; performance-based listener selection; and within-subject experimental designs.

We believe that others can use our methods for sensitive evaluations of any high quality audio system or device. More details about these methods are available in a recent report listed below.

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DIGITAL SIGNAL PROCESSING APPLIED TO THE EQUALIZATION OF THE LOUDSPEAKER ROOM INTERACTION

Claude R. Fortier and Pierre Côté

State of the Art Electronik Inc.
Ottawa, Ontario, K1J 9J3

1.0 Introduction

It was recognized by loudspeaker manufacturers that the loudspeaker and the room form an acoustical "marriage" which together determine the sound quality perceived by the listener in the room. Efforts by loudspeaker manufacturers in the last decade to further improve the quality of a loudspeaker by improving its free-field or anechoic response have resulted in very high standards of performance. This level of performance however is not available when the loudspeaker is placed in its intended environment: the listening room. This is due to the room modes and sound reflections coloring or altering the response perceived by the listener, no matter what position the loudspeaker or listener occupy in the room.

In the presence of reflecting surfaces, the direct or on-axis sound of the loudspeaker will be summed with the reflections of all the off-axis response and form the total response at the listening position. Room modes which occur in any closed room, no matter what the shape and degree of sound absorption applied, also affect the response at the listening position.

The desire to improve the measured as well as perceived response at any listening location from any typical loudspeaker position led a group of loudspeaker manufacturers to form a research consortium (CARC) and participate in a joint venture with the NRC. The collaboration was called the Athena project, whose purpose was to improve the overall quality of sound reproduction by loudspeakers in rooms with the use of digital signal processing[1] This paper sets out to demonstrate the technical performance of the current CARC/NRC DSP engine and its associated software.

2.0 Effect of Loudspeaker position

For the purposes of this experiment, a single loudspeaker, model CF-150 by State of the Art Electronik, was placed in 12 different locations in the new listening room at NRC. There were six different lateral positions and two different heights. Measurements were made at six different listener locations around a central area. The room with source (S0 to S5) and microphone (M0 to M5) positions is shown in Figure 1. The room at NRC is designed to emulate a typical good quality domestic listening room. The particular loudspeaker was chosen for its flat free field response and its small size, allowing it to be placed at different heights and lateral positions, near or far from the room boundaries. The source positions were chosen to mimic typical loudspeaker positions. The six microphone measurement positions were chosen to cover a usual listening area. Six measurements were made for each loudspeaker position, resulting in 72 total measurements for the un-equalized case. The same procedure was used for the equalized case. Based on methods found in the Athena project, all the measurements for a given source position were combined to yield a single response curve. This resulting curve is representative of the perceived response in the listener area, as based on psycho-acoustic testing.

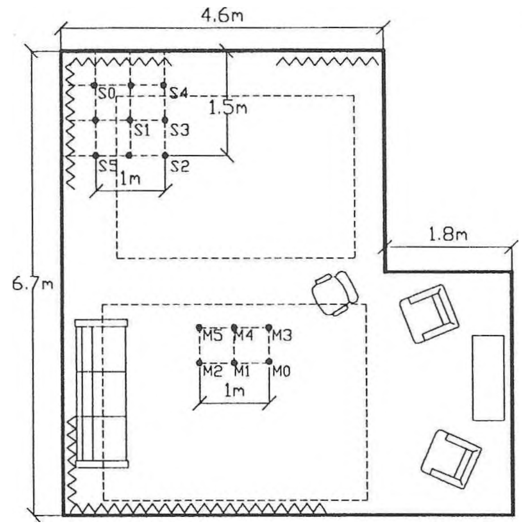

Figure 1. Listening room layout showing source positions S0 to S5 and measurement positions M0 to M5.

The 12 response curves for the unequalized case are shown in Figure 2. The absolute level is arbitrary, and all curves were matched to this arbitrary level using A weighting. The responses show a wide variation of ± 11 dB in the region below 400 Hz. This is due to the effect of room modes and of reflections by the boundaries of the off-axis portion of the loudspeaker output combining in and out of phase with the on-axis output. The loudspeaker is an omni-directional sound source in the low frequencies and its total output is important in determining the resulting response. It is obvious that the large variations in the measured response are due to the room's effect and not dependent on the loudspeaker's response curve. A change in the loudspeaker might tilt the overall envelope of the curves, but not change the deviations from the mean. The response above 400 Hz is quite uniform and independent of source position. Small deviations from a flat response are mostly determined by the loudspeaker and high frequency absorption characteristics of the room.

3.0 DSP Based Equalization

The large variations in response measured at the listening position from the same loudspeaker mean that the listener is completely dependent on the loudspeaker position for the perceived sound quality. This assumes that the loudspeaker has been selected and the room is in its final form. These conditions represent the vast majority of loudspeaker installations in recording studios and home listening environments. Listeners are very rarely at liberty to make geometrical changes or absorption changes in rooms which will improve the response below 400 Hz in a predictable manner. Thus a form of equalization is required which will compensate for the deleterious effects of the room on the loudspeaker. The current

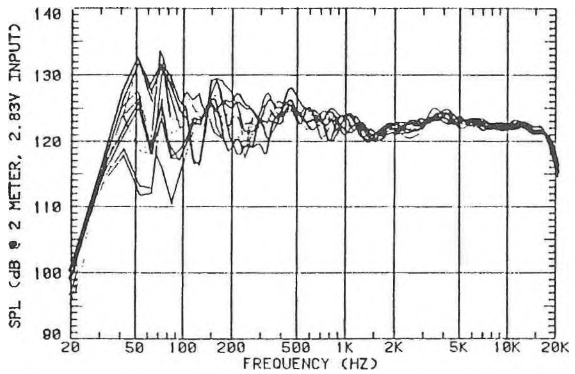

Figure 2. Frequency response measurements for 12 loudspeaker source positions for the unequalized case

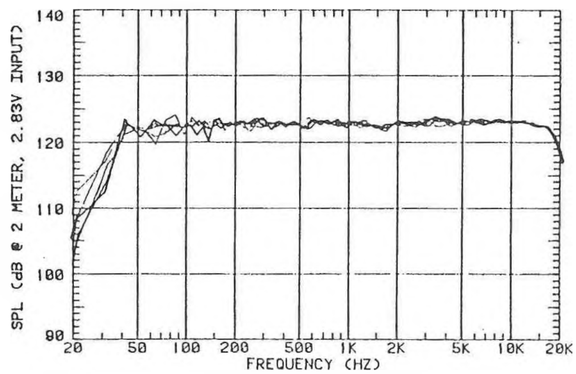

Figure 3. Frequency response measurements for 6 loudspeaker source positions for the CARC/NRC DSP engine equalized case.

method is to measure the response using a microphone and some form of measurement system. Then a 1/3 octave or parametric equalizer is used to set the response to a particular target. Measurements are usually performed at a single listening point and the equalization is adjusted for this single measurement. It was determined during the course of the Athena project that this is not a good solution as judged subjectively. In any case the 1/3 octave frequency centres and filter shapes do not coincide with the peaks and dips in the measured response and parametric equalizers do not have enough bands to accurately correct the measured response. A computer controlled equalization scheme was devised to correct the effects of the loudspeaker position.

The room response can be equalized in the time domain or in the frequency domain, using various weighting and windowing functions. Equalization in the time domain can preserve or improve the overall phase response, while equalization in the frequency domain effectively only equalizes the minimum phase part of the response due to the presence of all-pass phase components in the very complex room response. It has long been known by loudspeaker manufacturers that the frequency magnitude response is the best single element to improve for increasing the fidelity rating of loudspeakers. While listeners may be sensitive to phase effects under certain conditions, the most benefit is obtained by equalizing the magnitude response. This can be combined with some form of impulse response equalization, but only in the limit where the sound reflections being canceled with "inverse" reflections do not impair the perceived quality. It is very important to check the effects on the perceived quality of any added digital equalization, as the ear-brain perception system of the listener is very sensitive to some forms of time aberrations and very insensitive to others in small room acoustics. Equalization which appears good to the eye when the impulse response is viewed may sound very poor to the listener. The sound field is three dimensional and varies in frequency , time and direction of arrival. A purely engineering type of approach to response equalization was generally judged unacceptable by listeners during the course of the Athena project.

The equalization used for the results shown in Figure 3 was based upon 3 years of research in the Athena project. First a target response is determined. For the illustration of the equalizer performance in this paper, a flat target was used with a 4th order low-pass at 18.5 kHz in cascade with a 4th order high-pass at 39 Hz. The band limit characteristic of this target in the low frequency range is required not to overpower the woofer below its natural cut-off frequency while the upper frequency roll-off is required to match the anti-alias filter characteristic due to the 44.1 kHz sampling frequency used. The anti-alias filter response was not compensated for in the measurements. A flat target response in the room is not, *a priori*, the desired target for optimum perceived quality, but was used here for simplicity. Figure 3 shows the result of the responses for 6 equalized loudspeaker positions, S0 to S5, at one speaker height. As can be seen by comparing figure 3 to figure 2, there has been a vast improvement in the response, which now almost perfectly matches the target. The small unequalized frequency response deviations from flat above 400 Hz which were due to loudspeaker's response have been corrected, while the large response deviations due to the loudspeaker-room interaction below 400 Hz have been virtually eliminated. The entire equalization process for 12 speaker positions is completely automatic, requiring no user decisions, and thus eliminates the current practice of user controlled equalization, which has often resulted in overcompensation and colored response.

4.0 Conclusion

It was shown that variations in responses due to loudspeaker position, measured and combined over 6 listener positions were of the order of ±11 dB. The loudspeaker position was shown to be the dominant effect in the measured response below 400 Hz. It was then shown that the CARC/NRC DSP based engine could reduce the variation in the measured response to ±2 dB, independent of the loudspeaker position. The computations required were accomplished without the need for user interaction, thus permitting full automation of the system. Any target desired could easily be matched. The very high performance of this system will allow us to proceed to the next generation of studio monitor loudspeakers.

5.0 Acknowledgments

The authors would like to thank the Athena project sponsors : the Canadian Audio Research Consortium (Audio Products International, PSB Loudspeakers International, Paradigm Electronics Inc. and State of the Art Electronik Inc.), the National Research Council of Canada and the Industrial Research Assistance Program.

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REDUCING THE VARIABILITY OF LOUDSPEAKER PREFERENCE RATINGS THROUGH DIGITAL EQUALIZATION

James G. Ryan

Institute for Microstructural Sciences, National Research Council of Canada,
Ottawa, Ontario, K1A 0R6

1.0 Introduction

It has long been recognized that loudspeaker location and room geometry are sources of variability in listeners' ratings of loudspeakers [2][3]. When the intent of a listening test is to compare different loudspeakers, the effects of these variables are reduced through proper experimental design. In this manner, subtle differences in sound quality can be revealed.

Unfortunately, the environments in which consumer audio products are used do not always conform with those of a controlled listening test. As a result, the sound quality delivered to the consumer may not reflect the fideltiy of the audio product.

This paper reports some key results of the recently completed Athena project, a collaboration between NRC and the Canadian Audio Research Consortium. The purpose of Athena was to investigate the interaction of loudspeakers and rooms and to develop a means to lessen the deleterious effects on subjective assessments.

2.0 Variability of Loudspeakers in Rooms

An experiment was designed to examine the variability of loudspeaker preference ratings in different positions of a variable-geometry listening room specially designed for this purpose. The experiment compared the same loudspeaker in different positions as well as different loudspeakers in the same position.

Four loudspeakers, consisting of two different pairs, were placed at four different positions throughout the room as shown in Figure 1. The anechoic frequency responses of the loudspeakers in each pair were matched to within 0.25 dB. The two loudspeakers in a matched pair, therefore, were considered identical.

Listeners sat in a low-backed swivel chair which was rotated as required, to face the activated loudspeaker. The loudspeakers' positions and identities were hidden using an acoustically transparent but visually opaque screen. Of the 13 listeners participating in the experiment, only 4 had previous experience in listening tests.

The experiment was a four-way multiple comparison test where each listener performed four simultaneous ratings. Each trial consisted of a presentation of four selections of contemporary and classical music presented in a random order. Listeners could switch between the different loudspeakers at will and were asked to give separate ratings for each musical program. At the end of each trial, the listener left the room so that the loudspeaker positions could be changed and their loudnesses balanced. Six trials were conducted.

Listeners were instructed to rate the loudspeakers using a 10-point preference scale. A preference rating of 1 indicates that the listener "really disliked" the stimulus, wheres a rating of 10 indicates that the listener "really liked" the stimulus. The listeners were also

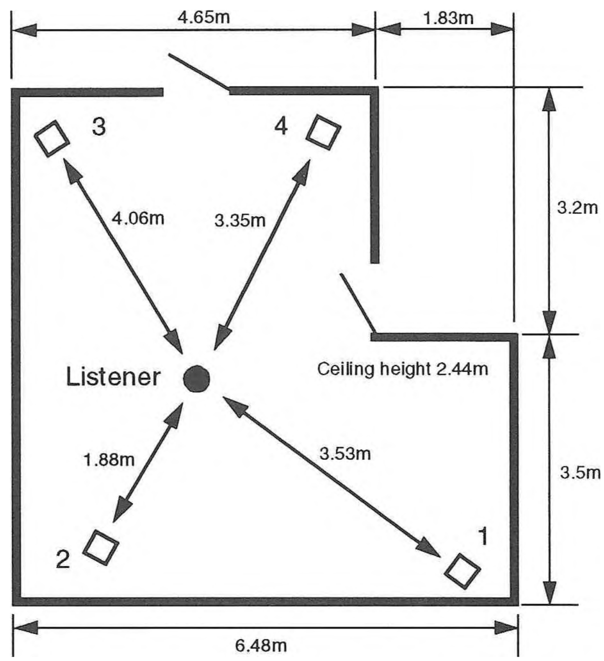

Figure 1. Layout of listening room illustrating listener and numbered loudspeaker positions.

encouraged to separate their ratings using the following guidelines:

Point Spread	Meaning
>2	strong preference
1.5 to 2	moderate preference
0.5	slight preference

Preference ratings were collected using a computer-controlled apparatus. Data were then analyzed using a repeated measures MANOVA model with SuperAnova 1.1. There are two significant findings from this analysis.

First, the most significant factor influencing listener preference ratings was loudspeaker location ($p=0.0001$). In fact, the listeners demonstrated a remarkable agreement in their preference for position 4 over position 1 or 2. The mean preference ratings for each location are shown in Figure 2.

The second finding is that there were no significant differences in listener preferences between loudspeakers ($p=0.4269$). That is, when measured in different room positions, the differences between the loudspeakers became insignificant.

3.0 Reducing Variability through Equalization

A second experiment was designed to examine the effect of a digital equalization scheme upon the variability of listener preference ratings across different room positions. The equalization is judged beneficial if it is capable of reducing this variability.

In this experiment, each member of a pair of well-matched loudspeakers were placed in two different room positions. Positions 1 and 4 were chosen because they were judged by many listeners as the most and least preferred positions, respectively.

The experiment was a four-way multiple comparison test where each listener performed four simultaneous ratings: two positions each with and without equalization. Each trial consisted of a presentation of five selections of contemporary music presented in a random order. Listeners could switch between the four stimuli at will and were asked to give separate ratings for each musical program. The order of presentation of the stimuli were randomized for each trial. A total of three trials was conducted. Listeners were instructed to rate the loudspeakers using a 10-point preference scale as in the previous experiment. Six of the ten listeners participating in this experiment had previous listening experience at NRC.

Preference ratings were collected using a computer-controlled apparatus. Data were then analyzed using a repeated measures MANOVA model with SuperAnova 1.1.

Data analysis shows that, even with equalization, loudspeaker location remains the most significant factor influencing listener preference ratings ($p=0.04$). As in the previous experiment, location 4 is preferred over loctation 1.

It was encouraging to also find that, on average, the equalized stimuli were preferred over the unequalized ones, however, the effect did not reach statistical significance ($p=0.1725$).

A contrast between location 1 equalized and location 1 unequalized was highly significant ($p=0.0053$). A similar contrast for location 4 did not reach statistical significance ($p=0.1693$). This is illustrated in Figure 3 which shows the mean preference ratings for position 1 and 4 with and without equalization. Note that the mean ratings for the equalized locations are close and within each others confidence interval while the unequalized means clearly differ.

4.0 Conclusions

The effect of loudspeaker placement upon listener preferecne ratings has been demonstrated. It was shown that the location of loudspeakers within a room can have a larger impact upon preference ratings than the type of loudspeaker. It was also shown that proper equalization can be used to reduce the variability of listener preference scores across different loudspeaker positions. Since the equalization scheme was implemented in realtime for these experiments, it has real potential for use as a consumer product.

5.0 Acknowledgements

The results in this paper were previously reported at the 12th International Conference of the Audio Engineering Society, Copenhagen, June 28-30, 1993 [1]. The author would like to thank the sponsors of this research: the National Reseacrh Council of Canada, the Industrial Research Assistance Program and the Canadian Audio Research Consortium (Audio Products International, PSB Loudspeakers International, Paradigm Electronics Inc. and State of the Art Electronics).

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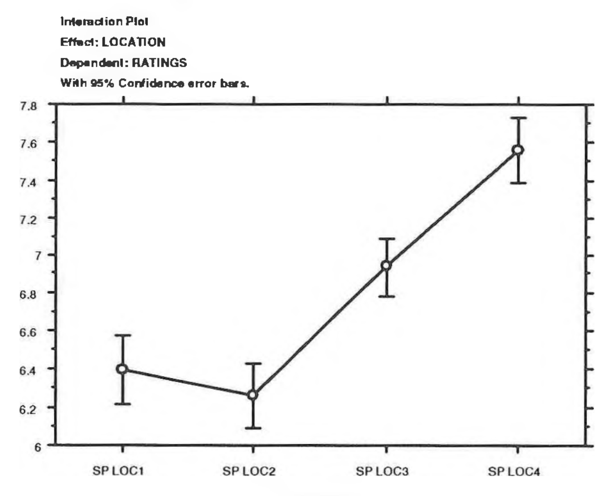

Figure 2. Mean listener preference ratings for various loudspeaker locations.

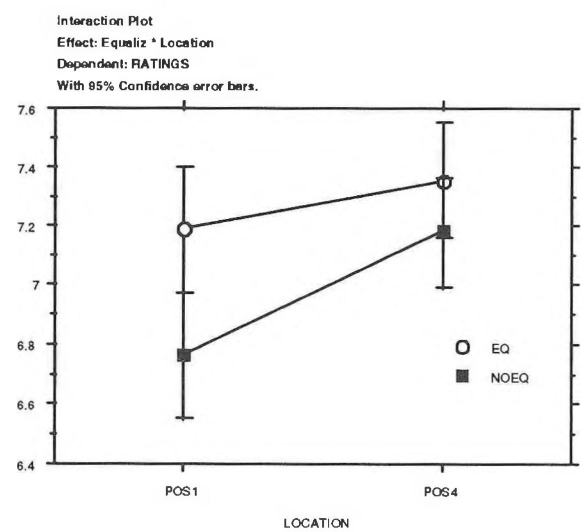

Figure 3. Interaction between equalization and loudspeaker position.

Ambisonic Sound for the Masses

by Jeffery S. Bamford
Audio Research Group, Department of Physics
University of Waterloo, Waterloo, Ontario
N2L 3G1

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In the past decade, television broadcasts have increasingly been available in stereo. The availability of stereo Video Cassette Recorders has also increased interest in home theatre. Films have used multi-speaker arrays to create better imaging and this has filtered down into videotapes of those films by using Dolby Surround encoding.

When a recording is made using Dolby Surround, four signals are encoded into two channels for storage. Upon playback, a Surround Sound decoder is required. Dolby Surround certainly sounds different from stereo but offers little imaging for rear sounds.

An Ambisonic system uses three or more channels to encode the sound for reproduction. By using more channels of information, a better reproduction of the image is obtained. Ambisonic systems also easily allow for the addition of more speakers. At the minimum, there must be as many speakers as channels. Adding more speakers will improve imaging.

The investigation is started by considering a plane wave. The plane wave is at an angle ψ with respect to the forward pointing x-axis. (In Ambisonics, it is customary for the x-axis to point forward and for the y-axis to point to the left). At a field (or listening) point, the plane wave can be described by $S_\psi = P_\psi e^{ikr\cos(\phi-\psi)}$, which is in polar coordinates (i.e. radius r at an angle ϕ). k is the wave number, ψ is the azimuthal angle the original plane wave makes with the x-axis, r is the distance from the origin of the co-ordinates (which is typically the centre of the listening area), i is $\sqrt{-1}$ and P_ψ is the peak pressure or amplitude of the wave.

The plane wave can be expanded in terms of spherical harmonics to be of the form: [1]

$$S_\psi = P_\psi(J_o(kr) + \sum_{m=1}^{\infty} 2\, i^m J_m(kr)[\cos(m\psi)\cos(m\phi) + \sin(m\psi)\sin(m\phi)]) \qquad (1)$$

where J_o and J_m are cylindrical Bessel functions of the first kind.

For a system with N loudspeakers that are equidistant from the centre location and in a regular array (i.e. equal angles separating the speakers), the plane wave signal from the nth loudspeaker is given by: [2]

$$S_n = P_n(J_o(kr) + \sum_{m=1}^{\infty} 2\, i^m J_m(kr)[\cos(m\phi_n)\cos(m\phi) + \sin(m\phi_n)\sin(m\phi)]) \qquad (2)$$

where P_n is the pressure from the nth speaker, and ϕ_n is the angle of the nth speaker.

If we sum over each of the N signals from the speakers and match this sum with the original plane wave, the following can be noted:

$$P_\psi = \sum_{n=1}^{N} P_n \qquad (3)$$

$$P_\psi \cos(m\psi) = \sum_{n=1}^{N} P_n \cos(m\phi_n) \qquad (4)$$

$$P_\psi \sin(m\psi) = \sum_{n=1}^{N} P_n \sin(m\phi_n) \qquad (5)$$

The above equations represent the matching conditions for the spherical harmonics. By matching all the orders of spherical harmonics, the original plane wave can be reproduced at the centre of listening area. Clearly, it is unfeasible to reproduce *all* of the spherical harmonics as there are an infinite number of them. For the present case only the first and second order limits are considered.

Equation (3) gives the first of the Ambisonic signals which is the pressure of the wave, which is denoted by W. The next order, m=1, gives the next two Ambisonic signals, $X = P_\psi\cos(\psi)$ and $Y = P_\psi\sin(\psi)$. The second-order Ambisonic signals are $U = P_\psi\cos(2\psi)$ and $V = P_\psi\sin(2\psi)$. With those five signals, a plane wave can be described up to second-order.

One way of analyzing this system is to look at the integrated wavefront error. This is done by integrating at a constant value of kr around the origin the difference between the original plane wave and the sum of the speaker signals. [2]

$$
\begin{aligned}
D \;=\; & \frac{1}{2\pi\,|P_\psi|}\int_0^{2\pi}\Bigg|\sum_{n=1}^{N} P_n J_o(kr) \\
+ & \sum_{n=1}^{N} 2\,i\,J_1(kr)\left(\cos(\phi)\cos(\phi_n)+\sin(\phi)\sin(\phi_n)\right) \\
- & \sum_{n=1}^{N} 2\,J_2(kr)\,P_n\left(\cos(2\,\phi)\cos(2\,\phi_n)\right. \\
& \qquad\qquad \left. + \sin(2\,\phi)\sin(2\,\phi_n)\right) \\
- & P_\psi\, e^{ikr\cos(\phi-\psi)}\Bigg|\,d\phi \qquad (6)
\end{aligned}
$$

With this equation, it is then possible to model various sized systems. By graphing the results for different numbers of speakers and layouts, it is possible to gauge how effective an Ambisonic Sound System could be. The above equation is for a second-order (five channel) Ambisonic System. By dropping the terms involving $\cos(2\psi)$ and $\sin(2\psi)$, the model is correct to first-order.

In order to compare the Ambisonic results with regular stereo and Dolby Surround, equations for these systems are required. For the stereo case, with speakers at $\phi_1 = -\phi_2$ and with $L = P_1$ and $R = P_2$, the signals are: $P_n = \frac{1}{2}\left(\frac{X}{\cos\phi_n} + \frac{Y}{\sin\phi_n}\right)$. [2]

Stereo only matches the velocity of the plane wave but *not* its pressure. Dolby stereo adds three more speakers driven as follows: a centre speaker with $\frac{1}{\sqrt{2}}(L+R)$ and two rear channels as $\pm\frac{1}{2}(L \mp R)$. It should be noted that Dolby Stereo reproduction is not quite this simple. For example, Dolby Surround Systems put a delay of a few milliseconds on the rear speakers. The rear signals are also limited to 7kHz. [3]

Ambisonic systems have as a general speaker signal: [4]

$$
\begin{aligned}
P_n \;=\; & \left(W + 2\cos(\phi_n)X + 2\sin(\phi_n)Y\right. \\
+ \; & \left. 2\cos(2\phi_n)U + 2\sin(2\phi_n)V\right)/N \qquad (7)
\end{aligned}
$$

where for a first-order system U and V are zero.

As a test of the various systems, a plane wave was modeled at an angle of 15 degrees. For reproduction, each of the systems would have to recreate a plane wave at 15 degrees. The stereo system had speakers placed at ±30 degrees. The Dolby System had the two speakers as in the stereo case, plus a centre channel and two rear channels at 150 and 210 degrees (i.e. 180 ± 30 degrees). The Ambisonic systems had speakers placed at 30, 90, 150, 210, 270 and 330 degrees. The plots of D versus kr for the various systems are shown in Figure 1.

Both stereo and Ambisonic systems offer fairly good imaging for low values of kr. This would correspond to low frequencies or for locations close to the centre of the listening area. The Ambisonic systems are clearly much better at localizations near the centre of the listening area than is the stereo

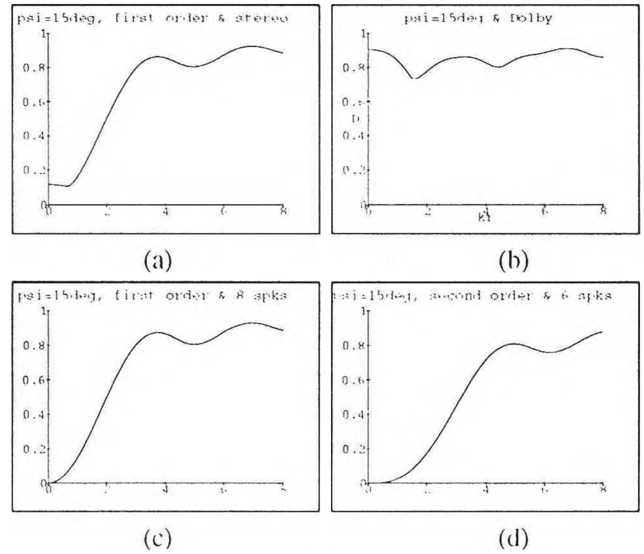

Figure 1: Plots of the D error versus kr for $\psi = 15$ for (a) Stereo (b) Dolby (c) Ambisonic (first-order) with six speakers and (d) Ambisonic (second-order) with six speakers

system. Dolby offers very little imaging anywhere. This is understandable as the Dolby System is mainly for ambient effects. The farther one gets from the centre of the listening area, the worse all four systems get. At kr=8 there is very little difference between the four systems.

The Ambisonic systems offer two important improvements over stereo and Dolby methods. The first is that they offer better imaging. An ambisonic system is able to image over 360 degrees, for example. The second benefit is that the effective area of listening is increased.

In conclusion, the second-order Ambisonic system offers improved imaging over a wider area than the first-order system and is suitable for larger rooms. A first-order system is better suited for the smaller home enviornment.

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An Overview of Audio Technologies in Teleconferencing: From Source to Receiver and Back

Bradley Basnett
Teleconferencing Department
Bell-Northern Research
Ottawa, Ontario

INTRODUCTION

In this paper, a typical round-trip teleconference audio path will be described. Figure 1 shows the most common components along the teleconference voice path.

SOURCE AND ROOM INTERACTION

A teleconference participant's mouth is located approximately 0.5-1.0 m from a surface mounted microphone exhibiting a half-hemispherical cardioid polar pattern. The acoustic energy arriving at the microphone is comprised of direct and reflected energy. The reflections degrade the intelligibility and overall transmission quality of the talker's voice. The arrival time and amplitude of each reflection are dependent upon the proximity and composition of the room boundary surfaces. Fortunately, the directional nature of the microphone reduces the pick-up of off-axis reflections. To further reduce the impact of the reflections, the well designed teleconference suite will be outfitted with strategically located absorptive treatment. Suggested acoustic treatments for teleconference suites are given in [1,2].

Early /Late sound ratios are an indicator of acoustic clarity [3]. Early/Late ratios with an early time limit of 50 ms have been measured to be 25-30 dB for a cardioid microphone at 0.5 m from an on-axis talker in a teleconference suite with a average decay time of 350 ms [2].

NOISE

Room ambient noise is an unwanted traveler on the teleconference audio path. Teleconference suites should not have ambient noise in excess of NC-30. The largest contributor to ambient noise is usually the HVAC system. Meeting tools such as overhead projectors and computers can increase the ambient noise by 5-10 dB . Often, these meeting tools are located on the conference table in close proximity to the teleconference microphones. Every effort should be made to reduce HVAC noise and to select meeting tools with quiet cooling systems. The conference room partitions should have an STC rating of 50 dB to insure that outside noise does not interfere with the teleconference.

In addition to the audible impact on the near-end and far-end participants, room ambient noise degrades the convergence performance of acoustic echo cancellers [4]. Fast converging acoustic echo cancellers are essential for full-duplex conversations involving teleconference suites.

Recently available teleconferencing systems have incorporated DSP based noise suppression. Active noise control for rooms is also available [5].

ELECTROACOUSTIC SYSTEM INTEGRATION

Microphones are positioned in the teleconference suite so that each participant is within +/- 30 degrees of the most sensitive axis of the cardioid microphone. A typical suite may have 6-8 surface mounted cardioid microphones. The transmission of room noise and acoustic echo is reduced by passing each microphone signal through an automatic microphone mixer. The mixer selects the microphone that it will turn "on" based on acoustic level and arrival time of the talker's voice. The mixer will turn "on" additional microphones as required. Care must be taken with the amount of attenuation applied to the "off" microphones as it negatively impacts the adaptation performance of acoustic echo cancellers. "Off" attenuation of 8-10 dB has proven to be effective.

SIGNAL PROCESSING

Subsequent components in the audio path may contain automatic gain control (AGC). The AGC circuitry attempts to bring the level of audio signal from the microphone to a pre-defined target value. This target value is chosen to ensure that a quiet talker and a loud talker are passed to the transmission circuitry at a nominal level.

The majority of teleconference systems operate partially or completely in the digital domain past this point in the audio chain. The audio signal from the microphone mixer is digitized and processed. Currently, the amount of processing that occurs in this portion of the audio path varies widely. One of the more elegant methods [6] will be examined.

In aid of later echo-cancellation, echo-suppression, double-talk detection, and 7 kHz transmission bandwidth, the microphone mixer signal is sampled at 16kHz and passed to a bandpass filter bank. The bandpass filters are numerous (29) and narrow (250 Hz). These bands are then subsampled at 1kHz. The subsampling enables efficient DSP to be performed.

The subsampled microphone signal then serves as one of the inputs to the acoustic echo canceller. The echo canceller is in the form of an adaptive filter. The adaptive filter subtracts that portion of the microphone signal that contains energy related to the receive signal as radiated by the loudspeaker. The other input to the adaptive filter is the subsampled version of the receive signal that is eventually radiated by the loudspeaker.

The loudspeaker(s) is typically positioned to provide good coverage of the seated teleconference participants while being off-axis to the teleconference microphones.

The size (number of taps) required of the adaptive filter is related to the decay time of the room in each band and any known non-linearity in the system [7]. The echo canceller described above can reduce the resultant acoustic echo by approx. 20 dB. In the case of audio for low-bit rate videoconferencing, transmission delays of up to 500 ms are inserted at the transmit codec to achieve lip-synch. 20 dB of echo reduction is not sufficient under this condition. Non-linear echo suppression is required to further reduce the acoustic

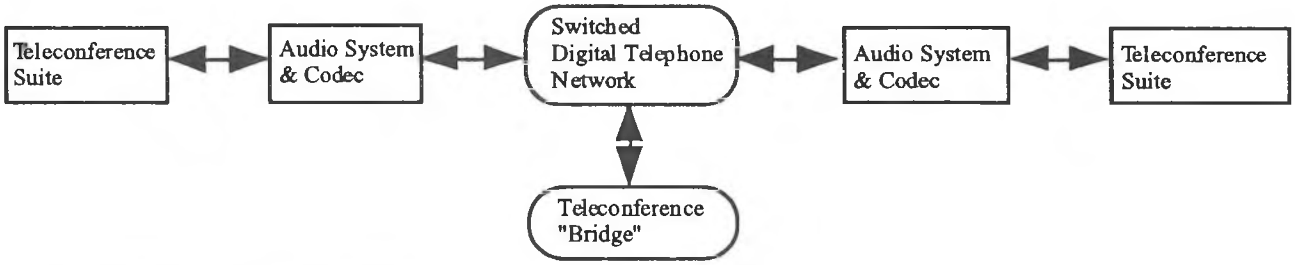

Figure 1. Typical Teleconference Audio Path

echo to an acceptable level. The suppression is in the form of subband transmit gain reduction. When the algorithm determines that the signal at the microphone is largely comprised of signals from the loudspeaker, the gain of that particular transmit subband is reduced.

TRANSMISSION

The processed microphone signal is then prepared for transmission. In some cases, this means that it will be converted back to an analog audio signal and enter the public telephone network. The increasingly more common scenario, whereby the signal enters and exits the telephone network in digital form, will be described here.

In order to transmit digital audio over digital telephone lines, some form of data compression is required to reduce the number of bits per second to a manageable and cost-effective number. Data compression is achieved through redundancy and irrelevancy removal techniques. Most commonly available digital telephone lines are 56kbit/s (switched 56) or 64 kbit/s (ISDN BRI). There are several compression (coding) techniques for transmitting audio over the digital telephone network. Standardized algorithms are now seeing broader use but several proprietary methods are also in active service. Proprietary techniques usually offer lower bit rates via greater compression. Standards offer compatibility while proprietary methods offer lower portions of the data stream being used for audio thus freeing up more bandwidth for video in the case of low-bit rate videoconferencing. G.711 and G.728 are ITU standards for narrow band (300-3.4kHz) transmission, and require 64 kbit/s and 16 kbit/s respectively. G.722 is the international standard for wide band (50-7.0 kHz) coding and it requires 48 or 64 kbit/s. Wide band transmission has become commonplace due to the ever expanding digital telephone network. It should be noted that there are devices available that can combine multiple switched 56 kbit/s or ISDN BRI lines to create a virtual higher bit-rate channel for audio bandwidth up to 22 kHz [8]. These devices are known as inverse multiplexers.

Now that the digital audio is coded and compressed it can make its way into the digital network. One scenario has the audio transmission arriving at a similar teleconference system at the far-end. The other is the case where the teleconference has more than two participants and the audio passes from one site to another via a "bridge". This is known as a "multipoint" teleconference. The "bridge" must broadcast audio from the active talker's site to all the other sites. The "bridge" also operates in the digital domain and usually incorporates automatic gain control (AGC), automatic talker recognition (gating), line echo-cancellation, and automatic mixing.

Audio from the active talker's site arrives at the far-end(s) via the bridge. It is then decoded and used as one of the inputs to the echo canceller and to the input of a D/A converter for amplification and reproduction by a loudspeaker(s) in the teleconference suite. The loudspeaker in a video teleconference suite is usually co-located with the video monitors. The loudspeaker output will reach the listener directly and indirectly via the perimeter surfaces of the room . The Early/late ratio (50ms) is 10-12 dB for most seats in a 12 seat video conference suite.

Depending of the acoustic distance between the loudspeaker(s) and microphone(s), the room treatment and directional characteristics of the microphone and loudspeaker, some of the speaker's acoustic radiation may reach the microphone(s). The acoustic echo cancellation and suppression process described above must also be in place at the far-end to facilitate full-duplex communication.

CONCLUSION

Current teleconferences employ many audio components and processes to achieve high quality, wide band, digital transmission from one acoustic space to another.

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Building a recording studio in a bank! A practical examination of some of the acoustical and audio design considerations for a new multi-track digital facility for Montréal's *Les Disques Star.*

Timothy Hewlings and Jean-Luc Louradour
Résonance TJL Inc., PO Box 5020, Ville St-Laurent (Québec). H4L 4Z6 (514) 745-8180

Introduction:

The owner of Les Disques Star, M. André Di Cesare, is one of Québec's most successful pop record producers. His work has taken him to some of the world's best known studios, both in North America and in Europe, so, when he was finally able to realize a longtime dream and build his own facility, he had some very definite ideas about what he wanted from a studio.

Site selection:

One objective was to keep the facility centrally located, so new construction was ruled out as being prohibitively expensive. The space had to be large enough to house the offices for the 30 or so, employees of a burgeoning record company, as well as the studio, but remain small enough to retain a certain amount of intimacy. The client also wished to own the property, rather than lease it.

After having evaluated several locations, the "bank" was found - counters, vault and all. The 40 year old, 3-story building had housed the offices of a local credit union and had been unused for several years since more modern quarters had been built. Situated in the City of Verdun, adjacent to downtown Montréal (and with numerous tax advantages), it has a store-front entrance on a street which, although relatively busy, carries mostly local traffic plus a few buses and a truck from time to time, but is situated far from any flight paths. It is in the middle of a city block and shares party walls with a community centre on one side and a small store on the other. The clear height of the ground floor is 3.5 m and the floors are wood-frame construction on 2 x 10 in. joists, supported by steel H-beams and columns - less than ideal conditions for building a recording studio.

Design Criteria:

The main room would be intended, primarily, for recording rock and pop artists/groups, but with the capability of recording larger ensembles such as brass or string sections, if necessary. Sessions for recording basic tracks often include acoustic and amplified guitars, acoustic piano, live drums and singers, so isolation from instrument to instrument would be important. The control room (CR) would require a large listening window behind a state-of-the-art 128-input mixing desk. Ample space had to be provided behind the console to install synthesizers. Noise levels were specified at < R.C. 18 for the CR and < R.C.15 for the studio.

Construction:

A large section of the ground floor of the bank was removed and Studio A built at basement level, to give greater interior volume to the room (finished height is over 6 m). The interior shell is completely self supporting. However, structural constraints made it impossible to remove the entire floor, so the control room was built at street level, overlooking the studio. Two large iso-booths were built, one on top of the other, facing the CR window, giving the interior a horseshoe-shaped appearance. The lower booth can be further sub-divided by a hinged partition. Access to the upper booth is via a catwalk approximately 2 m wide, which may also be used as a recording platform. The upper booth also supports one end of the ceiling trusses. The vaulted ceiling is a series of trapezoidal shapes for added rigidity - noise from the offices above was a major concern - and sound diffusion. Another booth, large enough for a grand piano, was built under the CR. Two other masonry "bunkers" were constructed outside the studio space. These have permanently installed instrument amplifiers for heavy guitar and bass tracks. The storefront windows were maintained, and a section of the studio inner shell wall was constructed in glass-brick to permit daylight to enter. Acoustical treatment consists of a combination of semi-rigid fiberglass panels and Schroeder diffusers. RT60 during and after completion of acoustical treatment (measured in 1/3 octave bands with a MLSSA system) are shown in figure 1. The first measurement was made before the installation of the low-frequency (LF) absorbers and diffusers, but after most of mid- (MF) and high-frequency (HF) absorption was already in place.

The main CR is essentially an irregular octagon, with lateral walls and ceiling splayed to direct first reflections away from the listening point. Monitor loudspeakers are flush-mounted in the front wall. Audio processing gear is installed in wings on either side of the mixing desk. The tape machines, amplifiers and power supplies are installed in an adjacent machine room.

A major addition was made to the rear of the building to accommodate a mechanical room, a second CR and a client lounge. The second CR and studio are essentially parallelepipedic and are intended primarily for pre-production and overdubbing.

HVAC:

To achieve the required sound isolation and ambient noise levels, four separate HVAC units were used. A system on the roof serves the office areas on the second and third floors, while another is used for Studio A. A third unit feeds the main CR, the second CR and machine room and a fourth smaller unit takes care of the overdub studio. Figure 2 shows the results of some recent measurements of the systems in Studio and CR A. (The slightly elevated noise levels in the CR measurements around 60 Hz and above 2 kHz were caused by the console computer and some audio gear which could not be shut off.) The mechanical engineers and contractors did an admirable job.

Electronic installation:

The installation includes audio, video, MIDI, AES/EBU, loudspeaker and control cabling throughout the building; there are over 13 km of audio cable alone. Four main audio patch bays allow access to any recording space from either control room - with their attendant special grounding and isolation requirements. Both CRs can access either the 24-track analogue tape machine or the 48-track digital recorder via a series of multi-pin connectors and the 66 microphone lines are also accessible from either CR through simple multi-pin connector patches.

There is an extensive CCTV network within the building with television monitors in both CRs so that visual contact can be maintained with areas that are not visible from the mixing desks.

From his office on the second floor, the studio owner can monitor (audio or video) and communicate with either studio or control room.

Foldback system:

A proprietary foldback system was designed and built. It has a 32 x 24 matrix which can be accessed by both CRs simultaneously and can be split between the two. The matrix can output with 12 individual stereo mixes accessible from either control room which can be patched to any pair of headphones. Each pair of headphones has its own amplifier with volume control and the whole system is powered by a single, central power supply.

How does it sound?

Figure 3 shows the magnitude of a typical transfer function in the MF and HF measured at the mixing position. The solid line represents the left loudspeaker and the dotted line the right. Similarly, figure 4 shows the LF response. No room equalization was used. There are some anomalies caused by console reflections, which tend to disappear with some spatial averaging. Several mixers have been using the room for the better part of a year and have been universally satisfied with the results. As well, the loudspeaker designer from Finland has conducted his own listening tests and given the room his benediction.

Figure 1
RT60 - Studio A

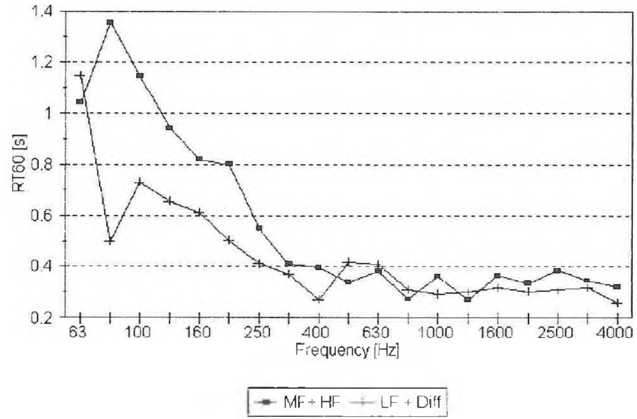
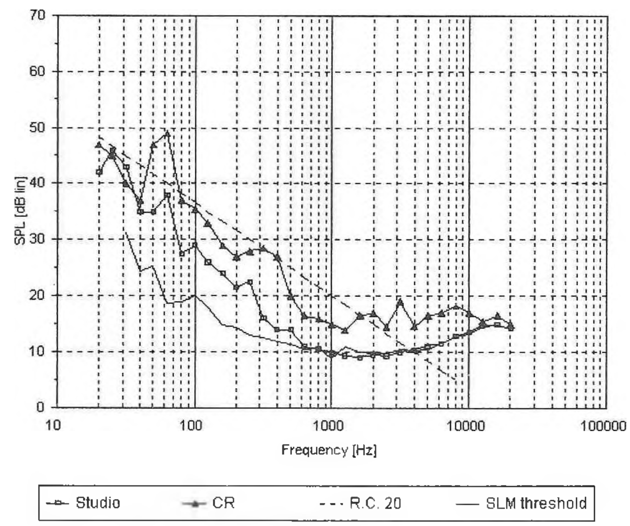


Figure 2
Ambient Noise



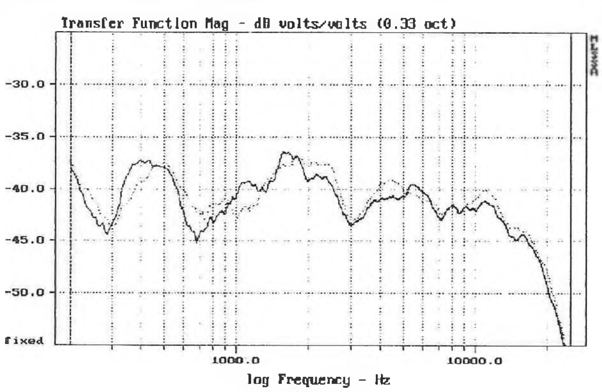


Figure 3

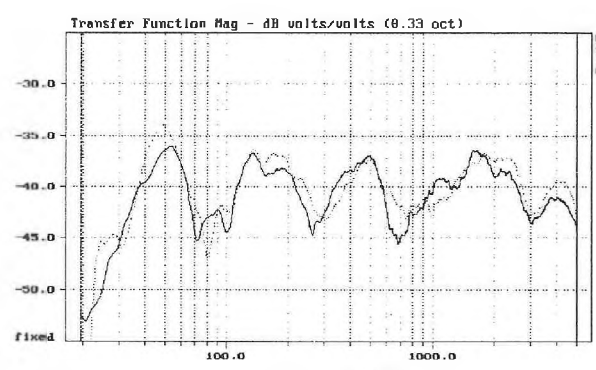


Figure 4

Sound System Modelling Software

Philip Giddings, P. Eng.
Engineering Harmonics
9 Edgewood Avenue
Toronto, Ontario
M4L 3G8

Introduction

Sound System Modelling Software is becoming commonplace in the design of sound systems for arenas, concert halls and other large rooms. There are a number of loudspeaker manufacturers that provide this software. They include: AcoustaCADD from Mark IV; CadP2 from JBL; Ease and Ease JR from Renkus-Heinz; and Modeller from Bose.

The software packages range anywhere from five hundred to several thousand dollars. All of the programs allow for basic room acoustics analysis and loudspeaker design. Most of the programs are specific in the loudspeaker components that they will allow you to use. However, the Renkus-Heinz system has a database that includes loudspeaker components from more than 20 different manufacturers.

The following is a description of the AcoustaCADD offering:

Entering the Model

A wire frame model of the room to be analyzed must first be entered. The wire frame model can be either open or closed. Rooms such as auditoriums and arenas are closed models as they have a roof (and a reverberation time). Football and baseball stadiums, with no roof, are open models. Obviously, reverberation time cannot be calculated in an open model.

The first step of the process is to determine all the planes needed to describe the model. For example, the walls, the ceiling, the floor, the balconies, the stage, the playing field, the ice, are all planes in space and have XYZ coordinates which are used to define the plane. Three XYZ coordinates define each plane.

Once all the planes have been entered, those surfaces that define the room must be identified and are called boundary surfaces. A boundary surface is defined by a plane and the planes it intersects.

With the planes and the boundary surfaces entered, the wire model can be viewed. Figure 1 shows the model for the Ottawa Triple A Stadium. There are a number of diagnostic devices to aid in correcting errors in entering the plane and boundary surface data. AcoustaCADD software has received some criticism about the difficulty of entering the wire frame model. The author has entered several models and, with experience, it takes less than three hours to enter a model for an arena or a stadium of less than 20,000 seats.

Acoustic Analysis

Once the wire frame model is entered, if acoustic analysis is desired, it is necessary to assign absorption coefficients to each of the boundary surfaces. A database exists for a number of typical surfaces; however, new data can be entered with its octave band absorption coefficients. Once all the boundary surfaces have been assigned an absorption value, the reverberation time is easily calculated. It can be calculated for any octave band. Another common acoustic analysis capability of these programs is ray tracing. Although this author doesn't use it commonly, it's available in the AcoustaCADD package.

Designing the Loudspeaker System

The initial step is to select, locate and aim the loudspeakers. The loudspeaker components include the manufacturer's model number of the horn, the driver and, in the case of AcoustaCADD, the filter module which would be used in the amplifier. AcoustaCADD includes loudspeaker data for Altec, EV and University products. The location is the XYZ co-ordinates in the model and the aiming is the elevation, azimuth and rotation. With this information entered, an isobeam is projected onto the surface of interest. The isobeams can be in the -3, -6 or -9 dB. down points of the horn at any octave band frequency from 125Hz. to 8kHz. The various suppliers of the software measure data in various ways for the isobeam and polar responses. AcoustaCADD uses measured data on five degree increments for every point on a sphere. Other suppliers of the software use ten degree increments and in some cases only measure data on a hemisphere and assume the other half of the sphere will be identical.

The projection of the isobeam onto the surfaces is where the usefulness of these programs becomes obvious. It becomes extremely obvious when the location of the loudspeaker, its type and the aiming have been properly selected. AcoustaCADD also allows a multiple isobeam mode where isobeams of six loudspeakers can be projected onto the model at a given time. Figure 1 shows two isobeams projected. This allows adjusting the relative aiming of various loudspeaker components. Once finished with this process the designer is confident that the aiming of the loudspeakers is accurate and the need to allow for field adjustments is minimized.

Once the loudspeakers have been entered and aimed in the model, we can then go to the next step which is to map the SPL. This involves specifying which surfaces are the critical ones to the sound system design. Typically, these include those where the patrons are seated. Once the surfaces have been selected, the resolution to be used in mapping the sound pressure level must be selected. Obviously, the density of the mapping grid determines the number of discrete points for which a sound pressure level must be calculated and has a large impact on the times taken by the program to determine a sound pressure level map.

Mapping the SPL can be time consuming as an great number of calculations are required. For each point selected on the grid, the program takes the contribution of every loudspeaker in the system and does a summation. This involves taking the sensitivity of each loudspeaker, the off-axis response and the loss with distance. The author uses a 486-66 PC and with 18 horns this calculation can take up to 20 seconds.

AcoustaCADD provides two forms of mapped SPL output. One is a colored map and the other is isobars. Either of these can be plotted at 1, 2, 3 or 4 dB per increment. See Figure 2 for an example of the output.

The performance of the loudspeaker system can be quickly determined in a mode of operation where a point on a surface is selected and the program then calculates the direct and reverberant sound pressure levels and the articulation loss of consonants. This applies to closed models and is obviously a key element to a good sound system design in a reverberant space such as an arena or a concert hall.

A Case Study

As seen in the figures, AcoustaCADD was used in the new Ottawa Triple A Baseball Stadium. A primary concern in the design of the loudspeaker systems was control of noise intrusion to a neighborhood just beyond left field. Several potential sound system designs were modeled and evaluated. The program was used to accurately compare the relative impact of the various designs at a distance of about 1000 feet from the stadium. The result of the final design is shown in Figure 3. With a sound system concept selected a detailed design was specified and tendered. The evenness of coverage of the installed system was good and no loudspeaker re-aiming was required during commissioning. There have been no complaints from the neighbours that the author is aware of.

Figure 1

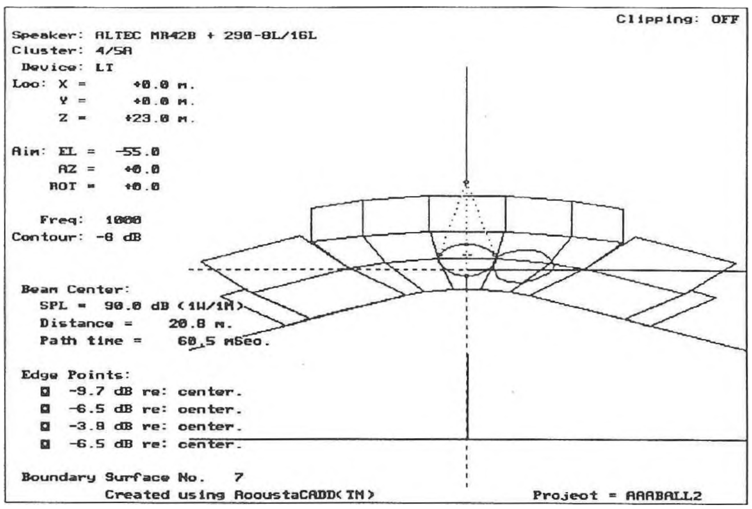


Figure 2

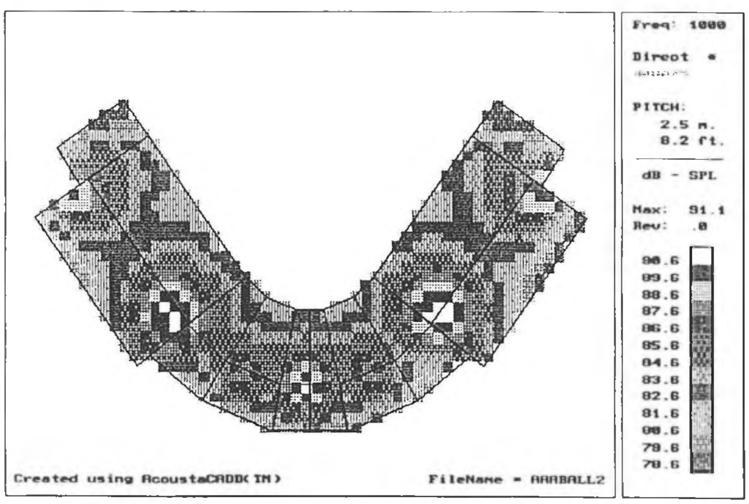
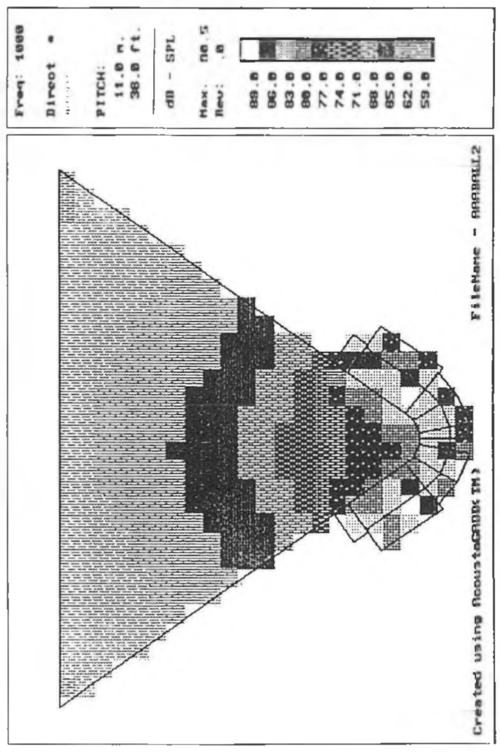


Figure 3



Room Acoustics Considerations for Loudspeaker Systems in a Large Reverberant Space.

John O'Keefe

Aercoustics Engineering Ltd.
50 Ronson Drive, Suite 127
Toronto, Canada, M9W 1B3

Introduction

The Galleria in Toronto is a monumental space with an internal volume of approximately 90,000 m^3. The acoustics in the building are comparable to very large rooms such as the cathedrals of Europe or domed stadia in North American and Japanese cities. As such, the Reverberation Times are three or four times longer than they should be for good speech intelligibility or musical clarity. This means that in most places in the room, useful early reflected sound is overwhelmed by detrimental late sound. Preferred Reverberation Times for speech and amplified music range from 0.7 to 1.5 seconds. Satisfying this criterion in a large venue such as the Galleria is simply not possible.

The Galleria has very low speech intelligibility and musical clarity levels. Like other very large spaces, perceived clarity decreases as the sound pressure level of the loudspeaker system is increased. To provide enough direct and early sound to listeners, a significant number of loudspeakers are required. The philosophy is that wherever one stands in the Galleria there should be a loudspeaker nearby. The loudspeaker(s) must be close enough that direct or early reflected sound is greater that the later reverberant sound. The danger with this approach however is that loudspeakers far away from the listener may generate sound that is interpreted by the ear as late or detrimental. This then is the fundamental dilemma associated with very large rooms: increasing the number of speakers means that some people will be exposed to better direct and early sound. For people located elsewhere in the room these same loudspeakers will introduce detrimental late sound.

Measurement Procedure.

The objective acoustic measurements we have performed in the Galleria include: Reverberation Time (RT60), Early Decay Time (EDT), and Distinctness (D50) and Speech Transmission Index (STI)

Measurements were performed with a MLSSA System manufactured by DRA Laboratories. The room was insonified with a dodecahedron source complete with 12 75 mm diameter loudspeakers and a single 12" loudspeaker in a separate bin. Room responses were measured with an omni-directional microphone at source-receiver distances doubling in distance, i.e. 1, 2, 4, 8, 16, 32 and 64 m.

At each location measurements were performed at four different source levels, increasing in amplitude at 10 dB steps. This was done to determine if increased sound levels had an effect the acoustics of the room.

Results

The results of our measurements are shown in Figures 1 and 2. For comparison, we have also shown the RTs in other large rooms such as St. Paul's Cathedral, London. The comparison confirms that the Galleria is similar to these very large spaces, none of which are known for good acoustics.

As pointed out above, practical experience in the Galleria suggests that speech intelligibility decreases as the sound level is increased. To investigate this phenomenon, a series of Speech Transmission Index (STI) measurements were performed at different sound levels. The results are not what some might expect. For the most part, speech intelligibility neither increases or decreases with sound level. The exception is the STI values for the lowest sound level where the ambient HVAC noise has affected the data.

The reason why the measurements do not agree with so called practical experience appears to be because we have used a single loudspeaker for the measurements as opposed to a distributed loudspeaker system like the one currently installed in the Galleria. A single source approach was chosen because it afforded the power of simplicity, not only for our measurements on site but, much more importantly, in our analysis that followed.

Analysis

In essence, the problem in a large room such as the Galleria is that there is too much late sound and not enough early sound. It would seem to make sense therefore to extract and examine the direct, early and late components of the sound measurements we have performed. This, by the way, is not a unique approach. It is the basis of the revised theory of sound in a room as proposed by Barron & Lee[1].

For the analysis, the following temporal thresholds have been assumed:

Direct	0 to 1 ms
Early	1 to 50 ms
Late	50 ms to ∞

The analysis was performed on broadband (unfiltered) data and the results are shown in Figure 2. Three simultaneous models are used to describe the sound in the Galleria, presented below with their R^2 values:

$$Direct = L1 - 15.43462r \qquad (R^2 = .96)$$
$$Early = L1 - 11.99859r - 1.74 \qquad (R^2 = .97)$$
$$Late = L1 - 2.717000r - 10.04 \qquad (R^2 = .92)$$

where: $L1$ is the sound pressure level 1 m. from the source
r is the distance from the source in m.

The first thing to notice in Figure 2 is that at short distances, that is less than 7 metres, the direct and early sound is greater than the late sound. If the early sound is louder than the late, one would expect good speech intelligibility. The STI measurements at less than 7 m indicate that this is in fact the case. Beyond 7 metres the late sound predominates and there is a corresponding decrease in speech intelligibility.

The second salient feature of Figure 2 is that the slope of the late sound is less than the slopes of the direct or early sound. This means that as one moves away from the source, the detrimental late sound persists while the important early sound dissipates. This begins to explain the reason why speech intelligibility and the overall acoustic impression can decrease as the sound level is increased.

The important difference between a single loudspeaker system and a distributed system with several loudspeakers is that the distant loudspeakers generate sound that a listener will interpret as late or detrimental.

To quantify the acoustical implications of a loudspeaker array, a simple mathematical model of the Galleria's acoustics has been developed. Using the three formulae shown above, the model calculates the 50 ms Distinctness coefficient (D50) for a given location depending on the number of loudspeakers and the distance from the listener to the closest loudspeaker.

The model shows that for 4 speaker array, spaced at 2 m intervals, a listener must be within 8 m of the nearest loudspeaker to obtain a D50 of 0.6 or better. If the spacing between the 4 speakers is increased to 10 m, the listener must be within 4 m of the nearest loudspeaker. When the number of speakers is increased to sixteen, the effect of loudspeaker spacing becomes insignificant for listeners located more than 2 or 3 m from the nearest speaker. Beyond that point D50 levels are unacceptably low.

This suggests the PA system design must ensure that no matter where one stands in the room, there will be a loudspeaker within 3 m (9 ft). To achieve this, a system of satellite loudspeaker stations has been proposed[2]. If implemented, it will be linked to the central sound system by a new infra-red broadcasting system similar to that used for hearing impaired systems.

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2. Philip Giddings, **Engineering Harmonics** Report, 17 January 1994.

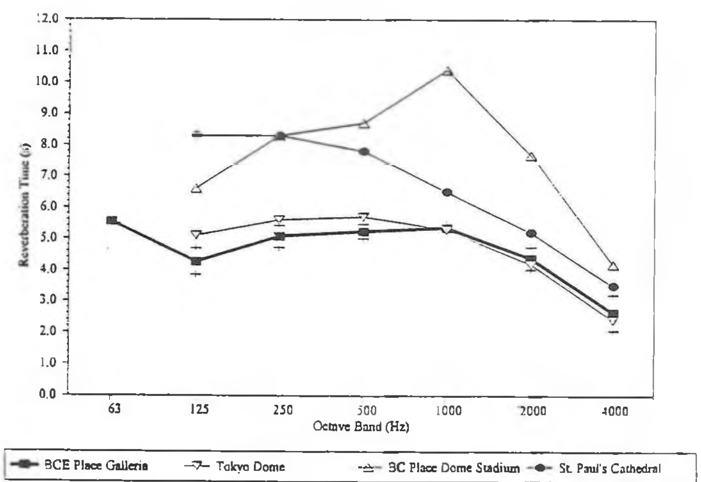

Figure 1 Reverberation Time vs. Frequency.

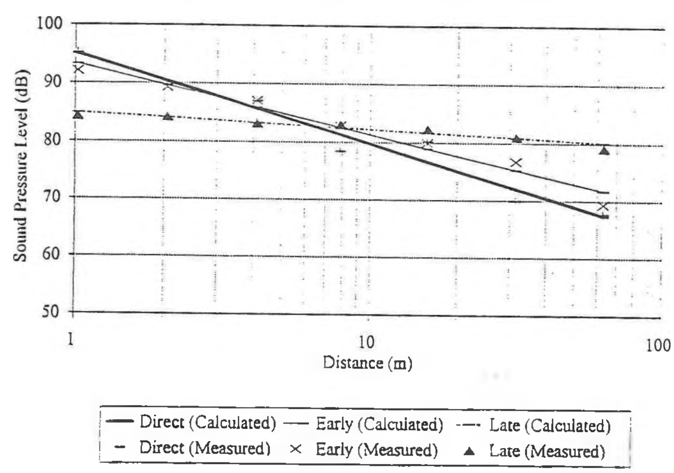

Figure 2 Regression Analysis of Direct, Early and Late Sound.

HOW ACOUSTICAL ENVIRONMENTS AFFECT PEOPLE

Murray Hodgson

Occupational Hygiene Programme and
Department of Mechanical Engineering
University of British Columbia
2206 East Mall, Vancouver, BC V6T 1Z3

1. ACOUSTICAL ENVIRONMENTS

The acoustical environment can be defined as the totality of sound to which an individual is exposed. Several generic points regarding the effect of acoustical environments on people are:

- the acoustical environment has a direct impact on normal-hearing and hard-of-hearing people;
- a non-optimum acoustical environment makes the world less accessible. This results in disability which, in turn, leads to handicap, especially for the hard of hearing;
- the magnitude of the impact / handicap depends on the exact characteristics of the environment, and on the individual impacted;
- this is an ergonomic/ecological issue - the environment must be adapted to the situation, taking into account human capabilities and the demands placed on people. There must be a good match between capabilities and demands.

What are the characteristics of acoustical environments that concern us? The main ones are:

- energy (loudness) - how much energy is contained in the sound (how loud is it)?;
- frequency (pitch) content - what frequencies (pitches) does the sound contain?;
- temporal variation (eg reverberation or echo) - how does the sound vary with time? How long does it take for the sound to die away?;
- direction - from what directions is the sound arriving?;

and, for a person with two ears:

- inter-aural differences - what is the difference between the sounds arriving at the two ears?;

and, in the case of a 'signal' (useful sound) in the presence of 'noise' (interfering sound):

- signal to noise ratio - what is the level of the 'signal' to that of the 'noise'.

2. AUDITORY FUNCTIONS

The act of hearing involves the following auditory functions, among others:

- detection - can the sound be heard?;
- discrimination - can the 'signal' be distinguished from the 'noise'?;
- recognition - can the signal be identified?; can the signal's meaning be understood?;
- localization - can the direction of the sound be perceived?

Here are some of the many complex acoustical activities that involve these functions:

- auditory scene analysis - the ability to contruct an image of a situation from auditory information;
- speech perception - the ability to understand speech;
- warning signal recognition - the ability to recognize warning signals;
- cocktail-party effect - the ability to 'pick out' one meaingful signal from a lot of noise.

3. CONSEQUENCES OF A NON-OPTIMUM ACOUSTICAL ENVIRONMENT

A non-optimum acoustical environment results in a mismatch between human capabilities and the demands placed on people. It may be detrimental to exposed individuals with respect to the following aspects of life:

- health
- safety
- communication
- social
- intellectual
- economic
- enjoyment

The effect may be significant for normal-hearing people, but is worse for the hard of hearing.

Let's look at this in more detail:

• Health	- fatigue, stress, anxiety - hearing loss
• Safety	- reduced ability to identify and respond to warning signals
• Communication	- compromised verbal communication - reduction of privacy
• Social	- compromised social development - compromised social interactions - isolation - reduced quality of life
• Intellectual	- compromised intellectual development - compromised learning in education
• Economic	- compromised (motor and intellectual) task performance - increased absenteeism at work
• Enjoyment	- annoyance, frustration - discomfort, dissatisfaction

Here are some examples:

- Industrial workshops
  - fatigue, stress
  - hearing loss
  - danger, accidents
  - compromised verbal communication
  - reduced productivity
  - annoyance, frustration
- Classrooms (students and instructors)
  - fatigue, effort
  - compromised verbal communication
  - compromised learning
  - annoyance, frustration
- Seniors' residences (staff and residents)
  - health decline
  - fatigue, stress, anxiety
  - compromised verbal communication
  - compromised social interaction, isolation
  - annoyance, frustration
- Movie theatres
  - compromised verbal communication
  - reduced revenue
  - annoyance, frustration

4. EFFECT OF HEARING LOSS

A hearing loss affects the auditory functions discussed above in the following ways:

- reduced sensitivity;
- reduced frequency resolution;
- reduced temporal resolution;
- reduced spatial resolution;

and, in the case of a unilateral hearing loss:

- reduced ability to separate signal from noise.

A hearing loss amplifies the impact of a non-optimum acoustical environment and the associated impairment / handicap. It amplifies the mismatch between capabilities and demands.

5. OPTIMIZING THE ACOUSTICAL ENVIRONMENT

The aim is to optimize the environment so that it is best adapted to the situation. The characteristics of acoustical environments must be matched to the human activity and auditory functions involved. The requirements are generally more stringent for hard-of-hearing than for normal-hearing people.

How to optimize acoustical environments is either quite well known (especially for normal-hearing people) or is the subject of research (especially for hard-of-hearing people).

The acoustical environment is optimized by administrative and engineering control measures.

Models exist for predicting acoustical environments. These can be used to optimize them.

A final practical point: it is crucial that we demonstrate that improving the acoustical environment is cost effective.

PERCEPTUAL AND COGNITIVE FACTORS AFFECTING SPEECH UNDERSTANDING

Bruce Schneider
Department of Psychology
University of Toronto

As Figure 1 illustrates, to understand speech in noisy situations, listeners must not only "extract" the speech signal from the background noise, but also determine the meaning of the words and phrases in the extracted signal, integrate this information with past knowledge, and store it for future use. Psychoacousticians, audiologists, and auditory physiologists typically have directed their efforts towards understanding the perceptual processes that are involved in signal extraction. Linguists and cognitive scientists, on the other hand, have focussed their efforts on how the words and phrases in the speech signal are processed linguistically, and how this processed information is integrated with past knowledge and stored in memory for future use. The endeavours of both groups have enhanced our understanding of how listeners process speech signals. However, for the most part, both groups have ignored the complex and often subtle interactions (indicated by the presence of a feedback loop in Figure 1) that occur between these two levels of processing.

To illustrate why such interactions cannot be ignored, consider the difficulties that the elderly often experience when attempting to understand speech in noisy conversational settings. These difficulties cannot be attributed solely to changes in auditory sensitivity because clinical tests of the hearing acuity of many of these individuals indicate that their hearing status is in the normal range. It could be that these clinical tests are insensitive to the perceptual deficits that are responsible for these comprehension difficulties, or that these difficulties are a consequence of age-related changes in more central or cognitive processes. What is more likely, however, is that age-related declines in bothperceptual and cognitive factors interact to produce comprehension difficulties.

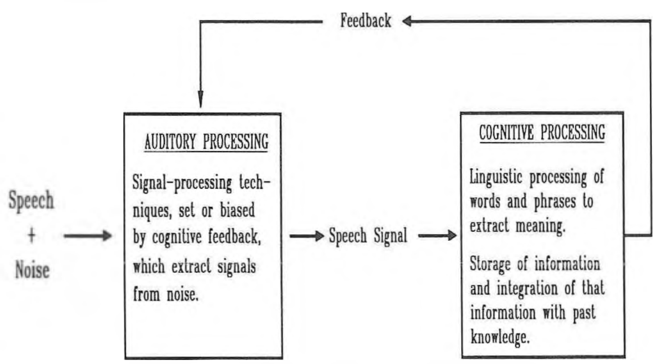

Figure 1. Simplified model of speech processing.

Age-related changes in perceptual factors such as the ability of listeners to use binaural cues to unmask signals in noise (Pichora-Fuller & Schneider, 1991, 1992), or the ability to temporally resolve two sounds (Schneider, Pichora-Fuller, Kowalchuk, & Lamb, 1994; Schneider, Speranza, & Pichora-Fuller, 1994) could contribute to difficulties in understanding speech. For example, old subjects with good hearing appear to have poorer temporal resolution than young subjects. Young and old subjects were asked to discriminate between two tone pips and a continuous tone of the same total energy and duration (Schneider, Pichora-Fuller, Kowalchuk, & Lamb, 1994, Schneider, Speranza, & Pichora-Fuller, 1994). In general, the size of the temporal gap required for discrimination was almost twice as large in older subjects. Moreover, as Figure 2 indicates, these thresholds are independent of the subject's audiometric threshold. Thus there are significant age-related declines in temporal acuity that are unrelated to the degree of sensorineural hearing loss. Because speech understanding requires listeners to process the rapid fluctuations in amplitude that are characteristic of speech, declines in temporal acuity may contribute to the speech understanding difficulties of the elderly by effectively degrading the speech signal and producing errors in speech perception.

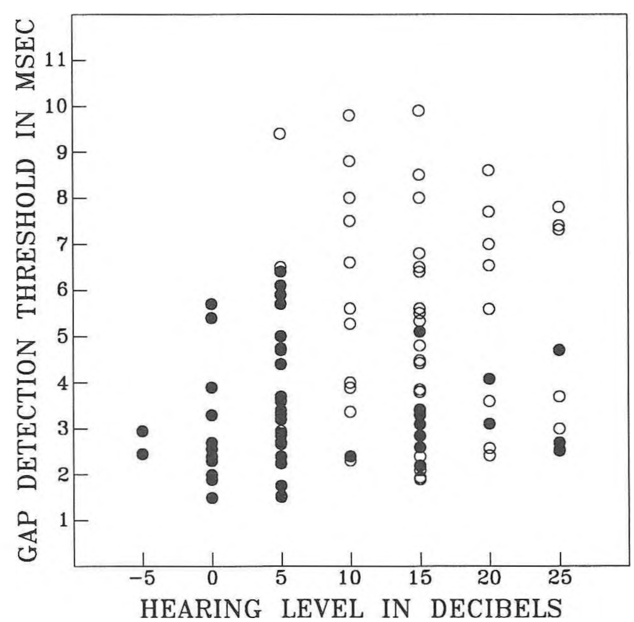

Figure 2. Gap detection thresholds for young (filled circles) and old (unfilled circles) subjects as a function of their hearing level.

Although age-related declines in perceptual processing may lead to errors in speech perception, part or all of the information lost in the early stages of auditory processing may be partially or completely recovered by higher-order processes. Listeners who "mishear" a word in a sentence can often recover the word from the context provided by the sentence. If the elderly, because of age-related declines in perceptual processing, are "mishearing" more of the words and phrases in everyday conversational settings, then they are forced to rely more often on context than their younger counterparts. Indeed, the elderly require a higher signal-to-noise ratio than the young in order to recognize words that cannot easily be recovered by the sentence context (Pichora-Fuller, Schneider, & Daneman, submitted). Therefore, in many conversational settings, the elderly have to depend on context whereas the young do not. As a result, speech processing is more effortful for them, and there is some evidence that they are more effective than young adults in utilizing the contextual information (Pichora-Fuller, Schneider, & Daneman, submitted), perhaps because they are more often forced to rely on it.

When the elderly have to use context to recover information lost or distorted by external or internal noise, it is likely that they do so at a cost. Although cognitive psychologists have seldom considered the toll placed on cognitive resources when listening occurs under degraded conditions, it is conceivable that when the signal is degraded, more resources will be required for listening, which might deplete the cognitive resources available for linguistic and cognitive processing. For example, speech understanding draws heavily on working memory, a system responsible for both the processing and the temporary storage of information during the performance of complex cognitive tasks (Baddeley & Hitch, 1974; Daneman & Carpenter, 1980, 1983; Craik et al., 1990). In order to integrate successively heard words, phrases, and sentences into a coherent representation, listeners must have access to the results of earlier processes. In addition to having access to previously stored information they must also be able to simultaneously manipulate the stored material during ongoing processing. If, in addition, to these tasks, working-memory resources are required to remove the ambiguity and recover the information in the signal that has been lost during perceptual processing, comprehension would suffer. For example, diverting cognitive resources to the task of recovery, may make them less efficient at storing information in memory (Pichora-Fuller, Schneider, and Daneman, submitted). Thus, perceptual deficits can have serious consequences for the linguistic and cognitive processing of speech.

If, in addition to perceptual deficits, the elderly also experienced difficulties when forced to divide their processing resources between the task of recovery of lost information, and comprehension of the message, they would be particularly disadvantaged. In such a case, perceptual and cognitive deficits would interact in such a way as to make it very difficult for some of the elderly to understand speech in everyday listening situations.

If, in order to function well in ordinary conversational settings, the elderly must divert cognitive resources to the recovery of information lost during perceptual processing, they will have fewer resources available for higher-order cognitive functions. Thus, they might not be as efficient or as fast at integrating this incoming information with past knowledge, or at storing it for future use, and therefore appear to have a comprehension deficit. At the very least, listening is likely to be more effortful for elderly listeners which might explain why some of them often observe that it is not so much that they cannot understand what is being said in these settings, but that they find it very fatiguing to do so.

Thus, in order to effectively study the speech understanding difficulties of the elderly, we are forced to consider the speech processing system as a whole because of the subtle interactions that can occur between perceptual and cognitive levels of processing when one is listening to speech..

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Engineering Aspects of Assistive Device Technologies for Hard of Hearing and Deaf People

Charles A. Laszlo, Ph.D.
Department of Electrical Engineering
University of British Columbia
2356 Main Mall, Vancouver, B.C. V6T 1Z4
Ph: (604) 822-3956 Fax: (604) 822-5949 e-mail: laszlo@ee.ubc.ca

Hearing aids are widely recognized as important prosthetic devices which are essential to the everyday functioning of hard of hearing people. It is less generally understood that hearing aids alone cannot satisfy all the requirements of the hard of hearing population. This population consists of people of all ages, with varied occupations, interests, hearing loss and needs. Their common characteristic is that while they suffer lesser or greater degrees of hearing loss, their primary mode of communication is speech.

Another, much smaller, group are deaf persons who communicate primarily by sign language, and who do not or cannot make use of audible speech. Consequently, the technologies required by the hard of hearing and deaf populations are different in many ways, although some devices are used by both.

Technical aids have a profound importance in the life of hard of hearing and deaf people. Many of them need several different devices to function independently in the home, at work, and at leisure. Even those who ordinarily do not use hearing aids may require specialized devices to use the telephone or to communicate in groups. The large variety of devices used by hard of hearing and deaf people may be divided into four major groups: assistive listening devices (ALDs), visual communication aids, alerting and sound recognition systems, and telecommunication devices and systems.

Acoustical aspects are important in the engineering design and application of each of these systems. These aspects may be quite complex, and many environmental, architectural, technological, and user-dependent issues arise.

Hearing aids. To appreciate the problems and the need for assistive devices we must first examine hearing aid technology. The primary purpose of hearing aids is to improve speech communication ability. In general, hearing aids receive sound waves, modify them, and then retransmit them to some part of the auditory system. The original concept of hearing aids is based on the view that hearing loss may be characterized by decreased sensitivities from the normal at distinct frequencies of sound in the range of hearing. The aim of hearing aids has been primarily to provide amplification such that at these distinct frequencies the amplified sound counteracts the effect of decreased sensitivity to sound. If this can be accomplished then the aided ear is considered to be approximately as sensitive as the normal ear.

During the last 40 years this simple concept has been extended by the addition of various improvements, including amplification in selected frequency ranges, compression of the intensity range of sounds, noise reduction, directional sound reception, and rugged and reliable components.

While these developments have resulted in better performance and in improved speech comprehension for some hard of hearing people, the hope for the all encompassing solution has never materialized. The main reason for this is that sensitivity loss is not the only problem. Many hard of hearing people also suffer from loss of ability to discriminate the details of speech sounds. This loss of ability means that hard of hearing people cannot recognize important speech components resulting in the partial comprehension of speech.

In particular, the goal of creating a few types of instruments performing amplification and relatively simple processing able to improve speech comprehension for most, if not all, hard of hearing people has remained elusive. It is now generally accepted that the hard of hearing experience a great variety of different auditory performance problems, that no simple processing of the auditory signal will suffice, and that complex signal manipulation must be tailored individually to achieve the best possible improvement in speech comprehension. While rapid developments in digital electronics and digital signal processing provide the tools needed for complex signal manipulation, we are still lacking the physiological, medical and audiological knowledge that could guide engineering design to create hearing aids that can significantly deal with the comprehension problem.

This problem for hard of hearing people is vastly compounded by degradation of the acoustical signal between the speaker and the listener. Such degradation eliminates some speech clues, resulting in additional loss of speech comprehension. This makes hard of hearing people non-functional in many common listening situations where the acoustical conditions are less than ideal.

Assistive Listening Devices (ALDs) are communication systems that bypass the acoustic pathway between speaker and listener. In general, the input to such systems consists of one or more microphones connected to a transmission system. On the receiving end, specialized receivers are coupled to the listener's hearing aid or to earphones for those who don't normally use hearing aids.

Assistive listening devices acquire their input either from a microphone or from another signal source (e.g. television set). The objective is to ensure that the speech signal suffers no degradation due to echoes, distortion or any kind of environmental and instrumentation noise. Since the acoustical interference is the most difficult to control, the length of the acoustical pathway is minimized by placing the microphone as close to the speaker's mouth as possible. The rest of the transmission is non-acoustical until the very end where the signal is transduced and fed to the listener's eardrum. Different transmission methods are available that work well under appropriate conditions, although each has its limitations and drawbacks.

Induction loop systems consist of a number of turns of wire connected to a suitable amplifier. The loop is placed so that it will enclose the area in which hard of hearing users will be located, for example a group of chairs in a lecture hall. Many hearing aids can receive transmission from magnetic induction loops directly via built-in pick-up coils. Induction loops are simple and reliable, but require either permanent installation or the placing of wires around the room each time it is used. In addition, the magnetic field cannot easily be confined to one room, so that cross-talk and privacy problems arise.

Infrared systems designed for large areas consist of a preamplifier/control unit and an amplifier/driver that powers one or more arrays of infrared emitting diodes mounted on a tablet-like enclosure. In use, the speaker talks into the microphone, generating the signal that modulates the high-frequency carrier driving the infrared emitters. The infrared light then travels to individual receivers which convert the light back into an electrical signal, demodulate and amplify the sound component, and provide acoustical, magnetic or direct input to the hearing aid of the hard of hearing person. These systems work quite well, but they require careful installation (either permanent of temporary), governed by the need to provide even illumination by placing the infrared emission panels in appropriate elevated locations in the room. The control and driver units must be wired to each of the panels and in turn the microphone(s) must also be wired to the control. Small portable IR units requiring no setup, wiring or electrical connections to use became available recently. These devices combine the power source, the microphone, the electronics and the infrared emitters in one package. Placed in the middle of a conference table, or on a speaker's rostrum, these devices can serve up to 50 people.

FM communication systems usually consist of a microphone, an FM transmitter, and FM receivers which are worn individually by every user. Sometimes the microphone and the transmitter are combined into a "wireless microphone". These systems require minimal technical expertise to set up, and are very small, usually the size of a cigarette pack. A major problem with these systems is that they provide no security, as the signal may be picked up outside the confines of the room or meeting hall. In some applications this is not a problem, for example churches, but in many meetings, lectures, and presentations the possibility of being overheard is not acceptable. For this reason FM systems are not nearly as popular as they deserve to be, and because of the relatively small number of systems in use, their price remains high. Additional concern is to satisfy the sound reproduction quality requirements which are higher for hard of hearing than for normal hearing people. This poses some difficulty because of the practical requirement to keep the transmission bandwidth as small as possible.

Over the past decade much progress has been made by the hard of hearing in ensuring that adequate communication systems are provided at public gathering places. An indication of the growing acceptance and societal commitment to providing hearing access to hard of hearing people is that the Canada Building Code now requires that one of the above described systems be provided in any meeting room larger than 100 square meters.

Visual communication aids are important for both hard of hearing and deaf persons. "Captioned" television programs, for which the written transcription of the speaker's words can be made to appear on the screen using special equipment, are now commonplace.

Newer systems allow similar captioning of live speech at any meeting, lecture or event, and greatly enhance the ability of hearing impaired people to work in groups. These systems comprise of a manual input device, a computer system, and a means of displaying the typed text. The input device may be a standard keyboard, or some form of stenographic keyboard used by court reporters, for example. The computers and software used range from simple word processors to sophisticated translation systems. Displays can be television monitors or LCD overlays placed on overhead projectors.

These systems can overcome the problem of communication for those who cannot make any use of the acoustical method. Visual methods are also useful to complement auditory comprehension especially in cases where precision and understanding of detail is important.

The design of these systems must take into account the human factors involved in translating spoken words into writing. Typing operators must hear the speakers with clarity and, significantly, these normal-hearing people often require assistive listening devices described above. An important engineering challenge is to design equipment that does not generate any acoustical noise (e.g. keyboard clicks) or electromagnetic interference.

Alerting and sound recognition systems make it possible for many hard of hearing and deaf people to work and enjoy leisure in environments where important acoustical annunciator signals may be generated (e.g. telephones and appliance timers). The increasingly independent and mobile hearing impaired population also demands improved safety, and the ability to receive warning sounds (such as fire alarms) in noisy environments, and/or when sleeping or not wearing hearing aids. Emergency vehicle siren recognizers also have the potential to contribute to driving safety.

The engineering challenges in the design of such systems are formidable. Since annunciator and warning signals are not standardized, and are greatly modified by the time they reach the listener, automated and intelligent recognition of signals is required. This is difficult to accomplish at a reasonable cost. For this reason, the current generation of devices is not very sophisticated and relies primarily on the detection of changes in intensity and the presence of a few characteristic frequency components.

Telecommunication devices and systems include telephone and hearing aid interfaces, TTYs (sometimes also called TDDs for Telecommunication Devices for the Deaf),, and visual image transmission.

An important technology is inductive coupling between hearing aids and telephones, enabling hard of hearing persons to use the telephone without removing their hearing aids. This technology is not influenced by ambient acoustic noise, and overcomes the usually poor acoustic match between hearing aids and telephone receivers. The significance of this is that it makes the telephone system accessible to hard of hearing people, making it possible for them to hold jobs, to live independently, and to fully participate in family and community life. A major engineering problem is that newer telecommunication technologies may create extensive electromagnetic interference, causing major difficulties with inductive systems.

TTYs are miniature computer terminals operating through the telephone network. They use acoustic couplers to connect the unit to the telephone set. For hearing people who wish to communicate with a TTY user, a country-wide Message Relay Centre service is available. Calling this centre hearing and hearing impaired people are able to communicate with the help of an operator who both listens/speaks and types on a TTY. This provides tremendous flexibility and freedom.

Accessibility issues

Acoustical design to provide hearing accessibility for hard of hearing people is faced with the reality that acoustical treatment can only achieve limited objectives under a given set of circumstances. Secondly, even optimal acoustical performance may not be sufficient for hard of hearing people.

Consideration of assistive devices as integral parts of the communication design of a given space or environment may result in improved solutions to the satisfaction of all.

However, the design and deployment of assistive device technologies cannot be separated from non-technical hearing accessibility issues for hard of hearing people. Their objective, simply put, is to achieve and maintain communication under varying circumstances. The means by which this achieved is of secondary importance, provided that some conditions are met. First, the methods used must allow hearing accessibility comparable to that of hearing people. Second, technologies should be easy to use, and should require minimal technical expertise to operate. Third, there should be minimal additional financial burden on the user.

In practice, existing technological solutions to the hearing accessibility problem do not satisfy all of these conditions. Such shortcomings often stem not from engineering inadequacies, but from other sources.

The lack of standardization is a major obstacle. For example, assitive listening devices from different manufacturers (but using the same transmission method) are not necessarily compatible which each other. In addition, in the absence of performance criteria consumers (and even engineers) are left to make difficult purchasing decisions without adequate information. The result is market-place confusion, high prices, and dissatisfaction.

The lack of standardization also slows the development of effective universal systems that would allow a wide range of hearing aids and assistive devices to work together. In particular, the coupling of hearing aids and assistive devices remains a major problem that cannot solved in isolation from evolving telecommunication systems and the problems created by an increasingly polluted electromagnetic environment.

This is in turn has a profound effect on the ability and expectation of hard of hearing people to achieve reasonably universal accessibility. In view of the fact that now 10% of the population has a hearing loss that affects their auditory functioning in some circumstances, any lack of accessibility has a tremendous social impact. The hard of hearing consumer movement is a vital force in promoting hearing accessibility, and the occasional collaboration between engineers and consumer groups has been very productive.

Clearly, hearing accessibility is an area where social forces and technological capabilities are closely interlinked. In particular, heard of hearing people cannot achieve hearing accessibility without the collaboration of the rest of society. This requires not only societal willingness and resources to create and install various technical systems, but also changes in individual attitudes towards the manifestations of hearing loss.

Electromagnetic Interference in Hearing Aid T-Coil Applications

Barry McKinnon
Barron Kennedy Lyzun & Associates Ltd., North Vancouver, BC V7M 3G4

0 INTRODUCTION

Our firm was retained to investigate complaints of high levels of electrical interference experienced by University of British Columbia staff wearing hearing aids in the newly constructed Brock Hall. The architect requested that we try to identify the source of the noise and see if it may be related to a deficiency in the construction of the new facility.

The noise was affecting staff members using the T-Coil setting on hearing aids. The T-coil is a coil of wire that is switched in place of the hearing aid microphone to allow the hearing aid to pickup the varying magnetic field at the earpiece of a telephone handset. It is essentially one half of a transformer, the other half being provided by the handset or an induction loop in a listening assistance system.

We often specify listening assistance systems in our sound system design work for theatres and other public spaces, and we are familiar with a number of typical causes of electromagnetic interference (EMI) in this application. We expected that the noise was being caused by devices such as "Power Smart" type fluorescent ballasts and other conventional electrical noise sources such as transformers and motors. We reviewed a set of electrical drawings to attempt to identify the most likely noise sources, and then arranged to meet with Ruth Warick, the Director of the Disability Resource Centre, located in the new building, and have the noisy areas located for us. Ms. Warick regularly uses a hearing aid equipped with a T-coil in her office where, ironically, the noise was reported to be the worst.

1 TESTING PROCEDURE

We used an induction loop field strength meter on the site to assess the levels of interference. The meter is a Magnatel™ MAG100/CAN, built by Assistive Listening Devices Inc., in Richmond, BC. The meter is normally used to measure the magnetic field strength of a telephone handset earpiece for use with hearing aid telephone pickup coils, and so is calibrated to the CSA standard of CAN3-T515-M85. The meter has a nominal bandwidth of 300Hz to 3300Hz where it exhibits linear response. The unit has ten Light Emitting Diode bars on the display, a "pass" condition is indicated when the unit is held tight to a telephone handset earpiece and shows 5 LED bars lit with a 392hz test tone, corresponding to a magnetic field strength of -19dB Amperes/metre.

The unit also has an audio output jack to allow monitoring of the signal being received at the meter. We do not have sufficiently accurate information on the frequency response uniformity or bandwidth limits of this audio output to know that what is heard or recorded on this output is an accurate representation of source spectrum. We used a portable Digital Audio Tape recorder to capture the signals from this output for later FFT analysis. The audio output was also used to feed a small powered "Walkman" type loudspeaker so that several people in attendance might hear the EMI effects. We found that it was extremely unpleasant to wear ear-bud type earphones or headphones while using the FSM, since the onset and level of noise could be quite extreme, presenting an indication of the effect on hearing aid wearers.

We carried the Field Strength Meter (FSM) through the area at ear level to determine what the EMI levels were in relation to the signal that was intended to be received (the telephone).

2 THE NOISE SOURCES

The most likely noise source, the fluorescent ballasts, were quickly eliminated as the cause of the noise. It was necessary to get within 300mm of the fixtures before a single LED would light, corresponding to a level of -31dB A/m. The noisiest area was in the immediate vicinity of the Director's office and the adjacent small conference room. The conference room was equipped with a VCR and a 28" TV monitor, which when powered up, produced a very broadband noise spectrum typical of the square waves produced in the switching power supplies of modern TV monitors. The FFT spectrum is shown below in Figure #1. The field was quite uniform around the monitor, and the receive level was very dependent upon the orientation of pickup coil. Those people with hearing aids could find a head orientation that would minimize the pickup level, but the field strength was as high as 5 LEDs within 1500mm of the monitor.

The PC type computer in Ruth Warick's office produced a similar spectrum of noise. The CRT monitor was responsible for a significant portion of the noise, but the CPU was contributing a continual background "hash" of noise as well. This was measured in excess of 5 LEDs at the operational distance of arm's length to the computer keyboard.

Over and above these two specific broadband noise sources, there was a sharply localized hum that was definitely related to the AC line frequency. We tracked the highest signal strength to a line at floor level that made a ninety degree bend in the corner of the office, and a second ninety degree bend just outside the office, along the wall of the conference room. Reviewing the electrical drawings once again did not present any likely culprits for electrical noise in this location. We went into the basement of the building to look for the electrical service that may be causing the EMI.

The main AC service entry to the building was in a location considerably removed from the Director's office. It did correspond to the location of a complaint of EMI in a seminar room in the basement. The three phase 600 Amp service to the building was a significant source of EMI across one corner of the seminar room. The noise there was noticeably different in harmonic content from the noise encountered in the Director's office, having a much stronger fundamental.

The underside of the slab was accessible throughout the area where the EMI was tracked, as the basement was undeveloped. The building heating services entered directly under the Director's office, tight to the underside of the slab. There was a steam pipe and a hot and cold water return following the path, but no electrical service evident. We were informed that there was no in-slab electrical in the building, which left only the pipes. When we measured up tight to the pipe wrap insulation we found that the pipes were definitely the source of the EMI. The FSM went off scale within 25mm of the pipe insulation on the steam pipe which seemed to carry the highest current. Calculating backwards from the FSM indicators (which are only bandwidth rated for 300Hz-3300Hz) the current in the pipe was in the order of 1 Ampere in the FSM bandwidth. The FFT display in Figure #2 shows the 60Hz AC line frequency fundamental some 15dB below the 300Hz harmonic which falls within the FSM bandwidth. The noise spectrum suggests the sum of ground currents from a variety of devices and phases, producing high harmonic content.

3 ATTEMPTED REMEDIES

Since the pipes passed through a mechanical room immediately adjacent to a large distribution transformer, we thought that there may be some induction into the pipes which formed a huge loop back to the University's Physical plant. Attempts to ground the pipes at the mechanical room actually increased the field strength, indicating that the current flow was from outside the building to the lower impedance ground point. Attempts to ground the pipe at the service entry point decreased the field strength along the pipe by approximately 4dB, but increased the field strength in the immediate vicinity of the Director's office.

The building ground in this new facility was likely superior to the buildings nearby that were connected together through the steam and water pipes. We were measuring ground current from other buildings flowing to the electrical ground for Brock Hall. The ideal solution would have been to install flexible, non-conductive isolators in the lines which would have provided the required break in the ground path. This was not possible at this stage of the building construction. The mitigating solution chosen by the contractor was a ground at the building entry, which did, at least, improve the condition for the largest number of people.

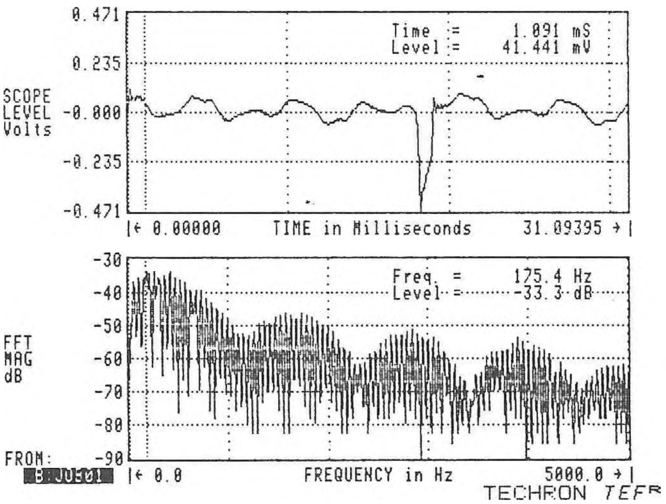

Fig. 1. Noise spectrum of TV monitor.

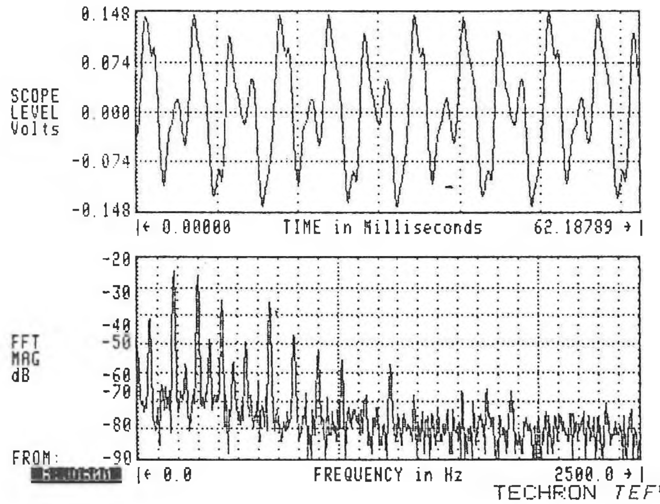

Fig 2. AC Line related noise spectrum.

The measures taken to alleviate the EMI were limited to those that were non-invasive to the building structure. There was no opportunity to try magnetic shielding above the piping carrying the current. Because the currents on the pipes were not extremely high, there was no concern for increased corrosion, so there was no secondary justification for having non-conductive isolators installed in the pipes to break the current flow.

The problem encountered in the basement seminar room was countered by using an FM listening assistance system. There was no practical approach to shielding the room from the magnetic field produced by the AC service to the building.

We suggested that UBC's Engineering Faculty may want to make this a student project, and investigate other avenues, such as active cancellation of the magnetic field.

4 SCOPE OF THE PROBLEM

The EMI problems in the new portion of Brock Hall led to a similar investigation in the older section of the building. The problem areas that were identified there were primarily a result of fluorescent lamp ballasts in need of replacement. The Disability Resource Centre pursued this investigation further and found that most buildings on the campus exhibited EMI in some locations, and in varying degrees of severity.

The EMI problem is quite different for this T-coil application than it would be for conventional audio system applications. EMI is usually a problem that can be alleviated by ensuring that the system has a carefully conceived and implemented approach to grounding and shielding, and to ensure that signal levels and transmission impedances are suitable for the environment. The T-coil in the hearing aid is specifically intended to pickup a varying magnetic field and translate that to a signal. It is, by design, very vulnerable to the type of noise encountered here.

5 IMPLICATIONS OF THE PROBLEM

The increased awareness of access requirements for those people with a hearing impairment has led to a number of changes in building design criteria. Where access has been identified as a factor, the permissible background noise and reverberation time in rooms have been lowered to accommodate moderately hearing impaired persons. The extent of the EMI problem that was identified here has helped make us aware of another aspect of the problem that was not previously apparent. The EMI factor should have some broad implications for accommodating hearing impaired access in new buildings.

The location of major electrical service entry, and major electrical rooms and transformers should not be close to areas where people will be required to use telephones, or the T-coil setting for use with an induction loop listening assist system. This is applicable to conventional building design, as well as large public spaces such as churches and theatres. This may affect the location of pay phones, offices, seminar rooms, and especially audiology clinics.

The problems encountered with TV and computer monitors, and computer CPU's may affect the choice and location of this type of electronic hardware. Flat screen LCD panels should have substantially less EMI than the CRT type, and may be a better choice in offices where the use of a T-coil hearing aid is required. TV monitors will always be a problem, as the CRT will be with us for quite some time before flat screen technology becomes affordable.

Unusual or unexpected sources of EMI such as steam pipes are hard to consider as problems at the design stage of a building. The important information to take away from the steam pipe scenario is that it takes very little current flow to produce a magnetic field large enough to cause serious interference for a hearing aid wearer.

For adequate speech intelligibility, English speaking people with normal hearing require a signal to noise ratio of up to 25dB. There are occasions where we can salvage understanding with as little as -6dB of S/N, but we are more comfortable with higher S/N ratios. In the case of the EMI outlined here we had a S/N of 0dB in most cases, a difficult situation for a phone conversation.

6 CONCLUSION

The design process is one of constant refinement as we learn of more parameters that affect the final outcome. In the case of disability access, we are continually finding impediments to access that we have been quite unaware of. A chance situation which combined an unusual source of EMI with the worst case scenario of location, and the people affected, has provided us with an opportunity to become aware of new parameters to consider in the design of a building to ensure that it is a safe, functional, and comfortable space to work or live in.

BUILDING PERFORMANCE FOR HEARING IMPAIRED PEOPLE

Jean-Rémi Champagne
J.R. Champagne Consulting Ltd.
7 Ossington Avenue
Ottawa, Ontario K1S 3B3

People compensate for hearing loss by relying on other senses, particularly that of sight. Visual elements and the quality of light are therefore important factors in the enhancement of visual information. Equally important is the acoustical environment. For persons using hearing aids it is essential to reduce background noises and to control reverberation in order to ensure speech intelligibility. Since hearing loss is limited to particular frequency ranges, the control of frequencies is important as well as the reduction of noise interference in environmental planning.

Our built environment must create settings for living, learning, working and recreation to be enjoyed through the senses of vision, touch, smell and taste but little or no hearing.

Deaf and hard of hearing people cannot rely on sound or spoken signals for orientation and wayfinding. Aural signals must be complemented by visual and other sensory signals. The success of communication depends on whether or not meaningful sound can be distinguished from background noise. Some ambient noise is useful for orientation and to provide the "feel" of space but, for people with hearing impairment, there are special requirements for loudness level, reverberation times, masking noise, background noise and, for those who use hearing aids, electro-magnetic interference with telecoil receivers. Noise levels of 47-50 dB can seriously interfere with the communication ability of hearing impaired persons without affecting those who hear normally. A speech-to-noise ratio of approximately 4.5 dB is about the lowest level acceptable (A.H. Suter, U.S. Dept. of Labor, Washington). According to research done in Arizona State University (Leshowitz, Dept. of Psychology), the ratio of speech to background noise level for hearing impaired persons must be increased by 10 dB over that measured for normal hearing persons. Most open concept office spaces will not accommodate this.

On entering a building, a hearing impaired person may not hear the sound of a door latch-release nor a message from an intercom or electronic signal to indicate the door operation; graphic instructions; closed circuit TV to communicate with someone inside the building; redundant signage and visual cues for orientation and wayfinding inside the building. Tactile information such as carpeting, floor textures, handrails may also be used.

The lobbies of public buildings, hotels and apartment buildings are generally designed for marketing considerations, aesthetics and ease of maintenance. Glass, polished stone and other hard surface materials are commonly used. Acoustical qualities are often ignored. The sounds of clicking heels, moving carts, motors and voices in these spaces can be more than somewhat disconcerting for people with minimal or even no hearing loss. Glare from building materials can cause disorientation and distract from he available visual information intended for wayfinding purposes.

Elevator lobbies need light signals to identify elevator car locations and the direction of car movements. Audible in-car signals need supplementary visual signals. Telephones in elevators as well as in public areas should have volume controls and be compatible with the telecoil mode of hearing aids. Some public telephones are installed with fluorescent lights just above the receiver and the ballast causes enough interference to make it impossible for a hearing aid user to decipher speech.

There is now great progress in evaluation techniques for the acoustical performance of rooms. This results in rooms that are more appropriately designed for speech intelligibility or fidelity in the perception of sounds of mechanical or instrumental origin. Often, however, the room shape or size and the distance of the receiver from the point of sound origin are still barriers to good communication. Fixed or movable deflectors, baffles, reverberation panels and absorption planes may be added to alter the design. Audio amplification systems may also be required.

Where practical, carpeting on floors may be used to control sound reverberation. A carpet with 5 to 15 mm pile and no underpad contributes to noise reduction and permits wheelchair access. Absorption panels on walls are best located between 750 and 2000 mm above the floor.

Sound transmission control requires an isolation of airborne sound to the greatest extent possible and this means: sealed baffles above partitions that stop at suspended ceilings; door and window assemblies installed in careful precaution to eliminate acoustical leaks; double glazing; walls and partitions insulated against sound transmission; enclosed self-contained stairways.

Special communication devices are required for deaf and hard of hearing persons. These may include: telephone-coupled teletype devices; flashing colour-coded lights activated by door bells, voices; closed circuit TV systems as visual intercoms; vibratory devices for beds, chairs or pagers; sound amplification systems including hardwire, induction loops, AM and FM radio frequency and infrared systems. Hearing aids amplify sounds, including sounds one does not want to hear. "Nuisance" sounds emit from ballasts in fluorescent lights, transformers, rheostats, motorized valves, solenoids, motorized appliances, computer or other electronic installations, magnetic fields around improperly shielded wires, and even induction currents such as that used in the audio loop itself. Where audio amplification systems are used these have the same problems as hearing aids and a few extra problems. Electro-magnetic interference is the main source of these problems. In New York City, where selected frequency bands are so numerous, there is sometimes overlap and one may pick up police calls in a session at the United Nations Building. The infrared system requires a direct beam from transmitter to receiver. Direct sunlight or an incandescent light source will cancel out the signal transmitted.

Could the building construction provide better shielding to prevent this? Could the transmitter or receiver be modified? Should there be stricter regulations? More stringent controls? International controls? The causes of interference from the building itself in audio amplification systems and from the installations and equipment in it need to be more clearly defined and documented so that appropriate alternative solutions can be examined.

Acoustics and Technology: A Hard of Hearing Perspective

Ruth Warick, Director
Disability Resource Centre
The University of British Columbia
1874 East Mall
Vancouver, B.C.
V6T 1Z1

At one time the emphasis was on the rehabilitation of the individual with the individual being required to adapt to the environment. Over the last two decades recognition has occurred that environmental adaptations are also required. Thus, it is recognized that both individual habilitation and environmental adaptations are essential.

The Canadian Hard of Hearing Association, a non-profit self-help organization consisting primarily of hard of hearing persons, works as an agent for change for both individual habilitation and environmental adaptations. CHHA numbers close to 2,000 members who have a hearing loss and rely on aural means of communication and sometimes the use of assistive listening devices. The prime concerns of CHHA members are hearing health care, better listening environments, public awareness about noise pollution, education, and employment. The latter two issues were identified through a national youth study undertaken by CHHA[1] and the others in a study undertaken by Marilyn Dahl last year.[2]

The CHHA survey about hard of hearing youth found that access is a major difficulty. In asking young hard of hearing persons about their experiences in everyday life, the survey found that about half had difficulty watching television, over half cited difficulties using the phone, and most cited difficulties hearing in small groups, in restaurants and public places.

The survey found that technical and environmental supports are not widely used in all settings. While use of FM or infrared hearing systems in the schools is quite good, with 50 percent of youth using such systems, in a public place the usage of such systems dropped to eight percent. However, 40 percent of survey respondents said they would welcome the use of such systems. This substantiates the claim that many more hard of hearing persons would use technical equipment if provided.

In fact, there is much technological knowledge available today but it has yet to be put into widespread use. There is a gap in the application of the technology, some of it due to a lack of information and some of it due to uncertainty about the unknown and concern over costs. This must be changed so that there is a greater application of existing technology.

At the same time, a watchful eye has to be kept on technical changes, as experience has shown that these are not always beneficial for hard of hearing persons. Some of the technologies nullify previous benefits for hard of hearing persons. This was the case when new telephone receivers were introduced which reduced the magnetic leakage required for T-switch hearing aid use. It was only through legal representation to the Canadian Radio Television and Telecommunications Commission (CRTC) that the principle of T-switch use of telephones was upheld. In this case, as in other situations involving technology, hard of hearing consumers worked together with experts in the field to ensure that the principle of hearing access was upheld.

More recently, the issue of environmental interference with use of hearing aids on a T-switch has surfaced although it is not a new problem. It is just that now it is receiving more attention as the impact of the problem is better understood. I have personally encountered this problem for a number of years; I've been in rooms where it has been impossible to use an FM system because of interference from the mechanical room a floor below. When using the telephone on my T-switch I've had to try to hear a conversation over the radio station inadvertently being picked up. Needless to say, it is frustrating when hard of hearing persons use technology which is intended to improve their acoustic environment, only to find it cancelled out by some other factor. Unfortunately, the solutions are very specific to a situation. What this means is that more engineers, architects and other professionals in related fields need to be aware of the problems and of ways to resolve it.

Fortunately, not all acoustic improvements are complicated. Carpeting, acoustical ceiling tiles and upholstered seating are a few of the ways to improve the acoustical environment. An acoustically-friendly room can reduce the need for other technical equipment. On the other hand, a room with poor sound quality may require extensive use of an assistive listening device. The preference obviously is to reduce the extent of use of external systems. One of the additional benefits of good acoustics is that persons without a hearing loss can benefit from it as well. As this fact becomes appreciated, undoubtedly more attention will be paid to acoustics and hard of hearing persons will benefit from the improvements. Yet, we are still far behind in good acoustics being recognized as a priority issue.

To conclude, I want to re-emphasize that the environmental issues which hard of hearing persons face are considerable. There have been technological advances and some increased knowledge of such advances. The challenge is to translate technological solutions into everyday action and to ensure that new advances do not reduce hearing access. To ensure this, hard of hearing persons and professionals need to work together, to share more knowledge, and to find cost-effective, qualitative solutions.

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Dahl, Marilyn. Report on a Study of the Comparative and Cooperative Roles of Canadian Deaf and Hard of Hearing Forum, Canadian Association of the Deaf, Canadian Hard of Hearing Association. March, 1993.

Validation of masked threshold predictions among people with sensorineural hearing loss

Raymond Hétu and Hung Tran Quoc,
Groupe d'acoustique de l'université de Montréal,
C.P. 6128, Succ. Centre-ville, Montréal, H3C 3J7

Auditory capacities are a given. This situation calls for efforts to adapt the work environment to the prevailing residual capacities of workers with a hearing impairment. Noise reduction is of course the first step required in order to improve signal-to-noise ratios. However, in many cases, these steps will not be sufficient to prevent performance impairment among workers with a hearing loss. Specific procedures for job accommodation are warrented.

With respect to auditory warning signal detection, given a certain ambiant noise, one needs to predict the signal level at a given frequency that will meet the individual's detection capacity. Previous work has been done to adapt a laboratory procedure for measuring frequency selectivity to the constraint of a clinical test [1-2], and also to adapt Dectect*sound*™ [3] to individual rather than statistical predictions of masked thresholds. The present investigation aimed at validating such predictions among people with various degrees and configurations of sensorineural hearing loss.

Methods

Participants

Participants were sampled in order to represent different degrees of sensorineural hearing loss lower than 75 dB HTL and different audiometric configurations. They were recruited among the clients of a regional audiological rehabilitation center in Montreal, using the following exclusion criteria: (a) air-bone gap greater than 10 dB between 0.25 and 4 kHz; (b) abnormal tympanogram; (c) interaural difference in hearing thresholds greater than 35 dB between 0.5 and 4 kHz; (d) maximum loss between 0.25 and 4 kHz greater than 75 dB HTL [53]; (e) the over-65 or under-18 age brackets; (f) presence of a disease associated with fluctuating hearing thresholds.

Inclusion criteria were determined according to audiometric configuration, that is, based on the difference between average hearing thresholds in the high frequencies (2, 3 and 4 kHz) and the low frequencies (0.25, 0.5 and 1 kHz). 'Descending loss' was defined as a 10 dB difference between the high and low frequency average threshold, the inverse being the case for 'ascending loss'. 'Flat loss' referred to a difference inferior to 10 dB. Four groups were thus recruited as follows:
- 13 individuals with a descending maximum loss of between 35 and 55 dB HTL;
- 21 individuals with a descending maximum loss of between 55 and 75 dB HTL;
- 6 individuals with an ascending maximum loss of 55 dB HTL;
- 12 individuals with a flat loss of between 30 and 55 dB HTL.

Procedure

The experimental setup was identical to the one used in a previous study [4] with white noise filtered by two lowpass and two highpass filters connected in series. The continuous notched noise was combined with a 250-ms pulsed pure tone repeated every 500 ms that was generated by a clinical audiometer and presented to the subject by means of a TDH-50 earphone. Masked thresholds were assessed by Bekesy tracking during 30 seconds per notch noise condition.

Auditory filters were assessed at 0.5, 1, 2, 3 and 4 kHz respectively in a random order. The masking noise conditions were also randomly presented except for the first condition, which was always the allpass noise ($g_l = g_u = 0.0$). Testing was initiated with the masking noise level set at 40 dB/Hz. When the allpass noise induced less than a 5-dB masking effect, the noise level was set at 50 dB/Hz. In order to avoid aberrant estimates of the filter slope on the high frequency side [2], masked thresholds were also assessed with six highpass noise conditions ($g_u = 0.0; 0.1; 0.2; 0.3; 0.4; 0.5$) when estimates of p_u resulting from notch noise testing were greater than 50. The auditory filters were characterized using the mathematical expressions proposed by Glasberg and Moore [5].

In order to test the validity of predicted masked thresholds from individual auditory filter characteristics, the 52 participants were asked to detect pure tone signals at 0.5, 1, 2, 3 and 4 kHz in three spectra of broadband noise (lowpass, highpass and bandpass [1]) at 85 dBA, in addition to the allpass white noise at 78 dBA used in auditory filter characterization.

Results and discussion

Considerable individual variations were observed in the detection thresholds with the four broadband noise conditions tested. Where the noise spectrum is flat, no relationship emerges between masked and absolute threshold level, as expected. There are nevertheless difference of approximately 15 dB between the higher and the lower masked thresholds.

Where the noise spectrum is sloping, there appears to be an association between absolute and masked threshold, above a certain value of absolute threshold: namely, above 35-40 dB HTL at 0.5 kHz and above 50 dB at 3 kHz. This is attributed to filter asymmetry and to upward spread of masking effects, which is more likely with poorer hearing sensitivity. With the lowpass noise showing a maximum slope around 3 kHz, the staggering of individual data is pronounced: a 41 dB difference is observed between extreme values. These large variations confirm the need for individual adjustment of auditory signals with respect to the residual capacities of hearing-impaired workers.

Table 1 presents the means and standard-deviations of the differences between predicted and observed detection thresholds within the four masking-noise conditions. Only those cases where the filter was actually characterized are included. As expected, the procedure generally leads to slight overestimations of the masked thresholds, the average error of prediction being smaller than 2 dB, with one exception, that is, 2.15 dB at 0.5 kHz with the white noise. The range in individual errors is relatively small (i.e., the standard-deviations of differences were smaller than 4 dB), with two exceptions: at 4 kHz with lowpass noise, and at 2 kHz with bandpass noise. With allpass noise, which actually served to assess the value of the fitting constant K, such errors are minimal.

As can be seen from Figure 1, the magnitude of individual prediction error is typically 2.5 or 3 dB in a majority of cases, and is rarely above 5 dB, a value that is compatible with the audiometric measurement error. In fact, the proportion of cases of underestimation by 5 dB or more is equal to 0, 3.5, 4.6 and 6.4% for the allpass, lowpass, highpass and bandpass noises respectively.

Table 1. Mean and standard-deviation (S.D.) of the differences between predicted and observed detection thresholds in four different masking noises. N refers to the number of individuals with whom the auditory filters had been measured.

		Signal frequency - kHz				
		0.5	1	2	3	4
Allpass noise	Mean - dB	2.15	1.78	0.93	1.36	0.63
	S.D. - dB	0.88	0.91	1.04	1.06	1.16
	N	40	48	40	32	28
Lowpass noise	Mean - dB	1.08	1.31	0.53	0.76	0.66
	S.D. - dB	2.69	2.36	3.25	2.88	4.10
	N	37	44	36	29	27
Highpass noise	Mean - dB	-0.01	-0.02	-0.44	1.66	0.47
	S.D. - dB	3.70	2.17	3.61	2.82	3.03
	N	37	44	36	29	27
Bandpass noise	Mean - dB	0.41	0.86	-1.11	1.48	0.99
	S.D. - dB	3.89	2.61	4.41	2.70	3.91
	N	37	43	36	29	27

The findings indicate that the errors tend to increase with signals located in the sloping portion of the noise spectrum. In those specific circumstances, the masked threshold depends not only on the overall width of the auditory filter but also on its shape. Assessing the slopes of the filters involves much more uncertainty than assessing only their width. This is especially true when the filters tend to be highly asymmetrical, as is often the case with descending audiometric configurations.

Guidelines for auditory warning signal design prescribe signal level adjustment at 15 dB above the estimated masked threshold in order to ascertain attention demand and facilitate signal recognition. As errors of underestimation in individual masked threshold predictions are equal to or less than 5 dB for a very large majority of cases (Table 1), the use of the present procedure would ensure signal detection for almost anyone whose auditory filters have been characterized. Some individuals might, however, be at a slight disadavantage with respect to signal recognition.

Generally speaking, the present endeavour demonstrates the feasibility of adapting the most common auditory demand in the industrial workplace, namely, sound warning signal detection, to the constraints imposed by a heairng loss. A clinical procedure allows one to characterize, in 15 to 25 minutes of testing time, the residual capacity for signal detection in noise in the better ear of an individual who suffers hearing loss. A computer model (*Detectsound*™) provides the required specifications in terms of signal level adjustment. The testing procedure together with the model constitute, in our view, a practical tool for job accommodation with people who sustain a hearing loss.

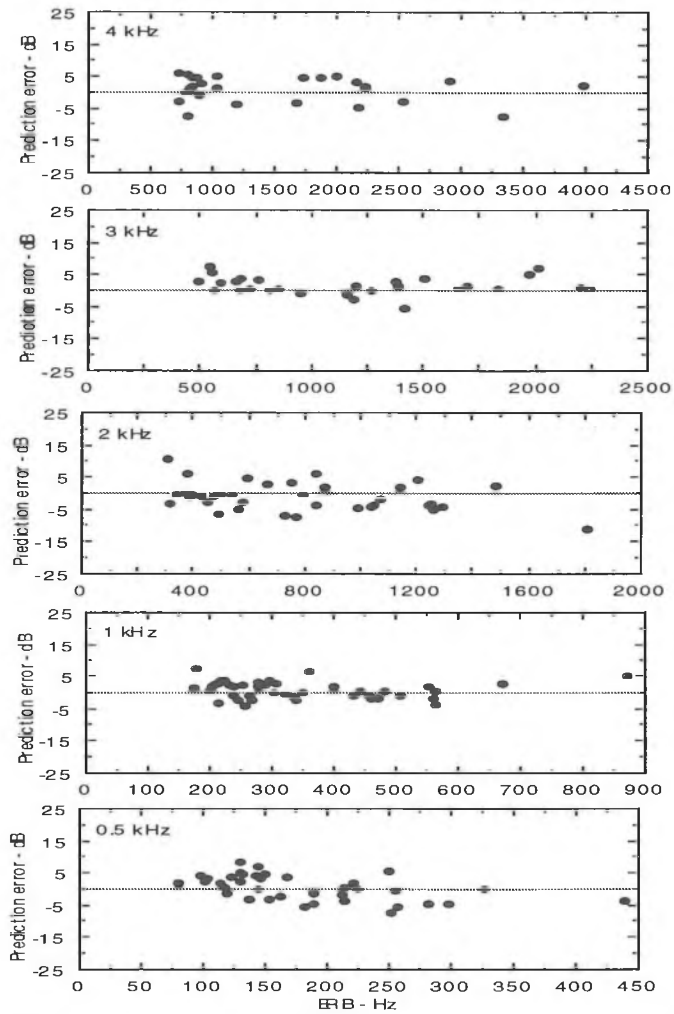

Figure 1. Individual values of the differences between predicted and observed thresholds masked by the bandpass noise.

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HEARING ACCESSIBILITY IN A HOME-FOR-THE-AGED

M. Kathleen Pichora-Fuller, PhD
School of Audiology and Speech Sciences
and Institute of Hearing Accessibility Research
University of British Columbia
5804 Fairview Ave., Vancouver, B.C. V6T 1Z3
Ph:(604) 822-4716 Fax:(604) 822-6569 e-mail: kpf@audiospeech.ubc.ca

Background: The majority of the institutionalized elderly have a clinically significant hearing loss. Even those with hearing thresholds within normal clinical limits often have sub-clinical declines in auditory processing such that, even though they have no trouble understanding speech in ideal listening conditions, they experience trouble understanding speech in the noisy conditions that are typical in everyday life (for a review see Willott, 1991). Nevertheless, few residents of homes-for-the-aged receive clinic-based audiologic services. Furthermore, even those elderly individuals who do receive clinic-based services often find that they continue to experience difficulty when trying to communicate in everyday situations (Health and Welfare Canada, 1988). Many of their activities of daily living are not 'hearing accessible' even after treatment.

Definition of Accessibility: An activity or facility is considered to be hearing accessible if hard of hearing individuals are able to function in the activity or facility as effectively as do people who have normal hearing.

Purpose of the Project: To achieve hearing accessibility for seniors in their activities of daily living at homes-for-the-aged, we believe that it is necessary to develop community-based (on-site) services as an alternative to the existing clinic-based audiologic services. Traditional clinic-based services focus on rehabilitating the hard of hearing person, most often by fitting them with a hearing aid. In contrast, the objective of a community-based accessibility approach is to develop solutions that could be implemented by hard of hearing seniors living in homes-for-the-age and/or by their regular communication partners (other residents, family, staff, volunteers), or by the management of the care facility. These solutions would extent beyond the typical provision of personal hearing aids and would include the application of appropriate assistive devices, administrative modification of programs and routines, and modifications of the physical and social environment. To demonstrate how such a program might operate, we designed, implemented and evaluated a hearing accessibility program at a model home-for-the-aged (Head and Jennings, 1994).

In the present paper we will describe the steps that were taken in developing an accessibility plan for the model home-for-the-aged, St. Joseph's Villa in Dundas Ontario. Examples of the solutions that were implemented and outcome measures of the degree to which the solutions were successful will be provided.

Assumptions and Objectives of the Project: The assumptions behind the design of the project were that most residents of homes-for-the-aged experience communication handicap and that communication handicap results in non-participation in desired activities or in communication performance that is below the potential of and/or unsatisfactory to the residents during participation in activities. The primary goals of our hearing accessibility program, therefore, were to increase the participation of residents in desired communication-demanding activities and to improve the effectiveness of communication during their participation in those activities.

Preliminary Survey: As the first step in developing the hearing accessibility program at St. Joseph's Villa, we needed to obtain a profile of the activities of daily living at the Villa that required hearing. In a pilot study, two meetings were held, each with 15 participants: 5 residents with known hearing loss, 5 residents who were considered to have good hearing, and 5 staff. The participants were asked, "In everyday life at the Villa, when is it important for a resident to hear?" Each person generated a situation and a list was compiled on a blackboard. The meeting continued until no further situations could be generated. The lists of situations generated were later reviewed by four experts (two audiologists, a speech-language pathologist who works with the elderly, and the nurse in charge of the clinic at the home-for-the-aged) who determined a final list of 33 key situations, excluding those that were considered to be duplicates or irrelevant to the project. The 33 situations were divided into categories: 17 primary situations in which a resident could initiate listening to speech communication and 16 supplementary situations in which a resident either listened to a non-speech signal (e.g. fire alarm) or could not chose whether or not to initiate speech communication (e.g. PA announcements). The 17 primary situations were the following: talking to familiar people, talking to hard-of-hearing people, telephone, chapel, meetings, exercise class, teas in the solarium, teas in the auditorium, teas in the tuck shop, dining in the main dining room, dining in floor-specific dining areas, TV, radio talk shows, taped books, taped music, movies at the Villa, and therapy.

Outcome Measurement Tool: The next step in the project was to determine the extent of participation by residents in each of the 33 situations and to determine whether or not they were satisfied with the effectiveness of their communication in the situations in which they participated. To obtain this information, we developed a questionnaire that would provide us with a profile of the scope and effectiveness of residents' communication. For each of the subset of 17 primary situations, the questionnaire asked a set of 10 questions that tapped the resident's interest and rate of participation in the situation, how much they understood and how satisfied they were with communication in the situation if they participated in it, and whether they employed and benefitted from technology or communication techniques when they were communicating in the situation.

Design of the Study: The questionnaire was administered twice at a six-month interval to provide us with baseline data about accessibility needs. An audiologist then implemented a service program that aimed to increase hearing accessibility at the Villa. The questionnaire was also administered six months and twelve months after the commencement of the service program. A comparison of the pre-program questionnaire results to the results obtained following the implementation of the service program were used to measure whether or not the service program had been effective in improving the scope and/or effectiveness of the residents' communication during primary communication situations.

Subjects: Thirty residents with relatively stable physical and mental health maintained their participation throughout the two years of the study. Subjects spoke English as a first language and had no communication disorders arising from causes other than hearing loss. At the beginning of the study, the mean age of the participants was 85 years (range 68 to 94 years). The length of residency in the home-for-the-aged ranged from 0 to 26 years, with over half having lived there at least 6 years before the beginning of the project. This is consistent with our impression that there was a well established community of residents at the Villa who knew each other well, thereby fostering motivation to communicate and to engage in activities that featured social interaction.

Pre-program audiometric tests were conducted to determine the extent of each participant's hearing loss. Using the rule of thumb that a person with a threshold loss of at least 40 dBHL at 2000 Hz is likely to benefit from wearing a hearing aid, we estimate that about half of the evaluated group had a degree of hearing loss that warranted a hearing aid fitting. The other half of the group demonstrated high-frequency hearing threshold loss consistent with aging (presbycusis) and even though the degree of their threshold loss was minimal, it is well known that such cases experience other auditory processing deficits that account for the frequent complaint of the elderly that they have difficulty understanding speech when there is background noise or multiple talkers (Willott, 1991). Even when there was no background noise, the best speech discrimination score obtained was fair (below 80%) for about 2/3 of the group. While hearing aids may correct for threshold loss, they do not overcome the latter type of auditory deficit that is characteristic of aging. In such cases where signal enhancement in noise is required, assistive listening technology may be more useful than hearing aids.

Pre-project Use of Technology: About half (16) of the evaluated group owned hearing aids at the outset of the project. Of those who had hearing aids, 15 used their hearing aid(s) at least some of the time, with most wearing their hearing aids all day long everyday. Prior to the beginning of the project, some public phones in the Villa were equipped with handset volume controls and four participants reported using them. No other public assistive listening devices were available. Nine participants used handset volume controls on private telephones. The only other assistive listening devices in use were jack-in earphones for television use that were owned but seldom used by two of the residents, and a one-to-one communicator that was owned but only tried once by one resident.

Questionnaire Results: In this paper, a selection of the results obtained during the baseline period will be reported to demonstrate how we appraised accessibility needs at the Villa. Specifically, the number of residents participating in the primary situations, the number of hours spent by residents in situations in which they participated, and the amount that they understood and the degree of their satisfaction with communication in the situations will be described.

One situation where there was a high rate of participation by residents but widespread trouble understanding speech was at the chapel. Therefore, the chapel situation will be used as an example of how the treatments were tailored to situation-specific requirements. Outcomes will be reported that indicate that the treatment was highly successful in improving the residents' communication function in the chapel situation.

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CHARACTERIZATION OF OCCUPATIONAL SOUND EXPOSURE OF PROFESSIONAL INVOLVED IN HIGHLY AMPLIFIED MUSIC REPRODUCTION

Martin Fortin and Raymond Hétu,
Groupe d'acoustique de l'université de Montréal,
C.P. 6128, Succ. Centre-ville, Montréal, H3C 3J7

Exposure to highly amplified music among listeners has not been demonstrated as a serious threat for hearing [1-3]. However, risk assessment for the professionals involved in the production and reproduction of highly amplified music has, as yet, received little attention. The present investigation aimed at characterizing the sound exposure within different professional categories, to estimate their risk of hearing loss and to explore the possibilities of limiting such a potential risk.

Methods

Participants

Two or three individuals were recruited to represent each of the following professional categories: sound engineer, sound technician (soundman) and disc jockey. They had to meet the following recruitment criteria: a minimum of 5-year experience and being currently employed by a well known entreprise in the trade.

Procedure

The participants were first interviewed individually concerning their work organization and the various factors governing their sound exposure. The interviews were tape-recorded and transcribed for analysis purposes. A second visit was later organized to record typical sound exposure conditions. Five 10-s samples of sound judged as being representative of each work activity were recorded using a BK-2231 sound level meter and a Sony PCM-1000 digital recorder. The samples were later assessed using a BK-2123 analyzer.

Results

The findings show that exposure could vary quite considerably from one week to another for sound engineers inparticular. Scenarios of representative exposure patterns were defined, and the level of exposure was computed accordingly. Sound engineers who work on tours are involved in an average of 6 shows a week during typically 3 weeks per month. An average exposure for each show was estimated at 2 hours for setting up and 3 hours for the show itself. For the sound technicians, a typical week was estimated to involve 50 hours during disc recording. Disc jockeys typically work 5 hours a day, 3 days a week.

The resulting L_{Aeq40h} are given in Table 1 for the three types of occupation. They ranged from 94 to 99 dB for the sound engineers. L_{Aeq40h} amounted to 89.5 dB for the recording technicians and to 93-94 dB for the disc jockeys. Based on ISO-1999.2 [4], significant hearing losses are predictable in the high frequencies even among individuals with an average sensitivity to noise-induced hearing loss.

Predictions indicate that the average hearing loss could amount to nearly 30 dB at 4 kHz after 10 years of work as a sound engineer (Table 1). Lesser degrees of loss are predicted for the sound technicians who are exposed to less powerful sound sources. Disc jockeys fall into an intermediate risk category.

Table 1. Estimated weekly exposure level (LAeq40h) and corresponding median permanent threshold shift at 4 kHz after 10 years (PTS50 - 10y) according to ISO 1999.2 [4] for 8 professionals involved in highly amplified music reproduction.

Occupation	LAeq40h dB	PTS50 - 10y dB
Sound engineers		
#1	99.0	28.8
#2	93.8	17.7
#3	94.2	18.4
Sound technicians		
#1	89.6	10.6
#2	89.5	10.5
Disc jockeys		
#1	94.3	18.6
#2	93.2	16.6
#3	94.9	19.8

During the interviews, the participants all mentioned that they felt signs of hearing impairment. One stated: "We all have more or less the same thing, this little dip around 6-8 k; but, apparently, this is normal...". Another said: "We are somewhat like miners who know they have lung problems or truck drivers who all have back problems. It is part of the job".

Furthermore, the actual sound levels during work sessions were high enough to induce temporary threshold shifts, which may impair work performance. An illustration is given below for a sound technician (Figure 1). The spectrum of the sound measured at the ear level during base tracks recording is depicted.

Also shown in Figure 1 is a reference curve for the mean lower limit of sound pressure level that will not induce temporary threshold shift. This so-called "effective quiet" curve [6] is derived from the mean free-field hearing threshold levels [5] elevated by 70 dB. It has been shown that exposure during 30 to 60 minutes at sound levels that are 80 dB above threshold induces 15 to 17 dB TTS_2 on an average [7]. Based on the data presented in Figure 1, a sound tehcnician with average sensitivity to TTS would sustain a 15 dB threshold shift at 0.5 and 0.6 kHz and over 20 dB shift at 4 kHz after 30 to 60 minutes of recording. Knowing that the daily work schedule in this trade often extend over 8 to10 hours, this means that asymptotic threshold shifts may be sustained while master tapes are being recording. This situation is paradoxical as TTS is associated with reduced frequency and temporal resolution [8]. Furthermore, sound levels such as those depicted in Figure 1 are by themselves highly challenging in terms of frequency resolution. In other words, the professional is working in a paradoxical situation where hearing acuity is both highly sollicited and deteriorated by the very signals that are being processed.

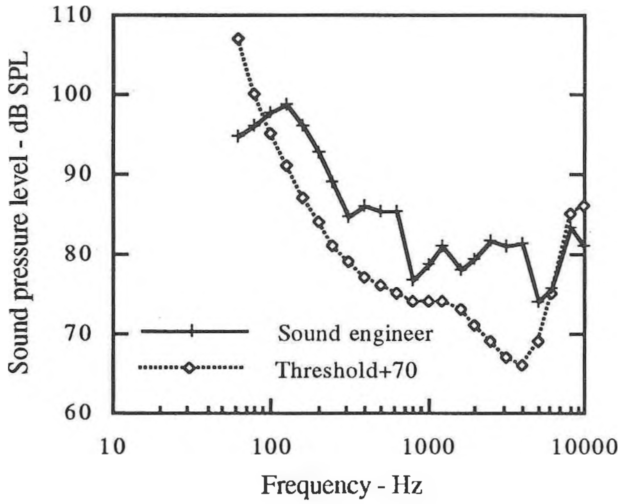

Figure 1. Spectrum of the music recorded at the ear level of a sound technician during base tracks recording, compared with free-field normal hearing thresholds [2] plus 70 dB.

Discussion

The above findings indicate that sound exposure among professionals involved in the production or reproduction of highly amplified mucic represent a potential damage risk to hearing as well as ergonomic problems. Possible means to reduce such exposures were identified.

For sound engineers, the major sound source is usually the percussion. A partial enclosure could possibly be used to reduce the contribution of this sound source. Controllers with digital interfacing and samplers could also be considered. The distorsion that is systematically sought for by guitarists could be obtained at lower sound levels using less powerful amplifiers. The monitor for this sound could be installed in a room behind the stage. Intra-aural monitors could also be used, provided that their sound power is limited to safe levels.

The disc jockeys are using a monitor that inform them of the sound environment on the dance floor. The acoustic monitor could be substituted by a visual monitor, such as a spectrum analyzer.

In both of the above cases, the potential solutions to sound overexposure would represent highly significant work organization changes. For this reason, their trial and implementation would require both a strong motivation on the part of the professionals involved and the active participation of the latter in the actual design and testing of the new procedures.

The problem of sound overexposure in the music industry is not restricted to the three job categories included in the present investigation. Musicians are at serious risk as well [9] and the recent introduction of intra-aural monitors on the market may not solve this problem if they do not meet the constraints involved in accurate sound monitoring with limited exposure. Furthermore, people who work in settings where highly amplified music is attended to may also be at serious risk of temporary and permanent hearing loss. This includes waiters and barmen in clubs and discotheques where occupational health standards do not appear to be enforced.

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Cases of possible job discrimination based on hearing loss

Chantal Laroche
Audiology/Speech-Language Pathology Program, University of Ottawa, 545 King Edward, Ottawa, Ontario, K1N 6N5

INTRODUCTION

Each year, complaints are filed at the Quebec and Canadian Human Rights Commissions by individuals who have been refused employment because their hearing sensitivity does not meet the standards required by the employer. This paper presents four recent cases. The first case deals with a young fireman who, after having completed his training in a specialized school, was refused a job based on a unilateral hearing loss which had not been detected before entering the school. The second case relates to a police officer having a unilateral hearing loss which does not meet the criterion of a federal employer. The two last cases refer to two railraod workers who failed the pre-employment hearing examination because they were not allowed to wear their hearing aids during the tests. These four cases will be discussed with the purpose of deriving a scheme of analysis which will take into consideration the Human Rights Acts and the current knowledge in the fields of audiology and ergonomics.

FIRST CASE: THE YOUNG FIREMAN

According to employers in the field of firefighting, job candidates must be physically fit, and their hearing thresholds must be within normal limits [1]. In fact, the rejection criterion is: "Hearing acuity loss by audiometric test of 20 dB or more for the speech frequencies (500-1000-2000 Hz) in either ear, or loss of speech reception of phonetically balanced words at or below 90 percent normal reception for either ear". The auditory tasks associated with firefighting are multiple. For instance, detection, identification, recognition and localization of sound sources in quiet and noisy surroundings should be performed in a reliable manner due to the possible threat to life if an auditory signal is missed. The young fireman in question filed a complaint after failing a pre-employment hearing test. He had not noticed his hearing loss prior to the test. In fact, the young man had a mild to moderate sensorineural hearing loss in the low frequencies, ranging up to 2 kHz. At the time of the test, the etiology of this hearing loss was not identified. He had passed all his theoretical and practical exams during his training and, by his account, had demonstrated that he could adapt himself and perform the task requirements of his job and those of daily life.

SECOND CASE: THE POLICE OFFICER

In this case, the subject suffered from a severe unilateral sensorineural hearing loss in the high frequencies from 2 to 8 kHz. The probable cause was exposure to gunfire during his training as a police officer or to tire explosion when the subject was a teenager. Like the fireman, this individual had never noticed his hearing problem prior to the evaluation provided by the employer's medical service. According to the employer's own regulations, the candidate did not meet the recruiting standard which stipulated that: "Hearing loss [should be] no greater than 30 dB in both ears in the 500 to 3000 Hz frequency range". The auditory tasks associated with police work duties are multiple and unlike the previous case, the employer has detailed in a document all the auditory tasks related to the job asked by the complainant. This case went to court last January. According to the E.N.T. specialist called by the employer to testify at the Canadian Human Rights Commission Tribunal, the audiogram is a reliable and valid tool to judge if candidates are able to perform all the auditory tasks detailed in the employer's document.

THIRD & FOURTH CASES: THE RAILROAD EMPLOYEES

In the last few years, the Canadian Human Rights Commission has received a number of complaints concerning the hearing standard included in Transport Canada's Order No. 0-9. All Canadian railway companies have to compy to the articles contained in this Order. This standard draws a distinction between new candidates and those already employed by the company[2]. For instance, the relevant articles of the Order stipulates that:

1) "No railway company shall accept for entrance to service in occupation referred to in Schedule I an applicant who was less than 20/20 hearing when tested by means of human voice or who has a hearing loss greater than 20 dB at frequencies of 500, 1000 and 2000 Hz when tested with an audiometer referred to in subsection 24(1)".

2) "No railway company shall retain in an occupation referred to in Schedule I an employee (a) who has hearing that is less than
(i) 15/20 in one ear and 5/20 in the other ear, or
(ii) 10/20 in each ear, or
(b) who has a hearing loss of 40 dB or greater in each ear at frequencies of 500, 1000 and 2000 Hz, except in assignments in which the hearing loss does not prevent the proper and safe performance of the assignments."

In an other part of the Order, it is clearly specified that: "No candidate shall wear a hearing aid during the hearing test conducted pursuant these Regulations".

Both of the railroad employees manifested a bilateral moderate sensorineural hearing loss and both had to wear hearing aids (one subject had one aid in the right ear and the other had two aids). With their hearing aids, their thresholds were in the range of 30 to 40 dB and the speech intelligibility scores were 80% in silence, at a conversational level of 45-50 dB. It is the complainants' position that, regardless of their uncorrected hearing, their corrected hearing enables them to perform all the duties associated with the jobs they are asking for. On the other hand, Tranpsort Canada holds that there are many problems in relying on hearing aids alone to overcome a handicap, and, therefore, it would not be prudent to allow railway service employees to use such aids. However, it could be noted that no detailed description of the tasks have been provided by the employers.

DISCUSSION

In the analysis of these four cases, different ethical and scientific aspects must be taken into consideration. First of all, it is important to clearly understand the Quebec or Canadian Human Rights Acts. Secondly, the analysis should be based on a good knowledge of the auditory abilities required to perform the job and the auditory status of the worker who is seeking employment. Thirdly, adaptation of the workplace or the use of assistive listening devices can be considered.

The Canadian Human Rights Act[3] clearly prohibits discriminatory policies and practices in matters related to employment. Section 7 states that it is a discriminatory practice to refuse to employ or continue to employ any individual or in the course of employment to differenciate adversely in relation to an employee on a prohibited ground of discrimination. Section 10 prohibits a policy, practice or agreement affecting recruitment, referral, hiring, promotion, training, apprenticeship, transfer or other employment-related matter if it deprives an individual or class of individuals of any employment opportunities on a prohibited ground of disrimination. Paragraph 14(a) of the Act provides exception to these prohibitions: it is not a discriminatory practice to refuse, exclude, expulse, suspend, limit, specify or prefer in relation to any employment if the employer establishes the practice to be based on a bona fide occupational requirement (BFOR). To insure that individuals may be assessed equally, the BFOR policy[4] does attempt to define parameters for the evaluation of an individual's performance or capacity to perform. In this respect, only the employer cited in the second example tried to prove that his hearing criterion was valid considering the list of auditory tasks the policeman had to perform. The final court judgement is not yet known however. All the other employers were convinced that the audiogram provided sufficient evidence of a candidate's ability or inability to meet the job requirements. According to these employers, the most important aspect to consider is the safety of the worker and his colleagues. The Commissions have stated that considerations of risk shall have diminishing weight according to whether they relate to the safety of others, the safety of the individual or the material loss.

With respect to the relation between auditory demands and capacities in the workplace, Hétu[5,6] has published two papers dealing with this topic. He states that job requirements involving auditory capacities are almost always based on medico-legal definitions of hearing that were adopted in order to compensate workers affected by noise-induced hearing loss. In fact, in such definitions, a certain amount of hearing loss demonstrated by the audiogram is the only tool used by most companies. It is now well known that if the auditory task is done in noisy surroundings, the frequency selectivity of the auditory system will be crucial. The temporal and spatial resolution are also important factors in many tasks. For instance, the localization of an alarm on a heavy truck moving in reverse on a noisy construction site is clearly important for the worker's safety. Moreover, speech perception in silence and in noise is not well predicted by the audiogram since it involves peripheral and central auditory processing. In short, it is impossible to predict all aspects of auditory performance based on a measurement of auditory sensitivity alone. In the four cases presented earlier, the audiogram was used to select candidates without considering the other auditory capacities except in the case of the fireman where the speech perception in silence was also considered. If we consider that most of the auditory tasks are performed in contexts where background noise is present in various levels, this tool is not necessarily the most useful means of auditory assessment.

With respect to the adaptation of the workplace, Hétu[5] notes that we should explore all the facilities which might compensate for the functional limitations associated with hearing loss. For example, the workplace may be adapted by reducing the background noise or the reverberation time and by selecting well designed warning sounds. Assistive listening devices such as hearing aids, FM system or infra-red system can also be considered. Concerning hearing aids, one important question often asked is whether or not hearing deteriorates if hearing aids are worn. A recent study by Hétu[7] demonstrated that hearing aids can be used as protective devices when the mold is closed and as a warning sounds detector by using the FM technology, for example. More research is needed in that field in order to apply the available technology. An other reason given by employers to refuse hearing aids is the possibility that the worker may loose it or that the batteries weaken with use. Hearing aid dealers and audiologists are aware of these risks but, it is now possible to find on the market hearing aids which are worn deep in the canal. When the battery fades, a warning signal informs the user to change it. Of course, each case has to be evaluated individually since some hearing loss can not be compensated sufficiently by intra-canal hearing aids.

CONCLUSION

In summary, the actual hearing criteria used by most employers should be revised in order to improve the selection process of new employees. Some audiological tools are now available that can provide a comprehensive assessment which is not restricted to auditory sensitivity. Measurement of frequency selectivity is one example[8]. Other tools would have to be developped to assess localization abilities, for example. It is also hoped that the cases presented here will enable the Human Rights Commissions to encourage employers to review their hearing criteria and to recognize that the audiogram is not a good predictor of the auditory capacities required in the real life situations. Employers will have to develop, with the help of ergonomists and audiologists, precise description of auditory tasks related to specific jobs so as to provide appropriate assessment of a candidate's abilities in fulfilling job-related tasks. Such assessment will also have to take into consideration possible adaptation of the workstation and the use of assistive listening devices. Ergonomists are used to deal with this type of relations between requirements and capacities in many fields such as muscular-skeletal problems and visual acuity. Unfortunately, very few studies have dealt with auditory capacities and job requirements. There is an urgent need for more studies in that field.

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FINITE-ELEMENT MODELLING OF THE NORMAL AND SURGICALLY REPAIRED CAT MIDDLE EAR

Hanif M. Ladak & W. Robert J. Funnell
Department of BioMedical Engineering
McGill University
Montréal, Québec, Canada, H3A 2B4.

INTRODUCTION

Discontinuity of the middle-ear ossicular chain results in conductive hearing loss. The most common type of ossicular discontinuity is caused by loss of the incudal long process; the second most common type is caused by loss of both the incudal long process and the stapes superstructure[1]. When the incudal long process alone is missing, a prosthesis (e.g., ossicular bone graft) can be fitted between the manubrium of the malleus and the head of the stapes; the resulting structure is called a malleus-stapes assembly (MSA). When both the incudal long process and the stapes superstructure are missing, a prosthesis can be fitted between the manubrium and the stapedial footplate; this structure is called a malleus-footplate assembly (MFA). The MSA and the MFA do not result in the complete elimination of hearing loss[7]. A quantitative understanding of the mechanics of the normal and surgically repaired middle ear should elucidate some of the reasons for this failure and would aid in the design of middle-ear prostheses.

This paper presents a finite-element model of the normal cat middle ear. The model was modified to investigate the effects of middle-ear surgery. The models are limited to low frequencies and sound pressure levels.

I. FINITE-ELEMENT MODELS

A. Normal Middle-Ear Model

Fig. 1 shows the normal middle-ear model which was developed by adding explicit representations of the stapedial footplate and cochlear load to an existing model of the cat eardrum[2]. The footplate is modelled as a thin plate with a thickened rim. Rigid plate elements are used to model the stapedial crura. The cochlear load acting upon the footplate is represented by in-plane and out-of-plane springs distributed around the periphery of the footplate. The malleus and incus are modelled as being effectively rigid with a fixed axis of rotation. The incudostapedial and incudomalleal joints are assumed to be rigid.

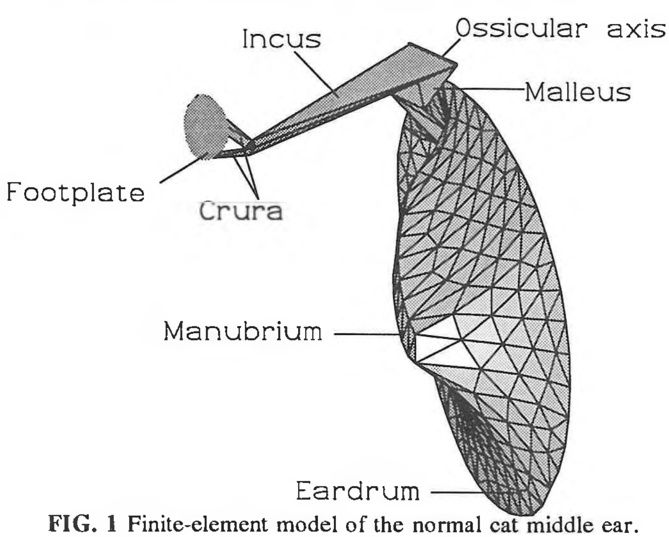

FIG. 1 Finite-element model of the normal cat middle ear.

B. MSA Model

It is assumed that the existing middle-ear structures being modelled have not been damaged by middle-ear disease or by the surgical procedure. Thus, the MSA model is identical to the normal middle-ear model except that the elements corresponding to the incus are replaced by a single brick element representing the prosthesis (shown in Fig. 2a).

C. MFA Model

As shown in Fig. 2b, the MFA model differs from the MSA model in that the crura are removed and the prosthesis makes direct contact with the footplate.

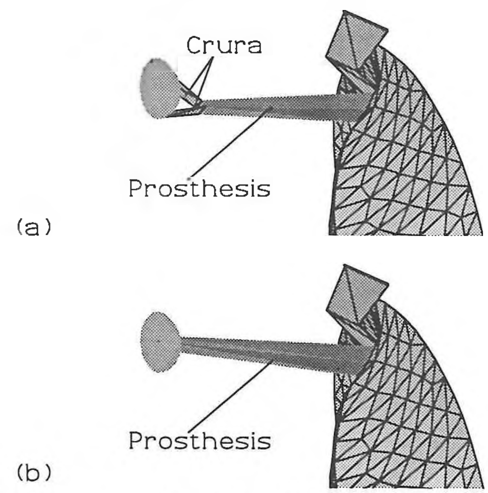

FIG. 2 Top portions of models of the surgically repaired cat middle ear. (a) MSA and (b) MFA.

II. RESULTS

A. Results for the Normal Middle-Ear Model

Displacement contours calculated for the eardrum in the normal middle-ear model are qualitatively similar to the low-frequency contours observed experimentally by Khanna[4]. The maximal displacement on the eardrum is 484 nm at 100 dB SPL and occurs in the posterior region of the pars tensa. Extrapolating the data of Tonndorf and Khanna[5] to 100 dB SPL gives a maximal drum displacement of 420 nm.

The umbo displacement calculated using the present model is 161 nm. The data of Tonndorf and Khanna[5] give an umbo displacement of 191 nm.

The out-of-plane component of footplate displacement at its centre provides an estimate of the volume displacement of the cochlear fluids. For the present model, the out-of-plane component has an amplitude of 93 nm. Guinan and Peake[3] measured the displacement of the footplate in anesthetized cats to be 85 nm which agrees well with the model.

B. Comparison of Normal, MSA and MFA Models

Fig. 3 shows the maximum eardrum displacement, the umbo displacement and the out-of-plane component of footplate displacement for the normal middle-ear, MSA and MFA models. The displacements for the normal middle-ear model and the MSA model with an intact ossicular axis are virtually identical. The displacement amplitudes for the MFA model with an intact ossicular axis are slightly larger than those of the normal middle-ear model; however, whereas the footplate in those models behaves as a rigid-body, that in the MFA model bulges as shown in Fig. 4. The displacement amplitudes of the MSA and MFA models increase when the ossicular axis is removed.

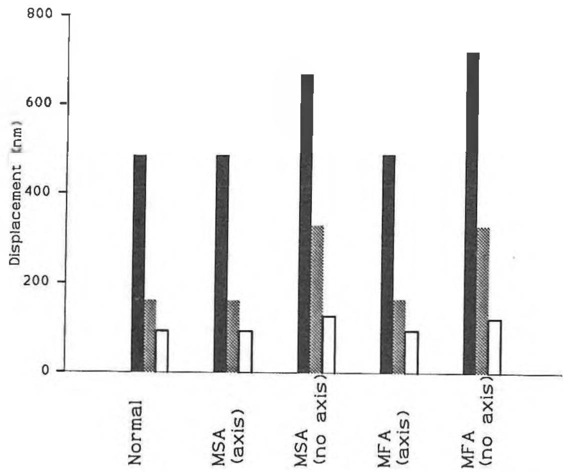

FIG. 3 Maximum eardrum (black), umbo (gray) and footplate (white) displacement amplitudes for the normal middle-ear, MSA and MFA models.

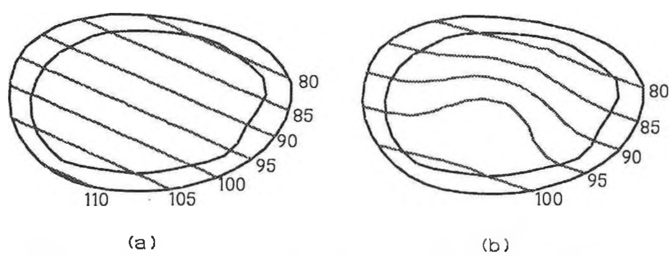

FIG. 4 Contours for the out-of-plane component of footplate displacement for (a) normal middle-ear model and (b) MFA model. All displacements are in nanometres (nm).

C. Parameter Variations

Displacements of the footplate in the MFA model are very sensitive to the stiffness and thickness of the footplate and to the stiffness of the annular ligament. They are not sensitive to the Poisson's ratio of the footplate.

D. Prosthesis Location

In the MSA and in the MFA, it is possible to position the prosthesis at various locations along the manubrium. (In the above simulations, the prosthesis was placed at the upper end of the manubrium.) Moving the prosthesis down the manubrium does not significantly alter the mechanical behaviour of the footplate; this result is consistent with the experimental results of Tonndorf and Pastaci[6]. This behaviour is expected since the ossicular joints in the model are rigid causing the manubrium, prosthesis and footplate to act as a single rigid body.

If the joints between the prosthesis and the bones are made flexible, the footplate's mechanical behaviour will be affected by prosthesis location. For example, consider an extreme case where pin joints are assumed, but the prosthesis is still rigid; this situation can be modelled using a truss element for the prosthesis. Fig. 5 shows the out-of-plane component of footplate displacement at its centre for the MFA model without an ossicular axis as the truss element is positioned at various locations along the manubrium; also shown are results for inflexible joints (i.e., using a brick element). When the joints are flexible, the largest footplate displacements are obtained when the prosthesis is located close to the upper end of the manubrium.

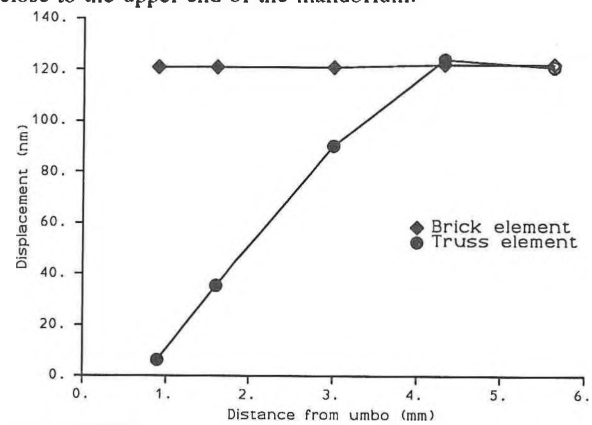

FIG. 5 Effect of prosthesis location (along the manubrium) on out-of-plane component of footplate displacement, in MFA model without ossicular axis.

III. CONCLUSIONS

The normal component of footplate displacement is virtually identical in the normal middle-ear model and the MSA model with an intact ossicular axis. For the MFA model, the mode of vibration of the footplate is somewhat different from that of the normal middle-ear model: direct contact between the prosthesis and the footplate results in bulging of the footplate.

In both the MSA and the MFA models, the position of the prosthesis along the manubrium does not affect the mode of vibration of the footplate as long as the joints between the prosthesis and the bones are rigid. When the joints are completely flexible, the largest footplate displacements occur when the prosthesis is closest to the upper end of the manubrium. More work is needed to characterize the rigidity of the joints.

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MIDDLE EAR STAPEDIUS MUSCLE ACOUSTIC REFLEX AND LARYNGEAL AMPLITUDE RESPONSE*

Brenda Orser
Queen's University, Kingston, Canada
orserb@qucdn.queensu.ca

1. Introduction

This study describes the ascending auditory pathway of the middle ear stapedius muscle acoustic reflex. The purpose is to localize the reflex's role in transmitting complex speech sound, specifically, resonant and prosodic elements — stress and syllables — produced in the laryngeal-pharyngeal region of the vocal tract. The corresponding acoustic components are amplitude and intensity. The processing locale of these speech elements apparently differs from that of segment processing which occurs in the peripheral auditory system. Acoustic and psychoacoustical data[1,4] suggest a system which, in addition to the peripheral auditory system, operates in sound identification; moreover, this system creates some form of representation of the laryngeal and prosodic elements in the human speech signal.[3] These observations are summarized here in reference to selection for a laryngeal and prosodic inventory in natural language.

Stevens asks how constraints imposed by the auditory system shape the inventory of sounds used in language. A second question, beyond this paper's scope, is how auditory processing imposes a classificatory structure on the phonetic sounds of language. The present, informal study contributes to deriving a system of non-peripheral bioacoustic constraints on the auditory representation of language.

2. Rapid change in the spectrum and its amplitude characteristics

Production of laryngeal-pharyngeal complexity occurs in the posterior region of the vocal tract, at/near the larynx. In addition to the resonant low-frequency spectral energy, and within a 50msec duration, this segment may contain periodicity within the same spectral region due to vibration of the vocal cords.[2] The rapid spectrum change which occurs corresponds to a consonantal sound produced with a narrow constriction along the midline of the vocal tract.[3]

In an utterance containing a laryngeal-pharyngeal consonant segment, rapid-spectrum change occurs at the segment's onset in the low-frequency area corresponding to an articulatory movement from relative constriction into that of vowel production, an unconstricted vocal tract;[3:64] rapid decrease in the first formant accompanies a decrease in the size of the constriction in the oral cavity.

There also may be an abrupt rise (or fall) in an utterance's amplitude; this if it occurs is independent of any spectral change. The class of speech sounds in effect are *interrupted* segments, mainly stops but also affricates and resonants. Studies remain inconclusive over whether there is a natural perceptual boundary when this relative amplitude, i.e., interrupted segment reaches a particular value; or, whether some other acoustic attribute forms the primary cue for distinguishing, for example, nasal resonants from stop plosives.[3:69]

Pharyngeal resonants occurring at syllable onsets and relative to stressed vowels are comprised of a relatively low first formant (200-600Hz) and within that a low db; nevertheless the amplitude energy and changes in that are present throughout the spectral range present during the pharyngeal's production. Difficulties arise in defining the perceptual effects acting on amplitude characteristics within the 200-600Hz frequency range. For example, rise-time is indeterminant; amplitude gain and slope are not measured psychoacoustically. Though conservation of amplitude gain can be resolved linguistically in laryngeal segmental and prosodic categories, e.g. syllable and stress, formal work is required in the parameterization and accompanying measurement of the amplitude change, i.e., amplitude gain, in the low-frequency spectrum. Further, how the amplitude is managed and its energy conserved is unclear, though there are suggestions of a transformation and significant diversion from the peripheral pathway.[1,3]

3. Peripheral System

The lower brain stem auditory pathway is a system of olivocochlear efferent tracts in a bundle, (OCB). It responds to phonetics segments, discriminating in terms of instant-by-instant speech perception. However, the response is not constant to adjacent pairs of phonetic sounds differing only by a fixed, specified distance along an acoustic dimension defined by a phonetic category, i.e., a limited acoustic range of stimuli. Responses are unpredicted by the physical quanta; they correspond instead to linguistic features — that is, natural classes of segments. This linguistic-perceptual disparity is unexplained by the OCB and its related responses.[1,3]

The OCB efferent system has an influence on the cochlea's response to sound. Further, the acoustic middle ear reflexes comprised of the stapedius (St) and tensor tympani (TT) muscles are important regulators of the auditory input to the cochlea. Relations between the middle ear reflexes — in particular, the St — and the OCB appear non-monotonic however, though it is widely held that the acoustic middle ear reflexes constitute one of the feedback loops controlling activity throughout the afferent auditory system.[1:101] The response to each of these as registered in the cochlear is nonlinear;[2,3] also, as described above, psychoacoustical data show the lack of a constant in subjects' ability to discriminate ordered pairs of phonetic categorical stimuli.

The following paragraph is from Borg's description of the stapedius reflex pathway.[1:115-9] The first neuron — the primary acoustic neuron from the haircells to the cochlear nucleus (CN) — synapses with the second-order neuron. This the first synapse of the St reflex pathway and is situated in the ventral cochlear nucleus (VCN). For pure tones, the dorsal cochlear nucleus (DCN) and posteroventral nucleus' neurons are not involved in the reflex. No primary fibres pass the cochlear nucleus (CN) in the trapezoid body (TB) to higher centres (in the auditory cortex), therefore, the VCN is the only possible location of the first synapse.

The second neuron passes in the TB and has contact directly with the ipsilateral St motoneurons in the nucleus of the facial nerve VII, (Nc7). Via interneurons in, or near, the medial superior olive (MSO), the second neuron relays to the third neuron, to the ipsi- and contralateral facial nucleus, Nc7. The 3rd neuron and the 4th, the motoneuron in Nc7, follows the facial nerve to the St in the middle ear.

There is no indication that the stapedial reflex, which in effect regulates sound transmission to the cochlea, occurs on a pathway directly between the auditory cortex and the cochlea, i.e., the peripheral auditory system. In fact, the facial nerve innervates the St. The observations point to the involvement of another system.[1,4] This system which may operate temporally as a parallel process to that occurring along the pathway in the OCB is similar to the OCB due to its function as an inhibitor. This, as Borg demonstrated could be the extrapyramidal system.[1] This, more so than the descending auditory system, may be involved in co-activating the middle ear muscles found in a number of motor reactions such as vocalization and body movements.[1:119]

4. Non-peripheral System

The extrapyramidal system might at best be derived formally.[5] Here, its description is preliminary only, from data on systematic observations. First, that cochlear potentials, in guinea pigs under light anesthesia, were changed in magnitude at the moment of a "spontaneous contraction" of only the TT (tympanic) muscle's reflexes. The change was greatest for low tones, and in the region of 100Hz amounted to a reduction of 35 to 40 db, with the reduction becoming progressively less as frequency was raised (to 0 db at about 1kHz). Further, transmission was enhanced from 1.3 to 1.8kHz (4 db @ 1.5kHz) and there was no effect at 2+kHz.[4] Wever & Bray, cited in the same study, experimenting on the cat, but on the stapedius muscle, indicated similar systematic changes in sound transmission, i.e., the cochlea potential changed as a result of tension in the normal direction of action on the tendon of the stapedius muscle. A new design corrected for the reduction in transmission being the result of a damping action by the muscle, more-so than as an effect of tension, i.e., neurological inhibitory action; excitation slightly above the threshold for each the TT and St reflex showed a reduction in the transmission of low tones, but with the reduced effect diminishing progressively and disappearing at around 600 Hz and enhanced between 600-1000Hz. A present concern is whether the spectrum's amplitude component and any gain at the slightly above-threshold intensities and in the low frequency range from 200 to 600 Hz is managed and conserved by the non-peripheral system. As mentioned, the amplitude change in this range is resolved linguistically, in feature detection according to natural classes of resonants (vs. stops) and in syllable and stress structure.

5. Concluding remarks

Conservation of the amplitude slope in the ascending, not only the lower brain stem auditory pathway is a problem of sound identification. The problem also arises in binaural localization using cues of interaural time and intensity differences. After Borg, it is likely that the lower brain stem organization of somamotor reactions to sound can be studied best as a function of the middle ear reflex system, e.g. the pathways at the VCN and the SOC, at least when pure tones are used as stimuli. With respect to complex stimuli and monaural localization, higher auditory pathways are involved.[5] In terms of sound transmission for identification, on the other hand the middle ear muscles and specifically the stapedius muscle is less connected to the descending auditory system than is generally assumed and appears more related to the extrapyramidal, non-peripheral system.

6. Notes & References

* Support for this research by the Strathy Language Unit, Queen's University, Kingston and by the American Philosophical Society, Phillips Fund is gratefully acknowledged.

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INSTITUTE OF HEARING-ACCESSIBILITY RESEARCH (IHEAR)

I am please to announce the creation of the Institute for Hearing Accessibility Research (IHEAR) at the University of British Columbia.

IHEAR will be a centre for research and training in the field of hearing accessibility -- improving the acoustical accessibility of the world -- a major issue affecting normal-hearing and, in particular, hard-of-hearing people.

UBC is in a unique position to establish pre-eminence in this field because of the research already under way here, the support of professionals and industry who want to participate in research and development, the support from the hard-of-hearing community, and Vancouver's world leadership in disability issues.

A unique feature of the institute is the involvement of the hard-of-hearing community, which was represented on the planning committee. It will continue to have a significant role to play.

The institute will capitalize on the broad spectrum of research related to hearing that already exists on campus in several departments and faculties. It will make it easier for resarchers to set up multidisciplinary projects, share equipment, coordinate graduate students and apply for grants.

Some of the topics the institute will address include acoustical conditions, rehabilitation of the hard of hearing elderly, psycho-social issues, hearing aids and assistive listening devices and hearing accessibility in the workplace, schools and health care.

Immediate plans call for seminars where researchers can present their work. The institute will also build links with professionals from the health care system, industry, education, and other fields related to hearing issues.

Long-range plans for the institute include funding new interdisciplinary projects, organization of research conferences and workshops sponsorship of scholarships for graduate students and support of stipends for visiting scholars.

Murray Hodgson, UBC

The Canadian Acoustical Association
l'Association Canadienne d'Acoustique

ANNUAL GENERAL MEETING / ASSEMBLEE GENERALE ANNUELLE

Date / Date:	October 20 / 20 octobre 1994
Time / Heure:	5:00 p.m. / 17h00
Place / Endroit:	Citadel Inn, Ottawa

AGENDA / ORDRE DU JOUR

1. Welcome (R. Hétu)
2. Review of the minutes of the last AGM (R. Hétu)
3. President's report (R. Hétu)
4. Executive Secretary's report (J. Hemingway)
5. Treasurer's (G. Bolstad)
6. Editor's report (M. Hodgson)
7. Membership report (D. Jamieson)
8. Awards Committee report
9. Acoustics Week in Canada reports:
   - AWC '93 Toronto (S. Abel)
   - AWC '94 Ottawa (T. Nightingale)
   - AWC '95 Québec (B. Gosselin)
   - Seminar organization
10. Other business:
    Business arising from the Board Meeting
    New business from the floor
11. Elections (D. Chapman)

Preliminary Announcement and Call for Papers

SDVNC '95

International Conference on Structural Dynamics, Vibration, Noise and Control

December 4-7, 1995, Hong Kong

SDVNC'95, An International Conference on Structural Dynamics, Vibration, Noise and Control, will be held in Hong Kong, December 4-7, 1995. The Chinese Society for Vibration Engineering (CSVE) in conjunction with the Hong Kong Polytechnic University is sponsoring this conference in cooperation with several cosponsor professional societies in the United States, Canada, and other countries. The scope of the conference includes: * Structural Dynamics; * Structural System Modeling, Identification & Control; * Launch Loads and Dynamics; * Rotocraft Dynamics; * Turbomachinery Dynamics; * Composite Structural Dynamics; * Computational Structural Dynamics; * Non-linear Dynamics; * Vibration & Noise Analysis & Control; * Vibration & Noise Measurement & Signal Processing; * Fluid-Structure Interaction; * Control-Structure Interaction; * Active- & Passive-damping; * Piezoelectric Actuator & Sensor; * Vibration Isolator & Attenuator; * Shock Isolation & Absorption; * Damage Detection; * Damage Tolerance of Composite Materials and Structures; * Computer Application & Algorithm; * Experimental Dynamics and Testing; * Sensors & Actuators.

Abstracts (three copies) of contributed papers are due Dec.31, 1994. The main text of the abstract should be approximately 300 words in length, and should contain the title of the paper followed by author's name, affiliation, address and tel. & fax. numbers. Preliminary acceptance will be notified before March 1st, 1995. Camera-ready manuscript must be completed before Sept. 1st, 1995. The registration form will be sent to the author as an enclosure of the final acceptance notice. The registration fee is $450.00 (U.S.) which includes the conference proceedings, reception, and banquet.

Abstract may be submitted to one of the following contacts in the United States and Canada:

Prof. Richard W. Guy, Center for Building Studies, Concordia University, Montreal, Canada H3G 1M8. Tel. (514)848-3191, Fax. (514)848-7965;

Dr. Ramesh B. Malla, Dept. of Civil Engineering, University of Connecticut, Storrs, Connecticut 06269-3037, USA. Tel. (203)486-3683, Fax. (203)486-2298;

Dr. Ji Yao Shen, Dept. of Mechanical Engineering, North Carolina A&T State University, Greensboro, NC 27411, USA. Tel. (910)334-7254, Fax. (910)334-7417.

Or, directly to the Conference Chair:

Prof. De Mao Zhu, Chinese Society for Vibration Engineering, Nanjing University of Aeronautics & Astronautics, Nanjing, Jiangsu 210016, the People's Republic of China, Tel. 86-25-4492492, Fax. 86-25-4498069.

CONTROLLING CONTROL VALVE NOISE: A CASE STUDY

Ramani Ramakrishnan

Vibron Limited, Mississauga, Ontario

Robert Gaspar

SPAARG Engineering Limited, Windsor, Ontario

1.0 INTRODUCTION

Transfer stations along long pipe line systems of gas utilities are used for isolating individual pipe lines for routine maintenance work and to supply local distribution networks. Control valves, located in short pipes, are operated in a throttled condition to empty and refill individual lengths of pipe lines. The noise level generated by the conventional valves, headers and associated piping during operations can be as high as 140 dBA. A case study, involving the design of control valves for a proposed transfer facility is presented in this paper. Applicable noise guideline procedures and feasible noise control measures are highlighted. The cost estimates of the noise control measures are also discussed.

2.0 NOISE CRITERIA

Transfer facilities are usually located in remote areas, considered rural and the applicable criteria are listed in the Ontario Ministry of the Environment and Energy (MOEE) publication NPC-132, "Guidelines for Noise Control in Rural Areas" [1]. In rural areas, within 30 m of a dwelling or a camping area, in any hour, the equivalent sound level (L_{EQ}) of a stationary source should not exceed the ninetieth percentile sound level (L_{90}) of the natural environment, by more than 10 dB. Further, L_{90} of the stationary source should not exceed the L_{90} of the natural environment, by more than 5 dB. The minimum MOEE noise limit is L_{EQ} and/or L_{90} of 40 dBA. The Energy Resources Conservation Board (ERCB) of Alberta have issued a Noise Control Directive for controlling noise from installations such as transfer facilities [2]. The directive is straight forward to apply and the limit is evaluated from a base night time sound level, L_{EQ}, in dBA. Unlike the Ontario limits, adjustment factors for day time operations and ambient environment can be applied.

For a typical valve site in rural areas, the MOEE limit is 40 dBA at the nearest receptor even if the blowdown or pipe equalization operations take place during the day-time hours and the operations take place for only a day. However, the ERCB limit at the nearest receptor is 70 dBA (40, base level + 10 dBA for day time operations + 5 dBA for absence of strong tonal components + 15 dBA for periodic one day only activity). It is seen that the Ontario limit is very restrictive and ERCB limits provide additional compensations. It is important to note that the Ontario limit of 40 dBA has been established with little consideration for the type activity undertaken at a typical transfer station. The Ontario Energy Board has reviewed MOEE guidelines and has indicated that the utility is to exercise a best effort to reduce noise from transfer stations. Using 55 dBA at a hypothetical receptor located at a distance of 100 m from the station, the applicable limit would be 70 dBA along the property line of the station.

3.0 NOISE LEVELS OF A VALVE SITE

The lay out of a typical transfer station is shown in Figure 1. The station consists of a pair of headers that are connected to different main pipe lines. The headers are used to purge gas from isolated sections of a main pipe line and for pressure equalization of an empty pipe line after the required maintenance is completed. The time required for the purging operations and for refilling usually varies between 8 to 24 hours.

The two headers in the existing station are connected to four main pipe lines. A small line connects the two headers and the blow-down pipe. Three small valves, plug type, control the purging and equalizations process. The valves numbers are shown in Figure 1.

The purging and equalization operations could only be simulated, since the four lines were in peak demand during the measurement period. The venting operation was simulated by: valve 1-closed; valve 3 (silencer valve) - open; and valve 2 was throttled. The pipe equalization was simulated by: (a) valve 1-open; valve 3-closed; and valve 2 was throttled and (b) valve 2-open; valve 3-closed; and valve 1 was throttled. The header (#2) used for the simulation had a residual low flow at a pressure of 200 psi due to a leaky main valve that connects the header to one of the main pipes. The header therefore could not be purged completely. The gas flow is always from valve 1 to valve 2 for pipe equalization operation. The exact location of the throttling valve could not be determined and it was between 1/4 open to 1/2 open.

The noise levels were measured (Locations are shown in Figure 1) at location A for purging; at location B for valve 1 throttling; and at location C for valve 2 throttling. The results are reported in Figure 2. The noise level from the venting operation through valve 2 for a line pressure of 350 psi was measured near the silencer. The overall sound pressure levels (SPL) were 101 dB and 88 dBA. The silencer performance increase with frequency and is shown in Figure 2. The valve throttling noise levels during pipe equalization process are also shown in Figure 2. The line pressures were approximately 800 psi before the valve and 200 psi after the valve due to the residual flow in the header. The overall SPL increased from 112 dBA to 124 dBA for the two valves. A general increase of the spectrum by about 10-15 dB was evident between the two conditions at all frequencies. The peak frequency range was from 1000 Hz to 4000 Hz for valve 1 throttling, whereas, the peak frequency range was from 400 Hz to 4000 Hz for valve 2 throttling. The level at 400 Hz increased from 85 to 105 dB and the level at 4000 Hz increased from 100 to 110 dB.

4.0 NOISE PREDICTIONS

Shock, shock-turbulence interaction, turbulence mixing and flow separation phenomena are common for the high pressure ratios and the resulting choked flow commonly encountered in control valves. This confined jet turbulent mixing process takes place after the valve-throttling element within a length anywhere from 4 to 10 diameters downstream of the valve and is the main cause of the aerodynamically generated sound field in the pipe. Details of valve noise generation process are reported in references 3 thru` 6.

The simplified prediction model from Reference 6 is applied in our study. In addition, the empirical procedure based on the ISA Standard [5], developed by Fishers Control Company is also used in the predictions. The models assume inline valves operating into a long downstream pipe without any sudden area changes. Most of the required inputs can be obtained from handbooks. However, two main inputs, valve flow coefficient C_v and valve pressure recovery coefficient F_l, are usually obtained from valve manufacturers and are only approximate empirical values. The following is assumed in the prediction model: upstream and downstream pressure are 800 and 200 psi; the valve were 1/2 open; and the valve pipes are very long with no sudden area changes. The predicted noise levels at a distance of 1 m from the valve are 127 and 128 dBA by the two procedures. It is seen that the two methods predict levels within one dB of each other.

The predicted levels must be further adjusted to reflect the actual operating conditions. It was pointed earlier that the exact valve position could not be determined exactly and hence the coefficients C_v and F_l are only approximate. The uncertainty factor is around 6 dB. The valve 2 was 1 m away from its junction with the header and hence additional dipole type sources were created by the turbulence/mixing region interacting with the sudden area change. The ring frequencies of the large header are excited which is reflected in the increase of the frequency range (from 400 Hz to 4000 Hz) for valve 2 throttling. If one used approximate adjustment factors for the above, the predicted noise level at 1 m from the valve for Valve 2 throttling is 126 dBA (127 - 6 dB factor for C_v + 5 dB due to sudden area change). On the other hand, the source region for valve 1 throttling was more than 20 feet away from its junction with the header and further, most of the connecting pipe from valve 1 and the empty header (#2) is buried under ground. A portion of the noise source region is therefore shielded. The predicted noise level at 1 m from the valve for Valve 1 throttling is 111 dBA (127 - 6 dB factor for C_v - 10 dB due to lagging effect by the ground). The predicted results are within 3 dB of the measured results. The simplified model can be used to predict potential noise impact from proposed transfer station with reasonable accuracy.

The lay out of a proposed transfer station is very similar to the existing station shown in Figure 1. Design details of pipe sizes, header-pipe connection details and valve selections were to be finalised depending on the noise assessment. The noise criteria for the proposed site was 70 dBA at the property line of the station. The simple Fisher Controls prediction model was used to predict the noise levels from two different valves. Valves of one manufacturer (B) was consistently producing 6 dB more noise than the other manufacturer (A). Valves of Manufacturer A was used for further analysis. It must be remembered that two valves (#1 and 2 of Figure 1) are needed for the throttling operations as well as for controlling pipe line flows in both directions.

The base noise level is evaluated with the assumption that the throttling valves are located sufficiently (minimum of 15 to 20 pipe diameters) away from the connecting headers. The property line was approximately 25 metres away from the nearest valve-header junction and a conservative 20 dB distance attenuation was assumed in the evaluations. The design pipe pressure is 900 psi and the valve is kept 1/2 open during throttling. The base noise level from a regular plug valve (large line) is 122 dBA. The noise level reduces by 4 dB to 118 dBA if the size of the valve (and the connecting line) is reduced by 2" (small line). However the flow rate is reduced by 30%, i.e., the time taken to fill the empty main line increases considerably. Similarly, if the valve is throttled by keeping the valve 1/4 open, the noise level reduces by 7 dB, but the flow rate is reduced by more than 50%. Even though the noise level is reduced at the property line with the above modifications, the level would last for a longer period. The evaluations show that the property line noise levels exceed the criteria by about 40 dB for the worst case condition. Feasible and practical noise control conditions are described in the next section.

5.0 NOISE CONTROL MEASURES AND COST

The proposed operating configuration requires 40 dB of noise reduction. Five possible measures can be used to reduce the valve noise. They are outlined below. (Cost of Regular Valves is CD$12,500).

A) Replace the valves with 'Quiet Venturi' type; (CD$8,000)
B) Bury the valves and as much as the connecting pipes;
C) Install a silencer in the connecting line with the condition that flow is from the throttling valve to the farther header (silencer insertion loss is 25 dB); (CD$30,000)
D) Lag the header and the entire exposed connecting line (lagging insertion loss is 20 dB);(CD$80,000)
E) Replace the valves with "smart" valves.(CD$150,000 for two)

Smart valves provide substantial noise benefit by applying the "tortuous path approach." The flow expands through a series stacked discs and thereby reducing the noise generation. The cost associated with the control measures as well as expected noise reduction of the measures are outlined in Table 1.

6.0 CONCLUSIONS

The noise emission from control valves were measured and compared with applicable prediction models. The suitable prediction models were used to estimate the noise output of control valves at a proposed transfer station. Various control options were evaluated and appropriate control recommendations were presented to satisfy a noise control limit of 70 dBA at the property line.

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Table 1. Noise Control Potential and Associated Costs (Noise Levels in dBA at 20 m from the valves)

Noise Control Combination	Noise Level (Large Line)	Noise Level (Small Line)
Regular Valves (2)	122	118
Item A (2)	115	113
Item B	117	113
Item C	97	93
Item D	102	98
Items A + B	110	108
Items A + C	90	88
Items A + D	95	93
Items A + C + D	70	68
Item E (2)	70	70

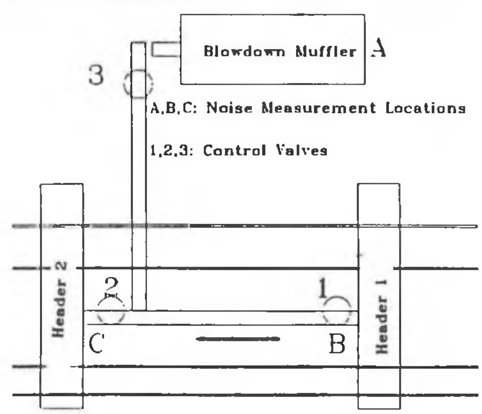
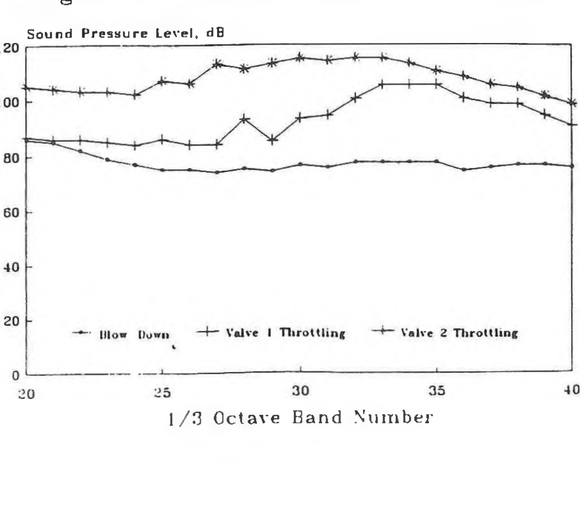


Figure 1. Lay Out of a Transfer Station

Figure 2. Control Valve Noise Levels



PIEZOFILM SENSORS FOR THE DETECTION OF PROPAGATING ACOUSTIC PRESSURE IN PIPES

B.V. Chapnik and I.G. Currie
Department of Mechanical Engineering
University of Toronto
Toronto, Canada M5S 1A4

INTRODUCTION

There are a number of situations in industry in which one may wish to measure the acoustic pressures in a piping system, but for which standard microphone methods are expensive or difficult to implement. These include cases of high fluid temperature, pressure or velocity, and material handling systems in which there are particulates or debris in the flow.

As an alternative, some researchers have investigated the possibility of using structural sensors attached to the piping, which is coupled to the interior acoustic field, to estimate the pressure [1,2]. This method has shown reasonable promise for axisymmetric waves well below the ring frequency of the piping.

This study describes the theory of the method, and the results of some experimentation performed at the University of Toronto using piezofilm ring sensors attached to a rubber tube, with air as the contained fluid. Some approximations which can be made for low frequencies are described, and some practical limitations of the method presented.

THEORY

The dispersion characteristics of free waves in infinitely long thin walled cylinders of finite shell impedance, including the effect of contained fluid, has been investigated by Fuller and Fahy [3], among others. The *in vacuo* shell dynamics are described by appropriate coupled equations of motion, and the pressure field in the pipe is assumed to take on the form of an acoustic wave equation in cylindrical coordinates. The wave dynamics of the shell and the fluid are then coupled through the boundary condition which ensures that the radial velocity of the shell and fluid at the pipe wall are equal.

Assuming travelling shell and pressure waves only, we may write:

$$u = \sum_s U_s \, e^{i(k_s^x - \omega t + \frac{\pi}{2})}$$

$$w = \sum_s W_s \, e^{i(k_s^x - \omega t)}$$

$$p = \sum_s P_s \, e^{i(k_s^x - \omega t)} \, J_0(k_s^r r)$$

where u, w are the axial and radial displacements of the pipe wall (torsional motion is uncoupled for axisymmetric waves and can be neglected), p is the acoustic pressure, and k_s^x, k_s^r are the axial and radial wavenumbers, respectively, related to the free wavenumber, k, through the Helmholtz equation by $k_s^r = \sqrt{k^2 - (k_s^x)^2}$.

Substitution of these forms into the equations of motion and coupling p, w through the boundary condition yields the characteristic equation of propagation, from which the axial wavenumber can be solved:

$$L = -\Omega^2 + 1 + \beta^2 \left[(k_s^x a)^4 + \frac{2+\nu}{2(1-\nu)} \right] + \frac{\Omega^2 \rho_f a J_0(k_s^r a)}{\rho_s h (k_s^r a) J_1(k_s^r a)} - \frac{\nu^2 (k_s^x a)^2}{-\Omega^2 + (k_s^x a)^2} = 0$$

The terms in L represent, in sequence, the pipe mass, membrane stiffness and bending stiffness (entire term in square brackets), the fluid loading term, and longitudinal effects coupled through Poisson's ratio. As the fluid loading term is non-linear, it may be expanded in a power series to linearize the equation to any desired degree of accuracy. At low frequencies, the solutions for k_s^x may be shown to include two real roots, one with energy predominantly in the fluid and characterized by relatively large radial motion, the other primarily a longitudinal shell mode. There are also two complex conjugate roots, representing solutions for a bending near field on the pipe wall, and an infinite number of imaginary roots representing evanescent acoustic modes.

Once the type of propagating mode is determined (i.e. an acoustic wave in the fluid will primarily excite the first mode while a structural excitation will excite the longitudinal shell mode, and shell discontinuities will excite the complex near field bending modes in that vicinity), the boundary condition can be used to solve for the pressure corresponding to a given radial displacement:

$$p = \frac{\omega^2 \rho_f a J_0(k_s^r a)}{(k_s^r a) J_1(k_s^r a)} \, w$$

Here the radial wavenumber corresponds to the propagating axial wavenumber, or a series can be used if more than one type of propagating wave exists at the cross section where w is measured.

EXPERIMENTS

Experiments have been performed to test the method, using a rubber tube down which acoustic energy is introduced via a 150W loudspeaker, coupled by a lined contraction to minimize the loading effect back on the woofer as the excitation frequency is varied. Piezofilm strips are bonded around the circumference to measure the axisymmetric ring strain, which is proportional to the radial displacement, w. These sensors also serve to filter out higher order circumferential motions. The corresponding acoustic pressure p calculated from the above equations are compared to the pressure measured inside the tube wall using a microphone probe. Figure 1 illustrates the experimental apparatus.

The experiments and theory are slightly different than described above, in that sensor pairs are used, and the average pressure between them calculated, while the actual pressure between the two is measured. This has the added advantage that the propagating part of the pressure can be separated out in a manner similar to an intensity measurement, but the theory is somewhat more complex, and is described elsewhere [1].

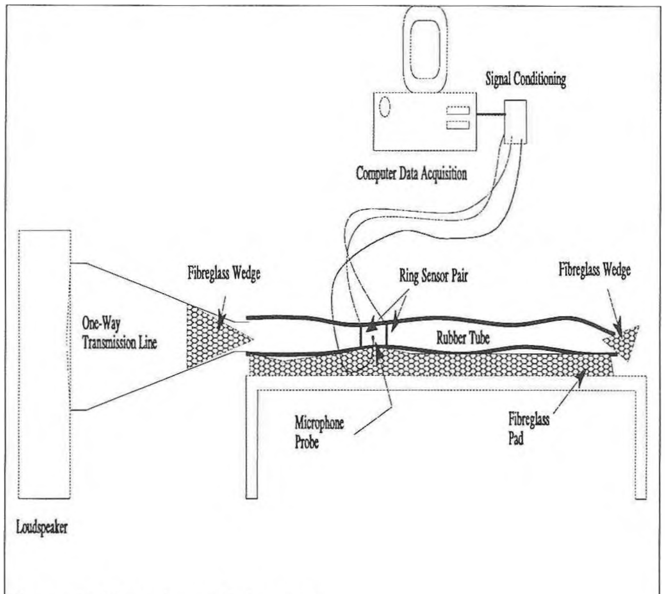


Figure 1: Experimental Apparatus

The experimental results indicate reasonably good agreement with theory for practical engineering purposes. At a number of frequencies below the ring frequency of the tube, the predicted pressure was on average within 6 dB of the measured pressure, and these differences remained highly constant through a range of signal amplitudes. Figure 2 illustrates the relative accuracy of the predicted pressure (re the measured pressure) at these frequencies, both unadjusted and corrected for the mean difference, which is approximately -3 dB, to account for such effects as imperfect bonding and imperfect shear transfer through the tube thickness. With the predicted accuracy so adjusted, the differences are within 3 dB, except at one frequency which corresponds to a full wavelength in the tube. That is, once a calculated or arbitrary figure for bonding losses is included, the sensors will provide a reasonably accurate estimate of the propagating pressure, provided that there are no significant standing waves at the frequency of interest, for which the above theory does not account.

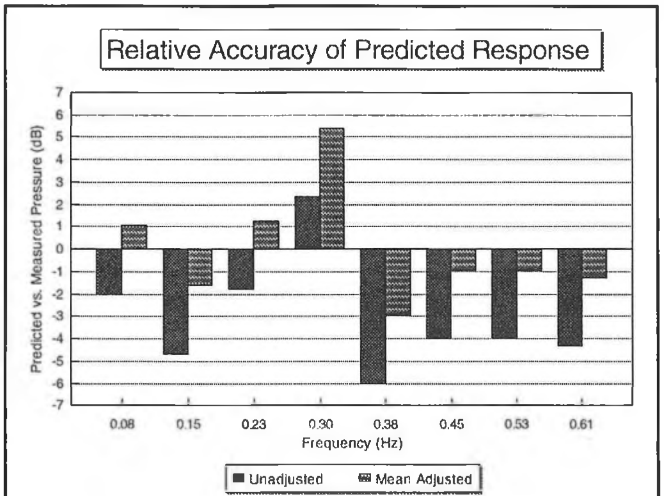


Figure 2

SIMPLIFICATIONS

For fluid borne excitation, if the forcing frequency is much smaller than the pipe ring frequency, then motion of the pipe wall is essentially controlled by its membrane stiffness. Provided that the sensor has negligible material properties in comparison with the pipe (which is **not** the case for the experiments described above, but will be the case in many industrial situations involving steel or hard plastic piping), then the pressure may be related to the displacement at the sensor location through the static membrane stiffness of the pipe [2]:

$$p = \frac{Eh}{a^2} w$$

Although this equation is useful only under the assumptions described above, it is much simpler to implement than the boundary condition equation described earlier. Also, for hard walled pipes, bonding losses are likely to be negligible when a stiff bonding adhesive such as epoxy or cyanoacrilate is used.

CONCLUSIONS

A method for estimating the internal acoustic pressure in piping systems using ring sensors attached to the pipe has been described. For arbitrary axisymmetric fluid borne waves, the internal pressure estimate is a function of the excitation frequency and wavenumber, which depends on mass and bending stiffness effects as well as the membrane stiffness. At frequencies well below the pipe ring frequency, the internal pressure estimate is a simple function of the membrane stiffness. Experimental results show that this method can be used to obtain reasonably good estimates when standing waves are not present, although imperfect bonding and shear transfer to the sensors should be considered. For long, hard walled pipes in particular, this method makes a simple and powerful alternative to in-pipe pressure transducers for acoustical measurements.

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reducing the noise of pressure pulp screen

Rémy Oddo, Jean Nicolas, R. Panneton
GAUS, Université de Sherbrooke,
Sherbrooke, Qué, J1K 2R1

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1- INTRODUCTION

A pressure pulp screen is a device able of classifying and separate fibers in paper pulp. The machine is made of two concentric cylinders closed at both extremities. The inner cylinder, called "basket" is perforated. The size and the form of holes is determined by the considerations on the quality of paper and the time where classification occurs during the process.
The pulp is injected under pressure in the centre part of the machine. The accepted pulp goes through the holes of the basket and is collected by a pipe connected at the part between the two cylinders. Refused pulp is collected by an other pipe in the centre part of the machine.
In order to clean the holes of the basket, the depression made by blades moving at less than 5 mm of the inner side of the basket is used. These blades are supported by a rotor running at 1150 rpm (fig. 1).
The noise of this machine is characterised by a discrete spectrum in middle and high frequencies (fig. 3). Levels depends of the consistency of pulp (1 to 4 % of fibers), the type of holes (vertical slots, profiled circular holes) and speed of rotation of the rotor.
Classical sound power levels in industry are 95 dB(A) for a basket with circular holes for a diameter of 90".

2- SOURCE CHARACTERISATION

The basket is excited by the rotor. Mathematical representation of rotating forces shows a temporal dependence of the form of $e^{-jN\Omega t}$ Ω, the speed of rotation. It corresponds to a discrete spectrum with lines $\beta\Omega/2\pi$ Hz away from each other, β corresponds to the number of blades.
A theoretical study made by R. Panneton [1] has shown that the response of a cylinder excited by this type of force shows a discrete spectrum with lines on the same frequencies as the excitation. The amplitude of these lines is much important when a coincidence between the frequencies of excitation and the modes of the cylinder occurs.
For our problem, the typical speed of rotation of the rotors and the typical modes of baskets implies that the response must be important only at the first ten harmonics of the frequency of rotation. The problem is that we have the greatest part of the acoustic energy around the 150-200 harmonics of the speed of rotation. In order to understand this phenomenon, R. Panneton studied the effect of an amplitude modulation of the force around the cylinder. He found that the response of the cylinder shows a new set of lines centred around the ξth harmonic of the speed of rotation, with ξ, the number of modulations around the cylinder. Figure 2 shows a numerical result with 75 modulations.
A typical basket for a small screen pressure presents 150 to 200 holes along its circumference. Analysis of the acoustic spectrum emitted by the machine shows that holes are effectively responsible of the modulation of the rotating force exciting the cylinder. This assertion is confirmed by the great dependence between the size and the form of holes and the acoustics levels emitted by the machine.

3- TRANSMISSION PATH CHARACTERISATION

The basket is vibrating under the excitation made by the rotor. These vibrations can be transmitted to the outer cylinder, which is the radiated part of the machine, along two paths:
- Liquid transmission through the pulp.
- Solid transmission through the clamped parts of the basket.

Some measurements are made in order to find the principal path of transmission. The general principle of these measurements is always the same: cutting one path and looking at the result on the overall noise emitted by the machine. Acoustic power is measured using an intensity probe. In order to get comparative results, all measurements are made in water (consistency of pulp = 0%). Tests were made on an ANDRITZ SPROUT BAUER PSV 2100 pressure screen.
At first, we tried to cut the liquid path. We used a "cushion" made of mineral wood enclosed in a plastic sheet hermetically closed. Note that in this case, the attenuation is not due to the absorption in the wool, but to the impedance discontinuity between the pulp and the air inside the cushion. Results of this first trial showed a difference of only 1 dB(A) with and without cushion.
For the second trial, we decided to cut the major part of the solid path. We have made a mechanical isolation between the basket and the rest of the machine by inserting a set of "o" rings. The high frequency problem allows us to use a quite hard suspension system. Results of this trial are shown in fig. 3.
For frequencies of interest, this system has reduced the global noise by nearly 10 dB. This result confirms that the major part of the vibro-acoustic energy is transmitted through solid path.

4- WAYS OF REDUCING NOISE

Like for all types of machines, there is three ways for reducing noise:
- Action at the source,
- dissipating energy during transmission to the radiator,
- action on the radiator.

A: Action on the source:
We know that the excitation is the depression force due to the blades of the rotor. this depression can be seen from the basket like a rotating force acting on the structure. High levels at high frequency are due to the presence of holes causing the modulation. Some production imperatives do not allow us to reduce the speed of rotation (and thus the force acting on the basket). Size of holes is also fixed by the kind of paper produced.
Only two actions are possible here:
- increasing the mechanical impedance of the basket,
- decreasing the phenomenon of modulation by local perturbations around the holes.

Some new types of baskets, with different types of profile which are made to increase productivity present modification on this type. For example, baskets with "Liehmann" profile presents vertical ribs on their inner face. These ribs act like stiffeners and provoke a perturbation in the pulp flow. The acoustical result is that there is a reduction of nearly 15 dB at high frequency. This type of baskets must be used when it is possible.

B: Action on transfers path:
Solid path is the principal one. The solution is a mechanical isolation between the basket and the rest of the machine. For an industrial application, it is very important to fix strongly the basket to the rest of the machine. The suspension must not involves large amplitudes of vibration because clearance between the basket and the blades is very low. The suggested solution is a set of "o" rings and a special device of fixation with mechanically isolated screws maintaining the basket in position. With this kind of device, noise can be reduce by 10 dB(A).
One of the advantage of this solution is that just a few modifications must be done on the machine but it will increase the number of manipulations needed to change the basket.

c: Action on the radiator:
It is possible to reduce the noise emitted by the outer cylinder by using a double shell with acoustic absorption. Note that this solution present the advantage of a thermal isolation of the machine, which is very important for paper makers.

5- CONCLUSION

A combination of solutions presented below can allow us to design a low noise pulp pressure screen (less than 85 dB(A) with pulp). Different actions are possible in respect with production imperatives:
- profiled basket when it is possible.
- mechanical isolation of the basket.
- acoustic barrier on the outer cylinder (with thermal isolation).

6- ACKNOWLEDGEMENTS

The authors would like to thank IRSST for financial support and ANDRITZ SPROUT BAUER for technical and logistical help, Christian Clavet for its technical support.

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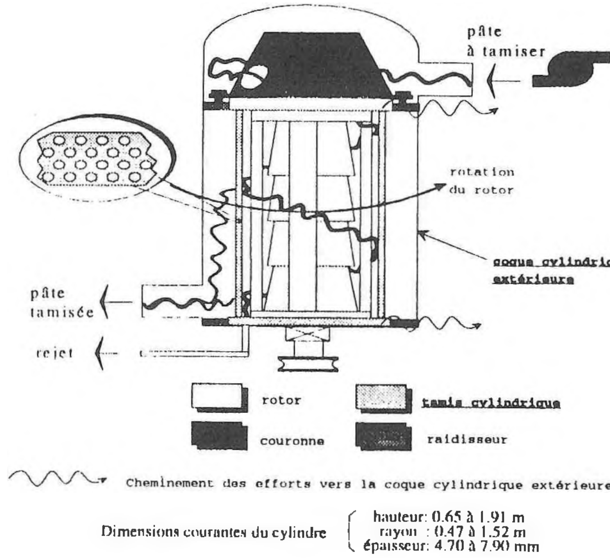

fig. 1 description of a pressure pulp screen

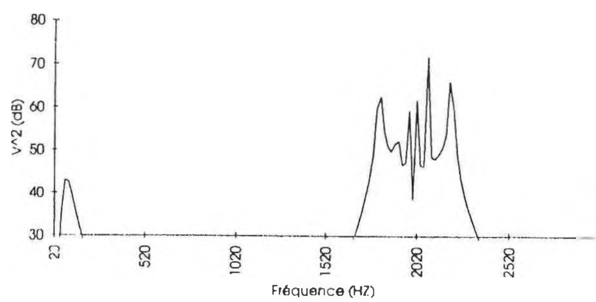

fig. 2 Quadratic velocity of a cylindrical shell
(numerical result with 150 modulations)

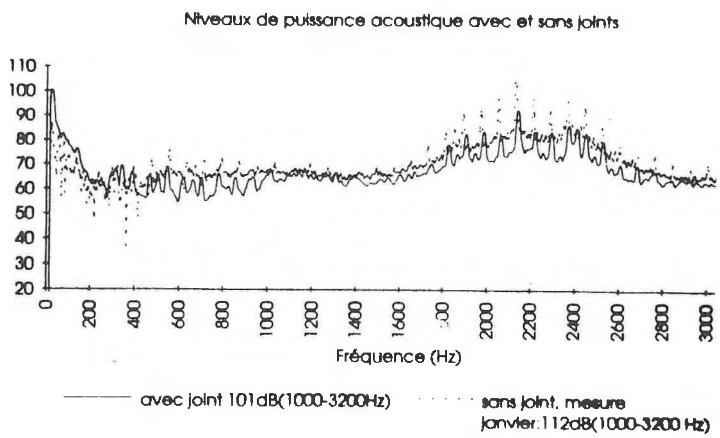

fig. 3: Effects on the global noise of a machinical isolation between the basket and the rest of the machine

Design of circular saw blades for quiet operation

Jean-Luc Wojtowicki, Olivier Beslin and Jean Nicolas
GAUS, Univresité de Sherbroke
Sherbrooke, J1K 2R1 (Québec)

July 1994

1- INTRODUCTION

This study presents the results of the last developments made on the reduction of the noise radiated by circular saw blades at the Acoustic and vibration group of Sherbrooke laboratory. The reduction of noise has been made on both the free running or idling noise and the cutting noise.

Our experiments showed that the free running noise is generated by the motion of teeth and gullets at high speed in the air which creates turbulences. Those turbulences can be significantly decreased as the dimensions of gullets are well chosen. The cutting noise is generated by saw blade vibrations excited by the cutting force. As the cutting force can not be easily reduced, the cutting noise has been reduced using modal analysis technics results and materials with high damping properties.

2- REDUCTION OF FREE RUNNING NOISE

The origin of the free running noise has been widely studied by several authors ([1],[2]). A tooth with an high speed motion creates turbulences phenomena in the air, the propagation of those aerodynamic disturbances which generates noise is closely dependant of gullets dimensions.

According to this hypothesis, an experimental study on free running noise versus gullets dimensions has been made using a rigid circular disk with four slots spread around the disk rim . The gullets characteristics are depth (d), width (w) and thickness (t). As the thickness (t) is equal to the saw thickness (see figure 1), a parametric study has been made varying the two other dimensions width(w) and depth (d).

According to [1], a critical ration $w/t \approx 1$ has been determinate. Under this critical value, vortex are kept inside the gullet and do not propagate, for higher values they become unsteady propagating and then generate noise.

From all those experiments and also several observations on existing saw blades, optimum dimensions for gullets have been extracted. For a 355 mm (14 inches) diameter saw blade and 3 mm (1/8 inch) thickness running at 3600 RPM, gullets should not excess the following dimensions: depth (d) = 9 mm and width (w) = 6 mm.

3- REDUCTION OF SAW BLADE VIBRATIONS

A classical approach of the problem have been applied to reduce saw blade vibrations. One of the most efficient solution to damp vibration is to use a "sandwich" structure with constrained damping layer. This solution have already been attempted by saw blade manufacturers, our experiments showed that those attempts were only efficient in soft wood, this solution had to be improved in hard-wood.

3-1 LOCATION OF THE TREATMENT

The modal behaviour of a circular saw blade has been studied for years by many authors ([3],[4]). The vibration of circular saws can be described as the sum of the motion of individual modes. The modal shapes are simply described by the loci of nodal circles and nodal diameters (see figure 2). As the saw blade is a disk clamped at its centre the maximum vibration levels are located at the free side of the structure *i.e.* at the edge of the saw blade beneath the tooth and gullets.

Our modal analysis experiments made on several saw disk showed that the modes with the highest vibrationnal levels have zero or only one nodal circle in the frequency range of interest. Moreover, those modes have from zero to a large number of nodal diameters. Hence, the vibrations to be damped are located near the edge of the saw as said above and along circles. A vibration damping treatment have to be located as close as possible to the teeth on a circular annulus (see figure 5).

3-2 CHOICE OF DAMPING TREATMENT

A software called ADNR [5] has been developed at the GAUS laboratory to calculate the sound emitted by rectangular plates with one or many layers including the effect of viscous damping. A circular saw blade is not a rectangular plate, however the software have been used to determinate the finest values of constraining layer *vs.* structure thicknesses; the damping material thickness and loss factor are given by manufacturers specifications.

Figure 3 presents the efficiency of a 310 x 310 mm composite structure *vs.* an homogenous rigid plate on the average quadratic velocity for the same dimensions and same thickness (3 mm). The reduction of peak levels is from 5 dB to more than 20 dB on all frequency range.

The great efficiency of the multi-layer plate described above can be applied to the vibration reduction of circular saw blade.

3-3 REDUCTION OF CUTTING FORCE

Based on experimental observations and intuitive considerations, the cutting force is obviously closely dependant of tooth sharpness. Experimental tests (measurement of motor electric consumption) on different type of tooth shape showed that teeth with high top bevel angle provides less cutting force than classical teeth (see figure 4).

4- APPLICATION ON A PROTOTYPE CIRCULAR SAW BLADE

An actual prototype (figure 5) has been built based on the specifications described above. The damping material and the stainless steel layer have been inserted in an annular groove in the saw body.

5- NOISE REDUCTION OF THE PROTOTYPE SAW BLADE

Lots of different circular saw blades available on the market have been tested on our fully controlled radial saw bench. Noise levels were measured in the same cutting conditions for every saw blades (feed speed = 9.96 m/min, rotation speed = 3600 RPM) at one meter from the cutting point and in front of the saw disk. These levels vary in cuting process between 90 and 98 dB(A).

The prototype circular saw has the lowest noise levels while cutting any material and free running. The average cutting noisereduction is 5 dB and a quite constant cutting noise level (under 87 dB(A)) in all type of wood.

6- CONCLUSION

The design of an actual and efficient quiet saw blade has been performed taking in account experimental and theorical studies and also using design tools (ADNR) to optimise classical but often mismatched damping solutions.

The prototype saw blade has the lowest noise levels both during cutting operations (≈ 87 dB(A)) and free running (82 dB(A)) among all circular saw blades tested at the GAUS laboratory. Those noise levels make it safely usable in workshops in an acoustical point of view. Moreover, an indirect effect of reducing vibration is the decrease of chip removing, an improvement of finishing quality and longer lasting cutting performance between resharpments.

7- ACKNOWLEDGMENTS

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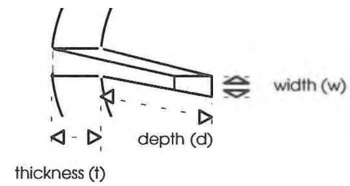

Figure 1 - Dimensions of the gullet

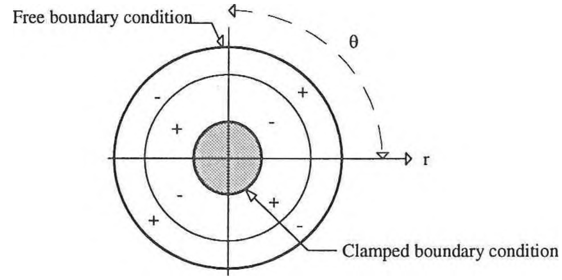

Figure 2 - Modal pattern of circular disk with 2 nodal diameters and 1 nodal circle

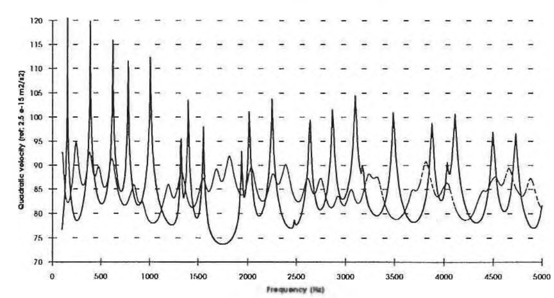

Figure 3 - Quadratic velocity of multi-layer structure (- - -) *vs.* homogenous plate (___)

Figure 4 - classical shape of alternate teeth *vs.* high top bevel angle alternate teeth

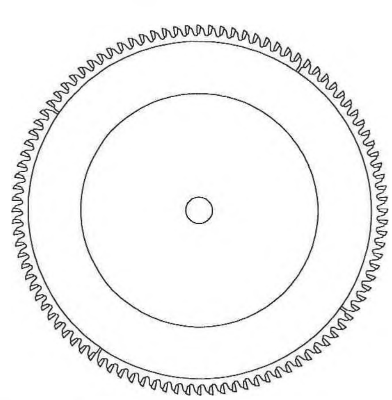

Figure 5 - general aspect of the prototype circular saw blade

Response of a non-baffled sandwich panel submitted to a reverberant chamber acoustic environment

H. Nelisse, O. Beslin, J. Nicolas
GAUS, Université de Sherbrooke,
Sherbrooke, Qué, J1K 2R1

July 1994

1- INTRODUCTION

It is well known, in the aerospace industry, that the acoustic pressure levels generated by the rocket launchers are extremely high. This strong acoustic pressure specificaly excites the payload in which a satelitte is located. Then, the satellites manufacturers have to submit their structural components to standardized tests in a reverberant chamber. We are then left with a situation where a panel is hung in a large reverberant room and immerged in a so-called diffuse field.. The excitation spectrum and levels are specified and measurements of acceleration levels induced in the structure are done. It leads to a vibroacoustical problem that presents many interesting challenges: (i) the panel is non-baffled (ii) the panel is excited on both sides by the acoustic field (iii) the coupling between the panel and the fluid has to be treated rigourously in order to obtain the pressure jump function across the panel (iv) the proposed formulation must take care of an extremely high modal density in the cavity, even at low frequencies.

In this paper, we are first presenting a new semi-analytical formulation to predict the behaviour of a non-baffled flexible plate in a rigid-walled rectangular cavity excited by an acoustical source. The second part of the paper presents some numerical results and discussions about the contribution of physical effects.

2- SEMI-ANALYTICAL FORMULATION

Consider a rigid-walled rectangular room (fig.1) of dimensions L_x, L_y, L_z, which contains a plate of surface $S_p = L_x \times L_y$ and enclosing a fluid of volume V. It is well known, using integral equation method, that the sound pressure P at any point point in V can be written

$$P(\vec{r}_0) = \iint_{S_S} dS\, G(\vec{r},\vec{r})\vec{\nabla}_{\vec{r}} P(\vec{r}) - \iint_{S_p} dS\, \overline{P}(\vec{r})\vec{\nabla}_{\vec{r}} G(\vec{r},\vec{r}_0)\cdot\vec{n}_p \tag{1}$$

where S_S denotes the surface of the of the piston and S_p the surface of the panel. The quantity $\overline{P}$ represents the sound pressure jump across the surface S_p while $G(\vec{r},\vec{r}_0)$ is the Green's function for the empty room. The first term of (1) represents the contribution of the source while the second term is the contribution of the plate to the sound pressure. By coupling the equation for the motion of the plate and the boundary condition between the fluid and the plate at the interface S_p with equation (1), the problem is completely and rigourously defined.

A: Green's function

Instead of expanding the Green's function over modes of cavity as it is usualy done, the function is expanded over a set of two-dimensionnal functions. It reduces the number of summations by one, a great feature for computer calculations. Such an expasion is written in the following form

$$G(\vec{r},\vec{r}_0) = \sum_{tu} g_{tu}(z,z_0)\, \psi_{tu}(x,y)\psi_{tu}(x_0,y_0) \tag{2}$$

where $g_{tu}(z,z_0)$ is a discontinous function at $z=z_0$. The basis functions ψ are just cosines along x and y and satisfy Neumann boundary conditions on the walls of the cavity.

B: Sound pressure jump

To compute the contribution of the plate to the pressure, we first expand the sound pressure jump function over the same basis functions used for the Green's function. One then write

$$\overline{P}(x,y) = \sum_{rs} \overline{P}_{rs}\, \psi_{rs}(x,y) \tag{3}$$

where $\overline{P}_{rs}$ are unknown coefficients.

C: Fluid-structure coupling

The fluid-structure coupling is being taken into account in two parts. We first introduce the classical boundary condition at the interface S_p which states that the velocity of the fluid must be equal to the velocity of the plate on S_p in order to write

$$-\frac{1}{j\omega\rho}\frac{\partial P}{\partial z}\bigg|_{z=z_p} = -j\omega\, w(x,y) \quad \text{on } S_p\,, \tag{4}$$

where w(x,y) is the deflection of the plate.

Since we have an acoustic excitation, we have a to consider the strong coupling limit in order to include *completly* the motion of the plate in the fluid. This motion is related to the sound pressure function, which is in fact the excitation force on the plate, by the following relation

$$\overline{P}(x,y) = Z(w(x,y)) \quad \forall\ (x,y) \in S_p \tag{5}$$

where Z is an operator that represents the mechanical behaviour of the plate. If we expand the deflection of the plate over *in-vacuo* modes of the panel, introducing a set of unknowns b_{mn},

$$w(x,y) = \sum_{mn} b_{mn}\ \varphi_{mn}(x,y)\ , \tag{6}$$

we can compute the operator Z and obtain

$$Z_{mnpq} = -\omega^2 M_{mnpq} + K_{mnpq}\ . \tag{7}$$

The matrices **M** and **K** are respectively the mass and the stiffness matrices of the panel.

D: Linear system of equations
Inserting equations (1),(2),(3) and (6) in equations (4) and (5) and integrating over appropriate domains, we obtain a linear system of equations solvable with standard algorithm

$$\begin{bmatrix} \mathbf{SYS^tZ - \omega^2 I} \end{bmatrix}\begin{bmatrix} \mathbf{b} \end{bmatrix} = \begin{bmatrix} \mathbf{-SW} \end{bmatrix} \qquad (8)$$

where **S** is a change of basis matrix, **Y** a vector that contains the acoustic information of the problem and where **W** is the source vector.

2- NUMERICAL RESULTS

To ensure convergence of our method,we have established a simple geometrical criterion : *the smallest wavelength of the acoustic basis functions* , λ^{ac}_{min}, *must be less or equal to the smallest wavelength of the in-vacuo modes,* λ^{struc}_{min} . Figure 2 shows the case of $\lambda^{ac}_{min} < \lambda^{struc}_{min}$ (sub-critical) and $\lambda^{ac}_{min} \geq \lambda^{struc}_{min}$ (critical and super-critical).
Results for the sound pressure jump and velocity of the plate are shown in fig.3 for a 0.7×0.5 m^2 panel in a $2.6\times2.0\times3.0$ m^3 cavity. It shows a stronger response of the plate at empty room modes frequencies comparatively to the response at *in-vacuo* eigenfrequencies.
The influence of the mechanical properties of the plate on the sound pressure jump is shown on fig 4. One can see the sound pressure jump remains unchanged for some values of the mass per unit area and flexural rigidity. It suggests the fact that the panel acts primarily as a if it is rigid so that the governing phenomena is the diffraction.

3- CONCLUSION

We have developed a novel semi-analytical approach that is able to to take care of many interesting features (i) acoustic excitation (ii) complete fluid-structure coupling (iii) non-baffled panel (iv) light and fast computer code. Preliminary results show good agreements with expected results and show the possibility to simulate aerospace applications such as the dynamic response of a composite panel with attached electronic equipments.

4- ACKNOWLEDGMENTS

This work was supported by grants from the National Science and Engineering Research Council of Canada (NSERC).

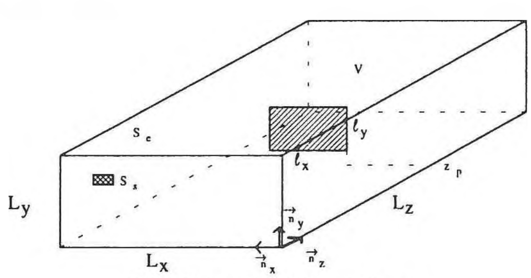

fig1 :Geometry of the problem

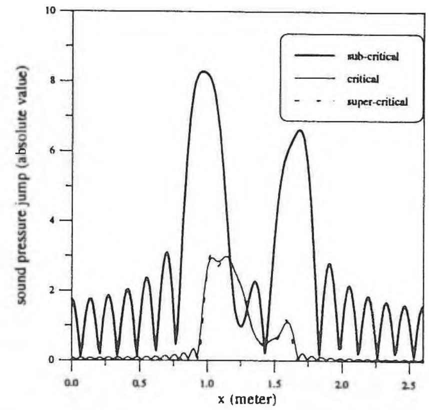

fig.2 : Sound pressure jump as a function of x (y being fixed) for three cases of λ^{ac}_{min}.

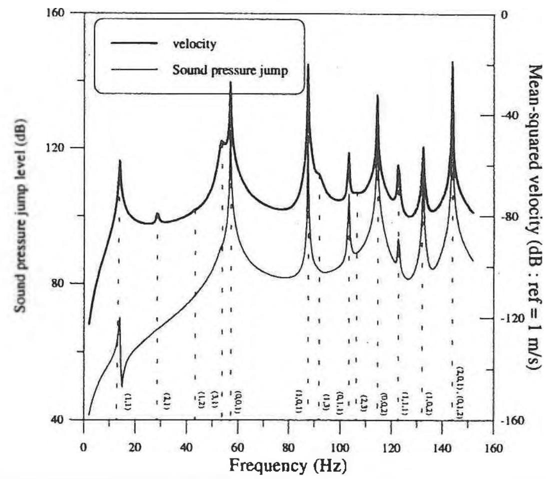

fig.3: Mean-squared sound pressure and velocity levels as a function of the frequency for a 0.7×0.5 m^2 panel. (ijk) denotes empty cavity modes and (mn) denotes *in-vacuo* modes

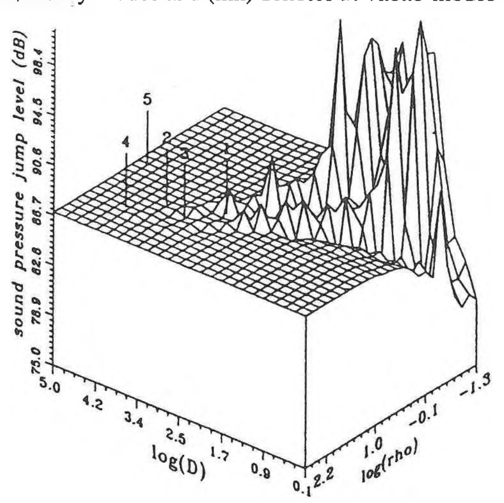

fig 4: Sound pressure jump level at 90 Hz as a function of log(ρ_S) and log(D) where ρ_S is the mass per unit area and D is the flexural rigidity. #1-#2 : 5 mm. and 1 cm. of aluminium, #3-#4 : 5 mm. and 1 cm. of steel, #5 : typical sandwich panel.

AN ANALYTICAL APPROACH FOR THE FREQUENCY RESPONSE OF A MULTILAYER DISC

Benoît Cournoyer, Noureddine Atalla, Jean Nicolas
Groupe d'acoustique de l'Université de Sherbrooke
Université de Sherbrooke, Faculté des sciences appliquées, Département de génie mécanique
Sherbrooke(Québec), J1K 2R1, Canada

Introduction :

The vibro-acoustic design of multilayer structures is becoming more and more important but also quite difficult to model due to the complexity of such structures. The first step to get over it is to find a simple yet accurate formulation to treat the case of a constrained viscoelastic layer.

Multilayer discs are commonly used in many mechanical systems : grinding wheels, gears, circular saw blades, ... This paper presents a model of such an arrangement and the determination of both its natural frequencies and mean quadratic velocity. For this purpose, a multilayer disc excited by a harmonic point force is used. The determination of the free and forced frequency responses is achieved through a variational approach combined with a Rayleigh-Ritz type approximation.

Statement of the problem :

A multilayer disc, clamped at its inner radius and free at its outer radius, is considered (figure 1). Each layer is described by its thickness, specific mass, Poisson's ratio and complex Young's modulus. The classical assumptions of Reissner[1] and Mindlin[2] for thin plates are used :

i) negligible axial traction-compression : $\sigma_{zz} = 0$;
ii) negligible torsion around the axis : $\tau_{r\theta} = 0$;
iii) pure bending;
iv) linear shear deformation;
v) each layer is made of a homogeneous and isotropic material.

Hence, the allowed motion of the disc comprises five *elementary displacements* :

i) a membrane effect described by two longitudinal displacements : u_r, u_θ;
ii) a pure bending described by a transversal displacement : w;
iii) a linear shear described by two agles : ϕ_r, ϕ_θ.

Therefore, the displacement field can be written :

$$(a)..\begin{cases} U_r^{cn}(L,t) = u_r^{cn}(L,t) + (R_n - z)\cdot\left[\partial w/\partial r(L,t) + \phi_r^{cn}(L,t)\right] \\ U_\theta^{cn}(L,t) = u_\theta^{cn}(L,t) + (R_n - z)\cdot\left[\partial w/\partial \theta(L,t) + \phi_\theta^{cn}(L,t)\right] \\ U_z^{cn}(L,t) = w(L,t) \end{cases}$$

where : - cn stands for layer n;
- R_n is the distance between the top surface and the neutral fiber of the n^{th} layer;
- L is defined as the triplet (r,θ,R_n).

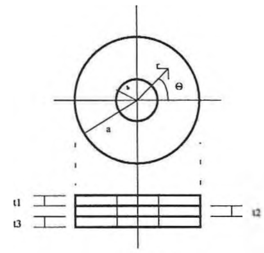

Figure 1

Methodology :

To calculate both free and forced vibration of the multilayer disc, the variational approach is used. The Hamilton's functional of the system is given by :

$$(b)..H = \iiint_{Vol} [T - V + W]\cdot dVol$$

where : - T is the sum of the kinetic energy of *each* layer;
- V is the sum of the deformation energy of *each* layer;
- W is the work applied to the entire disc;
- Vol is the total volume of the disc.

Using the continuity between the layers, the kinetic and deformation energies of each layer are written in terms of the displacement field of the *first* layer. To do so, the following assumptions are made, Guyader[3] :

i) continuity of the slopes throughout the layers :
- dw/dr and $dw/d\theta$ equal a constant;
ii) physical cohesion of the disc at the *interface* :
- the continuity of the displacement U_r, U_θ, U_z;
- the continuity of the shear stresses τ_{rz}, $\tau_{\theta z}$.

These continuity conditions can be written as a transfer matrix between the elementary displacements of the n^{th} layer (cn) and those of the *first* layer (c1) :

$$(c)..\begin{Bmatrix} u_r^{cn} \\ u_r^{cn} \\ \partial w/\partial r \\ \partial w/\partial \theta \\ \phi_r^{cn} \\ \phi_r^{cn} \end{Bmatrix} = \begin{bmatrix} 1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & K_1 & 0 & 0 & 0 \\ 0 & 0 & 0 & K_2 & 0 & 0 \\ K_3 & 0 & K_4 & 0 & 1 & 0 \\ 0 & K_3 & 0 & K_5 & 0 & 1 \end{bmatrix} \cdot \begin{Bmatrix} u_r^{c1} \\ u_r^{c1} \\ \partial w/\partial r \\ \partial w/\partial \theta \\ \phi_r^{c1} \\ \phi_r^{c1} \end{Bmatrix}$$

where :

$\cdot K_{1,2} = C^{c1}_{55,66}/C^{cn}_{55,66}$

$\cdot K_3 = 1/2\cdot(t_n + 2\cdot t_{n-1} + 2\cdot t_{n-2} + \ldots + 2\cdot t_2 + t_1)$

$\cdot K_{4,5} = C^{c1}_{55,66}/2\cdot\left[(t_n/C^{cn}_{55,66}) + 2\cdot(t_{n-1}/C^{c(n-1)}_{55,66}) + \ldots + 2\cdot(t_2/C^{c2}_{55,66}) + (t_1/C^{c1}_{55,66})\right]$

C^{cn}_{ij} being the coefficient (i,j) of Hook's matrix (stress-strain relations) satisfying the previous assumptions.

To find the disc eigenfrequencies and mean quadratic velocity, the method of assumed mode is used. A polynomial set of admissible trial function with unknown amplitudes is used. Introducing the displacement field [Eq.(a)] in the functional [Eq.(b)] and using the variational principle, one arrives to a system of linear equations for the unknown amplitudes.

Results :

All the results presented in this section have been computed for a 150 mm outer radius and 25 mm inner radius disc submitted to a 1 N force applied at a radius of 140 mm. Furthermore, every calculation has been conducted using 8 nodal diameters and 8 nodal circles.

Figure 2 shows the mean quadratic velocity of two identical discs : one is made of a single steel layer, the other is made by artificially subdividing the disc into three layers. This figure shows the accuracy of the model used.
Figure 3 displays a comparison between a disc made of one elastic layer placed between two free viscoelastic layers and an equivalent one layer disc. The theory behind the determination of the characteristics of the equivalent disc is taken from Mezache[4]. The agreement between the two approaches is excellent.
Finally, figure 4 presents the mean quadratic velocity of two identical steel base discs, the second having a supplementary 0.25 mm thick viscoelastic (3M ISD 112) layer constrained by a 0.5 mm thick steel layer. This figure shows the importance of damping achieved through a constrained viscoelastic treatment.

Conclusions :

- A new formulation for multilayer disc has been developed;
- an application to a three layer disc has been successfull;
- the efficiency of a constrained viscoelastic layer has been demonstrated, even, and surprisingly, in the low frequency range.

Acknoledgement :

This work is financially supported by I.R.S.S.T.

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Influence of the number of layers

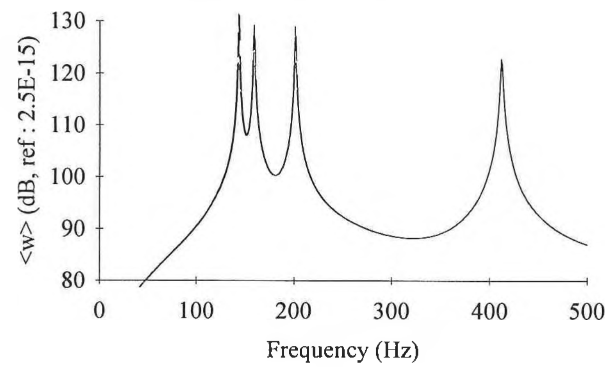

Figure 2

Comparison with an equivalent disc

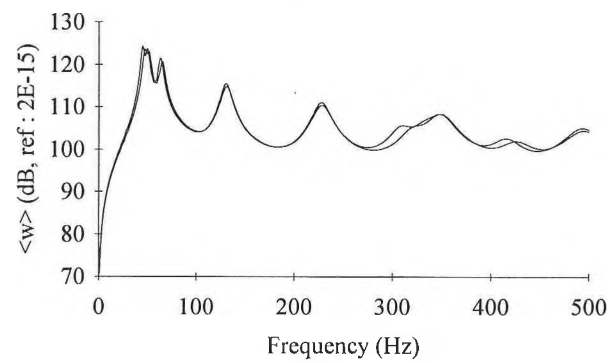

Figure 3

Influence of a constrained visco layer

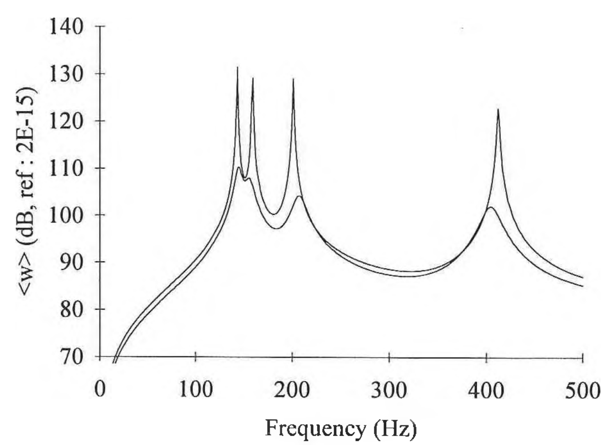

Figure 4

WHEEL SQUEAL IN RAIL TRANSIT SYSTEMS

Darron Chin-Quee, P.Eng., M.B.A.
Valcoustics Canada Ltd., 30 Wertheim Court, Unit 25
Richmond Hill, Ontario L4B 1B9 (905)764-5223

One of the most prevalent noise by-products of rail transit is wheel squeal. Due to high sound pressure levels (SPL) and pure tones, wheel squeal is increasingly of concern. It results in discomfort to passengers and unacceptably noisy environments for neighbouring land uses. Most environmental noise guidelines do not address this noise source. When it is addressed, the need to mitigate is tempered by the role of transit as a public convenience and essential service, and the limited success of mitigation techniques.

CHARACTERISTICS

Wheel squeal is often characterized by sound level peaks 20-25 dBA (or more) higher than the normal (rolling) sound from wheel rail interaction. Measurements of subway and streetcar wheel squeal conducted on Toronto Transit Commission (TTC) vehicles indicate that subway cars generate higher frequencies than that produced by streetcars. Typical plots of SPL (dB) in each ⅓ octave band are shown in Figures 1 and 2 before and during wheel squeal.

The spectral characteristics of wheel squeal vary from vehicle to vehicle and are affected by environmental factors such as temperature and humidity as well as physical plant related factors. Therefore, wheel squeal may be present for a given vehicle on one day and be absent or modified on others. This variability can complicate field study of the phenomenon.

CAUSES AND FACTORS

On straight track, wheel rotation is a pure rolling motion. On a short radius curve sliding motion occurs between the wheel and track surfaces resulting in the wheel vibrating at its natural frequencies which are often in the audible range.

Sliding speed affects the squeal noise produced. Studies of a simple cantilever on disk type sliding apparatus indicate squeal associated with the fundamental vibration mode varies little at higher speeds. At lower speeds, squeal associated with stick-slip phenomenon occurs and frequency decreases with sliding speed. Chatter noise occurs at very low speeds.

Any factor associated with the natural frequency of the wheel vibration and the frictional rail/wheel contact will affect squeal. Thus, wheel stiffness determined by the wheel material, shape, and damping are important. Lubrication, humidity and temperature also affect the stick-slip phenomenon associated with squeal. Contact forces affect rail and wheel vibration and hence wheel squeal.

However, the interaction of all these factors has not been studied extensively. In many cases where wheel squeal has been eliminated, the underlying reasons are not known. For example, recent renovations to a streetcar line prompted by maintenance requirements, drastically reduced the incidence of wheel squeal. It is unclear whether rail bed changes, new rails (profile or material), or a combination of both, were responsible.

MITIGATION TECHNIQUES

Mitigation techniques can be grouped into three categories: On Board, Structural and Wayside Treatments. To date, a universally applicable, consistently effective method, which addresses durability, operational safety and environmental concerns, has not been found. Some of the approaches used are:

On Board Treatment

Ring Dampers: In some cases squealing noise has been eliminated by fitting to the perimeter of the wheel disk, a damping ring consisting of an elastomer and a thin plate of steel.

Composite Wheel: A layer of elastomer is used between the metal wheel centre and the steel rim which carries the steel tire. Again, this technique has been successful on some systems and has failed on others. For some applications, the effectiveness is limited by the stiffness required for the proper operation of the wheel assembly, e.g. to negotiate track switches and curves.

Lubrication Blocks: Lubrication blocks apply solid lubricant to the wheel (throat and back-of-flange). Lubricants are not applied to the rail head for safety reasons. Success depends on matching the appropriate lubricant to temperature. A temperature insensitive lubricant has been difficult to find, resulting in inconsistent performance from season to season.

Structural Treatments

Rail: Rail segments have been replaced with a material that creates a solid lubricant effect. Success has been limited. Longevity is a concern.

Lubrication Systems: Grease, oil, and water lubricants have been tried. Grease and oil have safety and environmental concerns and are temperature sensitive. Water is effective but has limited seasonal use unless applied with anti-freeze which has environmental concerns.

Increased Radius: Some studies have shown wheel squeal is unlikely to occur on curves with a radius greater than 100 times the truck wheel base. For many urban transit applications e.g. streetcar loops, this could not be implemented due to space restrictions. In practice, loop radii of close to 200 m would be needed.

Turntables: Turntables have been proposed as an alternative to short radius loops. Problems with this method include potentially higher maintenance costs, restricted access at terminals, and level grade requirements.

Wayside Treatments

Wayside treatments are primarily limited to noise barriers. Effectiveness is limited by gaps for vehicle access in many cases.

ENVIRONMENTAL NOISE IMPACT

The potential environmental noise impact of wheel squeal can be significant. Typical A-weighted time histories of two streetcar passbys are shown in Figures 3a and 3b. The plots shown are for passbys made at the same location for two different vehicles (but same model), one exhibiting wheel squeal, the other with wheel squeal absent. Based on typical transit headways, the estimated daytime (0700-2300 hrs), nighttime (2300-0700 hrs) and 24 hour transit attributable L_{eq} for these passbys, with and without wheel squeal at this location, are:

	L_{eqDay}	$L_{eqNight}$	L_{eq24}
Wheel Squeal Present:	65 dBA	61 dBA	64 dBA
Wheel Squeal Absent:	58 dBA	54 dBA	57 dBA

The wheel squeal contribution to environmental noise is significant and in this case results in exceedences of most generally accepted environmental noise guidelines.

CURRENT RESEARCH

The above discussion highlights the need for further research into causes and mitigation of wheel squeal. The US Transportation Research Board has recently commissioned a study on Wheel/Rail Noise Control including wheel squeal. Hopefully, this will provide some further insight to an often overlooked problem. Research is needed within Canada. This will require a collective effort on the part of transit agencies, industry and universities.

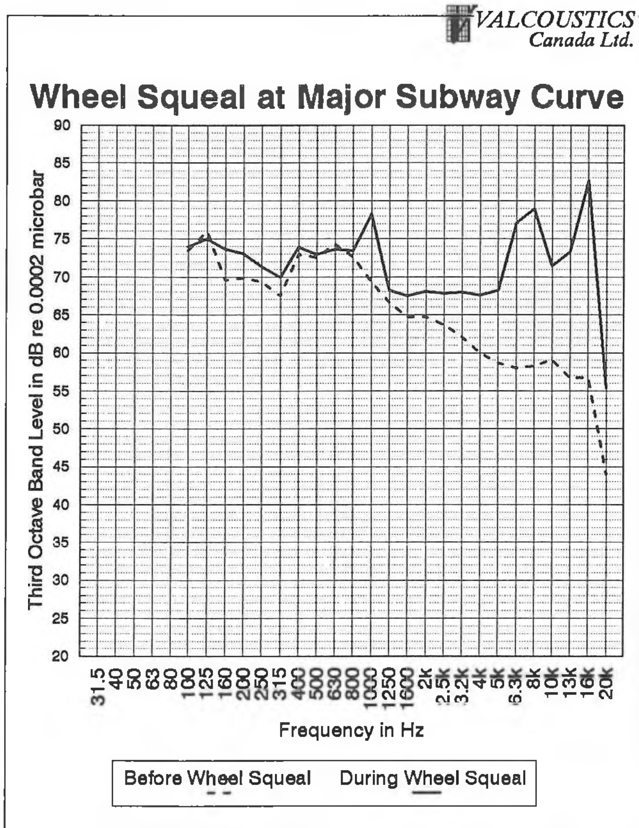

Fig 1: Spectral Comparison of Subway Before & During Wheel Squeal

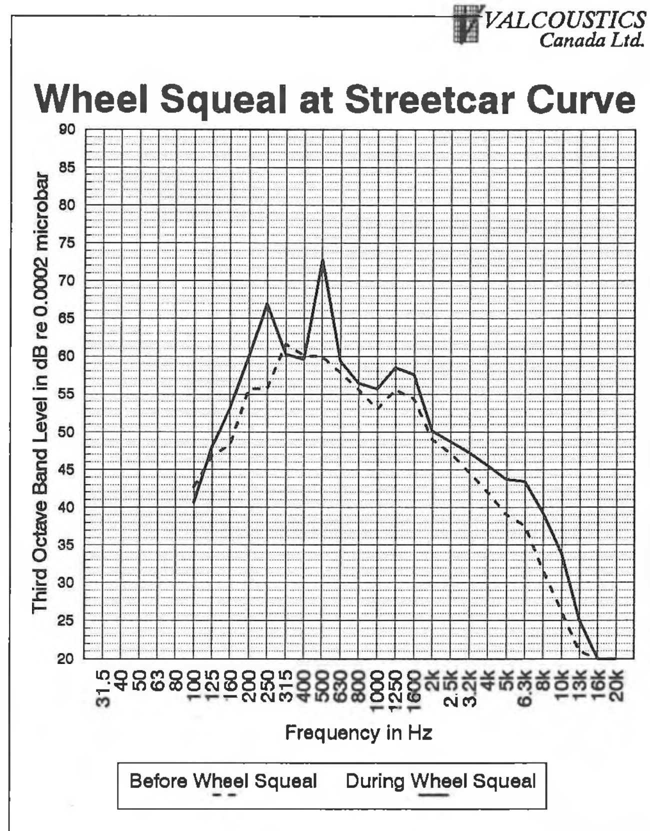

Fig 2: Spectral Comparison of Streetcar Before & During Wheel Squeal

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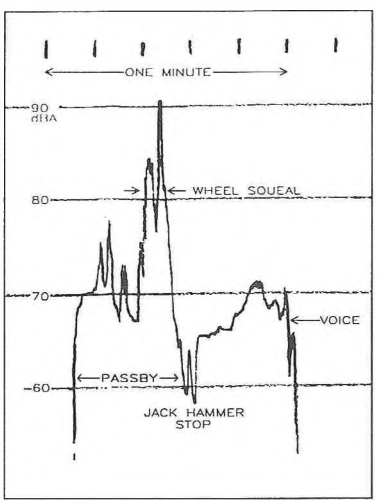

Event Data

SEL: 88 dBA
$L_{max}^{(a)}$: 90 dBA
$L_{MAX}^{(b)}$: 86 dBA

Ambient Data
(12:33 - 13:33)

L_{eq}:	72 dBA	L_{10}:	73 dBA
$L_{99.9}$:	57	L_5:	75
L_{90}:	61	L_1:	82
L_{75}:	65	L_0:	94
L_{50}:	70		

Notes

(a) Largest SPL (dBA) occuring during the homogeneous portion of the event, measured on fast response.

(b) RMS average of the plateau for the event's homogeneous portion (i.e. assumed portion of the event when the vehicle is directly in front of the microphone).

Location: 2nd Floor Deck, Residential Building
Distance to Centre-Line of Rail: 22 m
Other Noise Sources: Jack hammering in background

FIGURE 3a: Typical Streetcar Passby Time History
Wheel Squeal Present

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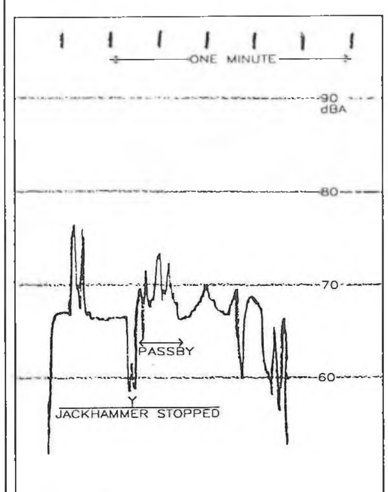

Event Data

SEL: 81 dBA
$L_{max}^{(a)}$: 73 dBA
$L_{MAX}^{(b)}$: 71 dBA

Ambient Data (12:33 - 13:33)

L_{eq}:	72 dBA	L_{10}:	73 dBA
$L_{99.9}$:	57	L_5:	75
L_{90}:	61	L_1:	82
L_{75}:	65	L_0:	94
L_{50}:	70		

Notes

(a) Largest SPL (dBA) occuring during the homogeneous portion of the event, measured on fast response.

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Location: 2nd Floor Deck, Residential Building
Distance to Centre-Line of Rail: 22 m
Other Noise Sources: Jack hammering in background

FIGURE 3b: Typical Streetcar Passby Time History
Wheel Squeal Absent

Broad-Band Active Noise Reduction In Communication Headsets By Digital Feedforward Control

G. J. Pan[1], A. J. Brammer[1], R. A. Goubran[2], J. G. Ryan[1], and J. Zera[1]

1. Institute for Microstructural Sciences, National Research Council, Ottawa, Ontario K1A 0R6
2. Department of Systems and Computer Engineering, Carleton University, Ottawa, Ontario K1S 5B6

1.0 Introduction

Communication headsets are frequently required to operate in environments in which acoustic noise interferes with the intelligibility of speech. The noise reduction obtained using conventional headsets, which commonly consist of an earphone mounted in a circumaural hearing protector, is known to be insufficient for some applications at low frequencies. For this reason, "active" noise reduction (ANR), in which a corrective sound is generated electro-acoustically to reduce the combined sound field at a desired location, has been proposed for reducing the noise at the ears [1, 2].

The purpose of the present work is to explore the performance obtainable with a digital ANR system for such applications, based on the use of a circumaural headset and a floating-point digital signal processor (DSP). This paper reports our initial results.

2.0 Apparatus and Method

A digital ANR system has been constructed by adapting a commercial circumaural analogue ANR headset, the performance of which in its original condition has been reported elsewhere [3]. For the purposes of these experiments, the ear cup and cushion, earphone and attached microphone were retained from the commercial device. A block diagram of the system is shown in Figure 1, where the headset is represented by one ear cup shown in cross-section. The cushion was sealed by spring pressure characteristic of the headband force against the base plate of a cylindrical enclosure (shown in outline), which represents the surface of the head. The enclosure has been developed for measuring the performance of ear cups at frequencies from 10 to 1000 Hz, and sound pressures of up to 140 dB [4]. For the purposes of the present work, a 1/3 octave-band spectrum shaper (Bruel & Kjaer models 1612/S1A & SP) was used to equalize the loudspeaker output.

The sound field surrounding the headset was sensed by a miniature electret microphone attached to the exterior of the ear cup, which generates the "reference" signal, X. The microphone (Panasonic, type WM-063), together with its preamplifier, possess a flat frequency response from 50 Hz to 10 kHz (within 1 dB), and dynamic range of from 30 to 120 dB (re 2×10^{-5} Pa). Control of the sound pressure within the volume between the ear cup and the base plate of the enclosure is maintained by the miniature electret microphone attached to the earphone, which monitors the "error" signal. This microphone is used to sense the residual sound pressure after generation of the control sound field by the earphone, and was also used in these experiments to monitor the performance of the ANR system. The anti-aliasing and reconstruction filters were 8-pole, 6-zero elliptic low-pass filters with cut-off frequencies set to 5 kHz (Stanford Research Systems model SR640). A/D and D/A conversion employed 16 bit successive approximation converters integrated with the DSP board (Spectrum PC/C31). The complete DSP system is hosted in an IBM 486 PC.

The control system employs an adaptive feedforward structure, and uses a normalized filtered-X LMS algorithm [1]. Figure 2 shows a block diagram of the control algorithm, in which X, W, U, H, and E represent the reference input signal, the impulse response of the controller, the controller output, the impulse response of the error path (from U to E in Figure 1), and the error signal, respectively. W and H were implemented as 100 tap finite impulse response (FIR) digital filters, and H was identified off-line in the present work. The step size used to adapt the controller weights was normalized with an estimate of the input signal power, P. Updating of the controller weights is then effected by:

$$W_k = W_{k-1} + \bar{\mu}\, E(k)\, R_k$$

where $\bar{\mu}$ is the normalized step size ($=\mu/P$), E(k) is the error signal at time k, and R_k is the filtered reference signal vector at time k

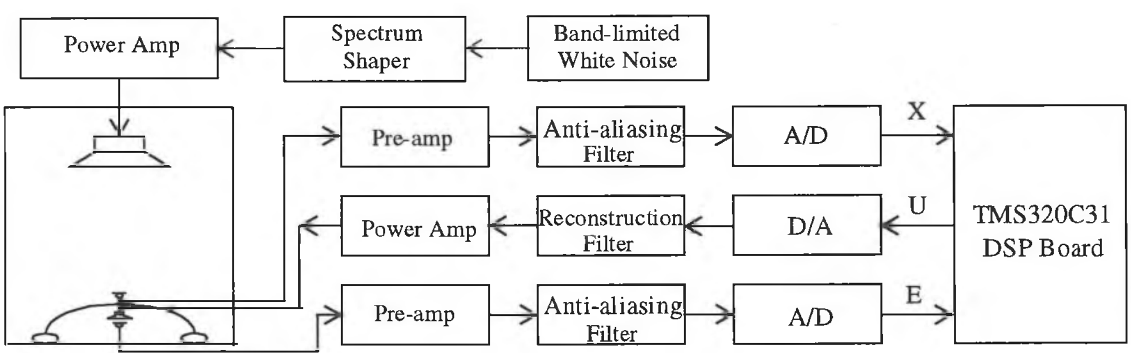

Fig. 1. An active noise control system for communication headsets.

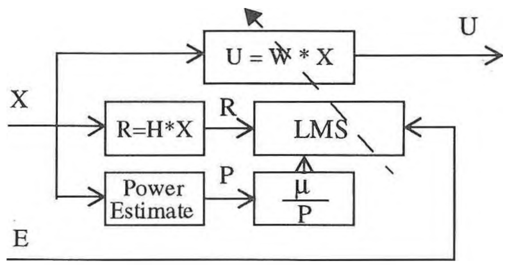

Fig. 2 Adaptive control algorithm

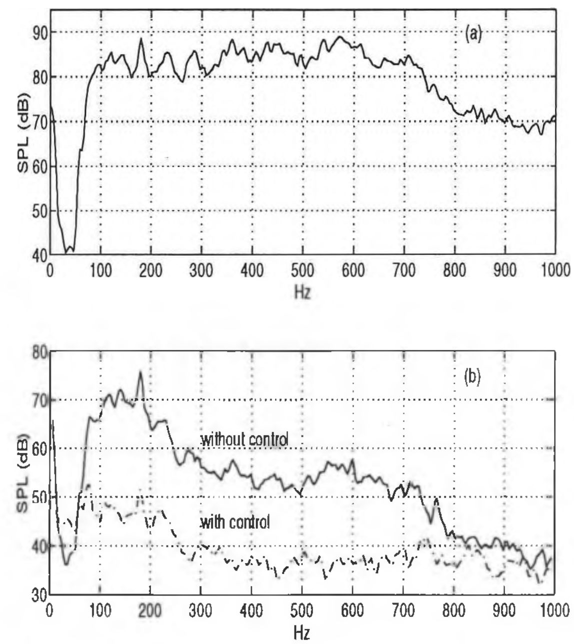

Fig. 3. (a) Spectrum of noise above headset; and (b) Active noise reduction performance.

obtained from the convolution equation R = H * X. The control signal was then generated by filtering the reference signal, using the on-line adapted controller weights.

A Texas Instruments TMS320C31 DSP was used to implement the control system in real-time. The sampling frequency was chosen to be 10 kHz for the present work, and at this rate the DSP board functioned in real time without overload. The performance of the control system was recorded by a FFT analyzer (Stanford Research Systems model SR770), using the Blackman-Harris windowing function.

3.0 Results and Discussion

The performance of the ANR system was determined with the band-limited noise shown in Figure 3(a). This noise spectrum was recorded outside the ear cup by the reference microphone, and is considered "flat" within the limitations of the apparatus, from 80 to 750 Hz. A sound pressure was chosen outside the ear cup at which all electronic and electro-acoustic components of the system were operating linearly.

With this environmental noise spectrum outside the ear cup, the corresponding sound pressure in the volume enclosed by the ear cup is shown by the spectrum labelled "without control" in Figure 3(b). This spectrum was recorded by the error microphone, and reveals, by comparison with Figure 3(a), that the passive attenuation of the ear cup is approximately 10 dB at frequencies between 100 and 200 Hz, and increases to close to 30 dB at 800 Hz. This attenuation is in agreement with results previously reported for this ear cup [3], and also indicates the extent to which the secondary source within the ear cup will influence the sound field at the reference microphone.

The performance of the adaptive control system can be seen in Figure 3(b). After an initial period of adaptation, the stabilized sound pressure in the volume enclosed by the ear cup is shown by the dashdot line labelled "with control". It is evident from this spectrum that the control system produces significant noise reduction at all frequencies of the environmental noise, and has somewhat reduced the predominance of the components at frequencies from 100 to 200 Hz. Since both spectra in Figure 3(b) were recorded at the error microphone, the difference between them gives the attenuation introduced by the control system. This difference is in excess of 20 dB at frequencies from 80 to 200 Hz, and in excess of 15 dB at frequencies from 250 to 650 Hz.

The signal labelled "without control" also displays the initial error spectrum that the control system experiences. This spectrum may be considered to be constructed from two band-limited noise signals of different amplitudes, with frequency ranges of from 80 to 200 Hz, and from 250 to 700 Hz. When viewed in this way, the results suggest that the current algorithm produces more attenuation when controlling the more intensive narrow-band noise. This inference suggests the use of parallel, frequency band-limited controllers to improve the performance of a digital ANR system for this application.

Acknowledgments

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THE ACOUSTICAL CHALLENGE OF QUARRY DESIGN

J. Emeljanow, B.Eng., P.Eng., A.D. Lightstone, Ph.D., P.Eng.
Valcoustics Canada Ltd., 30 Wertheim Court, Unit 25
Richmond Hill, Ontario L4B 1B9 (905)764-5223

INTRODUCTION

Quarries must be located where there are adequate supplies of the mineral resource (rock). Often they are in very quiet areas where there are no significant noise sources in the environment. However, there are often a scattering of permanent or seasonal residences around the proposed site where the applicable noise guidelines have to be met. The noise mitigation measures not only have to be effective acoustically, they must also meld with the operational design to make extraction economically viable.

NOISE GUIDELINES

In Ontario, the Ministry of Environment and Energy (MOEE) noise guidelines require that the hourly sound exposures (L_{eq} in dBA) produced by the quarry operations not exceed the existing ambient sound environment at neighbouring noise sensitive receptors (although mitigation to below 40 dBA is not required).

We have often found that the ambient sound environment is less than 40 dBA in the vicinity of proposed quarries. The very stringent sound exposure limit of 40 dBA that is triggered can be difficult to achieve at nearby receptors.

BASIC QUARRY OPERATION

A quarry operation is a heavy industrial operation using very large pieces of machinery, many of which have high noise emission levels. The operations typically involve:

- Drilling and blasting of rock;
- Transport of rock from the working face;
- Processing of the rock; and
- Shipping of the product off-site.

Prior to start-up of the quarry operations, some construction activities are needed, such as building of the scale house, storage building, processing plant, noise mitigation measures, internal haul roads, etc. These activities are not part of the chronic daily operations are often need not be considered if their duration is relatively small in relation to the life of the site.

NOISE MITIGATION

Many of the noise sources associated with the activities outlined above can be dealt with fairly easily. Most of the noise sources are located on the quarry floor, well below surrounding grade, taking advantage of the inherent screening provided by the working face. To maximize this screening, the operations can be designed so that the working face progresses towards the receptors and the equipment is located as close to the working face as possible, inherently screened by the embankment. Rock can be transported from the working face to the processing area by electrically powered conveyors, which are very quiet. Conventionally, large off-road trucks are used to carry blasted material from the working face to the primary crusher. Large, noisy pieces of processing equipment (secondary crushers and screens) can be located inside of buildings specially designed to provide appropriate noise attenuation.

Truly mobile crushers are being developed which travel with the working face. They can be loaded directly by an excavator. This method of mining can be very effective acoustically since this equipment can be located very close to the working face, maximizing the inherent screening.

However, the rock drilling operation is the one noise source that is very difficult to mitigate and is often the determining factor in terms of the noise mitigation requirements. The rock drill is always located on top of the working face. If there is very little overburden, the drill is exposed to the surrounding area, at least for the first lift. Large quarries often have to carry out rock drilling continuously even if the actual blasting occurs only once a week. Combine all of this with the fact that the rock drill is often the single most significant noise source and the challenge becomes obvious.

The most commonly used form of noise mitigation is the perimeter berm. The acoustical effectiveness of any barrier is greatest when either the source or the receiver is close to the barrier. For large sites, very high barriers are required since the rock drill would generally be far away from the perimeter berm as would the off-site receptors. If there is little overburden on the site, there may not be adequate resources to construct the perimeter berms.

To reduce the height of the berms, interim berms could be used. These are located fairly close to the operations and would be moved as required. In this way, the effectiveness of the barrier is increased, reducing the height requirements. However, there is the added operational complexity and cost to move the berms from time to time.

A further reduction in the height of the sound barriers can be achieved through the use of a portable barrier placed close to and around the drill. These barriers can be constructed on flat bed trailers fitted with skirts so that there are no holes or gaps. Again, complexity is added since the barrier needs to be moved frequently.

ROCK DRILLS

There are different types of rock drills available, with pneumatic or hydraulic drives. Percussive drill heads are common. Some models have dust collectors. The hammer type drills can be fitted either with an overhead hammer or with a down-the-hole hammer. Our experience has been that all models are relatively noisy, with sound emission levels in excess of 85 dBA at 15 m (some exceed 100 dBA). In some models, the drill head travels down the tower. In others, a down-the-hole head is used. The latter is reputed to be quieter. However, this does not appear to be the case as a result of radiation from the drill tower and above ground portions of the drill rod.

The best solution would be for mitigation to be added to at the manufacturing level to simply produce quieter rock drills. However, there are several difficulties with having this done:

- The major drill manufacturers, even today, are not fully aware that these stringent receptor based guidelines exist in Ontario. Their main noise concern is minimizing the sound level at the operator position.
- The Ontario market is too small to have enough clout with the manufacturers to cause design changes.

CONCLUSIONS

Quarries are often located in very quiet rural areas with a scattering of residential uses located around the proposed site. Currently the applicable noise exposure limit (one hour L_{eq}) can be as low as 40 dBA. Many quarry noise sources are fairly easy to mitigate since they are located on the quarry floor. The pit wall acts as a sound barrier by interrupting the line of sight between the operations and the neighbours, providing inherent screening. Most processing equipment can be enclosed. The rock drill operates on top of the working face and is often exposed to the neighbours. The drill is usually the single most significant noise source, producing in the range of 85 to 100 dBA at 15 m.

Mitigation measures other than the typical perimeter barrier often need to be considered. However, the best solution would be to manufacture quieter drills. To achieve this, quarry operators need to apply pressure and to educate the manufacturers about noise requirements in the province of Ontario.

REVIEW OF PHYSICAL MECHANISM AND COMPUTATIONAL MODELS

Gilles A. Daigle

Institute for Microstructural Sciences, National Research Council
Ottawa, Ontario K1A 0R6

INTRODUCTION

Propagation of noise in the atmosphere is governed by a number of interacting physical mechanisms including geometrical spreading, molecular absorption, reflection from a porous ground, curved ray paths due to refraction, diffraction by ground topography and scattering by turbulence. Accurate predictions of noise levels from a distant source must somehow account for all of these phenomena simultaneously. Although this goal is still beyond current capabilities, developments in computational tools for predicting sound propagation through the atmosphere have increased dramatically during recent years. The computational techniques now include analytical solutions for selected atmospheric profiles, ray tracing techniques which include interaction with the ground and meteorological conditions, and more sophisticated numerical solutions to the full wave equation; the fast field program (FFP) and the parabolic equation (PE). With modern computers, it is now becoming practical to incorporate some of these new computational tools into predictions schemes with advantages such as calculated noise contours based on observed meteorological patterns.

All noise prediction models include the attenuation due to geometrical spreading and, if required, molecular absorption. Where the empirical based models differ from computational models is in the incorporation of the other attenuation mechanisms. The empirical models tend to rely on general tendencies found in experimental databases. They often work well as long the specific situation of interest falls within the bounds of the databases. Computational models on the other hand rely on our mathematical ability to describe real-life situations. Recently the status of the computational methods was reviewed [1] and Benchmark cases to check performance and accuracy have been defined [2]. The purpose of this paper is to review the new computational models. The paper summarizes their limitations, their advantages, and shows a benchmark comparison of predictions. A complete and detailed description of each model, including comparison with experimental data, can be found in the cited references.

THEORETICAL BACKGROUND

The computational models assume simple harmonic time dependence $\exp(-i\omega t)$ and begin with the Helmholtz equation

$$[\nabla^2 + k^2]p(r,z) = -4\pi\delta(r, z - z_s) \quad (1)$$

where the wavenumber $k(z) = \omega/c(z)$, r is the horizontal range and z is the height above the ground. Reflection from a porous ground is described by the boundary condition

$$[\partial p/\partial z + ik\beta p]_{z=0} = 0 \quad (2)$$

at the ground surface where β is the normalized complex surface

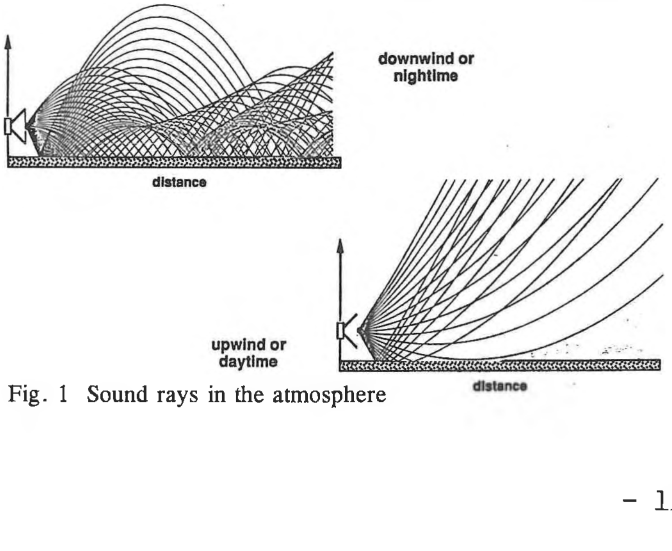

Fig. 1 Sound rays in the atmosphere

admittance. In general, the speed of sound varies with height resulting in curved ray paths due to refraction as shown in Fig 1. During nighttime or downwind propagation, ray paths are curved downward leading to multiple rays and favourable propagation conditions. During daytime or upwind propagation, ray paths are curved upward leading to an acoustic shadow with increased attenuation. The atmosphere is also turbulent, requiring the wavenumber to be separated into deterministic and stochastic parts, $k(z) = k_0(n_d + \mu)$, where n_d is the refractive index and μ a small perturbation. Turbulence scatters sound energy into shadows produced by barriers or refraction, limiting the amount of attenuation. Finally, terrain, such as berms or barriers, can be incorporated through boundary conditions or range dependence.

DESCRIPTION OF THE MODELS

The computational models describe below differ in their mathematical origin and in the way some of the physical mechanisms are incorporated. For example, some of the models are limited to a specific functional form for the sound speed profile. Some of the models do not include turbulence. All the models incorporate the boundary condition Eq. (2) in some way. The different approaches are presented following Ref. [2].

Analytical wave solutions

The Helmholtz equation (1) can be solved with a zero-order Hankel transform

$$p(r,z) = - \int H_0^1(Kr)\ P(K,z)\ K\,dK \quad (3)$$

Residue series solutions to Eq. (2) can be found when the sound speed is assumed to vary linearly with height. The residue solutions do not incorporate turbulence (i.e., $\mu = 0$). The downward refraction solution is called Normal Modes [3] while the upward refraction solution is described in terms of Creeping Waves [4]. The solutions usually converge rapidly. An example of how noise levels of a few hundred Hz are predicted to decrease with distance (Transmission Loss, TL) according to the Normal Mode solution is shown for a benchmark case [2] in Fig. 2(c).

In the case where the sound speed is constant with height above the ground, ray paths are straight and the solution Eq. (3) reduces to the more familiar sum of direct and ground reflected waves [5]

$$p(r) = A_d \exp(ikr_d)/r_d + Q(\beta,\phi)\ A_r \exp(ikr_r)/r_r \quad (4)$$

where r_d and r_r are the path length of the direct and reflected path, respectively, Q is the reflection coefficient and ϕ is the angle of incidence. In reality though, ray path are rarely straight and Eq. (4) is usually not valid at distances beyond a few hundred meters.

Ray tracing solutions

The effects of curved ray paths can be described from general principles. The curved ray modifies the angle of incidence and Eq. (4) can be used along with basic ray theory to construct heuristic physical solutions. Ray tracing solutions are computationally efficient. We note, however, that ray theory will not work beyond the shadow boundary in the case of upward refraction. The TL predicted from such a heuristic solution [6] for the downward refraction benchmark case is shown in Fig. 2(a). The heuristic model assumes the sound speed to vary linearly with height and can incorporate turbulence. The restriction of a linear sound speed profile can be removed if the ray solution is restricted to only one ground reflected ray. However this limits the validity to shorter distances [2].

The Gaussian beam approach [7] is another variation of ray tracing solutions. The basic concept of the theory is to launch a fan of ray-centered beams from the source and to trace the propagation of these beams through the medium. The wave equation is solved

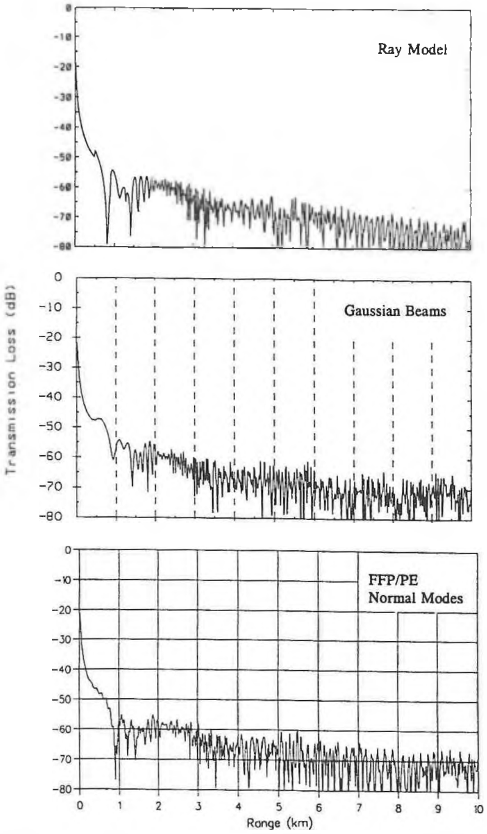

Fig. 2 Benchmark cases for downward refraction

in the immediate vicinity of each ray and the acoustic pressure at the receiver is obtained by summing the contribution of each of the individual beams. The Gaussian beam model can assume an arbitrary sound speed profile but the current implementation does not incorporate turbulence. The TL predicted by the Gaussian beam model for the benchmark case is shown in Fig. 2(b). We note that this model is effective in predicting the loss of performance of barriers where rays curve over the top of the barrier edge due to refraction [7].

FFP models

Fast Field Programs (FFP) performs a direct numerical integration on Eq. (4). They allow the prediction of noise levels in a horizontally stratified atmosphere where the sound speed is an arbitrary function of height. Current implementations are restricted to a flat ground with range-independent properties that assume a non-turbulent atmosphere. Four adaptations of the FFP are called CERL-FFP [8], SAFARI [9]. CFFP [10], and FFLAGS [11]. The CERL-FFP and CFFP treat the ground surface as a locally reacting impedance boundary while FFLAGS permits ground layering and elasticity in addition to porosity. The SAFARI FFP only allows layering and elasticity.

The three FFPs that assume porous ground predict the curve in Fig. 3(c) for the benchmark case. SAFARI predicts less Transmission Loss because of the assumption of a non-porous ground. The FFPs generally provide accurate prediction but are computationally time consuming. Further, the assumption of a non-turbulent atmosphere seriously restricts the use of the FFPs in the case of upward refraction.

PE models

The Parabolic Equation (PE) employs an assumption that wave motion for a particular problem is always directed away from the source or that there is very little backscattering. Writing $U = pr^{1/2}$, the Helmhotz equation in cylindrical coordinates is factored into propagation of incoming and outgoing waves. Considering only the outgoing wave leads to the *one-way wave equation*

$$\partial U/\partial r = i\sqrt{q}\, U \tag{5}$$

where $q = \partial^2/\partial z^2 + k^2$. Most implementations of the PE can be traced back to Eq. (5). The approach for advancing the field in range is the point of departure for the PE methods. Two popular software implementations are called FINITE-PE [12] and FAST-PE [13]. The FINITE-PE method numerically integrates Eq. (5) using a Crank-Nicolson approach. The boundary condition Eq. (2) must be satisfied at each step requiring several integration steps per wavelength to model the large variations of the field close to the boundary and results in computation times comparable to the FFPs. The FAST-PE uses a Green's function approach and a split-step operation that factors $\sqrt{q}$ into an operator for a homogeneous medium and another operator for propagation through the inhomegeneous perturbation. In addition, there are explicit terms for the field reflected from the ground allowing range steps of several wavelengths which results in dramatically decreased computation time.

The PEs allow the prediction of noise levels in a turbulent atmosphere [15] where the sound speed is an arbitrary function of height. Current developments are aimed at incorporating terrain [16,17]. In the case of the downward refraction benchmark case the PEs yield the curve in Fig. 2(c). In the case of upward refraction, models that neglect atmospheric turbulence predict large attenuations at longer ranges that are not supported by experimental data [18]. In the case of an upward refracting turbulent atmosphere the curves in Fig. 3 are typical levels [14] predicted for three values of the parameter μ. In essence, the relative sound pressure levels (SPL) in Fig. 3 represent the attenuation in excess of the transmission loss shown in Fig. 2 due to upward refraction.

SUMMARY

When experimental data is sufficiently documented to allow comparison with the computational models, good agreement is obtained for a wide variety of situations and conditions [8,18,19]. The speed of modern computers, the increase accuracy and reliability are making the use of computational models cost effective alternatives for noise prediction schemes. We are beginning to see effort directed at predicting the hourly, daily, or seasonal variations in noise levels due to changes in environmental conditions by incorporating local weather into predictions schemes using computational models.

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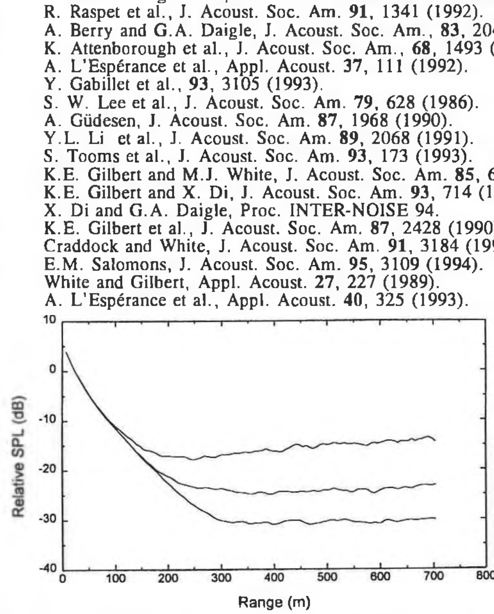

Fig. 3 Typical predictions in the case of upward refraction

EFFECTS OF EXCESS GROUND ATTENUATION ON AIRCRAFT NOISE CONTOURS

J.S. Bradley
Acoustics Laboratory, Institute for Research in Construction
National Research Council Canada
Ottawa, Ontario K1A 0R6

Introduction

Various airport noise prediction programs are used to calculate expected noise level contours around airports based on the details of the aircraft operations. One of the major factors influencing these contours is the excess attenuation of the sound from the aircraft when the sound is propagating at near grazing incidence to the ground. When considered in detail, this excess attenuation is a complex phenomenon, but airport noise prediction programs use quite simple approximations to estimate the effect of excess ground attenuation.

The attenuation of sound from an aircraft traveling close to the ground will depend on a number of factors that will vary as the aircraft passes by. The attenuation will be influenced by the spectrum and directionality of the aircraft noise source and these effects will vary with time as the aircraft passes an observation point. The resulting attenuation will also be influenced by the acoustical impedance of the ground as well as various meteorological effects. Thus, to accurately predict the attenuation of the sound from an aircraft, quite complex calculations would be required on a point by point basis as the aircraft passes an observation point.

Most airport noise prediction programs include only quite simple approximations to these complex effects. The influence of each aircraft fly-by is typically only calculated for the point of closest approach to the observation point and not as a complete point-by-point calculation for the complete fly-by. Usually, only overall A-weighted or PNL-weighted levels are considered. The excess ground attenuation is usually calculated in two separate parts: (a) ground-to-ground propagation, and (b) air-to-ground propagation.

Calculation Procedures

Figure 1 compares five different ground-to-ground attenuation algorithms. Of the five, the SAE model gives the largest attenuation at most distances. This SAE algorithm is used in the American INM and NoiseMap airport noise prediction models. Transport Canada's NEF_1.7 prediction program provides the lowest ground-to-ground attenuation. A Swiss procedure, an algorithm from an older version of NoiseMap, and an experimental

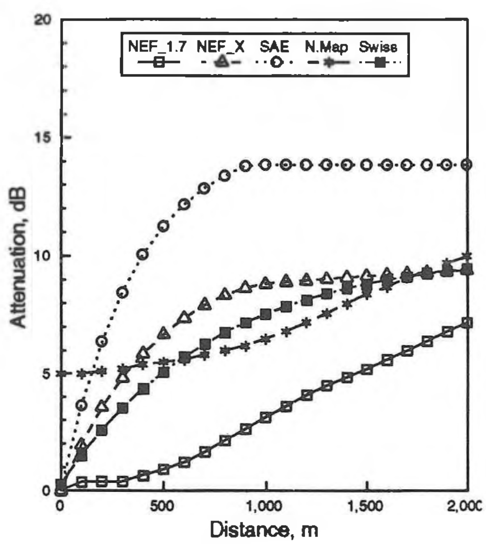

Figure 1. Comparison of ground-to-ground attenuation calculations.

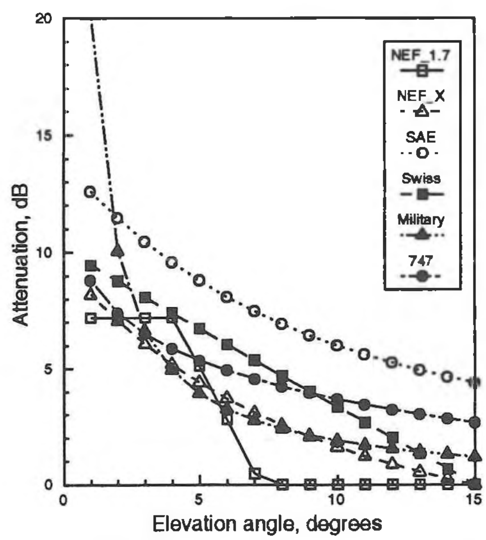

Figure 2. Comparison of air-to-ground calculations and measurements.

version of the Transport Canada program include intermediate values of ground-to-ground attenuation.

Air-to-ground attenuation calculations are compared in Figure 2. Again, the SAE model gives the highest attenuations and the procedures used in the NEF_1.7 program tend to give lower air-to-ground attenuations. As in the previous plot, the Swiss procedure and the experimental NEF_X program give intermediate results. This figure also includes an average curve of measurements of a Boeing 747 aircraft. This curve tends to approximate the intermediate predictions of air-to-ground attenuation. The curve labeled Military is a fit to the measured air-to-ground attenuations of various types of military aircraft.

Both Figures 1 and 2 indicate that there are quite large differences among the various procedures and that predicted aircraft noise levels could vary by several decibels. Figure 3 compares the combined effects of the SAE procedure with the NEF_1.7 program calculations. The differences in predicted attenuations are plotted versus elevation angle and distance from the source. Differences as large as 11 dB were found. The average difference for all distances and angles shown in Figure 3 is 4.85 dB.

Single Aircraft Examples

The large differences between excess ground attenuation calculations are expected to significantly effect the area of the airport noise contours. Contours for single aircraft types were first compared. Figure 4 compares four different calculations of NEF 20 contours for 100 take-offs of a Boeing 737-D17 aircraft. For this example the NEF_1.7 and NEF_X results had similar areas that were approximately double the areas calculated by the INM and NoiseMap programs. Although the NEF_1.7 and NEF_X programs tended to produce larger area contours, the differences between the different sets of contours varied with the aircraft type and the contour noise level.

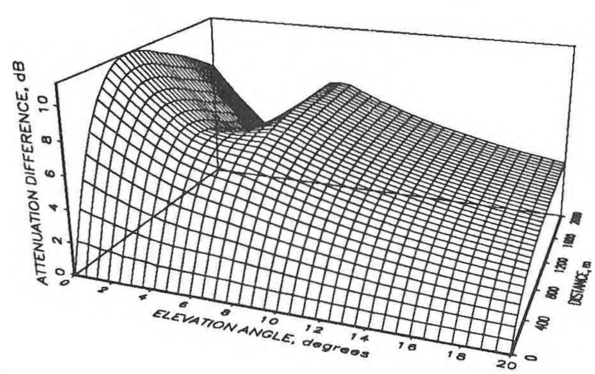

Figure 3. Differences between INM and NEF_1.7 excess ground attenuation calculations.

Complete Airport Examples

Comparisons were also made of complete airport noise contours resulting from the actual mix of aircraft operations at each airport. Figure 5 compares calculated contours for Ottawa airport. For the NEF 20 contour at this airport, the NEF_1.7 program produced contours approximately 60% larger than the INM program. The experimental NEF_X program with reduced excess ground attenuation produced contours with areas intermediate to the other two programs. The differences in the areas of these contours are thought to be largely due to differences in the calculation of excess ground attenuation and vary with both airport size and contour level.

Conclusions

Excess ground attenuation calculations included in commonly used airport noise prediction programs are only rough approximations to the actual propagation phenomena. There are considerable differences between the procedures used in different airport noise programs. As a result, the areas of the noise contours calculated for individual aircraft or for the mix of aircraft at actual airports can vary considerably between computer models. This is clearly an area where improvements to calculation procedures are necessary.

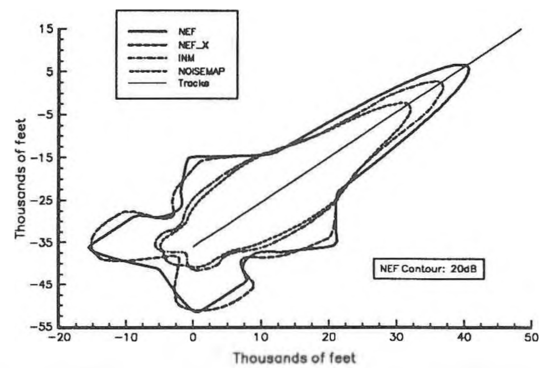

Figure 4. NEF 20 contours for 100 take-offs of a 737-D17 aircraft.

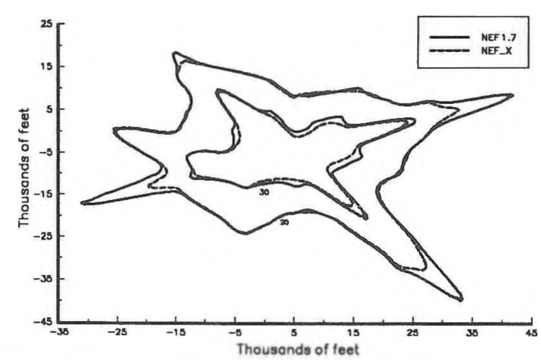

Figure 5. Calculated NEF 20 contours for Ottawa airport.

Insertion Loss Characteristics of Barriers and Berms

C.C. Harrison, K.R. Fyfe and L.J. Cremers
Department of Mechanical Engineering,
University of Alberta, Edmonton, Alberta, Canada

INTRODUCTION

Over the past 40 years, much effort has been placed into developing more accurate modelling tools for determining the acoustic performance of barriers and berms for road noise attenuation. Most of the work completed in this field has used ray-based diffraction models which ignore phase information of the acoustic waves [1]. This paper describes initial work completed in computing insertion loss characteristics of various barriers and berms as a function of frequency and distance behind the barrier using a wave-based model. A two-dimensional boundary element procedure has been used to calculate the sound fields of the presented results [2].

A cross-sectional slice of a flat roadside geometry and a simple thin barrier is depicted in Figure 1. Both the source (S) and various receiver (R) positions are shown.

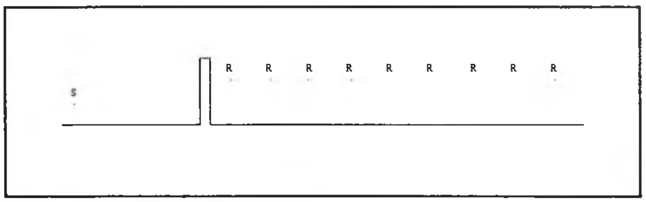

Figure 1 - A Typical Geometry

It has been most common to measure the performance of a barrier in terms of its insertion loss (IL). In many previous analyses [3,4], the insertion loss at one receiver point has been referred to as indicative of the general overall barrier performance. In another study, Hothersall et al. [5], the average of the insertion losses at various receiver points has been used as the basis for rating the barrier. For the current study, it has been chosen to use a range of equidistant receiver positions from 4m to 100m behind the barrier at a height of 1.5m off the ground. In this analysis, 3m high barriers are situated upon hard, level ground and a single line source is placed 15m from the center of the barrier and 0.5m above the ground.

RESULTS

Figure 2 displays insertion loss curves at octave band center frequencies for a simple hard thin barrier (0.2m in width) as a function of receiver distance from the barrier.

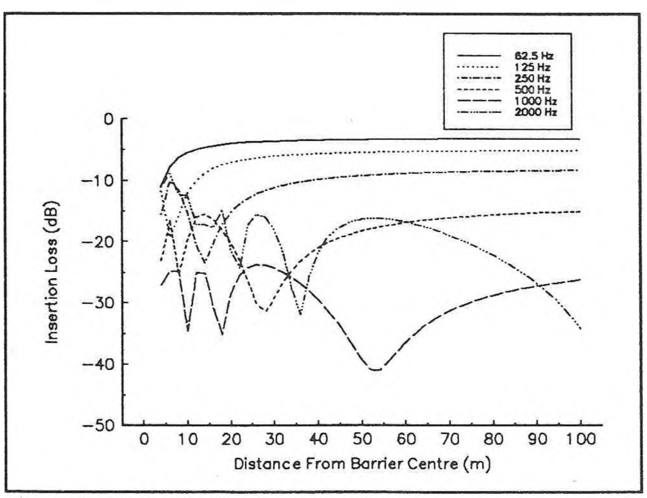

Figure 2 - Thin Barrier Insertion Loss - octave center freq's

It can be seen that at any given distance, depending on the frequency of the sound source, the barrier rating can vary anywhere between 5 and 40 dB's of insertion loss. Thus, selecting a barrier for a specific application or geometry can be quite simple, but designing an optimal shield for the wide range of combinations of the source position, frequency and amplitude, as well as the receiver location and the terrain can be difficult. To further emphasize this point, Figure 3 depicts the insertion loss as a function of both frequency and receiver distance from the thin barrier.

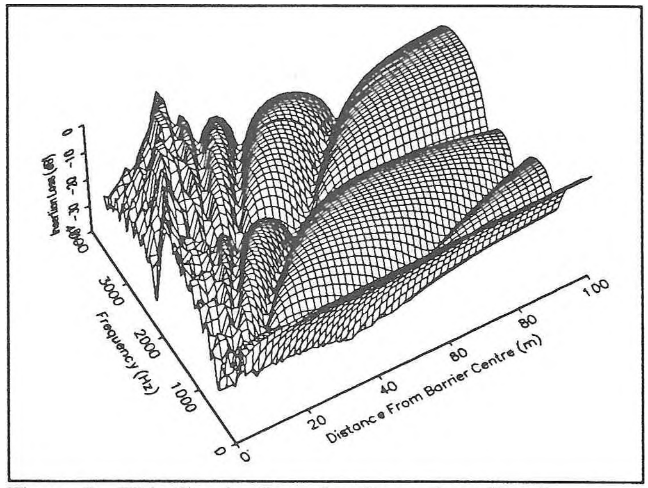

Figure 3 - Thin Barrier Insertion Loss - broadband

Depending on the choice of characteristic frequency or position, any one location on this surface could represent the barrier performance.

While it is practical to obtain a single reading from a physical measurement of the field produced by a broadband source, the wave-based boundary element modelling determines results at discrete frequencies. These results can be combined by making use of an input spectrum of the expected traffic noise. One such measured spectrum [6] is shown in Figure 4. The A-weighting curve and the resulting A-weighted spectrum are also shown.

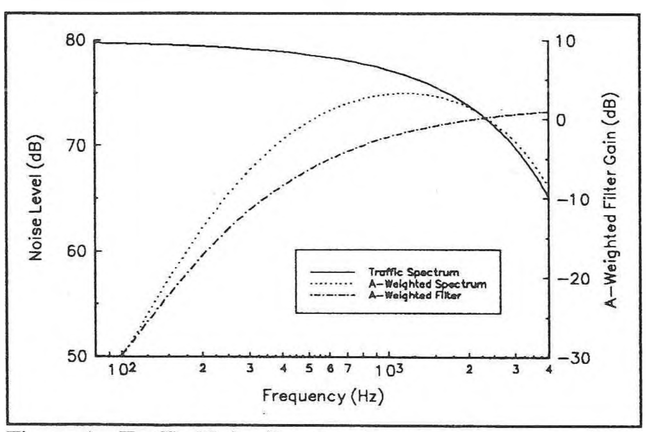

Figure 4 - Traffic Noise Spectrum

This weighted spectrum can act as a *filter* to combine the broadband measurements into one insertion loss curve that is now only a function of distance. Various spectrums, typical of different traffic configurations, could also be tested to evaluate the barrier or berm performance. Working with a single insertion loss curve, as a function of distance for each barrier, the selection of a particular barrier is now simplified. To take this one step further, various schemes are now being investigated to combine or integrate the insertion losses at the receiver positions. In this way a single insertion loss parameter could be used to quantify the barrier's performance.

In addition to the simple thin barrier, various shapes of barriers and berms have been modelled. Figure 5 shows some of the familiar barrier shapes.

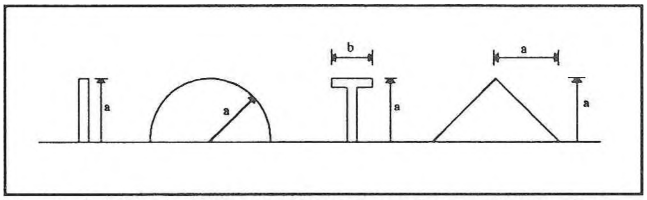

Figure 5 - Barrier Shapes with Dimension a=3m, b=2m

To compare these barriers, insertion loss curves are presented at a frequency of 500 Hz. Several national agencies [7,8] use an insertion loss curve at 500 Hz to be representative of a broadband insertion loss curve. The results of this test are shown Figure 6. The results show that the straight and 'T' shaped barriers are superior by about 3dB over most of the frequency range. More work is planned to investigate optimal barrier and berm geometries for various road configurations.

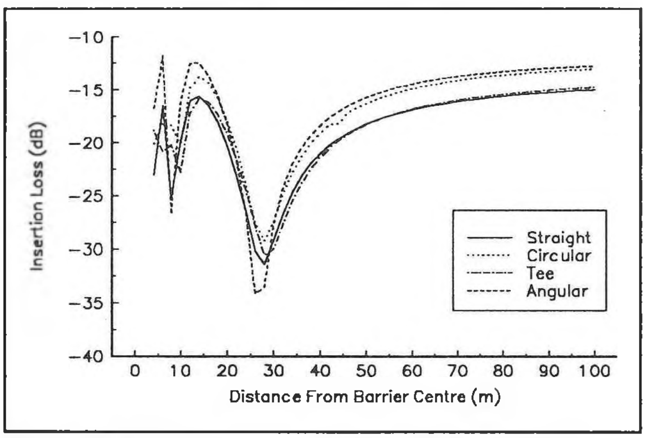

Figure 6 - Insertion Losses of Various Barrier Shapes

In addition to the insertion loss curves presented, sound level contour plots can be used for a quick and accurate visual comparison of fields with and without a barrier. Figure 7, for example, depicting a raised road geometry, is included as an example of these modelling capabilities. Sound levels, for this 500 Hz test, range from 100dB near the vehicles to 50dB in the shadow regions.

Ongoing studies are investigating the application of absorbative linings to both the barriers and on the ground surfaces in the shadow region of the barrier. As well, the effect of various road-side geometries on the performance of the barriers are being studied.

ACKNOWLEDGEMENTS

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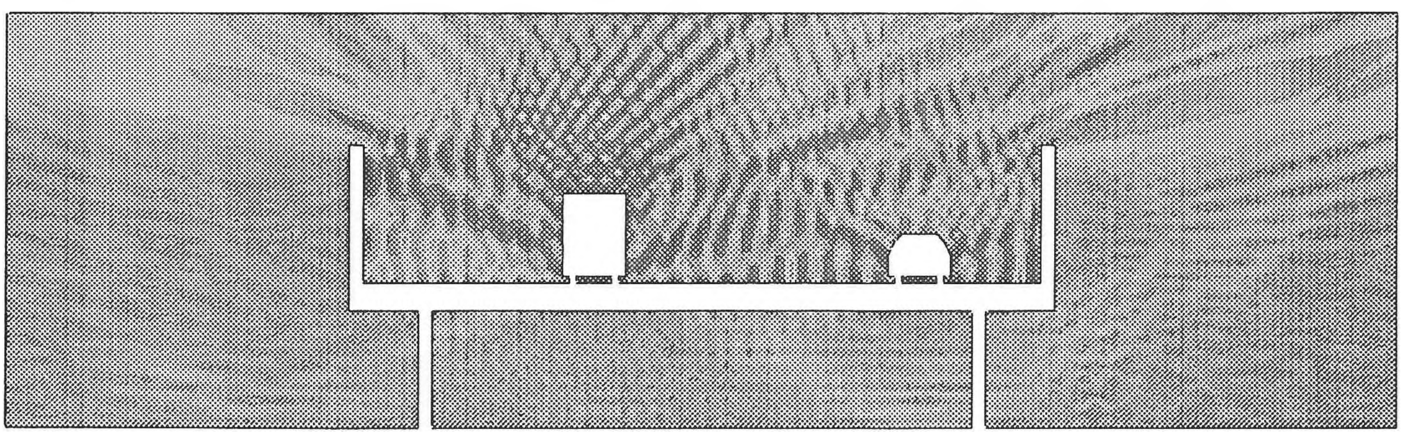

Figure 7 - Sound Pressure Contours for Raised Highway with Barriers

The Draft International Standard Method (ISO/DIS 9613-2) for Calculating the Attenuation of Sound during Propagation Outdoors.

Joe E. Piercy, IMS, National Research Council, Ottawa, Canada K1A 0R6

INTRODUCTION - how it came about

The International Standards Organization (ISO) technical committee TC 43 on Acoustics, sub-committee SC1 on Noise, in 1983 voted to form a new working group, WG24, with the objective "to specify methods for determining the attenuation of sound propagating outdoors. This information is necessary for determining sound pressure levels at a distance from a noise source, e.g. an industrial plant, a vehicle or an aircraft." This objective placed the project squarely between two sets of existing ISO standards - those specifying methods for determining the levels of sound power emitted by various sources, such as machinery and specified equipment (the ISO 3740 series) or industrial plants (ISO 8297), and the ISO 1996 series which specifies methods for the description of noise outdoors in community environments.

The response of Working Group 24 over the last 10 years has been to produce a potential standard in two parts, which are really two separate standards, to bridge the gap between these two existing types of standard, - ISO 9613, Acoustics - Attenuation of sound during propagatioon outdoors. Part 1, Calculation of the absorption of sound by the atmosphere, was published in 1993 as an ISO Standard, and Part 2, A general method of calculation, is currently being circulated for approval as a Draft International Standard. Together they enable levels of noise in the community, as described in ISO 1996, to be predicted from a variety of sources of known sound emission.

Now specifically how did these documents come about? Working Group 24 has 14 members from 13 different countries. At the first meeting in 1984 it became clear that a great deal of scientific work in this general direction had been proceeding, relatively independently, inside several countries - usually to back their noise regulations. The work was usually sponsored by the national government, and typically published only in a contract report to that government, usually in the local language, which tends to make it relatively inaccessible and unknown. Working group 24 initially sorted through this work, found much of it was complementary, and made the initial estimate that it could be integrated into the two part ISO document. Fortunately the work of integration had already begun inside individual countries in national documents such as standards and guidelines.

The proposed International Standard is derived mainly from the national documents listed in the references. Ref. 1 is the US contribution - the American National Standard method for the calculation of attenuation of sound due to atmospheric absorption. Part 1 of ISO 9613 is essentially an updated version of ANSI Standard S1.26 of 1978, which has been widely used. Ref. 2 is the "Guidelines" developed by the Netherlands, and ref. 3 the "general prediction method" developed jointly by the Scandinavian countries through their "Nordforsk Project", both for industrial noise. Ref's 4 and 5 are the "Guidelines" developed for the control and regulation of noise in Germany by the VDI, their engineering society. ISO/DIS 9613-2 is mainly an integration of the national methods given in these references 2 to 5.

It should be recognized therefore that the methods given in DIS 9613-2 are backed not only by the scientific and engineering studies given in these references, but also by the practical experience in the prior use of the methods in the country of origin.

SCOPE

ISO/DIS 9613-2 specifies an engineering method for calculating the attenuation of sound propagating outdoors, in order to predict levels of environmental noise in the community at a distance from sources of known sound emission. This method is applicable in practice to a great variety of noise sources and environments, including either directly or indirectly, most situations concerning road or rail traffic, industrial noise sources, construction activities, and many other ground-based noise sources.

The method aims to determine the average sound level $L_{Aeq,T}$ under meteorological conditions favorable to propagation - that is, moderately downwind propagation, or equivalently, propagation under a well-developed, but moderate, ground-based temperature inversion, such as commonly occurs at night. These conditions are chosen for stable propagation (i.e., accuracy of measurement and prediction), and also to provide an appropriate condition for a specific community noise limit - i.e., a level which is seldom exceeded.

The long-term average sound level $L_{Aeq,LT}$ may also be calculated from the above $L_{Aeq,LT,}$ using a small correction (0 to -5dB) based on elementary local weather statistics. This long-term average will normally cover a variety of weather conditions, as for example in the yearly day-night sound level YDNL.

The method in ISO/DIS 9613-2 consists specifically of octave-band algorithms (with centre frequencies from 63 to 8000 Hz) for calculating the attenuation of sound from a point source, or an assembly of point sources. Specific terms are provided in the algorithms for the following physical effects - geometrical divergence, atmospheric absorption, ground effect, reflection from surfaces, and screening by obstacles. The effect of atmospheric turbulence is included implicitly. Additional information concerning propagation through housing, foliage, and industrial sites is included.

ACCURACY

The estimated accuracy of prediction using the method of DIS 9613-2 for broad-band noise sources is shown in table 1. These figures have been obtained by comparing calculated values with an extensive data-base of measurements. The accuracy for propagation distances d less than 500m is seen to vary somewhat with distance and height of sound path h above the ground. For distances greater than 500m there is insufficient evidence to support an estimate of accuracy.

GROUND EFFECT

For downwind propagation from source S to receiver R the sound paths curve downwards, as shown in Figure 1 (a), and for distances long enough, there are several distinct paths 1,2,3 etc. as indicated. For each path there are a group of closely spaced rays from images I caused by ground reflection, as indicated at (b). Interference between these rays produces the ground effect[6]. For downwind propagation therefore the nature of the ground surface in the vicinity of the source and receiver is of prime importance, and that in between less so. The algorithms of DIS 9613-2 reflect this fact, and allow three surface conditions for each region - hard (asphalt, water, etc.), soft (any ground which will support growth), and mixed.

SCREENING

The attenuation due to screening may be calculated for diffraction over the top of a barrier (which includes a ground effect) or around its sides, as shown in Figure 2. There may be a single diffraction in each sound path when the barrier is thin, as shown in Figure 2, or the barrier may be thick, requiring a double diffraction per path. Thus the attenuation due to screening may be calculated for a variety of shapes of buildings as well as sound barriers.

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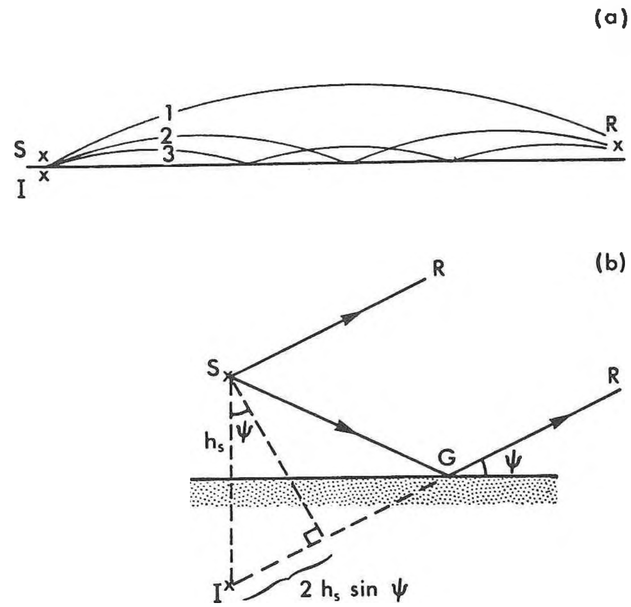

Figure 1. Sound propagation paths downwind

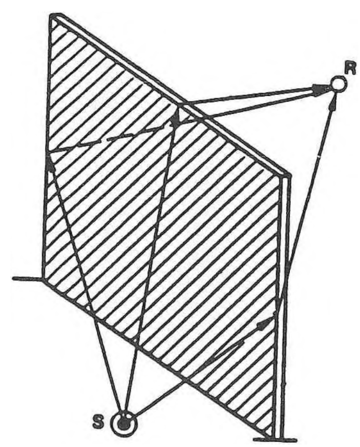

Figure 2. Different sound propagation paths at a screen

Table 1. Accuracy of $L_{Aeq,T}$(downwind)

Distance d	0 to 100m	100 to 1000m
h = 0 to 5 m	± 3 dB	± 3 dB
h = 5 to 30 m	± 1 dB	± 3 dB

The Estimation of the Linear Sound Speed Profiles Under General Meteorological Conditions

A. L'Espérance, Mechanical Engineering Department, University of Sherbrooke
Sherbrooke, Québec, Canada

G. Daigle, Institute for microstructural Sciences, National Research Council
Ottawa, Ontario, Canada

Introduction

In the case of medium and long ranges outdoor sound propagation, refraction due to temperature and wind gradients may increase (or decrease) the sound pressure due to a point source[1,2]. To fulfill the needs of practical outdoor sound propagation studies, it appears to be necessary to predict the sound speed profile (SSP) according to general meteorological informations.

The first part of this paper describes a practical method to estimate the sound speed profiles (SSP) under general meteorological conditions. The second part presents a method to estimate the corresponding linear SSP that can be incorporated in the heuristic acoustical model for outdoor sound propogation[3]. Finally, the third part of this paper gives comparisons between experimental and theoretical results.

1.0 Meteorological model

The sound speed profile between a source and a receiver could be expressed as a function of temperature, and wind characteristics. The exact determination of the wind speed and temperature profiles from general meteorological conditions is not possible. Using the surface-layer similarity scaling theory, it is however possible to obtain a good estimate of these profiles[4].

For this theory, it is necessary to evaluate *L*, the Monin-Obukhov length. This value depends mainly on the friction velocity and on the vertical heat flux at the surface. This last parameter is however difficult to measure and not easily available. To overcome this problem, a simple method to estimate *L* is proposed. This method use the relation observed by Golder[5] between values of Turner classes, the roughness of the ground and *L*.

2.0 Estimation of the linear sound speed profile

In a recent paper[6], an heuristic acoustical model for outdoor sound propagation has been presented. This model is an extension of the classical ray-theory that includes the effects of curved rays due to the refraction. This acoustical model assumes a linear sound speed gradient defined as:

$$c(z) = c(o)\,(1 + a\,z)$$

where a is the linear sound speed gradient, c(o) is the reference sound velocity on the ground and z is the height of the point. The assumption of a linear sound speed profile allows an analytical determination of the ray path parameters (travel time, angle of reflexion on ground, path length etc.)

The linear sound speed gradient *a* used in the acoustical model must be estimated from the SSP determined by the meteorological model. The method proposed is deduced from one used in radio communications[7]. Between a source and a receiver, the zone of space concerned with the propagation process is mostly defined by the first Fresnel ellipsoïd (Fig. 1). This is the zone of space where the path length difference between the direct path and any scattered path is lower than the half wave length.

$$\frac{\lambda}{2} = r_1 + r_2 - r$$

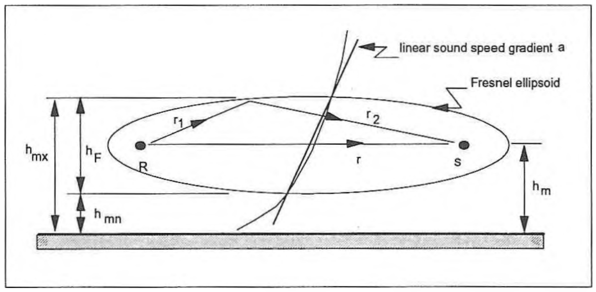

Figure 1 Schema of the the Fresnel ellipsoïd, the zone of space concerned with the propagation process.

The equivalent linear sound speed gradient is obtained by considering the mean of the real sound speed profile in the first Fresnel zone.

$$a = \frac{c(h_{mx}) - c(h_{mn})}{c(o)\,(h_{mx} - h_{mn})}$$

where $h_{mn} = h_m - h_F$

and $h_{mx} = h_m + h_F$

with $h_F = \sqrt{\frac{\lambda}{4}\left(r + \frac{\lambda}{4}\right)}$

(1) Thus, because the wide of the Fresnel ellipsoid (h_F) is function of the distance and of the frequency, the linear sound spreed gradiant *a* depends not only on the heights of the source and receiver, but also on the frequency and on the source and receiver.

3.0 Comparison with experimental results

To validate and determine the limitations of this approach, various acoustical and meteorological measurements of the noise emitted by strong and steady sources of an industrial plant have been done during the summer of 1993. These measurements were done during different days and at various periods of the days. The receiver's position considered was located at about 1 km of the sound sources. The data were analysed and show that there is an strong correlation between the increase of the SPL and of the linear sound speed gradient. Figure 2 shows the statistical distribution of the SPL measured during different periods and Fig. 3 shows these SPL as a function of the linear sound speed gradient.

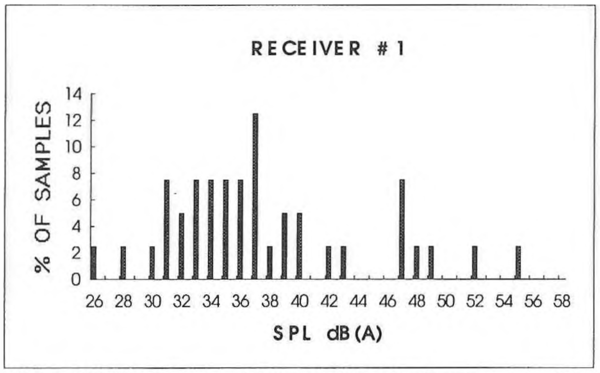

Figure 2 Distribution of the total Sound Pressure Level from two sources (320 and 440 Hz) of an industrial plant.

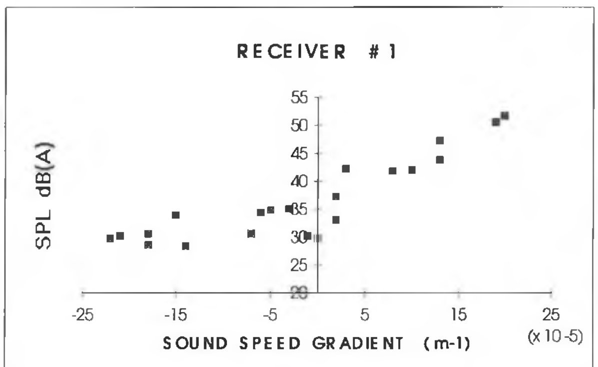

Figure 3 Sound Pressure Levels as a function of the linear sound speed gradient measured near an industrial plant (F=320 and 400 Hz).

To complete this analysis, the excess attenuation spectra (mean values, standard deviation and maximum values) measured at 2 089 m for 415 ground runup events[8] were compared to theoretical results. As the different meteorological conditions associated with these acoustical measurements were not known, mean values of linear sound speed gradient were used. The mean excess attenuation spectra predicted is in good agreement with the mean of the experimental results. Also the variations of the experimental data around the mean value can be explained by the variations of the various refraction conditions during the measurements.

Finally, it should be mentionned that, in combination with the heuristic acoustical model[6], this method gives a complet and practical scheme to predict outdoor sound propagation under various meteorological conditions, and only requires low computation time.

4.0 Conclusion

In this paper, a practical method to determine the linear SSP from general meteorological informations was proposed. In combination with a geometric ray acoustic model, a complete model for engineering purposes is obtained to predict variations of the SPL under different meteorological conditions. Comparisons with experimental results have shown that the general tendencies are well respected. From the experimental measurements, it was also observed that the heat flux conditions, function of the solar altitude and of the cloud cover, have an important influence on the increase of the sound pressure levels. Thus, the SPL's in the night period were statistically higher that those occuring during the day and the highest SPL have occured during the night when the wind direction was near the axis of propagation.

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MEASURING THE EFFECTS OF TURBULENCE

David I. Havelock

Institute for Microstructural Sciences
National Research Council, Ottawa, Canada K1A 0R6

Introduction

Turbulence plays an important role in determining the characteristics of sound propagating through the atmosphere. Only recently have acoustic theory, numeric modelling, and field measurements provided consistent values of SPL in the presence of turbulence. With the recent and rapid development of effective propagation codes, there is an increasing need for reliable and comprehensive field data against which to validate the codes and gain a better understanding of the effects of turbulence.

The effects of turbulence are perhaps most readily observed within an acoustic shadow region. A shadow region is generated near the ground by upward refraction of the propagating sound. It can also occur behind a sound barrier or terrain masking feature. Within an acoustic shadow region, there is no direct propagation path from source to receiver. Near the shadow boundary there is diffracted acoustic energy which decreases rapidly within the shadow region. In the deep shadow region, the meagre diffracted field is dominated by the acoustic energy which is scattered into the shadow region by the insonified turbulence outside of the shadow.

A schematic representation of the three major region is shown in Fig. 1 for a simple upward refracting geometry above a flat ground. Analogous geometries exist for curved ground (terrain masking) or acoustic barriers. The general form of this schematic has been observed repeatedly in outdoor measurements [1,2] and more recently in numeric simulation studies [3].

A large array of microphones was developed to investigate the effects of turbulence on a sound field. A large array is required because simultaneous measurements must be gathered over a large area to characterize the sound field before the parameters of the non-stationary turbulence change.

In this paper, the array is discussed and measurements of spatial coherence within a refractive shadow region are presented as a representative example of the effects of turbulence being measured.

The Microphone Array

The microphone array consists of up to 64 sensors and uses 1/4" electret microphones with matched preamplifiers. The microphone preamplifiers are connected to programmable line amplifiers (PGA) which can handle up to 6 channels. The preamplifier and PGA have a robust circuit design and are housed in a rugged enclosure. The microphones can be located 100 m from the PGA, allowing each PGA to service microphones which cover a large area. The PGA's are controlled remotely for gain settings and a power-saving sleep mode.

With over 70 Km of wire and 5000 connections, the cable set is a significant component of the array. A comprehensive grounding strategy is implemented throughout the cable set and the electronic components to provide good shielding and prevent ground loops. Cable specifications provide abrasion resistance, UV tolerance, water resistance, shielding and ample signal ground return. The main cable trunk lines connect the PGA's to the equipment trailer. They can be configured so that the PGA's are at distances of 100 to 1000 m from the trailer. Up to 96 channels of distinct signal can be carried by the trunk lines.

By using different combinations of cable lengths and clustering microphones at the PGA, tremendous flexibility can be achieved in the configuration of the array. In addition, up to 16 channels can be configured separately from the array using battery power and 4 mm DAT instrumentation recorders to provide satellite array capabilities.

The equipment trailer is equipped with battery and generator power systems and it is possible to switch between power sources without interrupting operation. A patch panel at the equipment trailer allows routing and sequencing of incoming signals and provides a single junction point to earth-ground. Inside the trailer, the signals are filtered using customized, programmable 5-stage analog bandpass filters with 18 poles per channel. The filtered signals are multiplexed using an array of solid state switches. The typical sampling rate is 8000 samples per second per channel and the switches can operate at up to 2 MHz. Data acquisition is performed by a 12-bit DAC hosted by an EISA-PC/486 class computer. Real time recording to disk is performed to the capacity of the disk and the data is subsequently backed up to 8 mm DAT for off-line storage.

The sound source consists of up to four single-driver enclosures tuned for 40-1000 Hz operation. A portable generator and a 2.4 kW amplifier are used to generate about 130 dB ref. 20 μP at 1 meter. A 10 m crank-up tower is used to mount meteorological instruments at heights of 2 and 10 meters. The temperature and wind data arecarried to the equipment trailer by the trunk lines where it is digitized. Most of the field trials have been conducted at a privately owned airport which has 700 m runways and is conveniently located near the laboratories. The flat, hard surface, and the variable weather conditions, have provided numerous interesting data sets.

Sound Field Characteristics

Through repeated tests and analysis we are investigating a number of characteristics of the sound field, particularly within a refractive shadow. The following is a representative list of the measurements and analysis being pursued. **1)** Average sound pressure levels as a function of range, **2)** vertical profiles of the sound pressure level near the shadow boundary, **3)** transverse and longitudinal spatial coherence, **4)** statistics of the fluctuations in the phase and magnitude at various ranges from the source, **5)** coherence between tones of different frequencies (frequency coherence measurements), and **6)** temporal coherence of the sound field.

Collection and processing of the data is ongoing. As an example of the sound field characterization being achieved, some results for spatial coherence will be presented.

Spatial Coherence Within an Acoustic Shadow

The spatial coherence length of the sound field is the separation distance at which the correlation drops from unity by 1/e of the drop at infinite separation. Outside of the shadow region, such as during downward refraction conditions, the transverse coherence length is quite short; it tends to be about 1 m at long ranges. The longitudinal coherence length outside the shadow region, on the other hand, is very long; it can be much greater than 100 meters [4].

Data for the transverse coherence within a shadow region is shown in Fig. 2. The signal frequency is 500 Hz and the range from the source to the receivers is 700 meters. The lower curve corresponds to stronger upward refraction conditions, for which the receivers are within the deep shadow region. The upper curve corresponds to weaker upward refraction and, with the receivers at the same range, they are closer to the shadow boundary yet still within the shadow region. The observed transverse coherence is 1 or 2 meters, which is similar to that expected outside of the shadow region.

The longitudinal correlation data is plotted in a similar format in Fig. 3, except that the sensor separation axis has a different scale. The longitudinal coherence length is greater than the transverse coherence length within the shadow region, yet much less than that expected outside of the shadow.

Summary

A large array of microphones provides a useful tool for investigating the characteristics of sound fields in the presence of atmospheric turbulence. Statistics other than the SPL, such as the spatial coherence, are strongly effected by the atmospheric turbulence. Characterizing these effects will improve our understanding of the relationships between turbulence and the sound field. This research is ongoing and is applicable to propagation modelling and simulation, adaptive beamforming for remote sensing, and acoustic probes for detecting meteorological effects.

Acknowledgements

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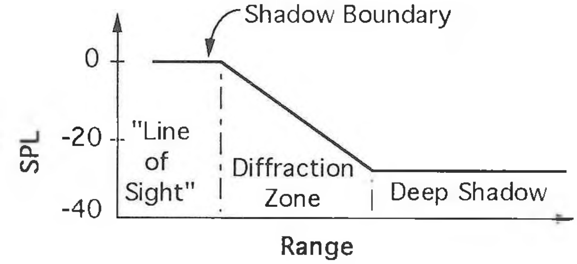


Fig. 1 Upward Refraction

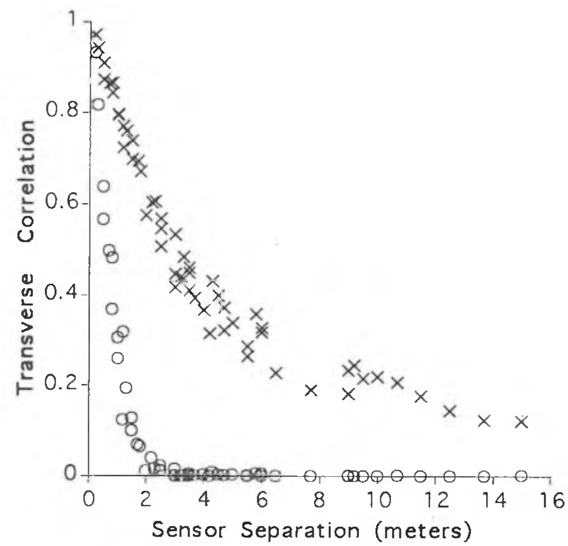


Fig. 2 Transverse Correlation

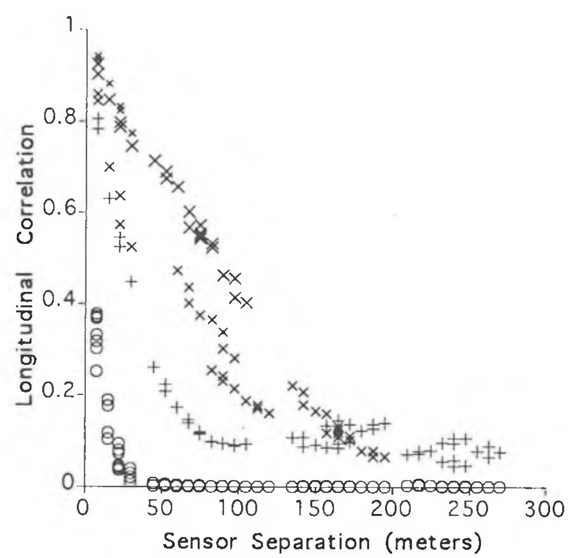


Fig. 3 Longitudinal Correlation

DSA (DIGITAL SPEECH AID) FOR STUTTERING PEOPLE

Marek Roland - Mieszkowski
Digital Recordings - Advanced R & D, 5959 Spring Garden Rd., Suite 1103, Halifax, Nova Scotia, B3H-1Y5, Canada

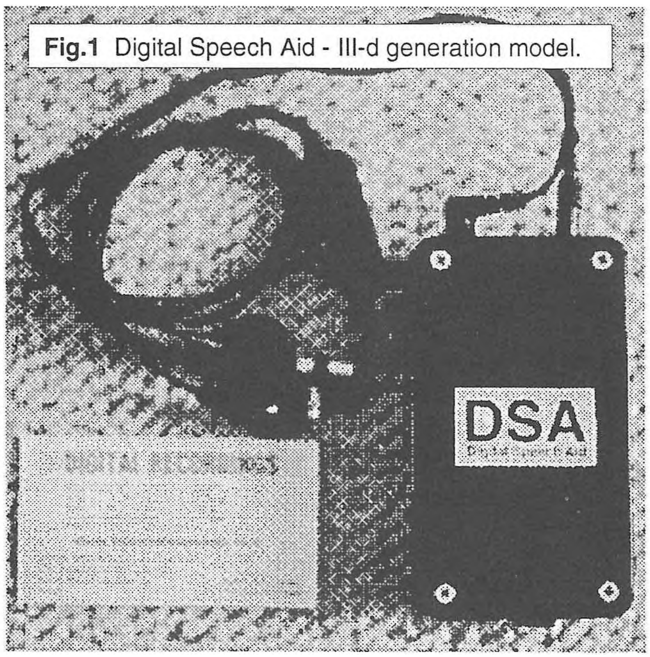
Andrzej Czyzewski & Bozena Kostek
Technical University of Gdansk, Sound Engineering Department, ul. Narutowicza 11/12, 80-952 Gdansk, Poland

Introduction

The **Digital Speech Aid** (Fig.1) is based on advanced digital signal processing technology. The software developed for the device exploits new approach to stuttering disorder. With DSA many stuttering people can speak fluently (or more fluently) in any fashion and at any rate.

Results of the **DSA** testing supports authors' theory, that stuttering is a physiological disorder, in most cases of a neurological basis. Stutterers become nervous because they stutter (not as believed by some, that they stutter since they are nervous).

Fig.1 Digital Speech Aid - III-d generation model.



Origins of Stuttering and Conventional Treatment

In many cases stuttering was believed to be caused by psychological disorders and nervousness of the person. This stigma still exists in the large part of the society and large part of the medical and health community. Many stutterers themselves believe in this very strongly. Outcome of this believe is not only great suffering on the part of the person, due to the fact of being labeled as "weird", "nervous" and "psychologically unstable" but also very often wrong approach to the treatment of the disorder.

Treatment offered by the Speech Pathologists involves various techniques to slow down the speech, coordinate speech production with breathing, change in various ways way of speaking and pronouncing words etc. It also involves some counselling and relaxation therapy, which very often overlaps with work and input from the Psychologist. These techniques work to certain degree and results depend very much on the particular case. Also they work often better in the clinical setting than in the real world, where person cannot concentrate as much on speech production. And unfortunately many of these techniques require a conscious effort on the part of the stutterer. Many people give up the speech therapy because in some cases they feel that fluent, but unnatural sounding speech is worse then stuttering itself.It is estimated that about 5% to 10% of stutterers are receiving some form of the therapy (indicative of the current treatments effectiveness) [1].

If stuttering is a physiological disorder (of neurological nature in most cases), telling a person to control it does not make much sense. It is like telling a person with faulty vision to take off glasses and to concentrate to see better.

New approach to understanding of stuttering

Several years ago, authors started a series of experiments to find what are the mechanisms of stuttering and whether one could use some techniques to compensate for this physiological (in authors' opinion) deficiency of speech.

Review of literature and consultations with several experts in the field reviled, that very little is known about stuttering and even less is understood. Authors were stunned by the many misinterpretations of the facts and experimental results [1].

It became obvious later why it is so - simply majority of the people in the field of the Speech Pathology have very little background in mathematics, physics, acoustics, electronics and signal processing. Therefore they are not equipped to interpret available data correctly. For this reasons authors' new approach and interpretations were met with skepticism in some cases and with hostility in others [2].

Digital Speech Aid (DSA)

DSA (Fig.1) is a small 11cmx6cmx3cm (LxWxH), sophisticated, electronic device with 256 different program settings. DSA uses a microphone and pair of earphones. It operates on batteries. The device is relaxing and non-disturbing. With DSA a person can speak in any fashion and at any rate. DSA is most effective in the case of "Classical Stutterers" who consist about 80% -> 90% of the stuttering population. Significant improvement or total fluency is observed in about 40% -> 60% of "Classical Stutterers". Rest of them also improve to various degrees. Device was and is still tested in real life, not artificial laboratory situation. Improvements were observed in all situations : in the office, at home, on the telephone, during public meetings, presentations, good and bad days, etc. Improvement is instant, however, we observe increase in effectiveness during the first 2 -> 6 weeks. After that it seems to remain the same. In majority of cases there is a carryover effect - person remains more fluent for 2 hours -> 2 weeks after using DSA. Significant improvement in self-esteem and self-confidence are observed. People like to use DSA and say that it is relaxing. Many people also indicate, that they feel, that they cannot stutter. Long term effects seem to support authors' theory and expectations - DSA is still effective (same level) after being used for 10 months.

First clinical trials - June 1992

First prototypes of DSA (II generation) were finished in May 1992. Some of the first clinical trials were very exciting and the device met with good approval from the stuttering people. Here are some of the remarks from this early study in Poland :

- *"....I think that Digital Speech Aid (DSA)is extremely effective in the elimination of stuttering, even in cases of very severe stuttering. ....During my 30-years long practice as a speech pathologist, no technique was as successful. I believe that we finally have the green light for people who have till now problems with elimination of stuttering ...We are waiting very anxiously for DSA to appear on the market and become available for all stuttering people......" Halina Stawikowska, Speech Pathologist, June 04, 1992.*
- *"....Speaking with DSA brought me big relief. I realized, that I can also speak like normal person, at relatively fast rate.All the sudden I got new surge of power.....This device helped me also in the inter-personal contacts. I am not afraid anymore !....It is fantastic to realize, that spoken word does not have to be "bumpy". Thank*

you , thank you, thank you !!!....", Anna Grudzien, 21 years old, June 05, 1992.

- *"....Speaking with DSA eliminates prolongation of vowels, speech becomes more fluent... Improvement is very good, much better then with the DAF (Delayed Auditory Feedback). In my case it was particularly noticeable in the German language, which I am studying and in which I stutter much more often then in the Polish language. While reading in German, with DSA I did not stutter at all....", Jacek Szot, 24 years old, June 03, 1992.*
- *"... I think, that use of DSA is relaxing.It is difficult to stutter. I could speak fast without stuttering. Normally I speak slow......", Prof. Antoni Stawikowski, Astrophysics, May 31, 1992.*

Long term effects - DSA (III-d generation model)

Long-term effects were and are tested right now.The initial results are very encouraging. It is expected that due to the relaxing and reassuring effect of DSA, the base-line stuttering level will decline (we observed this already in some cases). The effectiveness of the device is expected to remain the same for removal of the "natural level" of stuttering. In a sense DSA is expected to work as a prosthetic device.It does not mean however, that it has to be worn all the time.It can be only used in most difficult situations and as a backup. Below are some remarks from some patients who did use DSA for over period of 3 to 10 months :

- *"...my experience with DSA for past 7 months has been good....In past years talking on the phone was my reason to be nervous to speak to people. With DSA I now find it to be no problem to answer a phone and have a normal conversation. I recommend DSA for anyone with a speech stoppage and stuttering." John Dunphy, Dartmouth, Nova Scotia, Canada, March 25, 1994.*
- *"..Over the course of a few months, I became very comfortable with the speech device (DSA). ...I have less apprehension with "feared words" and stutter quite a bit less than before using the device. My confidence in speaking has improved over the time I used the device and the "carry-over" time has also increased. My fluent speech will last sometimes half a day without the device." 25 years old female stutterer, Halifax, Nova Scotia, Canada, December 19, 1993.*
- *"When I first began to use the DSA I noticed a dramatic increase in my fluency when speaking especially on the phone.....It has been so wonderful to have the freedom of not being afraid to use the phone, to call to make appointments etc.....My life has changed now since I have used DSA.....it is a wonderful machine to use." Lori Paruch, Halifax, Nova Scotia, Canada, April 04, 1994.*

Facts about stuttering and implications for treatment

About 4% of children and 1% of adults stutter. Stuttering changes with age. People stutter to a varying degree and in different ways. Often people stutter on particular sounds. Often rate of the stuttering varies for given individual (depending on various factors). Stuttering usually depends on language used by the person. Males stutter 3 times more often than females. Stuttering starts in the early age and in some cases goes away at later age. In many cases food and alcohol (or other chemicals) change stuttering - better or worse. In many cases exercise and physical activity change stuttering. In many cases stress changes the rate of stuttering - better or worse.

Becoming suddenly deaf leads to total fluency. With shadow speech (whispering) or choral speech (with other person) - majority of cases is fluent (90% ?). When singing or talking in noise (cafeteria, bar, music) - majority of cases is fluent (90% ?). Lowering or increasing pitch of the ones voice, assuming foreign accent and slowing down the rate of speech production also results in increased fluency .

Amplification or attenuation of the voice , delay of the voice in the range 1->100 ms, white or other types of noise, frequency shifting of the voice in the range -1 -> +1 octave, reverberation of the voice, combination of the above increases fluency.

It is obvious that hearing plays very important role in speech production and control. It is clear that stuttering is caused by physiological disorder, neurological in nature in most of the cases. Speed of propagation of neural signals seems to play important role, lower frequencies are more important than higher (facts and experiments), vocal tone is very important (facts and experiments). Stuttering seems to be correctable by the processing of sound (facts and experiments).

How to correct stuttering via Signal Processing ?

Methods of stuttering correction could be divided into three broad categories :

- **Masking-noise,** etc. make signal unusable for control in the Speech Control System (SCS). SCS relies on other afferent channels in this case (Fig.2).
- **Non-Masking** - DAF, FAF, Reverberation seems to be better, since signal is not as disturbing and it is comprehendible as voice by the higher levels of Speech Synthesis System (SSS), therefore helping in this synthesis (Fig.2). However SCS is probably not using this signal for the control (servo) purposes.
- **Correction** - signal is shaped via DSdP (Digital Sound Processing) in such a way as to correct for deficiencies and at the same time make it still acceptable by SCS for the control (servo) purposes. This is the preferred way of correcting, since it will be more effective and pleasant to use by the stutterer. Authors' hypothesis is that it is possible to use this type of systems in certain cases of stuttering. Further tests and experiments are required.

Hypothesis about SCS :

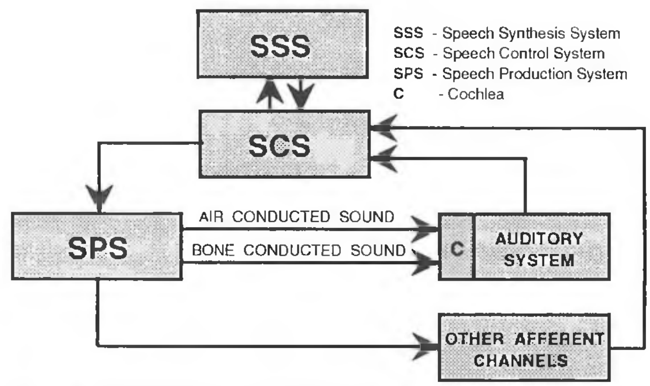


Fig. 2 Block diagram of the speech production system.

In stutterers the auditory signal is used by SCS, but from time to time the voice signal is not being accepted leading to prolongations and other observed stuttering effects (Fig.2). By manipulating signal via DSdP, one can obtain auditory feedback which will be on one hand acceptable by SCS for control (servo) purposes and on the other hand will lead to correction of speech production. This should in turn lead to fluency. Hopefully this could be done for all sounds produced by the stutterer. Also one would hope, that this correction will be working over the whole range of variability (stress, alcohol, etc).

Introduction of some kind of processing - ear plugs, amplification, equalization, filtering, DAF, FAF, DSA, noise, etc. changes auditory feedback. SCS is comparing the remembered signal (by the already formed and "fixed" neural network) with the produced signal and is correcting its shape. If SCS cannot do this, it will get stuck (stuttering effect) - system is trying to overcome a problem and is trying to do this over and over again. Since most of the SCS work is under a subconscious control, the stutterer has very little control over it.

Conclusions

Authors believe, that we are very close to explaining the stuttering disorder. Our hope is that scientists from different fields will join forces together in order to advance our knowledge of this disorder and its treatment. Without this approach this progress will be as slow as in the last several decades.

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Infant Dependence on Acoustic Cue Redundancy: Discrimination of the Word-Final Voicing Contrast, /t/-/d/.

Margret A. Orme and Linda Polka
School of Communication Sciences and Disorders
McGill University
1266 Pine Avenue West
Montreal, Québec
Canada H3G 1A8

INTRODUCTION

Several studies have investigated the perception of vowel duration as a cue to word-final stop voicing in children and adults. The findings have indicated that the ability to rely on a single cue in making such discriminations develops with age (Greenlee, 1980; Krause, 1982). Similarly, an increased reliance on redundant acoustic information has been observed for adult second-language learners of English (Crowther & Mann, 1992; Flege & Wang, 1989), which also suggests that acoustic redundancy is more important during language acquisition. Several studies that have examined discrimination of this contrast in infants and have suggested that, like adults, infants rely primarily on vowel duration differences in discrimination of final stop voicing contrasts (Eilers, 1977; Eilers et al., 1977). However, the role of acoustic cue redundancy and the prominence of the vowel duration cue over other cues in infant perception of final stop voicing distinctions was not been clearly established in this work. Therefore, it is difficult to relate these infant results to the findings from the adult and second language learners.

The present study was designed to investigate further perception of word-final stop voicing contrasts in infants and adults in order to test three hypotheses about infant phonetic discrimination skills. The first hypothesis was that, like second language learners, infants will show a greater reliance on acoustic cue redundancy than will native English-speaking adults. Thus, removing one or more of the multiple acoustic cues that distinguish this contrast would be expected to produce a drop in discrimination performance. The second hypothesis was that vowel duration would be the most salient cue for infants, as has been repeatedly demonstrated in native English-speaking adults and has been suggested by Eilers et al (1977). Accordingly, infants were expected to show their best discrimination performance when the vowel duration cue is present. Finally, previous cross-language studies have indicated that infants are becoming more sensitive to the phonetic structure of the native language in the latter half of the first year of life (Werker & Pegg, 1992). Given that the infants in this study were being tested on a native language contrast, it was hypothesized that older infants (10-12 month olds) may show better discrimination performance than younger infants (6-8 month olds).

METHOD AND DESIGN

To address these hypotheses, discrimination of a word-final stop consonant voicing contrast, /bid/-/bit/, was examined in English-learning infants at two ages, 6-8 months and 10-12 months, and in English-speaking adults. A set of natural speech tokens were edited to create three stimulus conditions which differ in the acoustic cues that signal the /t/ - /d/ contrast in word final position. The three conditions were: 1) the full cue condition (FC), in which vowel duration, burst cues and closure cues are present, 2) burst and closure cues neutralized condition (BCCN), and 3) vowel duration neutralized condition (VDN). Each subject was tested on discrimination of the /bit/ vs /bid/ contrast in each of the three conditions using the headturn procedure in the category change paradigm (Kuhl, 1987). In this procedure, the infant listens to a background stream of syllables that correspond to one category. At irregular intervals, the syllables change to another target category for a short period. All of the subjects were tested with /bid/ as the background category and /bit/ as the target category. The infant learns to respond to this change with a headturn; a visual reinforcer is used to condition a correct anticipatory headturn. Adults are simply instructed to raise their hand when the sound changes. The testing for each condition consisted of a short conditioning phase followed by a test phase. In the test phase, the subject was presented 25 test trials; approximately half are change trials and half are control (no change) trials.

RESULTS

Performance was evaluated in each condition with respect to three dependant variables: 1) percent correct (across the 25 test trials), 2) proportion of subjects in each group who met a pre-established discrimination criterion of 7 out 8 consecutive correct responses[1], and 3) number of test trials required to reach criterion (among those who subjects who met the discrimination criterion). Mean percent correct for each stimulus condition is plotted for each age group in Figure 1 below.

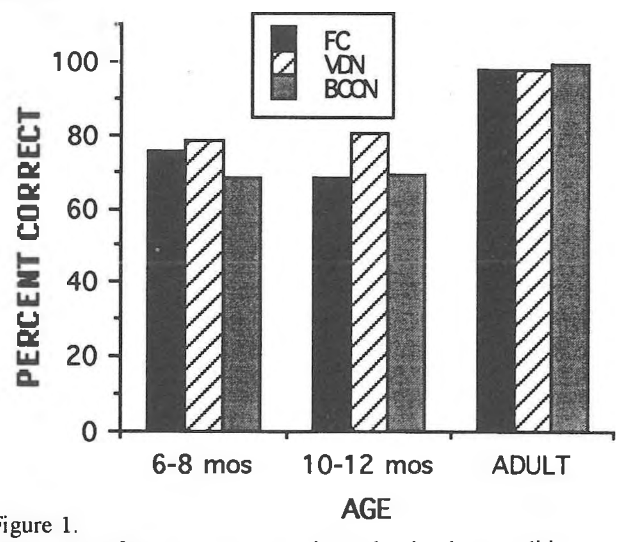

Figure 1.
Percentage of correct responses in each stimulus condition.

[1]The use of a preset criterion is common with the headturn procedure. This criterion corresponds to a significance level of $p < .05$.

Adults performed at ceiling levels for all three conditions with respect to each dependant variable. No infant age differences were observed with respect to any of the dependant variables.

To assess the effect of cue redundancy, performance in the full cue (FC) condition was compared to accuracy in each reduced cue condition (BCCN and VDN). Performance on the FC condition was greater than performance on the BCCN condition with respect to both the proportion of infants reaching criterion and the number of trials required to reach criterion performance level, though not for percent correct. However, performance on the FC condition was never better than performance on the VDN condition. When the vowel duration difference was removed (in VDN condition), performance was either maintained (re: proportion reaching criterion) or improved (re: percent correct scores) relative to the FC condition.

The relative saliency of the various kinds of acoustic cues that distinguish the /bid/ - /bit/ contrast was also examined by comparing performance in the two reduced cue conditions (VDN and BCCN). These analyses revealed that infants consistently performed significantly better in the VDN condition than in the BCCN condition. This pattern was evident in the analyses of all three dependant variables.

DISCUSSSION

Three hypotheses were addressed in this study. It was hypothesized that infant discrimination performance would show some improvement in the later half of the first year of life. The present findings failed to support this hypothesis. Although age-related changes in infant speech perception have been noted in previous work, similar age differences were not apparent in discrimination of final stop voicing in English-learning infants.

Another hypothesis which motivated this study was that, in perception of phonetic contrasts, infants would show greater dependency on acoustic cue redundancy than would adults. The present findings provide partial support for this hypothesis in that infants showed better discrimination performance in the FC condition than in the BCCN condition. However, infant discrimination in the full cue condition was less accurate compared to the VDN condition in which the vowel duration differences were absent.

The third hypothesis was that vowel duration would be the most salient perceptual cue for infants, consistent with previous findings with adults. Surprisingly, the findings failed to support this hypothesis and, instead indicated that burst and closure cues were more salient for the infants than were the vowel duration differences. This pattern of results differs from findings of prior research on adult perception of such contrasts which showed a prominent use of the preceding vowel duration cue by adults. Also, at least one infant study has suggested that infants of 6-12 months also make use of the vowel duration difference cue (Eilers et al., 1977). However, the present findings are consistent with findings from studies with older children and with second language learners in showing that a certain level of linguistic sophistication is necessary before vowel duration difference information becomes useful as a cue to final stop consonant voicing.

In conclusion, the finding that infants are better able to use burst and closure cues in their discrimination of final stop voicing, differs from the findings from previous adult studies which clearly show vowel duration to be a prominent perceptual cue. Thus, infants and adults appear to favor different acoustic cues in discriminating word final stop voicing contrasts. Consequently, considerable caution should be exercised in using research findings with adults as a basis for deriving infant aural rehabilitation regimes. Furthermore, infant responses and acoustic needs are likely to chnage through out language development. Therefore, further research is needed to describe the normal course of speech perception development, both to better understand normal language development and to facilitate determination of the specific clinical intervention needs of infants and children with communication delays and disorders.

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Voice Pitch as an Aid to Speechreading in Young Children

J.D.Fagg
School of Communication Sciences and Disorders
McGill University, Montreal, Quebec

Introduction

For the majority of the profoundly hearing impaired population, speechreading, or 'lipreading', is essential for the perception of speech in everyday communication. None of the aids to communication available to the profoundly deaf - whether conventional hearing aids, vibrotactile aids, visual aids or electrocochlear devices - are, as yet, able to provide good, consistent speech discrimination in the absence of speechreading.

However, for a number of reasons, the perception of speech through speechreading alone, even in ideal conditions, is extremely problematic [1,2,3]. The visual cues available in speechreading mainly give information about the place of articulation of a consonant. Even leaving aside the obvious problems caused by articulatory movements which are not visible to the speechreader, confusions arise because visible configurations of the lips, tongue and teeth are almost never unique to one particular phoneme. Manner and voicing information, required to allow consonants sharing the same place of articulation to be distinguished from one another, is not readily available through visual perception of the speaker's facial features. One would therefore assume that provision of this information to the speechreader should greatly improve his/her ability to directly identify consonants in everyday speech.

A number of recent studies have concentrated on trying to determine what sort of acoustic information could afford good speech discrimination to the profoundly deaf subject when presented alongside the visual information available through speechreading [4,5]. Because of the nature of profound, sensori-neural hearing losses it is generally neither possible nor desirable merely to present hearing impaired subjects with the whole amplified speech signal. These subjects will generally have little or no perception of higher frequency components of the speech signal, will have drastically reduced dynamic range and may have impaired perception of frequency, amplitude or temporal changes in the incoming auditory signal [5].

There are several reasons for considering fundamental frequency or voice pitch information as a potential aid to speechreading in the profoundly deaf [6]. It is a major and invisible cue to consonant identification. As well as providing segmental information through consonantal voicing contrasts, variations in fundamental frequency can provide a great deal of information about suprasegmental, prosodic aspects of the speech signal. There is also evidence that most profoundly hearing impaired people are able to perceive the temporal patterns which relate to changes in fundamental frequency, while many appear unable to detect more complex speech perceptual cues [4,7]. It has been suggested that the extraction and presentation of simple cues, matched to the perceptual abilities of the subject, may be a more effective way of aiding speechreading performance than presenting a "whole speech" signal in which cues such as fundamental frequency may be less explicit.

Studies using hearing subjects can allow experimenters to control the quality and amount of auditory information provided as a supplement to visual speech cues. Experiments with hearing adults have shown that presentation of voice pitch information along with speechreading can result in considerable improvements in the speed at which speech is perceived in a Connected Discourse Tracking task [8]. The primary aim of the present study was to investigate the effect of voice pitch information on the speechreading performance of young hearing children.

Investigations of speechreading skill in both adults and children have often produced confusing and conflicting results, usually as a result of the use of many different methodologies. Previous studies have differed widely in terms of the subjects tested, the speechreading materials used and the manner in which those materials were presented [9]. As yet there are no definitive answers on how speechreading skills should best be defined and tested. The methodology applied in the present study was largely dictated by the age of the subjects taking part.

Method

Subjects

22 children with normal hearing took part in the study. The age range of the subjects was 5.9 - 6.75 years (Mean age = 6.3 years, S.D. = 0.3 years). No subject had any history of hearing impairment, visual impairment or severe learning difficulty.

Test Materials and Procedure

The speechreading test was made up of 60 items. For each test item subjects were shown three black and white drawings, these drawings corresponding to the 'target' word, a visually similar 'distractor' word and a 'random' word (eg. 'book', 'bike' and 'fish'). Both the vocabulary and drawings used in this experiment were taken from the Manchester Picture Test (1984) [10], which has been validated with children of 5 years old and over. All of the words included in the test were monosyllabic.

A videotaped presentation of the target stimuli was used. An adult female speaker, whose head and shoulders only were visible on the television screen, presented each target word, preceded by the carrier phrase "point to the.....". Subjects were required to respond by pointing to the picture corresponding to the word spoken. The test was devised such that, for half of the test items, discrimination of the target word from the distractor would be expected to be facilitated by the ability to perceive voicing contrasts (eg. 'dog', 'duck' and 'ball').

Each subject was tested individually and performed the speechreading task twice, once in the silent condition and once with voice pitch information present. To control for order effects the subjects were randomly allocated to two groups, one group performing the silent condition first, the other performing the voice pitch condition first. The voice pitch information was extracted from the speech signal during the recording of the test stimuli using an electro-laryngograph. This auditory signal was presented simultaneously with the visual speechreading stimuli in the voice pitch condition.

Results

(a) Speechreading Ability Related t-tests showed that the difference between the number of words identified correctly and the number of errors made was significant both for the silent condition ($t = 5.4$, $df = 21$, $p < 0.001$) and for the voice pitch condition ($t = 10.8$, $df = 21$, $p < 0.001$) with subjects making more correct responses than errors for both conditions. When the errors themselves were analyzed, related t-tests showed that the difference between the number of distractor items chosen and the number of random items chosen was also significant for both the silent condition ($t = 11.42$, $df = 21$, $p < 0.001$) and the voice pitch condition ($t = 15.98$, $df = 21$, $p < 0.001$) with more distractor items being chosen than random items.

(b) The Effect of Voice Pitch Information A MANOVA showed that there was no significant effect on test scores of the order in which subjects performed the silent and voice pitch conditions. There was also no significant difference between the total correct scores obtained in the silent condition and those obtained in the voice pitch condition ($F = 2.83$). However, when scores with the two different types of test item ('voicing contrast' and 'no voicing contrast') were analyzed separately there was found to be a significant effect of condition ($F = 4.29$, $p < 0.01$). Related t-tests showed that there was no significant difference between scores obtained in the silent condition and those obtained in the voice pitch condition for items which did not involve a voicing contrast ($t = 0.78$, $df = 21$). However, a significant difference was found between scores obtained in the silent and voice pitch conditions for those items which did involve a voicing contrast ($t = 3.93$, $df = 21$, $p < 0.001$), with speechreading scores being higher when voice pitch information was provided.

Discussion

Hearing children of 5 - 6 years of age, with no known speechreading experience, have been shown to be able to identify familiar words using the visual speech signal only. It therefore seems probable that visual information has a role to play in everyday speech perception, even for those for whom the auditory modality alone can generally provide sufficient information for accurate speech discrimination. These young children have also been shown to be able to make use of voice pitch information to assist them in a speechreading task where the perception of voicing contrasts was required. However, it is not clear from these results that the benefits of this limited auditory signal would be enough to produce a significant improvement over speechreading alone in the perception of everyday conversational speech.

A number of studies have suggested that both visual and auditory aspects of phonology may be integrated in a common phonological store [11,12]. The apparent ease with which the young children in the present experiment were able to use visual information to access phonological knowledge certainly lends support to these theories. The results of this study also imply that the normal speech perception process involves the use of specific speech features as cues to the identity of speech sounds and that these speech features can be used in isolation from the whole speech signal.

It is not clear how beneficial the provision of simple speech pattern signals, such as the voice pitch signal, might be to a young, profoundly hearing impaired child. The enormous differences between the hearing subjects used in this study and the majority of profoundly hearing impaired children of a similar age, in terms of their language knowledge and experience of auditory speech perception, would make generalizations from one group to the other extremely problematic. It is the belief of some researchers that a hearing impaired child should be given as much auditory input as possible if he/she is to acquire language in a relatively natural way and develop auditory neural systems as fully as possible [13,14]. Others suggest that profoundly hearing impaired children may need specific auditory training in the use of a particular speech feature, such as voice pitch, if they are to benefit fully, or at all, from the provision of this kind of auditory signal [15].

However, we can surmise that any auditory information provided to a hearing impaired child must surely increase that child's chances of acquiring knowledge of spoken language, and we can speculate that for some profoundly hearing impaired children the provision of simple voice pitch information may be better matched to their perceptual capabilities than a more complex whole speech signal.

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BUILDING VOICE LINEUPS: ONE METHOD'S BIAS PROBLEMS

Ann Stuart Laubstein
Department Linguistics, Carleton University
Ottawa, K1S 5B6

1. Introduction

It is readily apparent that accurate voice recognition is a common phenomenon and misidentification, although less common, also takes place. How this occurs - which features of the voice are instrumental in this recognition or misidentification - remains an empirical question.

Nonetheless, despite this lack of knowledge, voice identification is being used as evidence in suspect identification by the legal systems of both the United States and Canada. There is no well established method for getting witnesses to identify suspects' voices but the most common method involves the presentation of a voice lineup. This poses further problems. The major problem posed by the so-called voice or earwitness lineup is the fact that a voice lineup is not simply a voice lineup but a speech lineup. And there is much more content to speech than the simple sound of the voice. The critical criterion for a good lineup is fairness. There must be nothing that distinguishes the suspect from the foils - something that is remarkably difficult to ensure in a voice lineup.

This paper describes an experimental study which investigates the feasibility of using what shall be termed the "transcript method" in the preparation of voice lineups - the currently accepted procedure in Canada and the United States (Mayor, 1989). This method has been used in an attempt to eliminate the speech content from the voice lineup. It was for instance, the method chosen by the Ottawa Police in a recent investigation. There are a number of problems with this method - in particular, it is not clear that it is capable of producing an unbiased lineup.

2.1 The Method

The procedure for preparing voice lineups used by the Ottawa Police is typical and will serve as the basis for the description which follows. First the suspect is interviewed by the police and this interview taped. Subsequently the interviewer's questions are spliced out, leaving only the suspect's speech on the tape. A written transcript is then made of the resulting monologue. Actors are chosen and they are asked to read the transcript; in the normal case they do not hear the tape of the suspect, however in the Ottawa case they did. The transcript and, in this case the tape, were used to prepare the foil samples. Because the suspect's tone was extremely aggressive and he spoke very fast it was felt that the actors should be afforded the opportunity of hearing the tape once, just to "get a feel" for the suspect's manner of speech. The actors were then asked to read the transcript aloud a number of times and their "best effort" was chosen.

2.2 Reasons for Using the Transcript Method:

The transcript method is used in an attempt to eliminate the variability found in a speech sample which is due to factors other than actual voice characteristics. Besides the more obvious information regarding gender and age, speech may point to diverse factors such as educational level, socio-economic background, ethnic heritage, occupation, health, and regional background. In addition, speech has different kinds of informational content - so-called "sentence meaning" and "speaker meaning", as well as affective content.

Controlling for all of these features, all of which may potentially threaten the lack of bias in a lineup, is fraught with difficulties. The solution was the transcript method.

2.3 Problems with the Transcript Method

As pointed out above relatively little is known about the identification of people on the basis of their voices. A set of characteristics has been identified which will lead to quite reliable machine identification (Hollien, 1990) but it remains an empirical question as to how this set relates to the parameters chosen by the brain.

Secondly there are all the problems associated with the physical characteristics of the machines used and the varying acoustical properties of the rooms in which the taping takes place. In addition, splicing out the interviewer's voice can have quite enormous effects on the recording.

These problems can be dealt with but it is not clear that the biggest problem - the inherent difference between the reading and speaking modalities is capable of solution. The experiments described below investigated two voice lineups prepared by the Ottawa Police in a recent criminal investigation. The investigation was aimed at determining whether the lineups were unbiased and hence whether their use by the police in suspect identification would be legitimate.

3.1 Experiment 1: Materials

A voice lineup was prepared using the transcript method detailed above. In addition to the suspect's voice there were seven foils' voices - actors from an Ottawa theatre company. The suspect's "monologue" lasted just under a minute and the actors' versions were all within five seconds in length of the suspect's tape. The versions judged "most natural" and the suspect's voice were put together on a single tape in quasi-random order with a two minute pause following each voice.

A questionnaire of sixteen questions was prepared relating to a number of sound/speech characteristics and possible judgements based on voice quality such as loudness, pitch, propensity to violence, suspect versus foil judgement).

3.2 Procedure

Subjects were told that they would hear a tape of eight voices one of which was the voice of a suspect in an ongoing police investigation. They were told that their participation would be of use to the justice system in that they would be helping to judge the fairness of a voice lineup. They were asked to evaluate each of the voices on a scale of 1 - 5 on each of the 16 questions asked. They were told that they would have two minutes following each voice to answer the questions but if they wished they could respond to any questions during the time that they were actually listening to a voice. Each subject was given a small booklet with the instructions on the front page and each of the subsequent pages labelled SPEAKER 1 (2,3,...8). The two runs of the experiment took place in two different classrooms at the university. Subjects listened to the tape on a ghetto blaster. Each run of the experiment lasted less than 25 minutes.

3.3 Subjects

Subjects (n=72) participated voluntarily for bonus points towards their final exam marks. They were from two introductory linguistics classes, n= 41 (Group A), and n=31 (Group B).

3.4 Results

Means were calculated for each voice with respect to each of the 16 questions. T-tests were then run comparing the suspect's mean to the foil mean closest to him on each of the 16 parameters. The suspect differed significantly (alpha $\leq .05$ (Bonferoni $\leq .003$)) from all the foils in both groups on only one of the questions, q8. In Group B the suspect also differed significantly on questions 1 and 16. In group A these two questions approach significance ($p = .028$ and $.018$).

3.5 Conclusion

Figure 1 indicates the two groups are virtually indistinguishable from one another and questions 1 and 16 have virtually the same means. The lineup appears biased regarding #1, #8 and #16.

q1 How difficult was it to understand the speaker?
Easy 1 2 3 4 5 Difficult
q8 Did you notice any sounds on the tape besides the speaker's voice?
None 1 2 3 4 5 Many
q16 How sure are you that this speaker is the real suspect and not a foil?
Foil 1 2 3 4 5 Suspect

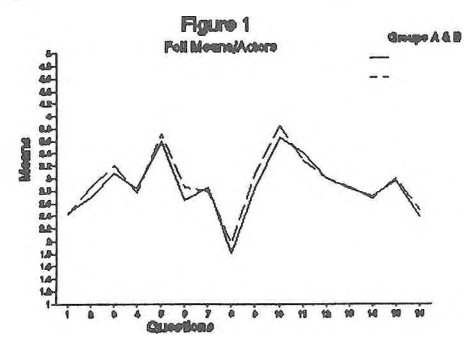
To overcome the biased nature of this lineup a new voice lineup was put together and a second experiment was run to determine whether it was biased.

4.1 Experiment 2: Materials, Subjects, Procedure

The tape was prepared in the same way as the tape for the first lineup except that the foils were Ottawa policemen and there were only six, not seven. The suspect was the 5th voice on the tape. The door and chair noises were included in all the foil samples. Subjects were again from two introductory linguistics classes (Group C n= 43, Group D n=47,) participating for bonus marks. The procedure was as in Experiment 1.

4.2 Results

The suspect's mean was significantly different from the closest foil on only two questions, #1 and #16 in both groups.



4.3 Conclusion

The lineup would again seem to be biased. It may be that this method is incapable of producing an unbiased lineup.

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Effects of Noise on Identification of Topic Changes in Discourse

G.A. Tidball, M.K. Pichora-Fuller, J.H.V. Gilbert, N.Lamb
School of Audiology and Speech Sciences
University of British Columbia
Vancouver, British Columbia, Canada

RATIONALE

Hard-of-hearing individuals frequently report that they are able to understand speech when the topic is known but that they experience difficulty when the topic changes. The motivation for the present study was to investigate how topic identification might influence listening comprehension in unfavourable listening conditions.

Listeners rely on contextual information in speech to aid comprehension. The richer the context, the less a listener must rely on the perception of the acoustic features of speech to comprehend what is heard.

An example of contextual knowledge which aids comprehension is knowledge of topic, or, simply stated, "what is being talked about." Previous studies have shown that comprehension and retention of language are aided by individuals' prior knowledge of topic (Bransford & Johnson, 1973; Larch & Larch, 1985). Knowledge of topic provides constraints on the language content of discourse. It may also help individuals to organize discourse content according to some pre-existing concept of language structure such as a hierarchical structure of topic and sub-topics.

While topic may be defined with respect to content, it may also be structurally defined. That is, in addition to knowing what the topic is, listeners are also aware of when a new topic is beginning or an old topic is continuing or ending. The boundaries of topics in discourse are indicated lexically, syntactically, and perhaps most importantly prosodically (e.g., variations in voice pitch, loudness, pausing, etc.). It follows, then, that in order for listeners to be able to identify a topic of discourse, they must be able to identify the boundaries of discourse topic, or when "what is being talked about" has changed. Inability to correctly perceive those structural cues which signal topic transitions would deprive the listener of important contextual information from which he/she could predict the content of the discourse.

The purpose of the present study was first, to determine how accurately normal-hearing individuals identify where topic changes occur in discourse under favourable listening conditions; and second, to determine how normal-hearing individuals' ability to identify topic changes in discourse is affected by competing noise.

METHODS

The stimulus materials used in the study were recordings of monologues spoken by a young, native English-speaking adult male. Monologues were employed to elicit natural discourse in which there was no turn-taking which might cue subjects to a change in topic.

The talker was asked to describe seventy-two photos chosen by both the experimenter and the talker from a large selection of the talker's family photos. Each picture was of a different event. The talker was instructed to describe the pictures so that future listeners might be able to imagine for themselves what each picture looked like. He was also asked to describe each photograph individually, without referring to any information mentioned in the descriptions of previous photos. Photographs were viewed one at a time. For the purpose of this study, each photograph was considered to represent a topic.

The talker was located in a sound-attenuating booth during the elicitation of the stimulus materials. The monologues were recorded digitally using SoundWorks (v.3) on a NeXT computer system.

Twelve normal-hearing listeners participated in the perceptual study. All were native Canadian English speakers between the ages of 21 and 35 years.

Prior to the test session, subjects were familiarized with the talker's voice by listening to a recording of him speaking on a topic unrelated to the test materials. Subjects then listened to the stimulus monologues in three signal-to-noise (S:N) conditions: +5 dB S:N (favourable listening condition), 0 dB S:N (less favourable listening condition), and a -5 dB S:N (unfavourable listening condition). Each monologue was comprised of eleven photograph descriptions, yielding a total of ten topic changes per condition. Half the subjects listened to the unfavourable listening condition (-5 dB S:N) first, half listened to the most favourable listening condition (+5 dB S:N) first. The monologues were presented monaurally to the subject's better ear.

The subjects' task was to indicate when they believed the talker in the pre-recorded monologue was about to begin talking about a new photograph. Subjects' indicated their responses by pressing a button placed on a table in front of them. This generated a response signal. The response signal together with the stimulus materials were recorded simultaneously onto an analogue tape cassette.

ANALYSIS

The analogue recordings of the stimulus materials and subjects' responses were re-recorded onto the NeXT sound processing system for analysis. We measured the latency of the subjects' responses in relation to the conclusion of one topic or photo description and the beginning of the following topic.

Three primary measures were taken:

1. the median latency of each subject's responses in each condition;
2. the number of false positive identifications of topic changes;
3. the number of times subjects waited until the following photo description or topic had begun before they indicated that they recognized a topic change.

An increase in one or all of these measures could indicate increased difficulty in subjects' ability to perform the experimental task. We measured the variation of subjects' ability to perform the experimental task as a function of increased competing noise.

RESULTS

Significant differences in subjects' median latency of response were found between the +5 dB and 0 dB S:N conditions ($p < .05$) and between the +5 dB and -5 dB S:N conditions ($p < .01$). No significant difference in median latency was found between the 0 dB and -5 dB S:N conditions ($p > .05$).

A significantly greater number of false positive responses were given in the -5 dB S:N condition than were given in either the 0 dB or the +5 dB S:N conditions ($p < .0001$).

The number of times subjects waited until the following photo description or topic had begun before they indicated that they recognized a topic change was also found to vary as a function of signal-to-noise condition. A significantly greater number of this type of response was given in the -5 dB S:N condition than were given in either the 0 dB and +5 dB S:N conditions ($p < .01$).

DISCUSSION

The above results show that in a favourable listening condition (+5 dB S:N), listeners are quite accurate in anticipating topic changes in discourse with some degree of accuracy. As the listening condition becomes increasingly unfavourable, listeners (1) require more time to identify the boundaries of topics, (2) rely more heavily on those cues which indicate initiation than those which indicate termination of a topic, (3) make more errors in their estimates of where topic boundaries occur.

We offer the following explanations as to why these results were obtained: that reduction of segmental and suprasegmental information in the unfavourable listening conditions reduces listeners' ability to perceive those cues necessary to identify topic changes in discourse; that decreased cues to topic boundaries results in increased processing time - the listener must listen longer to obtain enough information to identify topic boundaries; and that increased competing noise may mask parts of the speech signal to such a degree that the listener no longer detects the signal, with an apparent pause resulting in "miscues".

These results suggest that listeners have difficulty identifying the cues necessary to identify topic boundaries in discourse when the speech signal is significantly degraded by competing noise. Difficulty identifying topic boundaries decreases listeners' ability to organize discourse into topics, and thereby may reduce contextual information in speech.

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Figure 1 Mean of the Median Latency of Subjects' Responses

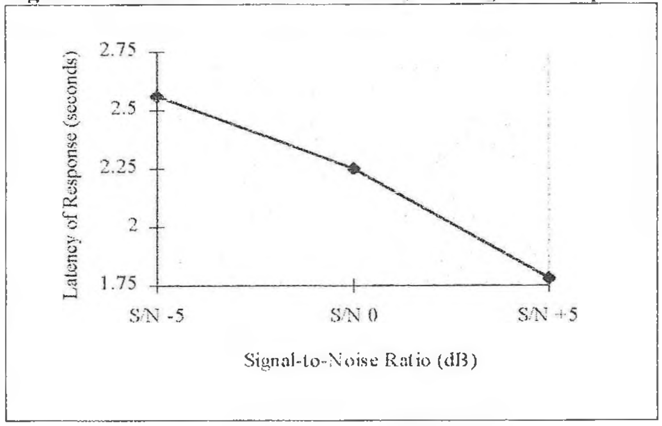


Figure 2 Mean Number of False Positive Responses

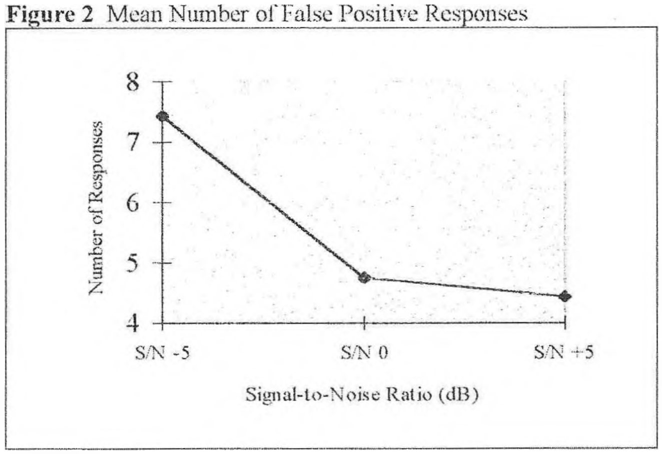
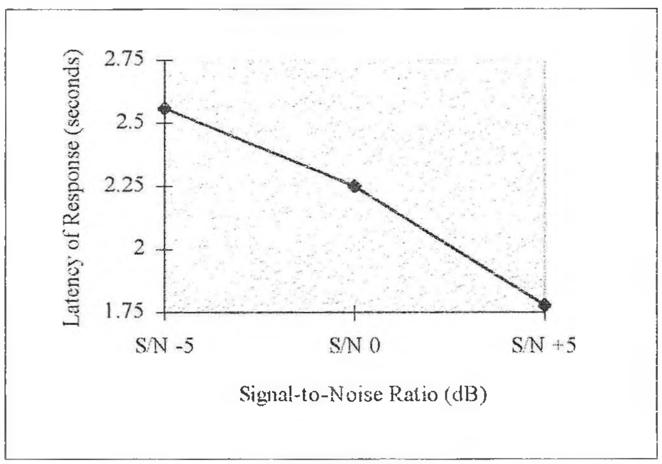


Figure 3 Mean Number of Times Subjects Waited Until Following Topic Began Before Responding



IS THE VOICED-VOICELESS PHONEMIC BOUNDARY INFLUENCED BY AN INTENSITY LEVEL OF THE PRESENTATION

Elzbieta B. Slawinski and Jane F. MacNeil
Psychology Department, The University of Calgary
2500 University Drive NW Calgary, Alberta T2N 1N4

1. Introduction

Multiple acoustic parameters contribute to the perceptual distinction of many phonetic contrasts. The importance of timing of voice onset relative to the plosive release and as well of other acoustic cues to the voiced/voiceless distinction for stop consonants has been studied in a number of phonetic environments. Many of voiced/voiceless phonetic distinction correlates for stop consonants in initial position have been examined using synthetic speech, including voice onset time (Abramson & Lisker, 1970), the presence of a voiced transition at the onset of voicing (Liberman, Delattre, & Cooper, 1958; Stevens & Klatt, 1974), a pitch change prior to voicing onset (Haggard, Ambler, & Collow, 1970), and the frequency of the first formant of voicing onset (Summerfield & Haggard, 1974; Kluender, 1991; Kluender & Lotto, 1993). All of these studies indicate that the duration of Voice Onset Time (VOT) is a dominant and decisive phonetic correlate of the phonemic (voiced) contrast for stop consonants in a word-initial position. Thus, the difference in the duration of VOT in naturally produced plosive consonants in the initial position of words serves to distinguish voiced and voiceless tokens spoken by native talkers of eleven languages (Lisker & Abramson,1964).

However, it was also found that beside the VOT other acoustical cues contribute to the phonemic boundary expressed by the VOT. For example, a longer VOT is required to judge a phoneme as voiceless when first formant (F1) transition is longer or FI frequency onset is lower (Summerfield & Haggard, 1977). Moreover, an analysis of speech production data demonstrated that voiced and voiceless stop-consonants vary also in the peak intensity and in the duration of the burst of frication noise. The frication noise is of a longer duration and of a higher intensity at the release of a voiceless plosive (Klatt, 1975). Thus, the perceptual categorization of bilabial stop-consonants in word-initial position, while mostly relying on a difference in the VOT, should also depend on an acoustical cue such as loudness of the noise burst. Therefore, the present study examined the influence of the intensity level (Sound Pressure Level) of a stimulus presentation on the voiced/ voiceless phonemic boundary between bilabial stop consonants in the initial position of words.

2. Method

2.1 Stimuli. Ten stimuli differing in duration of VOT and ranging from [ba] to [pa] were synthesized using a parallel/cascade synthesizer KLSYN88a implemented on a Macintosh II computer using a 10 kHz sampling frequency. The duration of 6 ms noise burst was constantly maintained for all stimuli. The first formant (F1) started at 200 Hz and increased in 40 ms to 720 Hz, the second formant (F2) started at 900 Hz and increased in 40 ms to 1240 Hz, the third formant (F3) started at 2000 Hz and achieved a steady state of 2500 Hz in the same time as F1 and F2. The fourth and fifth formants were constantly maintained at 3600 Hz and 4500 Hz, respectively, across the entire stimulus. Each synthetic syllable was 300 ms in duration and during voiced excitation the fundamental frequency was 120 Hz until the last 160 ms when the fundamental frequency fell to 100 Hz. Change from an initial voiced to an initial voiceless stop consonant was achieved both by delaying the onset of energy in F1 relative to higher formants (F2 and F3), and by exciting F2 and F3 with an aspiration noise prior to the onset of a periodic source. Change from an initial voiced to an initial voiceless stop consonant was accomplished by changing VOT value (the timing of periodic source onset relative to burst noise release) from 4 ms to 40 ms in 4 ms steps. Two series of stimuli differing by intensity levels (60 dB SPL and 80 dB SPL) were generated. Thus, these series differed by the intensity (loudness) of the noise burst.

2.2 Subjects. Twelve subjects with normal hearing (<10 dB HL for 250 Hz to 6 kHz range) participated in the experiment. Ten subjects learned English as their first language, and two were bilingual. All subjects except three were phonetically naive and had never participated in a speech perception experiment.

2.3 Procedure. Subjects were tested individually on an identification task in an anechoic room. Two series of stimuli differing by intensity levels (60 dB SPL and 80 dB SPL) were presented via headphones. Subjects were asked to identify syllables [ba] and [pa] and were instructed to press one of two buttons labeled either "ba" and "pa". Subjects were exposed to two randomized series of 100 stimuli (each of 10 stimuli presented 10 times) with a 15 minute break between series. Stimuli presentation and a collection of responses were controlled by a Macintosh II computer.

3. Results and Discussion

The mean percentage of the responses labeled "ba" pooled across subjects for 60 dB and 80 dB listening conditions are plotted in Figure 1.

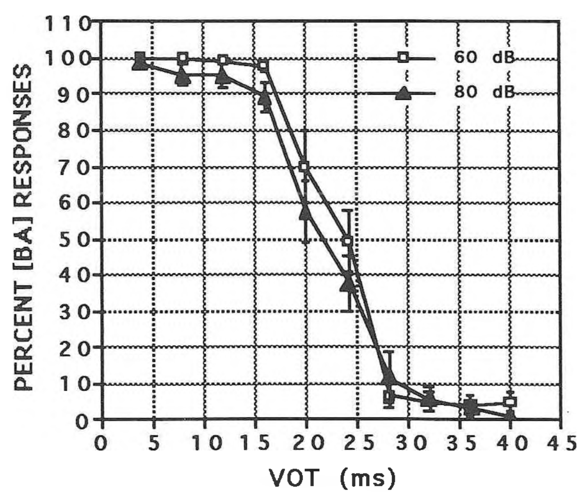


Fig. 1. Identification functions obtained for 60 dB and 80 dB conditions. Standard errors are indicated by errors bars.

In both listening conditions subjects have demonstrated categorical perception. However, it is possible to notice that the phonemic boundary, corresponding to 50 percent of identified [ba], is shifted towards shorter VOT values when stimuli were presented at 80 dB SPL compared to that obtained at 60 dB SPL. Moreover, subjects reported that they heard a more prominent noise burst when stimuli were presented at 80 dB SPL compared to that obtained at 60 dB SPL.

In order to assess the differences in the slopes of the psychometric functions for two listening conditions an interval of uncertainty (difference in VOT corresponding to 75% and 25% of [ba] responses) was measured. The obtained values of 5.3 ms and 7.9 ms for 60 dB and 80 dB respectively were significantly different ($p<.05$) and they suggest that subjects were more uncertain in their judgments when they were exposed to the higher intensity condition (Figure 2).

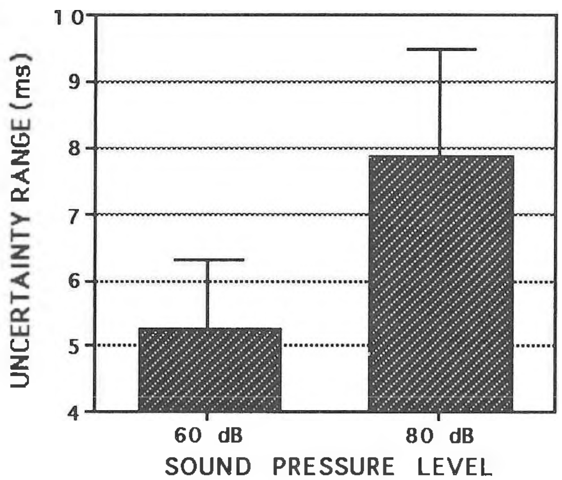


Fig. 2. Mean uncertainty ranges obtained for the 60 dB and 80 dB conditions. Standard errors are indicated by errors bars.

Individual subject phoneme boundaries were calculated using the best fitting logistic function to a continuum of identification probabilities, and subsequently condition-average phoneme boundaries were derived. Figure 3 displays phonemic boundaries obtained for the 60 dB (23.8 ms) and 80 dB (21.4 ms) conditions pooled across subjects.

The difference between these phonemic boundaries of 2.4 ms was found to be statistically significant ($p<.01$). Subjects were more inclined to identify stimuli as [pa] when they were exposed to the noise burst of higher intensity (loudness). Thus, it seems that loudness of the noise burst influences a judgment of the phonemic boundary between [ba] and [pa] categories. Thus, in order to assign a speech sound to the [p] category the shorter VOT was required in the presence of the louder noise burst. This result corresponds to Klatt's findings (1975) that the produced [pa] is characterized by the louder noise burst than that observed in the produced [ba]. Moreover, a study of Kobayashi and Honda (1991) has demonstrated that speakers with an electric larynx in order to make a distinction between voiced and voiceless plosives produce longer duration and higher amplitude of frication noise for the voiceless stops than for the voiced cognates. These researchers have also found that duration and amplitude of the noise burst were highly correlated with the perception of "voicelessness." Therefore, the magnitude of perceived loudness of a noise burst influences the categorization of the [ba-pa] phonemic contrast. Such impact of the noise burst is probably due to the effect of forward masking which in turn depends on the intensity of the noise burst.

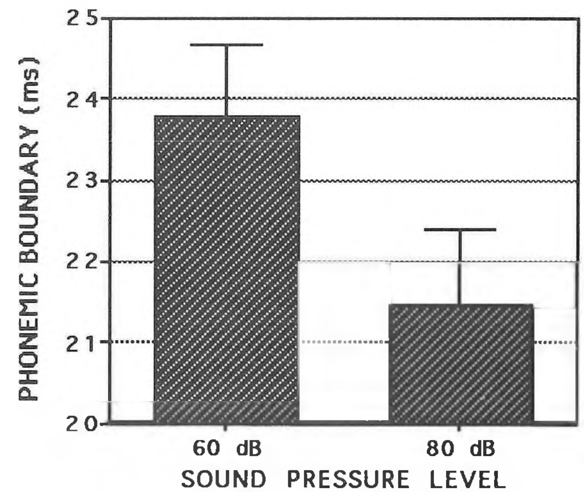


Fig. 3 . Mean phonemic boundaries obtained for the 60 dB and 80 dB conditions. Standard errors are indicated by errors bars.

The difference in the phonemic boundary observed between two intensity conditions may reflect a trading relation between VOT and loudness of a noise burst. Thus, in order to enhance perception of [b] and [p], the perceptual effects of changing one acoustic cues (VOT) may be offset by changing the other cue (loudness of a noise burst) in the opposing direction.

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ACOUSTIC AND PHONOLOGICAL FACTORS IN THE PERCEPTION OF ENGLISH /r/ AND /l/ BY JAPANESE LISTENERS

John S. Logan, Kathy BharrathSingh
Department of Psychology, Carleton University, Ottawa

1. INTRODUCTION

Japanese listeners generally have difficulty perceiving the difference between /r/ and /l/, even after years of exposure to English (MacKain, Best, & Strange, 1981). Such a finding suggests that substantial reorganization of the perceptual system is required in order for Japanese listeners to achieve fluent perception of /r/ and /l/. However, Logan, Lively, and Pisoni (1991) and Lively, Logan, and Pisoni (1993) demonstrated that Japanese listeners can learn to identify /r/ and /l/ more accurately after completion of a laboratory training procedure. Their procedure utilized a large ensemble of stimuli consisting of words produced by five talkers that contained /r/ and /l/ in five phonetic environments. Performance gains were modest but reliable after three weeks of training. Although this work indicated that adult Japanese listeners were capable of learning about /r/ and /l/, additional questions remained concerning how training affected the structure of phonemic categories.

The work of Logan et al. (1991) and others indicated that listeners can learn to redirect their attention to previously unattended portions of the speech signal. What factors affect the allocation of attention? One factor that seems to play an important role when Japanese listeners learn to perceive /r/ and /l/ is the phonetic environment in which /r/ and /l/ are embedded: performance is better when the phonemes are located in word-final positions than in word-initial positions. As a result of acoustic analysis, several researchers have proposed that this effect is due to differences in the duration of /r/ and /l/ in these two contexts. If the phonemes are located in word-final positions, they tend to be longer in duration relative to the same phonemes located in word-initial positions (Dissosway-Huff, Port, & Pisoni, 1982; Sheldon & Strange, 1982). As a consequence, listeners may have more opportunity to extract the critical information denoting /r/ or /l/ in word-final positions than in word-initial positions.

However, it is important to note that duration is not the only factor that may play a role in the ease with which listeners can direct their attention. Henly and Sheldon (1986) found that native speakers of Cantonese differed from Japanese listeners regarding the effect of phonological context on identifying /r/ and /l/. Whereas Japanese listeners had difficulty identifying /r/ and /l/ in initial positions, Cantonese listeners showed difficulty identifying /r/ and /l in final positions, indicating the critical role of the listener's phonological system. The difference between Japanese and Cantonese listeners suggests that one way to study how listeners redirect their attention is to experimentally manipulate the duration of the speech signal that corresponds to /r/ or /l/ and present the stimuli to Japanese and Cantonese listeners. By directly comparing the performance of these two groups, it should be possible to determine the relative contribution of external acoustic cues and internal phonologically-determined attentional mechanisms.

As a first effort towards directly investigating the effect of /r-l/ duration on perception, a group of English and Japanese listeners were presented a series of English words. Stimuli with word-initial /r-l/ had the /r/ or /l/ portion elongated, whereas stimuli with word-final /r-l/ had the /r/ or /l/ portion shortened. If duration of the /r/-/l/ segments is the primary determinant of performance, then the Japanese listeners should identify the elongated stimuli in word-initial position with greater accuracy than normal duration word-initial stimuli. Conversely, they should find the shortened versions of the stimuli containing word-final /r/-/l/ more difficult to identify than the normal versions. If their performance is not positively affected by the duration manipulation, it would suggest that the phonological/attention system is so firmly entrenched that simple efforts to make the acoustic features underlying /r/ and /l/ more or less salient are unlikely to succeed.

2. METHOD

Stimulus preparation and experimental control utilized CSRE software (Jamieson, Ramji, Neary, & Baxter, 1990). A male speaker of Canadian English (CE) recorded tokens of the following /r-l/ minimal pairs: late-rate, lead-read, lock-rock, loot-root, fear-fill, hair-hell, more-mole, and tall-tar. Tokens were low-pass filtered at 4.8 kHz and then digitized at 10 kHz using a 12-bit A/D converter. Tokens were then modified in one of two ways. Tokens containing word-initial /r-l/ had the duration of the /r/ or /l/ portion of the signal elongated, whereas tokens containing word-final /r-l/ had the duration of the /r/ or /l/ portion of the signal shortened. In the case of tokens containing word-initial /r-l/, the longest steady-state portion within the /r-l/ portion of each token was determined using a waveform editor. This section was iteratively reduplicated to produce two versions of each token in addition to the original, one that was 40 ms longer, the other that was 80 ms longer. In the case of words containing word-final /r-l/, 20 ms and 40 ms sections of the steady-state portion within the r-l portion of each token were removed. Altogether, 48 unique stimuli were produced using this procedure.

The stimuli were first tested with a group of six native speakers of CE, all of whom reported no history of a speech of hearing disorder. The pretest consisted of a two-alternative forced choice (2AFC) identification task in which each stimulus was presented once. Stimuli were presented over a Radio Shack loudspeaker at approximately 75 dB SPL. Listeners were seated in front of a computer monitor that displayed two words from a minimal pair. 250 ms later the auditory stimulus was then presented. Subjects were then required to move a mouse to indicate which word from the minimal pair had been presented. As well, each CE listener rated the quality of the 48 tokens using a 7-point scale, where 1 was "very clear", 4 "clear", and 7 "distorted".

Two Japanese subjects were recruited from the Carleton University community. One was a graduate student, the other an undergraduate. Both participants were first exposed to English in a formal setting at age 13 when they began English classes in school. Each had lived in Canada for approximately five years at the time they were tested. Both reported that although English was their primary language for educational purposes, they used Japanese in their non-student activities. Both listeners reported no history of a speech or hearing disorder.

The procedure used to test Japanese subjects was essentially the same used to test the CE listeners. The only changes were the use of headphones (Sony MDR-P1) to present stimuli, an increased number of stimulus repetitions (8 x 48), and no requirement to rate the stimuli. In addition, the Japanese listeners were asked to complete a language experience questionnaire.

Table 1 – Mean quality ratings of stimuli by CE listeners as a function of stimulus modification (original, moderate, or extreme), position (initial and final), and identity of r-l segment (/r/ versus /l/).

	Original	Moderate	Extreme	Mean
/r/-Final	2.3	2.1	2.5	2.3
/r/-Initial	1.7	2.3	3.3	2.4
/l/-Final	2.0	2.2	2.0	2.1
/l/-Initial	1.9	2.1	2.5	2.2
Mean	2.0	2.2	2.6	2.3

Table 2 – Mean identification performance for Japanese listeners as a function of stimulus modification (original, moderate, or extreme), position (initial and final), and identity of r-l segment (/r/ versus /l/).

	Original	Moderate	Extreme	Mean
/r/-Final	98.4	75.0	73.4	82.3
/r/-Initial	75.0	90.6	78.1	81.3
/l/-Final	92.2	87.5	90.6	90.1
/l/-Initial	57.8	48.4	42.2	49.5
Mean	80.9	75.4	71.1	75.8

3. RESULTS

Overall, each token was accurately perceived by the CE listeners, with no subject making an identification error. However, ratings appeared to be more sensitive to differences among the stimuli than the accuracy data. Table 1 shows the mean ratings across duration (original, moderate, & extreme), /r-l/ position (initial & final), and segment identity (/r/ or /l/). Tokens were generally perceived as clear, with a mean rating of 2.3. An analysis of variance revealed no reliable effect of /r-l/ position (initial vs. final) nor any effect of whether the word contained /r/ or /l/ ($p>0.05$). However, a significant main effect of duration, $p<0.05$, $F(3,10)=4.695$, $MSE = 2.02$, was obtained. As the duration of the change increased (by either the addition or deletion of a portion of the stimulus), subjects rated the modified stimuli slightly more distorted. The effect was small, but it was observed across all six listeners. No other reliable effects were obtained.

Mean identification accuracy for the Japanese listeners is shown in Table 2. Overall, their performance was substantially worse than the CE listeners, with 75.%of the stimuli correctly identified by the Japanese listeners compared to 100% correctly identified by the CE listeners. Because of the small number of Japanese listeners, it was not possible to carry out any statistical tests on the data. However, an examination of Table 2 indicates several interesting findings. First, identification performance is poorer for /r-l/ in word-initial environments (63%) than for word-final environments (86%), replicating earlier work. Second, performance for stimuli containing word-initial /r-l/ is substantially worse than in any other environment (50% compared to between 81% and 90%). Finally, the results indicated that subjects' performance did vary as a function of duration, although not always in the direction predicted. Only for /r/-initial words did lengthening the signal appear to have any positive effect, and that was limited to the 40 ms increase. For the length decrease in word-final position, the only noticeable effect appears in /r/ final words. In the other two environments the duration modification appeared to have either little effect (/l/-final) or an effect opposite to that predicted (/l/-initial).

A comparison of the CE and Japanese listeners requires rating data to be compared to accuracy data. Though indirect, the two tasks should each be measuring some aspect of how listeners perceive the stimuli. Overall, the two sets of data were not correlated ($r=-0.026$). Although a global effect of duration was found in both groups, the effect is likely attributable to different sources for each group. For example, the most poorly perceived category for CE listeners is the extreme (long) duration /r/ in initial position, receiving a rating approximately twice as poor as the original stimulus, while for the Japanese listeners, the same category was perceived approximately as accurately as its original counterpart. In general, the results suggest that the CE and Japanese listeners are sensitive to different aspects of the signal.

4. DISCUSSION

The results of the present experiment provide mixed support for the hypothesis that a sufficiently salient acoustic cue is capable of over-riding the attentional focus produced by a listener's native language phonological system. Some caution is necessary before completely accepting this conclusion, however. First, results from the CE listeners suggest that the duration manipulation may have produced slightly degraded stimuli. Despite the fact that the stimuli were identified perfectly by the CE listeners, their quality ratings indicate that the duration changes may have also changed the acoustic structure in other, more subtle ways. Thus, the methodology itself may require further investigation.

A second qualification of the present results is that only a small sample of Japanese listeners were used. The problem with small samples is that additional listeners can change the pattern of results. The two subjects who participated in the present experiment performed consistently across the different types of stimulus categories. Nevertheless, additional subjects would be useful to reduce the variability in the data even further.

Setting aside the above mentioned caveats, the present results do suggest that simple laboratory procedures may not provide a panacea for difficulties that listeners may have in perceiving certain phonemes. As Logan et al. (1991) noted, substantial amounts of training with naturally produced stimuli from multiple talkers may be the only way to reliably facilitate the perception of difficult-to-perceive nonnative phonemic categories.

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ÉTUDE ACOUSTIQUE COMPARATIVE DES TONS "hoi" - "nga" EN VIETNAMIEN DU NORD ET DU SUD

Nguyen Ngoc-Quang
Université du Québec à Montréal, Département de linguistique
C.P. 8888, succ. A, Montréal, Québec CANADA H3C 3P8

1. INTRODUCTION

Dans le système d'écriture en vietnamien moderne, il y a six signes diacritiques qui représentent six tons distincts de la langue. La principale distinction se manifeste aux modifications des hauteurs dans les courbes tonales. Selon les régions géographiques, certains tons dans ce système hexa-tonal diffèrent de l'un à l'autre [1, 2, 3].

Sur le plan phonologique, le vietnamien du nord possède six tons phonologiques tandis qu'en vietnamien du sud, il n'y en a que cinq [4, 5, 6]. Cette différence de nombre résulte du processus de fusionnement des deux tons "***hoi***" - "***nga***" que nous appelons ton 4 et ton 5 respectivement.

Notre étude vise à décrire et à caractériser, sur le plan de la réalisation phonétique, les propriétés acoustiques (les courbes tonales) des deux tons "***hoi***" - "***nga***" en vietnamien du nord et du sud. La normalisation de la fréquence fondamentale (F0) permet enfin de comparer ces deux tons en question.

2. MÉTHODOLOGIE

Le corpus d'analyse contient 54 mots monosyllabiques tirés de deux enregistrements réalisés au laboratoire de phonétique de l'Université du Québec à Montréal. Ces mots, enregistrés dans des phrases porteuses, contiennent chacune des neuf voyelles du vietnamien dans trois structures syllabiques (V, CV et CVC) ainsi que les deux tons "***hoi***" - "***nga***" (C1 = [s] et C2 = [m]). Deux locuteurs de sexe masculin, l'un du nord et l'autre du sud, appartenant à la classe moyenne, ont participé à l'étude.

Le corpus a été ensuite numérisé à 10 kHz à partir du logiciel d'analyse Speech Station. Un extracteur des valeurs de la fréquence fondamentale a été utilisé pour générer les valeurs de F0 de la rime, soit V ou VC [**Figures 1, 2**]. Les durées de ces segments ont été d'abord converties en pourcentage (100%) puis les valeurs de F0 ont été mesurées dans chaque intervalle de 5% de la durée convertie [7].

La compilation des données comporte deux étapes. La première étape consiste à trouver les courbes tonales caractéristiques de chaque type de syllabe en faisant la moyenne des valeurs de F0 des neuf voyelles [**Tableaux 1 & 2**]. La deuxième étape, quant à elle, consiste à trouver la fréquence fondamentale normalisée de chaque ton en faisant la moyenne des valeurs de F0 des trois types de syllabes [**Tableau 3**].

3. RESULTATS

La normalisation de la courbe tonale des tons 4 et 5 produit par les locuteurs de chaque région [**Figure 3**] nous permet de ressortir les observations suivantes:

@ Deux facteurs qui jouent un rôle décisif dans la caractérisation des tons sont la hauteur et le mouvement relatifs de F0.

@ Les tons 4 et 5 du vietnamien du sud ont une courbe tonale presqu'identique i.e. ils ont une même hauteur ainsi qu'un mouvement relatif analogue, descendant-montant.

@ les courbes tonales des tons 4 et 5 du vietnamien du nord ont cependant un mouvement quasi-ressemblant, descendant-montant-descendant, mais leurs hauteurs sont tout à fait distinctes entre elles; l'écart maximal de la hauteur du ton 5 est 49 Hz.

4. CONCLUSION

Nous avons fait ressortir l'importance de deux facteurs -la hauteur et le mouvement relatifs- que sont à la base de la différence entre les tons "***hoi***" - "***nga***" de deux régions dialectales du Vietnam.

Nous retenons également que, sur le plan phonologique, les tons "***hoi***" - "***nga***" du vietnamien du sud ont subi un processus de fusionnement de la hauteur relative et ils ont une même réalisation phonétique, un mouvement descendant-montant. Comparativement au vietnamien du nord, il apparaît que la neutralisation consiste à une perte de la hauteur relative du ton "***nga***".

Cependant, les tons 4 et 5 du vietnamien du nord gardent précieusement, chacun de leur part, leurs propriétés acoustiques particulières, descendant-montant-descendant et une distinction de hauteur relative.

Il serait sans doute intéressant de chercher les indices psycho-acoustiques, par le biais des tests perceptifs, qui permettent de différencier ces deux tons "hoi" - "nga" du vietnamien du nord.

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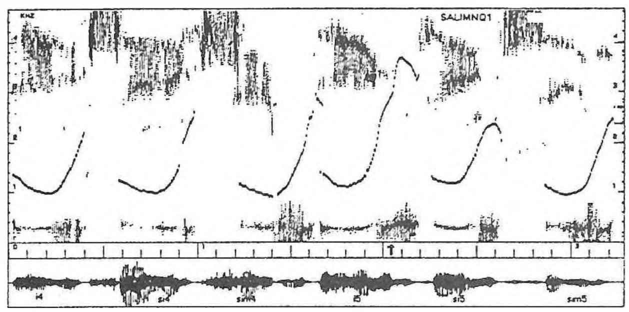

Figure 1: Les courbes tonales des tons 4 et 5 dans les mots [i, si, sim] produits par le locuteur du vietnamien du nord

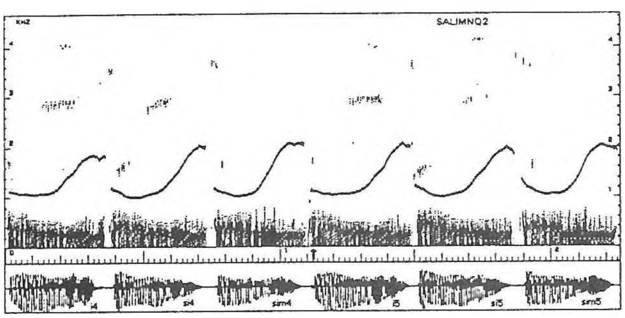

Figure 2: Les courbes tonales des tons 4 et 5 dans les mots [i, si, sim] produits par le locuteur du vietnamien du sud

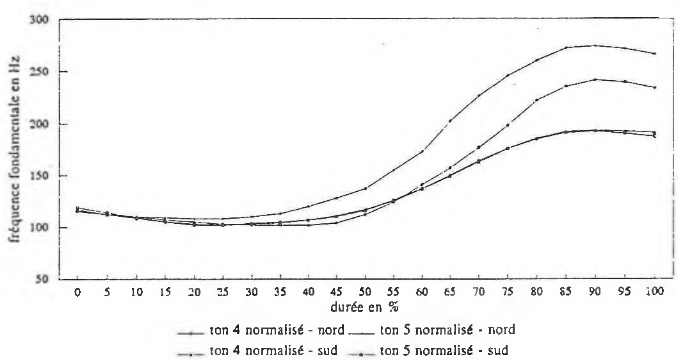

Figure 3: Les tons "*hoi*" & "*nga*" du vietnamien du nord et du sud

F0 / Durée en %	Ton 4			Ton 5		
	V	CV	CVC	V	CV	CVC
0	116	112	119	121	114	110
5	114	108	114	120	111	106
10	111	106	112	119	108	104
15	108	104	110	116	107	103
20	105	102	108	115	106	103
25	103	100	106	115	106	104
30	102	99	105	117	108	106
35	101	100	104	119	111	110
40	102	101	102	125	119	115
45	105	104	104	133	130	122
50	111	112	112	140	142	129
55	120	128	125	163	154	148
60	134	149	139	188	166	166
65	150	166	154	214	197	196
70	169	184	178	244	217	216
75	188	203	202	266	237	233
80	217	229	221	282	254	245
85	230	241	235	294	272	249
90	236	247	239	296	277	250
95	233	247	237	288	276	249
100	224	245	230	277	274	246

Tableau 1: Les fréquences fondamentales des tons 4 et 5 du vietnamien du nord associées à trois types de syllabes

F0 / Durée en %	Ton 4			Ton 5		
	V	CV	CVC	V	CV	CVC
0	111	118	121	111	123	124
5	110	111	114	109	116	116
10	107	107	110	107	110	111
15	105	103	106	105	104	107
20	104	100	103	104	101	104
25	103	100	102	103	101	103
30	104	102	103	105	102	104
35	105	104	104	106	104	106
40	108	107	106	109	106	107
45	112	112	109	113	110	109
50	117	118	115	119	116	114
55	125	126	123	126	123	128
60	136	138	136	137	135	140
65	149	151	148	150	145	154
70	160	165	163	162	159	170
75	173	176	180	175	172	181
80	183	186	190	187	182	187
85	191	189	196	195	186	192
90	193	187	199	196	188	192
95	192	185	200	193	187	191
100	188	184	202	189	186	187

Tableau 2: Les fréquences fondamentales des tons 4 et 5 du vietnamien du sud associées à trois types de syllabes

F0 / Durée en %	nord		sud	
	Ton 4	Ton 5	Ton 4	Ton 5
0	116	115	117	119
5	112	112	112	114
10	110	110	108	109
15	107	109	105	105
20	105	108	102	103
25	103	108	102	102
30	102	110	103	104
35	102	113	104	105
40	102	120	107	107
45	104	128	111	111
50	112	137	117	116
55	124	155	125	126
60	141	173	137	137
65	157	202	149	150
70	177	226	163	164
75	198	245	176	176
80	222	260	186	185
85	235	272	192	191
90	241	274	193	192
95	239	271	192	190
100	233	266	191	187

Tableau 3: Les fréquences fondamentales normalisées des tons "*hoi*" & "*nga*" du vietnamien du nord et du sud

Relâchement et abrègement des voyelles: phonétique ou phonologie?

François Poiré
Université du Québec à Montréal, Département de linguistique
C.P. 8888, succ. A, Montréal, Québec (Canada) H3C 3P8

0. Introduction

Nous voulons dans ce travail jeter un nouveau regard sur la relation qui existe entre le relâchement des voyelles hautes et leur abrègement en français du Québec. Nous allons arguer que le relâchement est un phénomène phonétique qui semble toucher toutes les voyelles, au même titre que l'abrègement et l'allongement de ces segments.

1. Problématique

Dans la section sur les lois de durée de Marchal (1980), on peut lire:

«(...) les voyelles (...) dans une syllabe accentuée fermée par /p/, /t/ ou /k/ (...) deviennent très brèves (on pourra souvent observer une tendance à l'ouverture des voyelles hautes.)» p.84

D'autre part, il ajoute à la page 163, lorsqu'il aborde plus spécifiquement le relâchement des voyelles hautes:

«Ce phénomène n'est pas gênant pour la communication tant qu'il n'y a pas réduction d'une opposition de timbre.»

Il semble donc que d'un côté, le phénomène du relâchement soit un phénomène régulier en français du Québec, et d'un autre côté, on ne l'associe qu'aux voyelles hautes en raison d'une interférence communicationnelle causée par une opposition vocalique menacée /i, e/.

De plus, s'il y a relâchement, c'est en raison de la présence en position post-vocalique de certaines consonnes dites abrégeantes. S'il s'agit de consonnes dites allongeantes, on ne parle plus de relâchement, Il y aurait alors distribution complémentaire parfaite. Le tableau qui suit synthétise ce que Marchal (1980) dit explicitement au sujet de cette distribution concernant l'effet des consonnes sur les voyelles hautes.

	relâchement	durée
occlusives sourdes	oui	abrè.
occlusives sonores	non	all.
fricatives sourdes	oui	abrè.
fricatives sonores	non	all.
nasales	?	?
l	?	?
r	non	all.

Tableau 1. Effet des consonnes entravantes sur les voyelles hautes selon Marchal 1980 (abrè. = abrègement, all. = allongement).

La distribution complémentaire semble parfaite. Elle le demeurera jusqu'à ce que l'effet des nasales et de /l/ soit élucidé.

La question qui nous intéresse est donc la suivante: le relâchement des voyelles est-il une conséquence normale de l'abrègement (ou bien: l'allongement préserve-t-il du relâchement)? Si oui, peut-on alors parler d'un phénomène phonologique?

2. Méthodologie

Pour tenter de répondre à ces questions, nous avons construit un corpus dans lequel les voyelles du français /i, E, a/ se retrouvent en syllabe ouverte ou fermée par les consonnes /p, b, f, v, l, r, m/. Chaque mot a été enregistré trois fois à tempo normal dans la phrase porteuse «répète x deux fois» par un seul locuteur. Chaque phrase a été numérisée à 10k par échantillon pour qu'ensuite soient extraites les durées des voyelles ainsi que les valeurs de F1 et F2 en leur centre, à l'aide du logiciel Speech Station. Nous ne parlerons ici que de durée et de F1.

Afin de comparer les valeurs de durée et de formants, les voyelles en syllabe ouverte servent de point de référence. Ainsi, la durée de /i/ en syllabe non entravée vaut 100%. Si une consonne a un effet allongeant de 5% sur cette voyelle, nous dirons que cet effet vaut +5. Tous les tableaux de la section suivante suivent ce principe.

3. Résultats

3.1 La durée

Le tableau 2 montre l'effet des sept consonnes du corpus sur la durée des trois voyelles. Les consonnes sont ordonnées non pas selon les classes naturelles mais selon l'importance de l'effet, de la plus grande valeur négative à la plus grande valeur positive.

	i	E	a
p	-35.01	-31.33	-33.05
f	-22.04	<u>-07.45</u>	<u>-16.97</u>
m	-13.18	+09.00	+17.50
b	<u>-03.06</u>	+19.02	+19.95
l	+13.42	+40.84	+47.45
v	+86.52	+55.09	+58.34
r	+95.43	+88.38	+72.98

Tableau 2. Effet des consonnes sur la durée des voyelles exprimé en %.

Les valeurs soulignées montrent le passage de l'abrègement à l'allongement. Ce tableau montre que les trois voyelles subissent les mêmes effets quant à leur durée. La seule différence notable concerne les effets de /m, b/ sur le /i/.

Nous sommes donc en présence d'une hiérarchie où les valeurs changent de manière graduelle et non catégorielle

3.2 Les valeurs de F1

Le relâchement est habituellement associé à l'ouverture des voyelles même si certains auteurs essaient encore de lier le phénomène à une baisse de la force articulatoire (Ostiguy et Tousignant, 1993). En suivant Delattre (1951), l'augmentation des valeurs de F1 serait la conséquence acoustique de l'ouverture de la cavité buccale. F2, toujours selon le même auteur, diminuerait à mesure que la cavité frontale s'allonge (à mesure que la langue recule).

Si le relâchement est réellement associé à l'abrègement, les valeurs de F1 des voyelles abrégées devraient augmenter.

Le tableau 3 montre les valeurs de F1 de nos trois voyelles sous l'influence des consonnes abrégeantes du tableau 2.

	i	E	a
p	+08.19	+01.10	+09.62
f	+04.92	+03.29	+08.87
m	+09.83	+06.68	0.00
b	+08.18	-01.10	+10.48

Tableau 3. Effet des consonnes sur le F1 des voyelles exprimé en %.

On peut voir que les valeurs négatives du tableau 2 sont corrélées à des valeurs positives du tableau 3. Mais pour que cette corrélation soit concluante, les valeurs positives de durée du tableau 2 devraient être associées à des valeurs négatives de F1. Le tableau 4 donne ces valeurs d'effet sur F1 des consonnes allongeantes du tableau 2.

	i	E	a
l	+08.19	-01.10	+05.65
v	0.00	0.00	0.00
r	-06.56	+24.17	+09.68

Tableau 4. Effet des consonnes sur le F1 des voyelles exprimé en %.

On retrouve dans le tableau 4 et des valeurs positives et des valeurs négatives. C'est donc dire que la corrélation n'est pas présente.

4. Discussion

Nos résultats montrent qu'il est très difficile d'établir une corrélation directe entre l'abrègement des voyelles et le relâchement. De plus, l'allongement ne semble pas préserver ces mêmes voyelles des changements de valeur de F1. Ces deux points vont à l'encontre de ce qui est dit habituellement sur ces phénomènes. Autrement dit, les effets des consonnes sur le F1 des voyelles (tableaux 2 et 3) ne permettent pas de dégager deux ensembles de données distincts justifiant la présence d'une règle de relâchement au niveau de la phonologie. Seul le comportement de /i/ en ce qui concerne sa durée (tableau 1) va dans le sens d'une distinction phonologique au niveau des classes de voyelles: si ce segment est plus facilement affecté dans sa durée par les consonnes entravantes, c'est qu'il appartient à une classe de voyelles dites ultra légères (Cedergren et Simoneau, 1985).

5. Conclusion

Que F1 soit ou non le corrélat acoustique du relâchement, nos données montrent que les voyelles subissent presque toujours les mêmes modifications formantiques selon la consonnes entravantes. Le relâchement serait alors un simple phénomène phonétique général du français, du moins dans sa version parlée au Québec. Il aurait obtenu son statut de vedette de la prononciation de cette variété en ce qui concerne les voyelles hautes en raison de la distribution de l'espace phonologique de notre système vocalique en interaction avec les particularités de la réalisation phonétique propres à cette variété.

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Le focus et l'intonation en français parlé à Montréal

Louise Levac
Université du Québec à Montréal, Département de linguistique
C.P. 8888, succ. A, Montréal, Québec (Canada) H3C 3P8

INTRODUCTION

Notre recherche comporte deux volets. D'une part, nous voulons connaître quelles sont les caractéristiques linguistiques du focus et ce qui le distingue comme catégorie prosodique des autres catégories comme, par exemple, l'intonation terminale, l'intonation continuative, etc. D'autre part, nous voulons savoir si les variations de la fréquence fondamentale expliquent le fait que les auditeurs québécois ont l'habileté de discriminer un focus des autres types catégoriels.

1. METHODOLOGIE

Notre corpus est constitué de quatre extraits d'environ cinq minutes puisés dans l'enregistrement de quatre locuteurs (deux femmes et deux hommes) du *Corpus Sankoff-Cedergren* (Le français de Montréal, 1971). Ces extraits sont filtrés et numérisés à 8000 sur un Texas Instrument. Ils sont analysés de deux façons: par une segmentation perceptuelle et par une segmentation acoustique.

1.1 Segmentation perceptuelle

Deux locutrices québécoises ont segmenté à l'écoute les enregistrements en quatre catégories prosodiques: soit, 1) l'intonation complète, 2) l'intonation partielle, 3) l'accentuation secondaire et 4) le focus. Cette catégorisation prosodique s'est faite en deux séances; la première pour identifier les trois premières catégories et la seconde pour identifier le focus. Avant la séance d'écoute, des critères impressionnistes généraux ont été remis et la consigne a été de marquer distinctivement sur un texte orthographié de l'enregistrement les endroits où elles percevaient une catégorie. Le tableau 1 ci-haut présente ces critères que nous avons élaborés et le tableau 2 fournit des exemples de segmentation perceptuelle.

TABLEAU 1: Méthode d'écoute et critères impressionnistes

Première séance: Ecouter tout l'extrait d'un locuteur, puis à la réécoute et en vous servant du texte écrit

1) Marquer d'un] une intonation complète. (Mettre dans cette catégorie les intonations suivies d'une pause.)

2) Marquer d'une } une intonation partielle, interne à l'intonation complète. (Mettre dans cette catégorie les hésitations qui ne sont pas suivies d'une pause.)

3) Marquer d'une) une proéminence accentuelle secondaire, qui est interne à l'intonation complète ou à l'intonation partielle.

Deuxième séance: Ecouter tout l'extrait d'un locuteur, puis à la réécoute et en vous servant du texte écrit

1) Marquer d'un + une proéminence accentuelle (très) forte et (très) haute.

TABLEAU 2: Exemples d'énoncés segmentés à l'écoute (Les marques entre parenthèses indiquent que les juges ont des interprétations variées.)

2.23 C'est mon frère } qui l'a acheté } quand elle est morte.] Elle est morte}+ au mois ( )) de février.]

7.2 Puis ensuite] l'école a été changée } pour Basile-) Routhier.]+

70.8 Il faut pas } que je m'étende} (+) là-dessus]+

1.2 Segmentation acoustique

La durée en ms de la voyelle marquée d'une marque prosodique et la valeur en Hz de son fondamental sont calculées et comparées entre elles dans le but de faire ressortir les différences acoustiques générales qui opposeraient le focus aux autres catégories prosodiques. La segmentation de la voyelle se fait prioritairement sur le signal. Le changement dans l'onde oscillographique sert à fixer les bornes initiales et finale du segment. A l'écoute, nous décidons de l'étiquette phonétique du segment. A l'aide de l'extracteur du programme *Speech Station*, les valeurs du fondamental pour un énoncé sont calculées automatiquement et affichées sous la forme d'un contour intonatif. La valeur en Hz au 2\3 de la voyelle est retenue pour l'analyse comparative des différentes marques prosodiques. Le mode de comparaison des valeurs en Hz est le demi-ton, il est calculé en mettant en rapport la valeur de la voyelle marquée prosodiquement à celle de la voyelle adjacente de gauche. Le tableau 3, à la page 2, donne un exemple des valeurs obtenues par la segmentation acoustique.

2.RESULTATS

2.1 Analyse linguistique

Les résultats de la segmentation perceptuelle montrent que le focus peut être entendu indépendamment des autres catégories, mais qu'il peut aussi être cooccurrent aux catégories] et }. Il est optionnel: de grandes parties des extraits ne contiennent aucune marque de focus alors que toutes les phrases des extraits comportent au moins la marque].

Une analyse discursive des lieux d'ancrage des marques du focus montre qu'elles peuvent être interprétées comme des marqueurs prosodiques qui ont une fonction déictique: le focus fait ressortir un item énoncé de façon à y attirer l'attention de l'interlocuteur; il s'agit d'un marqueur extralinguistique qui oriente l'attention sur des sujets d'intérêt pour l'un ou l'autre des interlocuteurs. On peut paraphraser cette fonction par les phrases suivantes: "Ceci, ça m'intéresse d'en parler" et "Cela, ça t'intéresse d'en parler".

2.2 Analyse perceptuelle

La mise en commun du travail d'écoute de l'une et l'autre des auditrices indique qu'il existe un taux d'accord moyen de 55.77 % entre les catégories prosodiques distinguées et les syllabes marquées. Il existe un taux d'accord partiel moyen de 11.81 %, c'est-à-dire que les locutrices ont toutes les deux perçu une marque prosodique mais qu'elles ne l'ont pas identifiée par la même étiquette. Le taux de désaccord moyen est de 32.42 %; dans ces cas, une seule des locutrices a perçu une marque prosodique et l'a catégorisée.

2.3 Analyse acoustique

Nous présentons les résultats préliminaires de l'analyse acoustique pour lesquelles les juges ont un taux d'accord. Le tableau 3 présente les deux paramètres acoustiques (durée et pitch) et le rapport en demi-tons. Dans l'énoncé 2.6 du locuteur 2, *Il n'y avait pas de transports*, la dernière syllabe du mot *transports* a été perçue comme portant une marque d'intonation complète] et une marque de focus + . La voyelle [ɔ] de cette syllabe a une durée de 48.9ms et un pitch de 135.1 Hz. Le rapport des valeurs du fondamental entre la pénultième [ɑ̃] et la dernière syllabe est de 3.73 demi-tons.

CONCLUSION

Dans la parole spontanée, les marques prosodiques sont entendues et catégorisées par les locuteurs natifs. Avec des critères impressionnistes et des consignes d'écoute, nous avons obtenu des données qui montrent l'existence de notions linguistiques comme l'intonation complète, l'intonation partielle, la proéminence secondaire, le focus. Dans notre recherche, le focus a été critérié comme une proéminence forte et haute. Les résultats des données d'écoute montrent que le focus est une marque distincte des autres marques prosodiques, qu'il est optionnel et qu'il est cooccurrent ou indépendant. Les résultats de l'analyse acoustique mettront en lumière les corrélats physiques propres à chacune des catégories prosodiques.

TABLEAU 3: Exemple d'analyse acoustique

2.6 Il n'y avait pas ( )) de transports.]+

étiquette i a v ɐ p ɑ t r ɑ̃ sp ɔ w

durée 48.9

Hz 109 135.1

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PHARYNGEALS IN TIGRINYA

John Anderson and Henry Rogers

Department of Linguistics
University of Toronto
Toronto, Ontario M5S 1A1

INTRODUCTION

This paper focuses on the phonology and phonetics of /ʕ/, a voiced pharyngeal approximant. While the Tigrinya pharyngeal is like pharyngeals in other language families in which they occur (Afro-Asiatic, Caucasian, and Salishan) in their effect on vowel formants, they differ in that they are realized as a break in the formant patterns of an adjacent vowel segment.

Tigrinya is a South Semitic language spoken in Eritrea. It has the following phomenic inventory:

Consonants

	t t'		k k'		ʔ
b	d		g		
f	s s'	ʃ		ħ	h
	z			ʕ	
		tʃ tʃ'			
		dʒ			
m	n	ɲ			
	l, r				
w		j			

Vowels

i		u
e	ə	o
ä		a

The phonemes /k/ and /k'/ are fricated to the uvular [χ] and [χ'] (sometimes [ʁ]) after vowels.

The speaker for this study is a 27-year old graduate student at the University of Toronto.

BREAKING

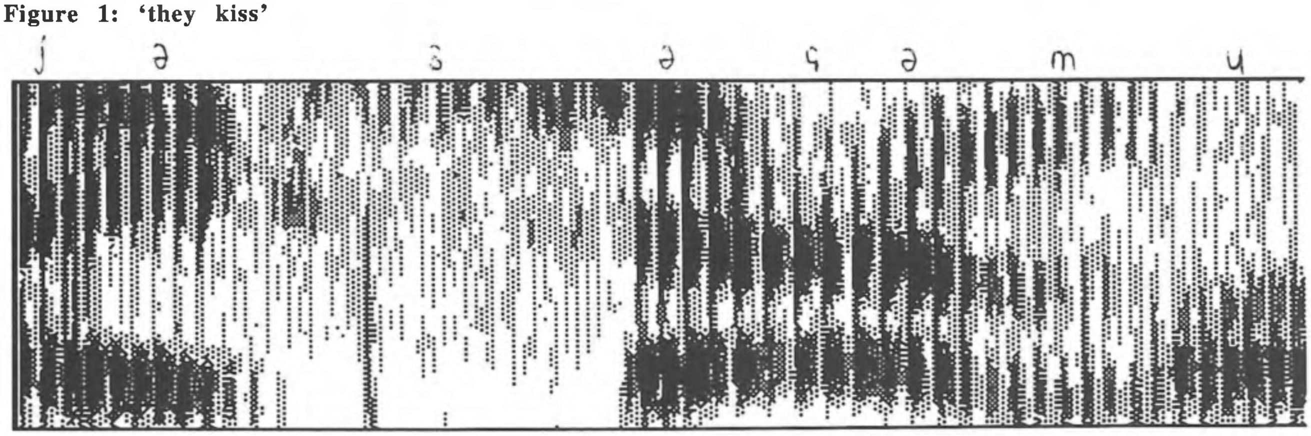
The results of this study are for the most part consistent with the findings reported in the literature and are given below. However, this study uncovered a remarkable feature of Tigrinya that is not reported in detail elsewhere. When, phonologically, the voiced pharyngeal consonant is followed by another consonant, the pharyngeal is not realized where one expects, but instead in the middle of the adjacent vowel segment. Consider the example 'they kiss' shown in the spectrogram in Figure 1. The phonological representation /jəsəʕmu/ is phonetically manifested as [jəsəʕəmu]. Vowel formants are clearly present on both sides of the pharyngeal [ʕ]. Bessell (1992) mentions that [ʕa, aʕ] sequences in Salishan are difficult to distinguish phonetically, but does not go into detail. Perhaps her observation is a result of the same phenomenon noted here.

VERB STRUCTURE

Evidence that the pharyngeal is phonologically at a syllable edge is available from Tigrinya nouns and verbs. In Tigrinya, as with other Semitic languages, the basic verb consists typically of three consonants (e.g. /nbb/ ' to read'). Inflected forms are formed by inserting vowels between these consonants and by prefixation and suffixation. Leslau (1939) states that the imperfect of a verb has the template form /CäCCä/; when a suffix is added the second vowel is deleted. The informant for this experiment pronounced the first vowel in the form as /ə/ rather than /ä/. The third plural is formed by the discontinuous affix /jə...u/ or /jä...u/, depending on the verb. The third plural form of the root /fk'r/ 'love' is /jäfək'ru/ 'they love'. For the verb root /sħk'/ 'laugh', there is /jəsəħk'u/ 'they laugh'. For /sʕm/ 'kiss', the phonological representation is /jəsəʕmu/ 'they kiss'; phonetically, however, this is realized as [jəsəʕəmu], with the voiced pharyngeal consonant being realized in the middle of the adjacent vowel.

Additional evidence is provided by possessive suffixes on nouns. The first person plural possessive suffix is /na/. The word for 'cow' is /lam/, and 'our cow' is /lamna/. The word for 'idea' is /ħasab/, and 'our idea' is /ħasabna/. For /səmamməʕ/ 'agreement', the phonological form is /səmamməʕna/ 'our agreement', but phonetically it is [səmamməʕəna].

Figure 1: 'they kiss'



VOWEL DURATION

The duration of a vowel not adjacent to a pharyngeal is an average of 116 ms. The pharyngeal in 'they kiss' is 33.1 ms, with a vowel of 55.2 ms preceding it and a vowel of 44.8 ms following it. When the duration of the two vowels preceding and following the pharyngeal are added together, the result is 100 ms. This suggests that the pharyngeal is realized in the middle of one vowel, as opposed to a second, epenthetic vowel being realized after the pharyngeal.

EXPERIMENT

The purpose of the experiment was to study the acoustics of pharyngeals compared to glottals, velars, and uvulars.

A total of 136 words were recorded from the speaker; these were elicited in a frame /ʔəndägäna ... bäl/ 'say ... again' to control for rate and rhythm. Of these, 65 were real Tigrinya words and 71 were nonsense forms. The words were recorded four times, giving a total of 536 tokens. The forms were selected to illustrate pharyngeal consonants as well as velars, uvulars, and glottals. The forms were digitized and analyzed using the acoustic analysis program Signalize. Measurements were made of duration of the pharyngeal consonants and the adjacent vowels.

RESULTS

Pharyngeals

The voiced pharyngeal approximant /ʕ/ is realized as a pharyngealized segment of the preceding or following vowel. A vowel - voiced pharyngeal cluster is realized as a normal vowel portion which is slightly pharyngealized, followed by a strongly pharyngealized portion. The strongly pharyngealized portion (/ʕ/ in Figure 1) corresponds to the voiced pharyngeal.

Before /ʕ/, the formant transitions of vowels show a rise of the first formant and a fall of the third formant. The amount of rise and fall varies depending on which vowel is being affected.

When the measurements of the centres of the pharyngeals are compared to measurements of the centres of the steady state vowels, the following generalizations emerge. These findings are consistent with those reported in the literature (Alwan, 1986 (Arabic); Ghazeli, 1977 (Arabic); Butcher and Ahmad, 1987 (Arabic)).

Pharyngealized /i/ shows an average lowering of the third formant of 285 Hz and an average raising of the first formant of 122 Hz, when compared with steady state /i/. The forms with pharyngealized /u/ show an average lowering of the third formant of 184 Hz and an average raising of the first formant of 92 Hz. The forms with pharyngealized /a/ show an average lowering of the third formant of 265 Hz and an average raising of the first formant of 61 Hz.

The second formant lowers from steady state to pharyngealized for /i/ and /a/ and rises for /u/.

Uvulars

The uvular consonants (derived from velars) are usually fricatives, but sometimes they are approximants. Alwan (1986) found that the F1 transition for /i/ and /u/ falls from uvular to vowel, but that it rises for /a/. This study compares formants immediately preceding uvulars with steady state vowels, and finds that the differences in F1 are consistent with Alwan's findings. F2 lowers from vowel to uvular for /a/ and /i/, and remains steady for /u/, which is consistent with Alwan. F3 rises from vowel to uvular for /a/ and /u/, and lowers for /i/, which is also consistent with Alwan.

Alwan (1989) found that F1 is the perceptual cue as to whether a sound is uvular or pharyngeal. Instances of the vowel [a] with a F1 that is at least as high or higher as that of a steady state vowel were perceived as following a pharyngeal. If F1 is 90 Hz less than a steady state F1, the vowel was perceived as following a uvular. If F1 was between these two values, the vowel was perceived as following a glottal.

The results of this study show F1 transitions for pharyngeals, uvulars, and glottals that support Alwan's 1989 findings, except for the fact that F1 for /u/ was shown to

The second and third formants come together at the edge of a velar. This is consistent with findings in the literature (Kent, 1992).

CONCLUSION

The findings in this experiment are for the most part consistent with the literature, except for the transposition effect, which has not been reported for any other languages.

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DIGITAL AUDIO LABORATORY STATION

Marek Roland - Mieszkowski & Danuta Roland - Mieszkowski
Digital Recordings - Advanced R & D, 5959 Spring Garden Road, Suite 1103, Halifax, Nova Scotia, B3H-1Y5, Canada

Andrzej Czyzewski & Bozena Kostek
Technical University of Gdansk, Sound Engineering Department, ul. Narutowicza 11/12, 80-952 Gdansk, Poland

Introduction

A new **Digital Audio Laboratory Station (DAL)** system (Fig. 1) was developed under **NeXTSTEP & UNIX** operating system. DAL system is available now on NeXT and IBM PC 486 /Pentium computers and will be soon available on HP, SUN, DEC and Canon computers. Research in the areas of acoustics, psychoacoustics, audio engineering, audiology etc, requires in many cases screening of the hearing abilities of the involved subjects and investigators. Required tests are often non-standard and need much better flexibility and precision than what is available from standard analog and digital equipment. With the DFG (Digital Function Generator) software [1,2], Digital Audiometer becomes a sophisticated Digital Audio Laboratory Station for performing any type of audio experiments. The fact that signal synthesis is implemented in software makes modifications very fast and simple, allowing for a rapid development of new experiments and testing procedures.

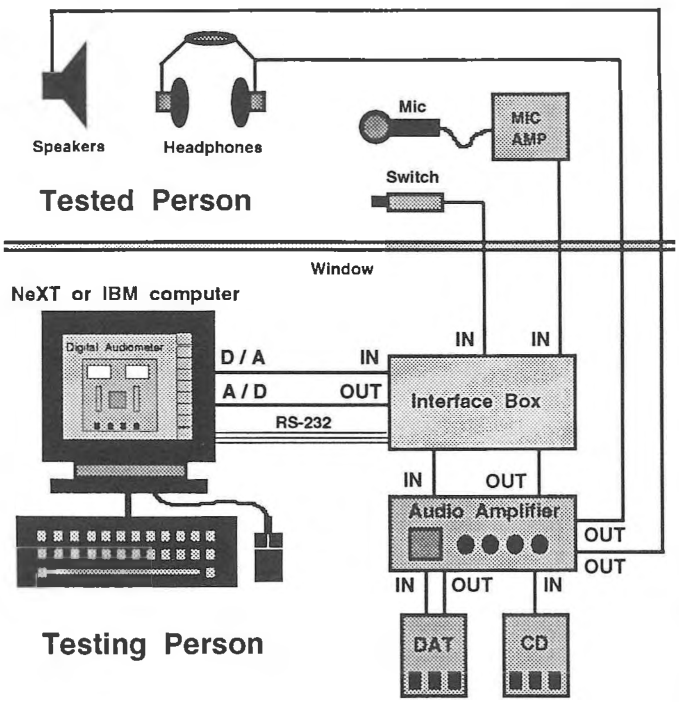
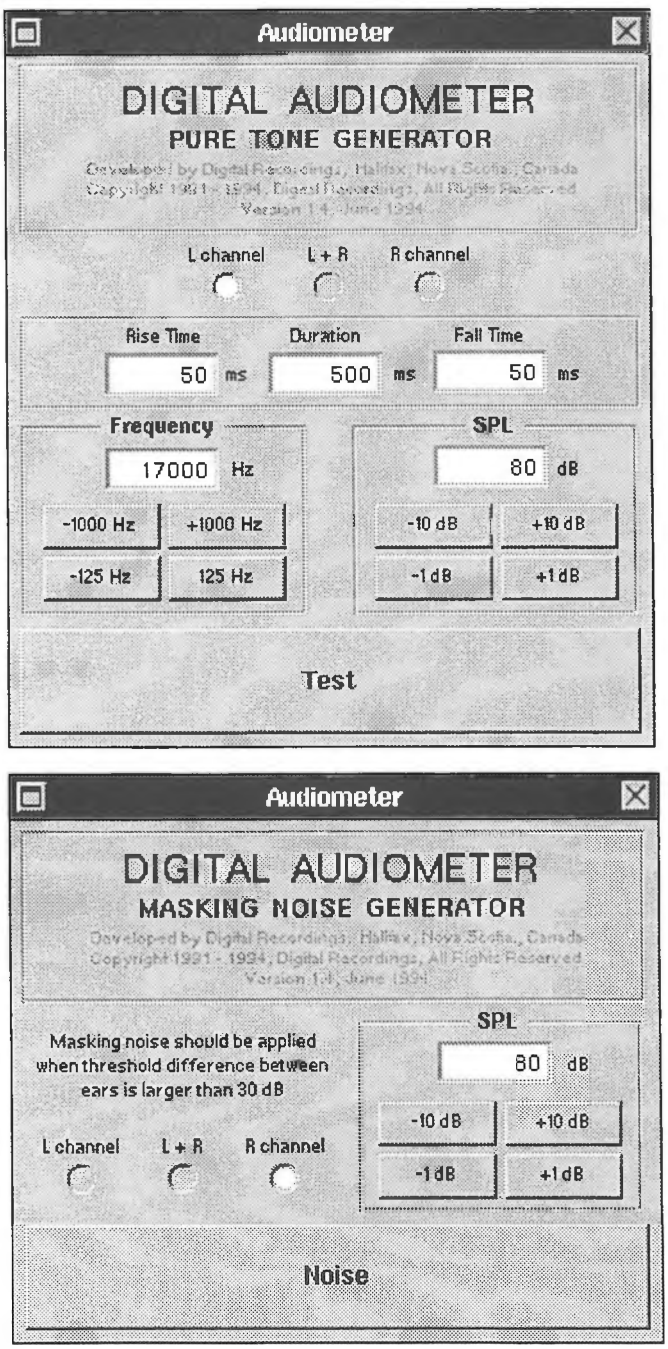


Fig. 1. Schematic Diagram of the Digital Audio Laboratory

Digital Audiometer

Audiometer.app was written for precise testing of hearing and advanced research in acoustics / psychoacoustics (Fig. 2). The quality of test signals is much higher than in the commercially available audiometers, since signal synthesis is performed in real time in digital domain and signals are dithered [3, 4]. This application is very useful for standard audiometric testing in the field of acoustics, psychoacoustics, music, sound engineering, audiology, speech pathology, etc. Signals are generated in real time according to algorithms developed by the authors [1, 2] and do not require a DSP processor (which is not available in some computers).Also the Digital Audiometer software works very well with the Digital Oscilloscope and Spectrum Analyzer (both applications come with the NeXT computer) and any recording / editing software. Playback and recording/analysis of the signal can be done simultaneously since DSP chip is not used and the main processor is used only 2->3% during signal reproduction.

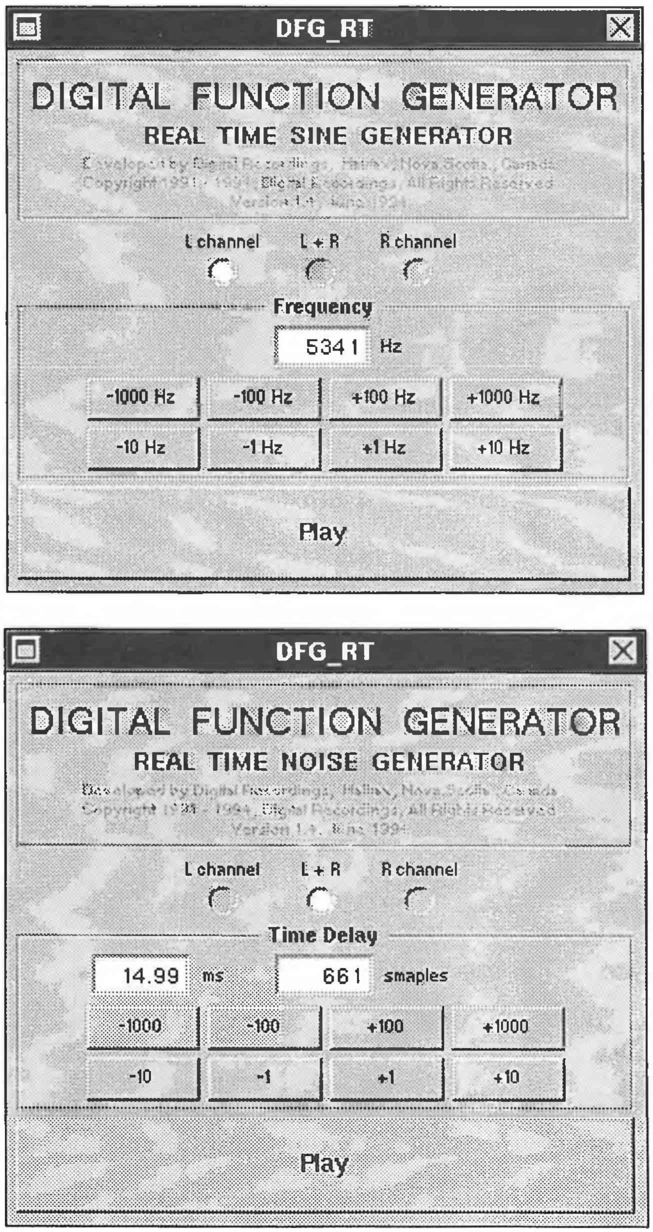
Fig. 2. Two software modules of Digital Audiometer



Below are listed main advantages of Digital Audiometer:

- frequency range: 20 Hz -> 20,000 Hz
- frequency stability +/- 0.001 Hz (at 1000 Hz)
- S/N = 95 dB (16 bit system)
- amplitude stability +/- 0.0003 dB (for 16 bit amplitude)
- no harmonic or intermodulation distortions (due to used algorithms and dithering)
- easy implementation of standard and non-standard tests
- easy modification of tests and procedures
- automatic and manual testing
- storage of all test signals in digital format
- high precision, consistency and no need for calibration
- archiveing of test results on hard and optical disks
- further processing and statistical analysis of raw data
- display and printing of test results / data
- communication and exchange of results via E-mail / Ethernet
- low price, since this is software for popular computers
- easy upgrade and technical support, since it is software

Fig. 3. Two software modules of DFG_RT



Digital Function Generator

At the present time the DFG software [1, 2], which is part of DAL, consists of 8 modules.:

- **The Principles of Digital Audio module** allows synthesis of pure tones and white noise. It can also be used to illustrate concepts of signal amplitude, frequency, phase, interference, coherence, incoherence, signal ramping, additive synthesis, beats, virtual pitch as well as to demonstrate quantization, dithering, aliasing / hard clipping / harmonic / intermodulation distortions, etc.
- **The Modulation (AM, FM & AFM) module** allows synthesis of pure tones which can be Amplitude Modulated (AM), Frequency Modulated (FM) or Amplitude and Frequency Modulated (AFM).
- **The Additive Synthesis module** allows very flexible synthesis of complex sounds from their Fourier components.
- **The Sweep Generator (AS, FS & AFS) module** is a very flexible tool for generating arbitrary amplitude (AS), frequency (FS) or amplitude and frequency (AFS) sweeps.
- **The Function Generator module** can be used to synthesize sine, square, triangular, sawtooth, pulse and white noise signals.
- **The Sound Sequencer module** can be used for construction and playback of arbitrary sound sequences.
- **DFG_RTsine** module was written for real-time synthesis of the sinusoidal (pure tone) signals (Fig.3). Signals are generated in real time and do not require the DSP processor. Playback and recording/analysis of the signal could be done simultaneously since DSP chip is not used and main processor is used only 2->3% during signal production.
- **DFG_RTnoise** module was written for real-time synthesis of the noise signal (Fig.3).

DFG is used with great success in teaching of acoustics, psychoacoustics, audio engineering, and in various research projects at many universities in Canada, USA, Europe and Asia.

Conclusions

The most precise digital-domain method for the generation of arbitrary audio signals was successfully implemented on the NeXT and IBM-PC 486 computers [1, 2]. Since NeXTSTEP / UNIX is a multitasking environment, many software applications can run simultaneously.Digital Audiometer and DFG software by-design put very little demand on the hardware and do not require a DSP processor for sound synthesis. This allows the Digital Oscilloscope, Spectrum Analyzer and Sound Recording software to run simultaneously with DFG [1]. This in turn allows very sophisticated tests and demonstrations in acoustics,psychoacoustics, audiology, electronics, physics and engineering to be performed on a single NeXTSTEP / UNIX computer.

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ACCOUNTING FOR MUSICIANS' SUPERIOR AUDITORY SERIAL-ORDER IDENTIFICATION: AUDITION OR NOTATION?

Jocelyn A. Lymburner and Annabel J. Cohen
Department of Psychology, University of Prince Edward Island
Charlottetown, PEI C1A 4P3

1.0 Introduction

Short-term memory for sequences of tones is typically better for listeners who have had musical training than for listeners who lack such training (e.g., Cohen, Trehub & Thorpe, 1989). It would seem that superior auditory skill would account for the the musicians' advantage. However, several experiments in which listeners match visual patterns to short auditory sequences (e.g., Parncutt & Cohen, in preparation) suggest that facility with a notation system may be an important factor. Since there is no definitive answer as to whether the musicians' superiority for auditory serial-order identification is due to audition or to familiarity with notation, the present study examines various possibilities by presenting subjects with both visual and auditory cues to aid recognition of sequential patterns. Evidence for the relative ease with which musicians and non-musicians use such cues enables us to explore the effects of musical training and determine the extent to which such effects are confined to or extend beyond the auditory realm. If auditory memory rather than knowledge of musical notation accounts for the musicians' advantage, then musicians' performance on an auditory task should exceed that on a visual task which entails the same sequential notational matching. In this case, only auditory, not visual, task performance should be higher for the musician as compared to the non-musician. If, however, superior notational ability accounts for the musicians' advantage, then music'ans' performance on both tasks should be equal and, at the same time, better than that of the non-musician.

2.0 Method

It was the task of the subject to match the order of eight items of an audio, visual, or audiovisual stimulus sequence to one member of a closed set of nine orders shown in Figure 1.

The tone sets from which 8-item sequences were comprised were the major (successive intervals 2212221), minor (2122122), and chromatic (1111111) scales, as well as a scale of successive intervals of four semitones (4444444) - augmented triadic, as used by Cohen & Frankland (1990). The lowest tone in each scale was 256 Hz (C_4). Tones were 200 msec in duration with an intertone interval of 200 msec. Sequences of visually presented numbers corresponded with the successive intervals of the four scales used. One numerical step equalled one semitone; hence, the chromatic scale was represented by 1 2 3 4 5 6 7 8 while the major scale was represented by 1 3 5 6 8 10 12 13 and the augmented by 1 5 9 13 17 21 25 29.

The set of nine different sequences used by Cohen and Frankland (1990), and Parncutt & Cohen, (in prep.) represented a range of complexity with respect to organization and processing difficulty (see Figure 1).

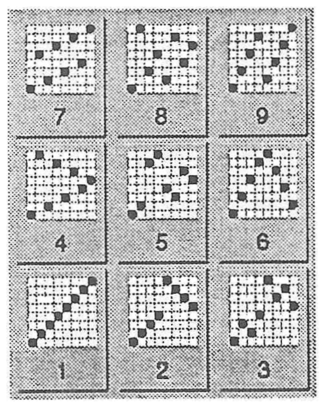

Figure 1. Graphic representation of nine sequential patterns

There were 36 sequences generated from all possible combinations of the 4 sets (of tones or numbers) and 9 orders. In audiovisual and visual conditions, at the base of the screen, there was displayed simultaneously the series of numbers corresponding to the scale notes. In audiovisual and audio conditions, the sequences were presented through headphones. The experiment consisted of 3 blocks (Audiovisual task, Audio task, or Visual task) of 36 randomly ordered trials each.

Twenty-four students participated. One half of them, designated musicians, had received an average of 12.6 instrument-years of formal music training. The other half, designated non-musicians, had received an average of 0.5 instrument-years of formal training. Groups were matched for age and sex.

3.0 Results

Overall, musicians significantly outperformed non-musicians but this was attributable primarily to the audio task, as seen in Figure 2 , $F(2,44) = 6.35$; $p<.005$, mixed model ANOVA.

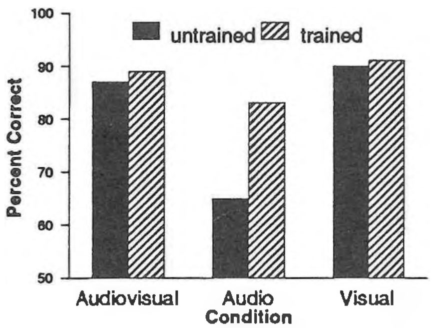

Figure 2. Mean percent correct on the three tasks by untrained (non-musicians) and trained (musician) subjects.

Response time was also significantly influenced by the interaction of condition and training, $F(2,44)=7.61$;$p<.005$ (See Figure 3). For musicians, the presence of the auditory stimulus (audio and audiovisual task) decreased response time by almost 6 seconds as compared to the visual task.

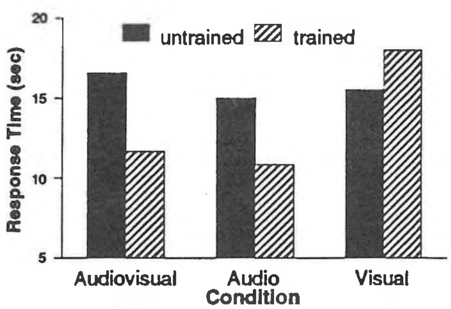

Figure 3. Mean response time on the three tasks for untrained and trained subjects.

Other effects of scale and order, unrelated to training, were in general comparable to those observed in previous studies and though of interest, are not discussed further here.

4.0 Discussion

Musicians produced better scores on the audio task and were faster than non-musicians on both tasks with an auditory component. As musicians were not superior to non-musicians on the visual task it seems unlikely that their advantage in auditory serial-order identification is due to skill in converting sequential information to visual patterns. Performance in the audiovisual condition was equal for both groups but the pattern in time was not (musicians were faster). Hence, it appears that the auditory code of musicians is both fast and accurate.

A practical application of the above findings is that an auditory code can be more efficient than an equivalent visual sequential code in a sequential pattern-tracking task. However, only musically trained listeners seem able to exploit this auditory code. This is the case, even though the auditory sequences used in the experiment were generally not musically structured. The present results suggest that the musicians' advantage in auditory sequential tasks is by virtue of auditory memory rather than familiarity with musical notation; such memory is not specific to musically structured sequences *per se*, though it may be specific to nonverbal auditory information.

5.0 Acknowledgements

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6.0 References

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AN EXPERIMENTAL APPROACH TO EVALUATION OF ACOUSTIC MASKING OF BELUGA COMMUNICATION BY SHIP NOISE

Christine Erbe, University of British Columbia, Dept. of Geophysics and Astronomy
2219 Main Mall, Vancouver BC, V6T 1Z4
David M. Farmer, Government of Canada, Fisheries and Oceans, Inst. of Ocean Sciences
9860 West Saanich Road, Sidney BC, V8L 4B2

1 Introduction

Out of concern about the impact of industrial noise on marine mammal communication, the Canadian Coast Guard initiated a project to develop quantitative techniques for determining the degree of interference between various kinds of noise and marine mammal calls. Particular emphasis lies in the study of icebreaker noise.

During a cruise in the Arctic Ocean, vocalizations of different animals, for example seals, humpback whales, killer whales and beluga whales, were recorded as well as icebreaker related ramming noise and bubbler noise. If the icebreaker rams an ice-ridge at full speed, it might crack the ice immediately or be lifted onto the ridge and crack it due to its weight. Also the ice might withstand the ramming such that the ship is stopped with its propeller still turning at full speed. The corresponding noise is a broadband signal consisting of short bursts of high intensities. Bubbler noise is generated when a ship passes through cracked ice and uses so-called bubblers along its sides that blow air at high pressure into the water in order to push ice debris away and leave a clean passage for the boat. The corresponding noise is a loud signal of long duration but narrow bandwidth.

2 Previous Research

Preliminary interference studies were carried out in three different ways. The animal calls were digitally mixed with the noise signals in various signal-to-noise ratios. First, the spectrograms of the mixed signals were plotted and the threshold noise level at which the original animal call could not be recognized was determined by eye. Second, the mixed signals were converted to analog form and the human ear was used to detect the threshold. Third, matched filter techniques were applied in which the cross-correlation coefficient of the mixed signal and the call was evaluated and a threshold noise level was defined. Discrimination of signal from noise cannot simply be expressed in terms of the signal-to-noise ratio. The masking effect depends on both frequency and temporal properties of call and noise. The overall result was that animal calls of long duration and temporal coherence, as used by the bearded seal, are very robust to noise of short bursts like ice ramming even if the noise level far exceeds the call. This is because identification is possible from the short undistorted pieces emerging through the gaps in the noise field. Even though bubbler noise is not as loud as ice ramming noise, its masking effects are more severe due to its long duration and frequency overlap with most of the animal songs. For comparison, naturally occurring, thermal ice cracking noise was also investigated. It has the least impact on the chosen animal vocalizations. Although these three applied methods yield qualitatively similar results, they exhibit subtle differences. In order to assess their accuracy and to develop reliable models for the animals' auditory abilities, experiments with trained mammals are necessary.

3 Current Experiments

The Vancouver Aquarium houses various marine mammals, for example killer whales, beluga whales, porpoises, sea-lions and seals. As a first step, experiments are conducted only with beluga whales, because they are easily trained. Right now they are learning to identify three different beluga calls, one is a short chirp, the other two are pulsating calls. One of the latter consists of short pulses at a high repetition rate, the other consists of long pulses at a low repetition rate. The spectra of the three calls are shown in figure 1. The songs are transmitted into the pool by a J9-hydrophone. The whales have to station against a pole held into the pool by a whale trainer and whenever they hear one of the three songs they are trained to move away. Once this works reliably, noise is introduced. The different calls are mixed with different kinds of noise at different signal-to-noise ratios and played to the animals in a random sequence. Each sound has a duration of about 2 seconds. The animals are allowed a 2-second-reaction time. If they do not respond to a signal, i.e. if they do not move away from the pole, it is assumed that they were unable to identify the call. If they do move, the event is counted as a successful discrimination of call from noise and the animals are stationed again for further listening. The experiments are carried out in a small pool adjacent to the main pool. There is always only one animal in the small pool at a time. In order to make sure that the investigated animal does not respond to calls coming from the other animals in the main pool, their acoustic activity is observed continuously with a recording hydrophone. For obtaining reliable data, it is essential that no cues are passed to the animals. Therefore, neither the whale nor the whale trainer knows when to expect a signal and which one it will be. The experiment conductor sits out of view

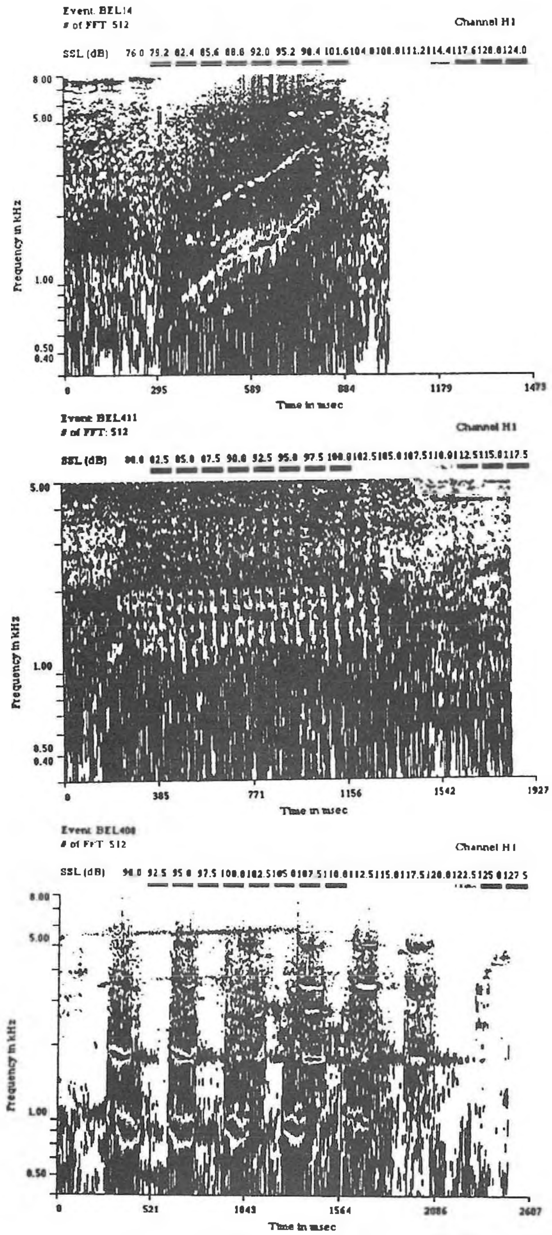

Figure 1: Spectra of Beluga Calls

of the whale at the side of the pool. He controls the transmission of the signals with a portable notebook computer and observes the response of the recording hydrophone. He can only see the whale trainer who reports the movement of the whale. A sketch of the pool is shown in figure 2.

4 Resonance Frequencies of the Pool

During any acoustic experiments, the resonance frequencies of the pool have to be taken into account. As a closed chamber, the pool has an infinite number of distinct eigenfrequencies. It acts as a filter on any incoming signal by amplifying these frequencies and damping the others. By solving the three-dimensional wave equation under specified boundary conditions for the dimensions of the pool,

$$\Delta P = \frac{1}{c^2} \cdot \frac{\partial P}{\partial t^2} \qquad ,$$

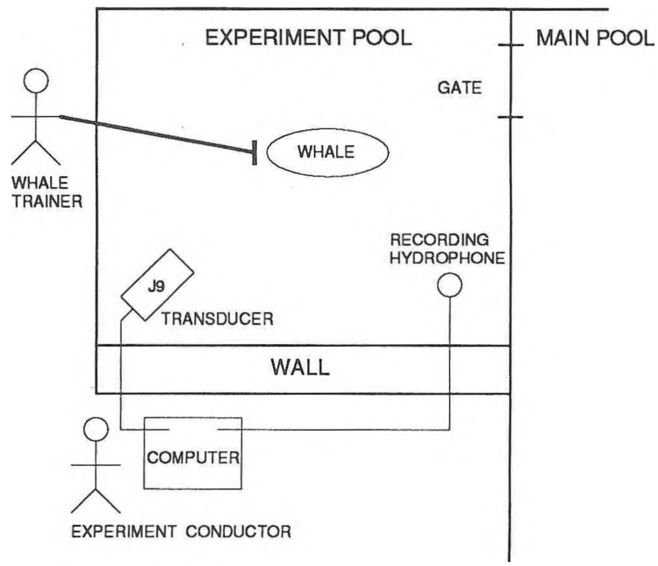

Figure 2: Experiment Set-up

where $P = P(x, y, z, t)$ denotes the pressure, Δ represents the Laplacian operator and c stands for the sound speed, the theoretical resonances were calculated. With one transmitting and one receiving hydrophone, these frequencies were experimentally verified. The presence of the whale alters the acoustic response of the pool such that the resonance frequencies are shifted and the spatial characteristics of the modes are changed. This was also measured and led to a proper positioning of the whale and transducer.

5 Neural Network Modeling

Neural networks as a fourth method for modeling discrimination of signal from noise have recently been introduced. A two-layer back-propagation network was trained with four different seal calls [1] by applying the pure calls and varying the weights and biases such that it gives a desired output. Then the net was tested with calls mixed with icebreaker noise and thermal ice cracking noise. It successfully identified the calls even for high signal-to-noise ratios.

This technique will also be applied to the beluga calls. The call recognition of the neural net will then be compared to that of the trained animals. This provides a reliable assessment of the accuracy of the neural net model. The long term goal is to use the aquarium tests to develop increased confidence in such models so that they may be used to determine the impact of any noise on marine mammal communication for situations where direct experiments are impossible.

References

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Relationship of Underwater Acoustic Intensity Measurements to Beamforming

Gerald L. D'Spain

University of California, San Diego
Marine Physical Laboratory (MPL)
Scripps Institution of Oceanography
San Diego, California 92106

Introduction

The standard method of determining the directional properties of underwater acoustic fields is to coherently combine the weighted pressure field measurements made by spatially distributed sensors. An alternative approach is to make measurements of other properties of the acoustic field, in addition to pressure, at single points in space. For example, measurements of acoustic particle velocity can be combined multiplicatively with those of pressure to determine the vector acoustic intensity. The purpose of this presentation is to discuss the relationship between the information obtained on the directionality of the sound field by the standard beamforming techniques and vector acoustic intensity.

I. Standard vs Single Point Beamforming

The link between the standard approach to beamforming and beamforming using single point measurements is provided by the Taylor series expansion of the acoustic pressure field. The expansion in space about the measurement point $\underline{x}_o$ is:

$$p(\underline{x},t) = p(\underline{x}_o,t) + \nabla p(\underline{x}_o,t)\cdot\Delta\underline{x} + \frac{1}{2}\Delta\underline{x}^T\begin{bmatrix}\text{Matrix of}\\ \text{2nd Derivatives}\end{bmatrix}_{\underline{x}_o}\Delta\underline{x} + \ldots \qquad (1)$$

where $\Delta\underline{x} \equiv \underline{x} - \underline{x}_o$. What Eq. (1) says is that the measurement of acoustic pressure and its higher-order spatial derivatives at a single point in space is equivalent to the measurement of acoustic pressure in a volume about the measurement point. Therefore, the techniques used in beamforming with spatially distributed pressure measurements can be applied to single point measurements of pressure and its spatial derivatives. In particular, since acoustic particle velocity at a given frequency is proportional to the first order spatial gradient of pressure, then high resolution beamforming techniques can be used with simultaneous pressure and particle velocity measurements. Application of minimum variance (Capon) beamforming techniques to these types of single point measurements has been discussed previously [Acoust. Soc. Am. meetings, May, 1992 and May, 1993]. Some examples using real ocean data also will be given in this presentation.

The distance, $\Delta\underline{x}$, that the pressure field can be extrapolated from the measurement point with a given error and with an expansion to a given order is dependent upon the degree of spatial variability of the pressure field. The maximum spatial variability in any direction at a given frequency cannot exceed that determined by the acoustic wavelength, at least for non-evanescent acoustic fields. (Otherwise, the velocity of the energy flow required to support the spatial structure would exceed the speed of sound in the fluid, which is physically impossible). Therefore the effective spatial aperture of a single point array is related to the acoustic wavelength, rather than being determined by a fixed inter-element spacing as with conventional arrays. Thus, single point arrays are frequency-adaptive; their effective aperture decreases with increasing frequency. The result of this property is that the plane wave response (or beampattern) for single point arrays is independent of frequency. This result also implies that no grating lobes exist, i.e., spatial aliasing cannot occur since a sampling in space is not being performed.

The phenomenon of superdirectivity is directly related to the Taylor series expansion of the pressure field. That is, superdirectivity arises for a spatially distributed hydrophone array when the directivity index is maximized as a function of the element weights, and the interelement spacing becomes smaller than half the acoustic wavelength [Pritchard, J. Acoust. Soc. Am., Nov, 1954]. One can show that, as the ratio of the interelement spacing to the acoustic wavelength approaches zero, the weights for a "linear point array" approach the finite difference approximations to the spatial derivatives of pressure given in Eq. (1). The instability that results when the weights become large and of opposite sign can be avoided by the use of alternative transduction methods suggested by the physical interpretation of the spatial derivatives of pressure, e.g., when the measurement of a component of acoustic particle velocity replaces the measurement of pressure at two closely-spaced points.

Note that high resolution beamforming techniques have been applied to single point measurements in other fields, e.g., the estimation of ocean surface gravity wave directional spectra from "pitch-and-roll" buoys [Oltman-Shay and Guza, J. Phys. Oceano., Nov, 1984].

II. Acoustic Intensity and Beamforming

The central quantity in array processing is the data cross spectral (or cross correlation) matrix. For simultaneous acoustic pressure and particle velocity measurements at a single point, the cross-spectral matrix is:

$$[S(f)] \equiv \begin{bmatrix} S_p(f) & z_x S_{px}(f) & z_y S_{py}(f) & z_z S_{pz}(f) \\ \cdots & z_x z_x^* S_x(f) & z_x z_y^* S_{xy}(f) & z_x z_z^* S_{xz}(f) \\ \cdots & \cdots & z_y z_y^* S_y(f) & z_y z_z^* S_{yz}(f) \\ \cdots & \cdots & \cdots & z_z z_z^* S_z(f) \end{bmatrix}$$

where the symbol "*" indicates complex conjugation. Conversion factors, indicated by z_x, z_y, and

z_z, must be defined in order to convert the terms involving particle velocity into units of pressure. Typically, the conversion factors are set equal to the $\rho_o c$ of the medium. The output autospectrum of the single point beamformer under application of the conventional beamforming method is:

$$D_B(f) = \underline{e}^H\ [S(f)]\ \underline{e}$$

where the plane wave steering vector, $\underline{e}$, is given in terms of the direction cosines as:

$$\underline{e}^H = \frac{1}{2}[\ 1\ \cos(\beta_x)\ \cos(\beta_y)\ \cos(\beta_z)\ ]$$

Applying the minimum variance approach introduced by Capon [Proc. IEEE, 1969], the single point beamformer autospectrum becomes [D'Spain *et al*, Oceans 92 Conf., Nov, 1992]:

$$D_C(f) = \left[\underline{e}^H\ [S(f)]^{-1}\ \underline{e}\right]^{1}$$

One part of the link between acoustic intensity and single point beamforming is now clear. That is, the active acoustic intensity at a given frequency is equal to the real part of the cross spectrum between pressure and particle velocity, and the three orthogonal components of this cross spectrum are the off-diagonal elements in the first row and column of the data cross spectral matrix.

Physical interpretations can be provided for the other quantities in the data cross spectral matrix, i.e., its trace is proportional to the total acoustic energy density and the properties of the 3-by-3 particle velocity submatrix are describable in terms of the polarization of acoustic particle motion. Relationships among the various elements of this matrix can be derived from the basic acoustic principles of conservation of mass and momentum [D'Spain *et al*, J. Acoust. Soc. Am. Mar, 1991].

The second part of the link between single point beamforming and acoustic intensity can be obtained by integrating the conventional beamformer output autospectrum weighted by the look direction vector, $D_B(f)\hat{l}$, over all look directions, $\hat{l}$. The result, for purely real conversion factors, is the vector sum of the active intensity components, $C_{pj}(f)$. For example, for zero elevation angle, the integration over azimuth θ gives:

$$\frac{1}{2\pi}\int_0^{2\pi} D_B(f)\hat{l}\ d\beta = z_x C_{px}(f)\hat{x} + z_y C_{py}(f)\hat{y}$$

where $\hat{l} = \cos(\theta)\hat{x} + \sin(\theta)\hat{y}$ and when the conversion factors, z_x, z_y, are purely real.

In summary, beamforming decomposes the sound field into spatial frequency components. The decomposition is performed at first order in the acoustic variables where the principle of superposition is valid. Acoustic intensity, on the other hand, measures the total, net flow of acoustic energy at a single point in space. It accounts for the full interaction of all the wave field's spatial spectral components.

III. MPL's Vertical DIFAR Array

MPL's Swallow floats, freely drifting sensor systems that measure both acoustic pressure and the three components of acoustic particle velocity, thereby permitting the calculation of acoustic intensity, are discussed elsewhere at this meeting [Desharnais and D'Spain, this meeting; also D'Spain *et al*, IEEE J. Ocean Engin., Apr, 1991]. Another MPL system capable of making underwater acoustic intensity measurements is the vertical DIFAR array [Nickles *et al*, Oceans 92 Conf., Nov, 1992]. The array is composed of 16 elements, with an inter-element spacing of 15 m. Each of the elements contains three orthogonally-oriented geophones to measure acoustic particle velocity and a hydrophone to measure pressure. The array elements also contain a flux-gate compass for measuring the orientation of the horizontal geophones, a programmable high frequency data acquisition for acoustic element localization, a preamp with programmable gain from 0 to 120 dB, and a 16-bit A/D converter that provides additional dynamic range. Data collected by this array will be used to illustrate some of the preceeding points of this presentation.

IV. Extension of Single Point Beamforming and Energetics to Higher Order

A physical interpretation of the term at second order in the Taylor series expansion in Eq. (1) can be made by realizing that in an acoustic field in an otherwise stationary fluid, the spatial gradient of the acoustic particle velocity is equal to the acoustic rate of strain. Referring to the second rank strain rate tensor as $\dot{\varepsilon}$, the following relation is true:

$$p\dot{\varepsilon} = \nabla(p\underline{v}) + \frac{\partial}{\partial t}\left\{\frac{1}{2}\rho_o(\underline{v})(\underline{v})\right\}$$

It can be shown that the trace of $p\dot{\varepsilon}$ yields the conservation of acoustic energy equation. For stationary, stochastic fields, in the frequency domain:

$$S_{p\dot{\varepsilon}}(f) = \nabla \underline{C}_{p\underline{v}}(f)$$

Therefore, the cross spectrum between the acoustic pressure and the acoustic rate of strain provides information on the local spatial heterogeneity of the active acoustic intensity.

The second order Taylor series term also improves the performance of the single point beamformer. Whereas the beamformer at first order has a maximum directivity index (DI) of 6 dB, a main lobe width of 105 deg, and can distinguish two sources in azimuth, the second order beamformer has a maximum DI of 9.5 dB, a main lobe width of 65 deg, and can distinguish four distinct sources in azimuth.

Proposed modifications to MPL's vertical DIFAR array to permit approximate measurements of the acoustic rate of strain will be presented.

V. Summary Recommendations

With simultaneous measurements of acoustic pressure and particle velocity, single point beamforming is most applicable in cases where the directionality of a source of interest is to be estimated in the presence of another loud, interfering source. The use of the active acoustic intensity vector is appropriate when the number of sources is large, as in ocean ambient noise studies.

References

A complete set of references for this presentation is available.

ACOUSTIC INTENSITY MEASUREMENTS WITH SWALLOW FLOATS

Francine Desharnais[1] and G.L. D'Spain[2]
[1]SACLANTCEN, Viale S. Bartolomeo 400, 19138 La Spezia, Italy
[2]Marine Physical Laboratory, 291 Rosecrans St, San Diego, CA 92152-6400, U.S.A.

INTRODUCTION

A Swallow float is a deep water drifting float that contains four acoustic sensors: one hydrophone that measures acoustic pressure, and three orthogonal geophones that measure particle velocity in three dimensions. Each unit is free-floating, and also contains: a compass to derive the true heading of the float, all the data recording hardware, a ballast to control the float depth, and a high-frequency transducer for communication between the float and a nearby receiver. More information on the floats themselves can be found in D'Spain et al. [1]

The advantages presented by the Swallow floats are two-fold: first, the free floating ability reduces the flow noise to a minimum, producing very good quality data down to 0.5 Hz; second, the information from the four sensors can be combined to give measurements of vectorial acoustic intensity. Only one float is therefore needed to obtain both the magnitude and the directionality of the ambient noise field at one point in space.

ACOUSTIC INTENSITY

With the basic assumption that the acoustic field is stationary and ergotic, the following frequency domain representation of the mean acoustic energy density per frequency can be derived [2]:

$$E_{\text{tot}}(\mathbf{x},f)=\frac{1}{2}\frac{1}{\kappa_s(\mathbf{x})}\left(\left[\rho_0(\mathbf{x})c(\mathbf{x})\right]^2\sum_{j=1}^{3}S_{v_j}(\mathbf{x},f)+S_p(\mathbf{x},f)\right) \quad (1)$$

where $\rho_0(\mathbf{x})$ = ambient density;
$c(\mathbf{x})$ = sound speed;
$\kappa_s(\mathbf{x})$ = adiabatic incompressibility;
$S_{v_j}(\mathbf{x},f)$ = one-sided spectral density function of geophone data;
$S_p(\mathbf{x},f)$ = one-sided spectral density function of hydrophone data.

The hydrophone autospectrum in equation (1) is proportional to the acoustic potential energy density spectrum, and the sum of the three geophone autospectra is proportional to the acoustic kinetic energy density spectrum. Also, the three-geophone spectrum can be used to derive an equivalent pressure autospectrum. In areas where the basic assumption is valid (as in the deep water column), both spectra should be equal.

The one-sided cross-spectral density function between the pressure signal $p(\mathbf{x},t)$ and the geophone velocity signals $\mathbf{v}(\mathbf{x},t)$ is defined as:

$$\mathbf{S}_{pv}(\mathbf{x},f)=\mathbf{C}_{pv}(\mathbf{x},f)-i\mathbf{Q}_{pv}(\mathbf{x},f) \quad (2)$$

where the real part $\mathbf{C}_{pv}(\mathbf{x},f)$ is the coincident spectrum and the imaginary part $\mathbf{Q}_{pv}(\mathbf{x},f)$ is the quadrature spectrum. The coincident spectrum (or active acoustic intensity) gives the direction and magnitude of the mean energy flow. The quadrature spectrum (or reactive acoustic intensity) represents the small scale spatial heterogeneity of the sound field at position $\mathbf{x}$.

IONEX 92 EXPERIMENT

In June 1992 a joint sea experiment took place (IONEX 92) between SACLANTCEN (Italy) and MPL (Marine Physical Laboratory, Scripps Institution of Oceanography, USA) on board NRV Alliance. A group of approximately ten MPL Swallow floats was deployed on two occasions in the Mediterranean Sea. For each deployment, six or eight floats were deployed at depths between 200 and 1500 m, and three floats were moored at the bottom (3000 m).

A typical ambient noise measurement for the band of 0 to 25 Hz is shown in Figure 1 for a selected three-minute period of the first deployment. The dashed line is the pressure spectrum derived from the hydrophone data; the solid line is an equivalent pressure spectrum computed from the three component geophone data (both spectra have units of dB // µPa²/Hz). The deep water acoustic field being fairly homogeneous, the two spectra are about equivalent (the differences above 20 Hz are probably due to small calibration errors). Below 0.5 Hz, the geophone data is potentially contaminated by mechanical noise. The associated active intensity measurements are shown in Figure 2, in a true geographical plane. An intensity vector is associated to each frequency bin. The direction is given in degrees relative to true north. The magnitude in dB // µWatt/m²/Hz is obtained by measuring the length of each individual vector along the y axis, starting at -80 dB. The vectors align themselves with the resultant acoustic flow at any particular frequency.

Above 10 Hz, the acoustic field is dominated by shipping noise. A main shipping lane is located south of the float field, and several ship lines can be seen (e.g. harmonic lines from a ship NW from the float field are shown with arrows in Figure 1). The background noise levels in that frequency range are approximately constant (apart from the individual ship lines) at around 85 dB, which is in agreement with noise levels of areas with high shipping traffic [3] The intensity plot also shows that the resultant acoustic flow at these frequencies is coming generally from the south.

Around 4-5 Hz, the noise levels display a minimum at approximately 75 dB, which is often observed at these frequencies. Below 5 Hz, the noise is dominated by wind noise and wave-wave interaction effects. During the second deployment, the wind increased from light to 17 kn (9 m/s), and the background levels below 4 Hz increased by approximately 10 dB during the same period. This increment also agrees with other published data [4, 5].

During the first float deployment, the winds were light throughout, and a cycle was visible both in the hydrophone spectral amplitude in the frequency band of 1-3 Hz (Figure 3, middle cluster), and in the amplitude of the vertical geophone signal (Figure 3, lower cluster, scale on the left of the plot). The period of the cycle was in the order of 12h, although only 20 hrs of data were collected, which increases the measurement uncertainty. Figure 3 also includes the bearing of the horizontal acoustic intensity vector (upper cluster, scale on the right of the plot) for the same float. The angle indicates that the acoustic flow generally comes from the Italian coast (NW) during the loud part of the cycle, but from the Mediterranean deep water area (SW) during the quiet part of the cycle, although there is an offset in time between the amplitude cycle and the directionality cycle. Notice that the directionality data is also much more scattered. Moreover, the cycle was present in the data of all floats, with a slightly higher amplitude in the bottom float data. Similar cycles have been observed by Bourke and Parsons at frequencies of 5, 10 and 32 Hz [6]; they associated the cycle with tidal forces. Their measurements however were made on ice floes at shallow water sites in the Arctic; also their measured noise may have been generated by tidal-induced stresses acting on ice floes. Other data from other deep water sites are at present investigated for similar cycles at frequencies below 5 Hz.

Another interesting effect could be seen during the experiment. To the south-west of the experimental site there is a subduction front area, or an area particularly rich in seismic occurrences. Throughout both deployments, a succession of transient signals were recorded by the floats. The signals were characterized by an increase in spectral levels below 5-10 Hz, and the same directionality for the active intensity vectors in that frequency band. Of all the catalogued events (>50), ≈95% were coming from the subduction area. Some of these events were also recorded by nearby seismic land stations in Italy and Greece. A comparison of the time series from the Swallow floats and some land stations demonstrate that Swallow floats have a higher sensitivity than land stations to earthquakes occurring below the sea bottom.

CONCLUSIONS

Acoustic intensity measurements have been made with several Swallow floats. The floats give an accurate measurement of ambient noise, both in amplitude and in directionality. They can be used to obtain intensity vectors from discrete sources such as ship lines, or more broadband sources like small earthquakes, with similar accuracy.

Some particular features of the sound field were measured. An increase in wind speed from 0 to 9 m/s increased by 10 dB the background spectral levels below 4 Hz. The effect was not observed above 4 Hz since the field is dominated by shipping noise at these frequencies.

The floats also detected a very low frequency signal in the band of approximately 1 to 3 Hz, which is potentially linked to tidal forces. Similar cycles have been observed at higher frequencies in shallow water sites covered with ice floes. The noise generation mechanisms in both environments might be different, and the tidal cycle hypothesis is still under investigation for the deep water sites.

Finally, the Swallow floats were used for the detection of seismic events from an active subduction front area located nearby. It was found that the Swallow floats have a greater sensitivity to these small earthquakes than the land-based seismic stations in the area.

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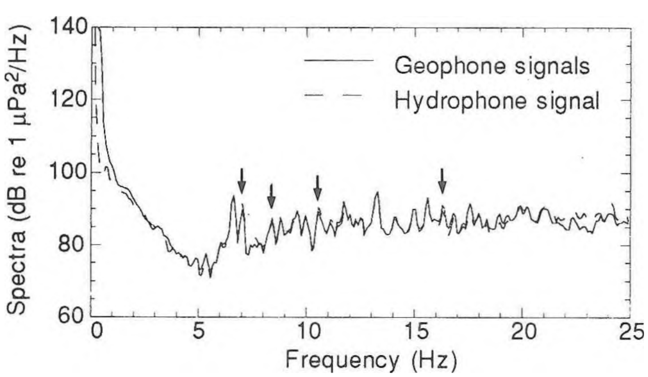

Figure 1. Ambient noise pressure spectrum from the hydrophone (dashed line), and equivalent spectrum from the three geophones (solid line). The time average is three minutes.

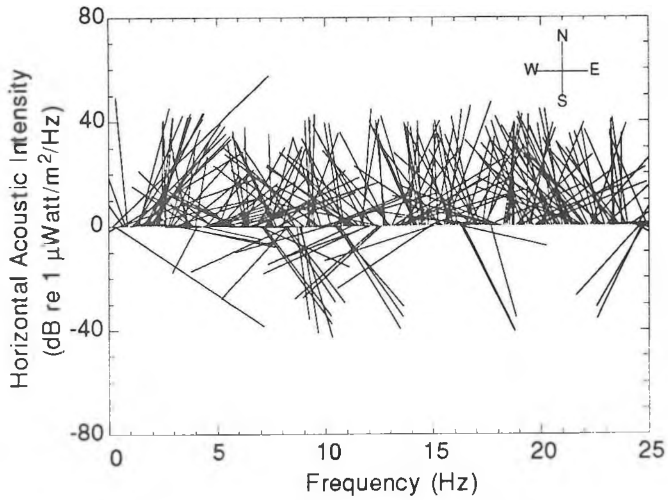

Figure 2. Active acoustic intensity measurement for the same time period as in Figure 1. Each vector is aligned with the resultant acoustic flow at a specific frequency bin (true geographic plane). The magnitude is obtained by measuring the length of each vector along the y axis, starting at -80 dB.

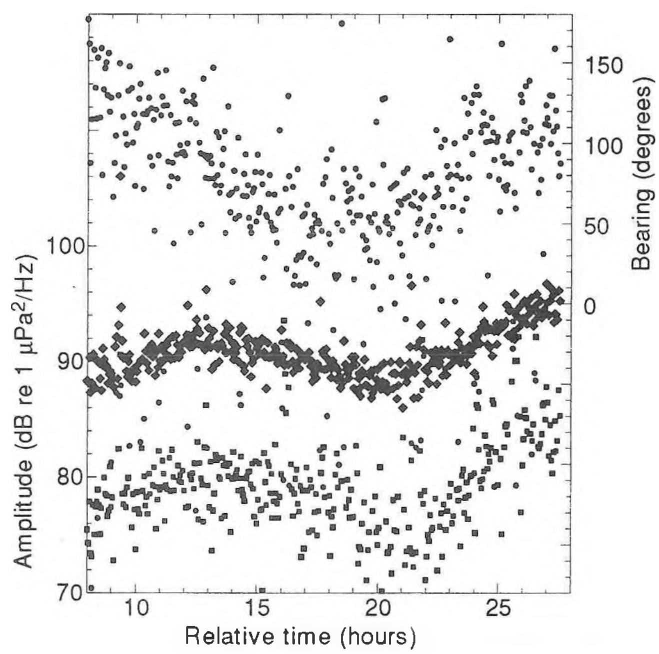

Figure 3. Left scale: middle cluster: hydrophone spectral amplitude; lower cluster: vertical geophone spectral amplitude. Right scale: upper cluster: bearing of the horizontal acoustic intensity vector (direction opposite from noise source). The frequency band is from 2 to 2.5 Hz.

MINIMIZING INSTRUMENT EFFECTS IN AN OCEAN BOTTOM SEISMOMETER

D. J. Dodds
GeoAcoustics Inc., Box 790, Aurora, Ontario, Canada L4G 4J9
D. M. F. Chapman, J. C. Osler, and W. C. Risley
Defence Research Establishment Atlantic, Box 1012, Dartmouth, Nova Scotia, Canada B2Y 3Z7

1. Introduction

Ocean acousticians have traditionally made measurements of the pressure component of the sound field. An earlier paper[1] described initial tests by one of the authours on an existing ocean bottom seismometer (OBS) which uses velocity sensors (geophones). Geophysicists have used OBS instruments primarily to time discrete seismic events. This paper describes a new OBS designed to measure sea floor vibration amplitudes, down to the ambient noise level, in the 1-50 Hz range. The design goal was to minimize both the self noise (an additive effect), and the multiplicative effect of the transfer function which relates sea floor motion to the motion recorded by the OBS.

2. Self noise of an OBS

A geophone consists of a "seismic mass" suspended in a geophone case. Relative motion between them is detected by a magnet and coil. Mechanical energy is coupled from the sea floor motion into the mass-spring system of the geophone, transformed into electrical energy by the magnet and coil, and input to an amplifier. Electrical energy is also generated by noise sources: thermal ("Johnson") noise in the resistance of the wire forming the coil, Brownian noise (also a form of thermal noise) in the mechanical parts of the geophone, and electrical noise in the amplifier.[2] Thermal noise is a fundamental phenomenon which cannot easily be reduced. So, to achieve a specified signal-to-noise ratio, some minimum mechanical power must be coupled from the sea floor motion into the seismic mass and transformed into electrical power. This puts a lower bound on the seismic mass, magnet mass, and coil mass, and an upper bound on resonant frequency. Geophones with a resonant frequency <10 Hz must be precisely leveled to function properly, which adds to the mass of the geophone system. Consequently, a self-noise specification puts a lower bound on the geophone mass.

3. Coupling to the Sea Floor

For practical reasons, an OBS must be recoverable, so it must sit on the water side of the sediment-water interface, and be denser than water. When the sea floor moves vertically, the system will act like a mass (the mass of the OBS, less the mass of water it displaces) on a spring (representing sea floor compliance). Near to, and above the resonant frequency of this system, the relative motion between the sea floor and OBS may be significant compared to the motion of the sea floor in the absence of the OBS. When the sea floor moves horizontally, there are more complicated effects. Horizontal motions are discontinuous across the sea floor-water interface. Under horizontal movement of the sea floor alone, the system will again act like a mass on a spring. However, the mass of the OBS is augmented by the effective mass of its entrained water, which increases the relative motion between the sea floor and the OBS by comparison to the case of vertical motion. The OBS may rock (since its center of mass must lie above the sea floor), and horizontal motion of the water alone can excite motion of the OBS. For these reasons, coupling of the OBS to the sea floor can be expected to be poorer in the case of horizontal motion than in the case of vertical motion. Analyses in the literature show that the ideal OBS package should have a low profile, substantial anchor area, a small mass, and a density near that of the surrounding water or sediment.[3,4] A further requirement is that the OBS be effectively rigid, so that the geophones follow the motion of the anchor.

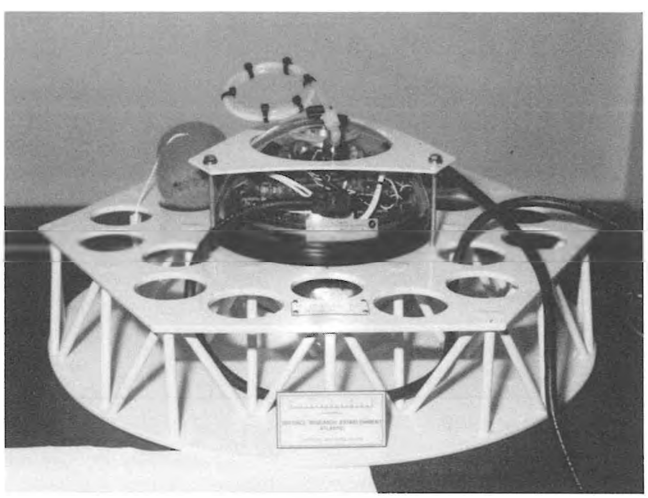

Fig. 1. The geophone package. The spherical glass pressure vessel is supported in the middle of the aluminum anchor disk.

4. Design of the OBS

The OBS was intended to have the capability to measure ambient noise. Its geophones are a triad of the smallest commercially available models which give a predicted self-noise spectrum 20 dB below typical ambient noise in the 1-50 Hz range. They have a resonant frequency of 4.5 Hz, a seismic mass of 52 g, a total mass of 0.35 kg (per geophone), and require leveling to an accuracy of about 1°. Their outputs are digitized by 24-bit delta-sigma analog-to-digital converters, giving a dynamic range of more than 120 dB. The function of the remainder of the OBS is to level the geophone triad, protect it from ambient pressure, couple it to the sea floor, and transmit its outputs in digital form.

To minimize the mass coupled to the sea floor, the components of the system related to deployment and recovery are installed on a frame connected to the geophone package by cords which are designed to lie slack when the OBS is on the sea floor. The geophone package with its anchor disk is shown in Fig. 1. The anchor is an aluminum disk 0.6 m in diameter, which is large enough to prevent excessive rocking of the OBS in response to horizontal excitation. To achieve rigidity, the disk is stiffened by a space frame based on a regular tetrahedral cell. (A simple disk of sufficient thickness to achieve rigidity at 50 Hz would be so massive as to impair coupling to the sea floor.) The 0.25 m spherical glass pressure vessel (which is rated for operation to 6,700 m depth) sits in a cradle in the anchor disk.

The geophones are mounted on a block which rests on the inner surface of the pressure vessel. The leveling mechanism functions by raising the geophone block so that it is freely suspended from a point at the center of the spherical pressure vessel. When the geophone block has settled in a level attitude, it is lowered again to rest firmly on the inside of the pressure vessel. The geophones are arranged in the Gal'perin configuration, meaning that they are mutually perpendicular, but each is inclined equally to the vertical direction. This results in a compact, symmetrical configuration which fits conveniently within the spherical pressure vessel. The three components of motion measured by the geophones can be resolved into horizontal and vertical components by the digital electronics within the pressure vessel.

Table 1 shows the mass of various parts of the OBS. The self-noise requirement puts a lower bound on the geophone mass, but that is only a small part of the total mass coupled to the sea floor. The leveling mechanism contributes significant mass, and it also occupies a large volume which contributes to the mass of the pressure vessel. A more integrated design (e.g. integrating the leveling mechanism into the geophones) could further reduce the mass, thereby improving coupling to the sea floor.

Table 1. Mass of OBS components	
Seismic mass of geophones (3)	0.16 kg
Geophones (3)	1.1 kg
Leveling mechanism, electronics	3 kg
Pressure vessel	4 kg
Anchor disk	12 kg
Total mass coupled to the sea floor	20 kg
Deployment/recovery system	≈200 kg

5. Measuring Sea Floor Coupling

To provide a means of evaluating the quality of coupling achieved, a small vibrator (consisting of a miniature dc motor driving an eccentric mass of about 1 g) is mounted to the inside of the pressure vessel. The motor axis is horizontal, so that the excitation has both vertical and horizontal components. The motor can be driven at frequencies from 2 Hz to 50 Hz. The vibrator is driven at selected frequencies with the OBS on the sea floor, and the response is compared to the response with the OBS suspended in water. If the OBS is stiffly coupled to the sea floor, its motion will be less than when it is suspended in water.

6. Results

Typical results of a coupling test are shown in Fig. 2. The "suspended" response show a slope of about 20 dB/decade, indicating that the OBS is constrained by inertia, as would be expected. In the horizontal axis, the "grounded" response is less than the "suspended" response, showing good coupling, up to about 8 Hz. A resonant peak occurs at 10 Hz, and above 20 Hz there is little difference between the "grounded" and "suspended" cases, indicating that the OBS is poorly coupled to horizontal sea floor motion at these frequencies. In the vertical axis, the "suspended" response is substantially less than in the horizontal axis because the anchor disk entrains a large mass of water (about 100 kg) in vertical motion. A resonant peak occurs in the "grounded" response at 20 Hz. The effect of the entrained water mass must be considered when evaluating the results. Without the entrained mass, the vertical "suspended" response would be similar to the horizontal "suspended" response, and therefore greater than the vertical "grounded" response. Thus good coupling is achieved for vertical motion up to 50 Hz. Results have been obtained at several sites, and show significant variations in the quality of coupling depending on sea floor sediment type. It may be possible to improve horizontal coupling by attaching vertical vanes to the bottom of the anchor disk.

7. Conclusions

This OBS uses coil-and-magnet geophones and has a noise floor below typical ambient noise levels in the 1-50 Hz range. The design achieves good coupling to vertical sea floor motion, but the quality of horizontal coupling depends on sea floor characteristics. Horizontal coupling may be improved by vertical vanes attached to the bottom of the OBS anchor disk.

8. Acknowledgment

This work was funded by the Government of Canada through Defence Research Establishment Atlantic.

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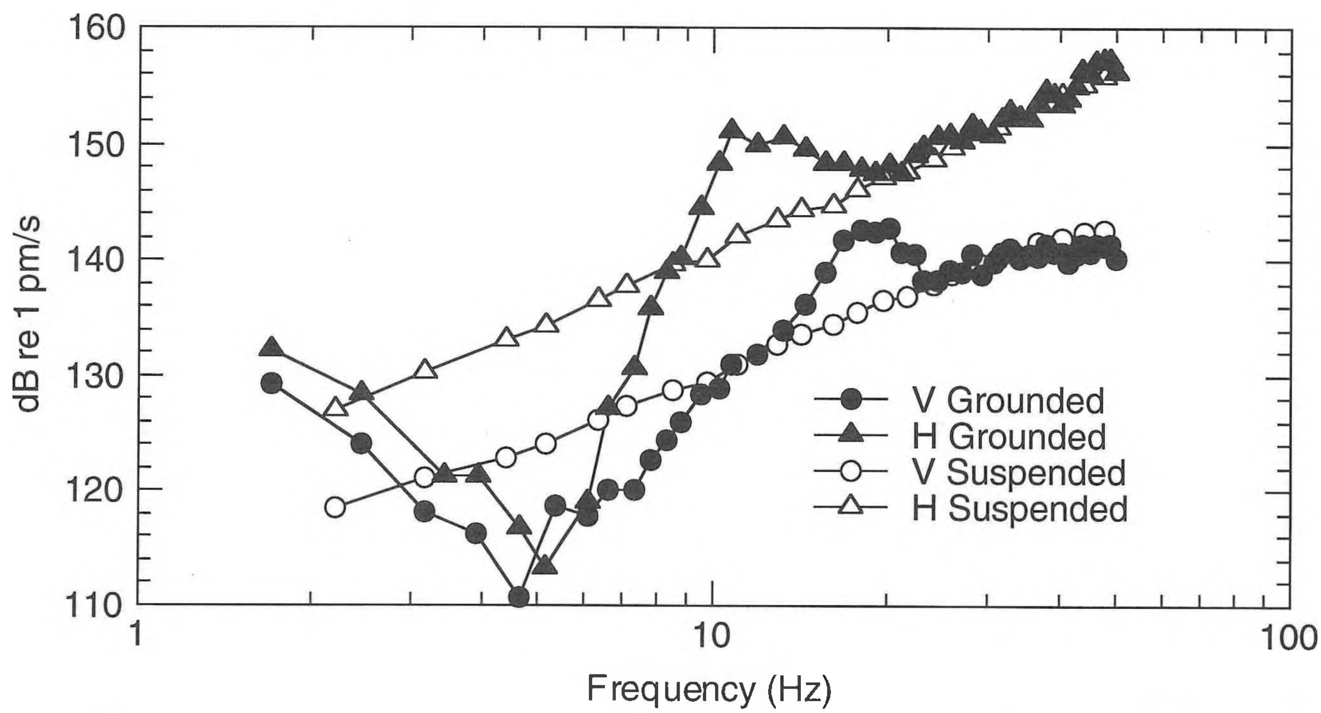

Fig. 2. Results of a coupling test. The values shown are the vertical and horizontal amplitude response of the OBS to an integral vibrator in two conditions: suspended in water, and grounded on a typical sea floor.

Seismo-acoustic measurements using an ocean bottom seismometer in the high Arctic

S. E. Dosso, G. H. Brooke and D. G. Baade

Defence Research Establishment Pacific

FMO Victoria B.C., Canada V0S 1B0

INTRODUCTION

Geoacoustic properties of the upper tens of metres of ocean bottom sediments can significantly influence low-frequency acoustic propagation in the ocean. Therefore, knowledge of these properties is required for reliable propagation modelling and matched-field processing. To date, however, few measurements of geoacoustic properties have been reported for the high Arctic at sufficient resolution for underwater acoustic applications. This paper describes a high-resolution seismic experiment designed to measure ocean-bottom compressional and shear properties on the continental shelf of the Lincoln Sea, north of Ellesmere Island, Canada.

EXPERIMENT

The seismic experiment was carried out at a research camp located on a multi-year ice floe in the polar ice pack. The geometry of the experiment is shown in Fig. 1. A specially-designed three-component ocean bottom seismometer (OBS) was deployed by melting a hole through the sea ice with a hot-water drill, and lowering the unit by cable to the sea floor. The OBS signals were transmitted to the surface via the cable and recorded on a multi-channel digital seismograph at a sampling rate of 4 kHz. Sources consisted of explosive charges detonated on the sea-floor. A total of 17 sources of a variety of sizes (0.5–27 kg) were deployed through nine holes in the ice at ranges of 35 to 900 m from the OBS. Fig. 1 also shows the acoustic and seismic waves of principal interest: the direct and reflected waterborne waves, the head wave (refracted compressional wave), and seismic interface (Scholte) wave.

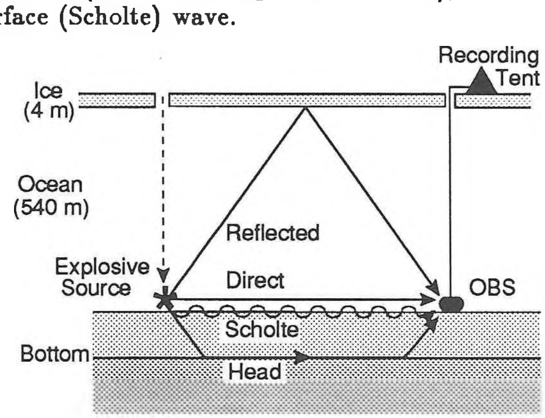

Fig. 1. Experiment geometry.

ANALYSIS

Fig. 2 shows the recorded vertical-component seismograms plotted in the format of a seismic section. Refracted, direct, and first and second surface-reflected arrivals are evident on all seismograms (labelled P, D, R and RR respectively), and good examples of Scholte waves appear as the low-frequency arrivals on seismograms recorded for ranges less than 400 m. First-break arrival time picks were made for all seismograms. These fell on two well-defined line segments indicating that the refracted arrivals can be interpreted as head waves, and that the data set is well suited to interpretation by the classical slope-intercept method [1]. Fig. 3 shows the two-layer compressional-speed model determined by this analysis, with the envelope of the model parameter uncertainties indicated by dotted lines.

Shear properties of ocean sediments are perhaps most readily determined from the propagation characteristics of Scholte waves [2, 3]. The Scholte-wave speed is closely related to the shear speed over a depth of one to two wavelengths into the sea-bed, but is relatively insensitive to the compressional properties. Since long-wavelength components of the Scholte wave penetrate a greater depth into the bottom than short wavelengths, the dispersion characteristics of this wave provide information about the shear-speed profile of the upper sediments. Fig. 4 illustrates the dispersion of vertical and horizontal seismograms for a source at 82-m range in terms of Gabor diagrams, which show contours of energy, in arbitrary decibels (maximum level: 99 dB), as a function of

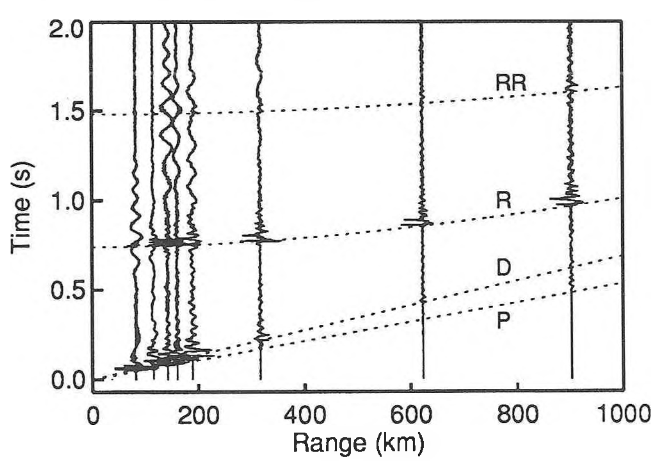

Fig. 2. Seismic section.

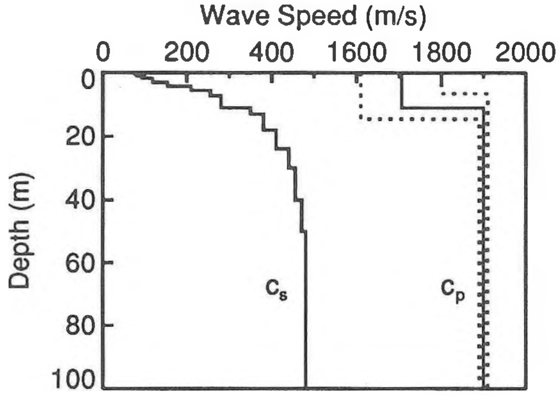

Fig. 3. Shear (c_s) and compressional (c_p) speed models (note the 1000-m/s discontinuity in the wave-speed axis at 500 m/s).

group speed and frequency. The dispersion properties shown in these diagrams are representative of all the recorded seismograms. The Gabor diagram for the horizontal (y) seismogram indicates that the energy is propagating in three discrete modes: M_0, the fundamental (Scholte) mode, and M_1 and M_2, the first and second shear modes. Shear modes appeared most strongly on the horizontal seismograms and were evident only for short-range shots (< 140 m). The z-component Gabor diagram shows only M_0 and M_1. At all ranges it was found that the Scholte mode was more clearly defined for z than y.

A shear-speed model for the bottom sediments was constructed by matching the observed dispersion characteristics of the Scholte and shear modes [2, 3]. The solid white lines superimposed on the energy contours in Fig. 4 represent modal dispersion curves computed for

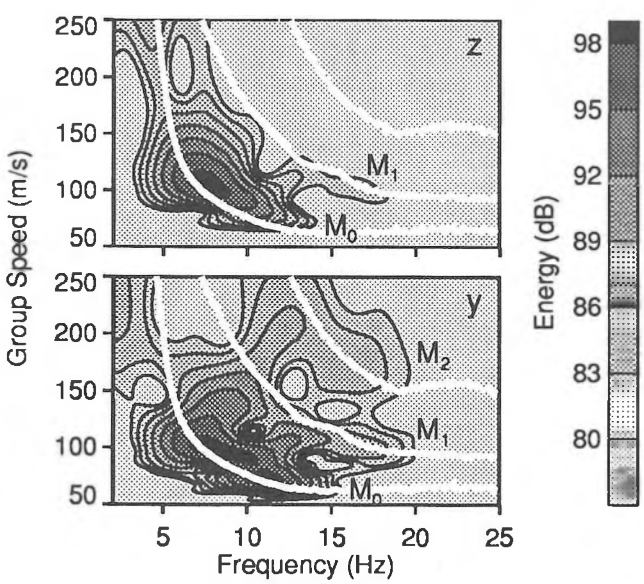

Fig. 4. Gabor diagrams for vertical (z) and horizontal (y) seismograms recorded for a source at 82-m range.

the shear-speed profile shown in Fig. 3 using the full-wave numerical propagation model SAFARI [4]. This profile was determined by systematically varying the parameters of a layered model until good agreement was achieved between the computed dispersion curves and the measured Gabor diagrams. Considerable care was taken to ensure that the shear-speed model correctly reproduced the dispersion characteristics of the seismograms recorded at all ranges.

The (volume-average) attenuation of bottom sediments was estimated from an analysis of the exponential decay of signal amplitude A with range r:

$$A(r) = A(0)\, S(r)\, \exp[-\alpha\, r], \tag{1}$$

where α is the attenuation coefficient and $S(r)$ represents the geometrical spreading appropriate to the particular wave type. According to (1), the attenuation coefficient can be determined as the slope of a plot of $\log_e[A(r)/S(r)]$ as a function of r; by transforming to the frequency domain, this procedure can be carried out at a number of frequencies to investigate the frequency dependence of the attenuation. This procedure was used to estimate frequency-dependent compressional and shear attenuation coefficients, $\alpha_p = 0.1$ dB/km/Hz and $\alpha_s = 5$ dB/km/Hz, using head-wave and Scholte-wave arrivals respectively

SUMMARY

A high-resolution seismic experiment was carried out in the Canadian high Arctic to measure ocean-bottom geoacoustic properties. Compressional- and shear-speed models and (frequency-dependent) compressional and shear attenuation coefficients were determined. All of the geoacoustic parameters are in good agreement with accepted values [5].

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TUNED MASS DAMPERS FOR HIGH RISE BUILDINGS - A CASE STUDY

Mal P. Sacks, Ph.D., P. ENG.

TACET ENGINEERING LIMITED
111 Ava Road, Toronto, ON M6C 1W2

John C. Swallow, M.A.Sc., P.ENG.

JOHN SWALLOW ASSOCIATES
250 Galaxy Road, Etobicoke, ON M9W 5R8

1. INTRODUCTION

The Chifley Tower in Sydney Australia is a 46 story, 275 meter high steel structure, which is rare in Australia. Being steel it is lighter and more flexible than an all-concrete design. The building is wedge shaped with curved external surfaces and numerous setbacks and transfers. The design wind condition was 155 km per hour. These winds could drive the building in any one of three coupled translation and torsion vibration modes whose natural periods ranged from 4 seconds to 6 seconds.

The predicted horizontal acceleration on the upper floors was 0.3 m/second^2 and 0.5 m/second^2 in 5 and 50 year return. winds. The horizontal displacement in the 50 year wind was 0.47 meters. The assumed damping of the structure was approximately 1.5%. Calculations indicated a 2% modal mass TMD with 14% of critical internal damping would result in 0.15 and 0.26 m/second^2 accelerations in 5 and 50 in return winds. The effective damping of the structure would be 4.2%. The function of the TMD is to provide greater comfort for building occupants. It is not required for the structural integrity of the building.

The TMD is required to operate in any direction but since the torsion mode was coupled to the translational modes no twisting motion of the TMD is required.

A passive TMD was chosen because of its inherent simplicity and reliability. The passive TMD requires no external power and its operation is initiated by the wind-induced motion of the building. (*See Figure 1*).

2. DESIGN CONSIDERATIONS

The TMD must be tunable to different frequencies so that it can be matched to the finished building natural frequency and also so that it can be adjusted as the natural frequency changes over the life of the building. For a pendulum type TMD this requires changing the effective length of the cables. As it is difficult to raise or lower a large mass block or to raise or lower the pendulum upper support points, the design features a cable supported pendulum with the upper cable support point fixed, the lower support point attached to the mass block and a moveable tuning frame which fixes the cable laterally, much as a violin string is fixed by the players finger.

The mass, at 395 tones is the largest in the world for a single TMD. The TMD mass is set by the desired damping to be added by the TMD, the building model mass and the maximum permissible mass block amplitude. To provide a compact design the mass block was made 4 x 4 x 3 meters high and consisted of a simple box of 50 mm steel plate filled with 100 mm thick plates. Several layers of plates were welded to the sides of the box resulting in a compact and rigid structure. The mass was required to have an amplitude (swing) of 1.5 meters with a working range of 0.9 meters.

Maximum damping force required was 1150 kilo newtons to be applied over the full 3.0 meter excursion of the mass. This total force was provided by 8 hydraulic cylinders, mounted inclined at 45° which reduced the stroke requirement and increased the stability. For redundancy, 8 hydraulic cylinders were used, 2 on each side of the TMD. Each was required to dissipate 60 kilo watts peak power, 15 kilo watts continuous. To achieve this, each cylinder was provided with an oil-to-air heat exchanger and approximately 200 liters of hydraulic oil for thermal capacity.

Smaller TMD's can be tested with a simple pull-and-release but the loads in this case were too large. Consequently, a powered hydraulic system was designed to allow the damper to be driven at its natural frequency. To reduce the power requirements a system of valving to bypass the energy dissipation cylinders was provided.

3. COMPONENTS

The mass block was made as a simple box of 50 mm steel plates open at the bottom and the top except for a small shelf angle on the bottom. The mass was constructed by adding 100 mm plates and welding in the bottom two layers and top two layers so as to make a rigid and virtually solid steel assembly. The welding was done on site as the components had to be kept within the capacity of the construction crane.

The mass block is supported by steel cables which provided a simpler flexible support than the bearings and the gimbals used in other designs. Special radiused housings were used top and bottom to allow the cables to bend without stress concentration. Eight cables were used instead of four for redundant support - in the four cable design if one fails, two will take the entire load. In the eight cable design, if one fails the load is still shared by seven. The tuning frame was a simple steel beam structure supported by four building columns. It was designed to be raised and lowered at 300 mm steps which corresponded to a change of cable length 300 mm and 0.1 second change in natural period. The tuning frame was designed for loads of 400 kilo newtons laterally and has special fittings for the cable to pass through without binding on the cable or causing stress concentration.

For possible over-travel a snuber system was provided which consisted of a post attached to the bottom of the mass block which was designed to impact a circular snuber assembly. The snuber assembly consisted of 80-30 mm rods such that the rods would plastically deform.

The damping force was provided by hydraulic cylinders forming part of the energy dissipation system. Each cylinder had 4" rod couplers, 6" bore, and was 3.8 meters long with a 1.8 meter stroke. Each end of the cylinder was mounted with a ball joint to allow damper motion in any direction.

4. HYDRAULIC SYSTEM

An integrated hydraulic system was designed to provide several functions. The energy dissipation system is passive and is interlocked to the powered hydraulic test system. The power unit was also used to operate eight jacks for lifting the mass block and also a maintenance test system located in the building. The eight dissipative cylinders are permanently connected between mass block and building floor. Lock-out valves allow this system to be shut down and locked in place so that motion of the mass block is prevented. The hydraulic test system consists of two hydraulic cylinders connected to two perpendicular sides of the mass block and which can be operated either individually or simultaneously by the hydraulic system power unit. This allows driving the mass block in several different directions for testing. While the mass block is being driven, the dissipation system circuitry is bypassed so that minimal energy is dissipated thus reducing the size of the hydraulic power supply. Once the mass block is driven to an appropriate amplitude a switch over occurs in which the dissipation system is brought back into play and the hydraulic test system is bypassed. After the switch over the TMD becomes a passive damped system whose properties can be determined from the decay of the vibration amplitude time history.

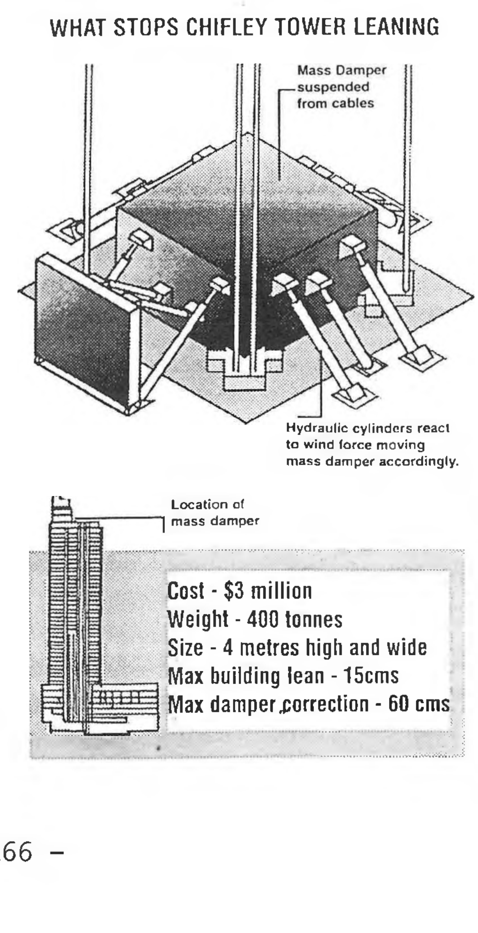
The maintenance test system consists of a frame into which one of the dissipation cylinders can be mounted and mechanically driven at various stroke velocities. The rod and cap ends of the dissipation cylinders are connected by a hydraulic circuit containing a flow control valve at each end. By setting the flow control valves to various positions, and then measuring stroke velocity and cylinder pressure, the effective flow (or orifice) coefficient of the hydraulic circuit could be calculated. The relationship between orifice coefficient and TMD equivalent linear damping, at the various control valve settings, provides the over all calibration of the energy dissipation system.

The maintenance test system is a permanent on-site accessory of the TMD. This system proved invaluable during commissioning as it allowed for on-site diagnosis and replacement of out-of-spec components.

5. CONCLUSIONS

The TMD was commissioned in February 1994 and is now fully operational. During commissioning tests the TMD was shown to increase the damping ratio of the building by approximately 2% to 3%. The TMD is functioning as expected and will produce significant reductions in wind-induced building vibration amplitudes.

Figure 1. The TMD as explained to Sydney residents. Froi the Sydney Morning Herald, 12 Feb 94.



Multi-Celled Liquid Dampers to Eliminate Annoying Floor Vibrations
by

Ronald L. Shope and Thomas M. Murray
Virginia Polytechnic Institute and State University
Blacksburg, Virginia USA

INTRODUCTION

It is often quite difficult to eliminate annoying floor motion that results from human activity. Many past methods have proven to be either ineffective or economically and architecturally prohibiting. Although tuned mass dampers (TMDs) have been used to control floor movement, their success has been limited. This is primarily because peak floor vibration amplitudes are usually less than 1 mm (0.040 in.) and dampers require a much greater amplitude to dissipate energy. This paper presents a multi-celled liquid damping device supported on a steel plate together with the necessary mass that appears to be an effective solution.

BASIC CONCEPTS

A TMD consists of an additional mass attached to the structure by a parallel spring and damper. This acts as a single degree of freedom system and its natural frequency is tuned so that it matches the frequency of the original structure. TMDs control a structure by the reaction at the spring and damper. This reaction is a time dependent force acting in the opposite direction of structure movement. In the case of floor vibrations, there routinely exists more than one mode which is excited by human activity. Therefore, in order to eliminate all annoying vibrations, it is necessary to provide multiple TMDs.

TEST FLOOR

The test floor used in this study was designed and constructed to simulate floors commonly found in office and retail buildings. The floor consists of a 3-1/2 in. concrete slab on metal deck supported by open web steel joists. The joist span is 25 ft. and the spacing is 30 in. The width of the floor is 15 ft.

PROTOTYPE DAMPERS

The prototype damper, shown in Fig. 1, uses a 12 in. wide steel plate as the spring. The length of the plate ranges from 6 to 8 ft. and the thickness ranges from 5/8 to 1 in. The plate is supported at each end by an angle which simulates a pinned connection. The angle is positioned on a steel bearing pad which rests on the floor. Two stacks of 1/2 in. thick steel plates serve as the additional mass. This allows the mass to be adjusted in 10 lb. increments. Damping is provided by multi-celled liquid filled bladders confined in two rigid containers. This deviates from conventional TMDs which have a dashpot or damping element connecting the additional mass to the original structure.

TEST METHODS AND TUNING

The free vibration response of the floor was measured by conducting an impact test commonly known as a Heel Drop test. The standard Heel Drop test is performed by first standing on the balls of one's feet then leaning back allowing the heels to impact the floor. An accelerometer is placed in close proximity to the impact. The impact triggers the data acquisition system and the floor response is recorded for eight seconds. A Fast-Fourier Transform (FFT) is then performed on the acceleration history to extract the frequency response spectrum of the structure. In addition to the Heel Drop tests, acceleration histories were recorded while a person walked along the midspan of the floor perpendicular to the joists.

As with conventional TMDs, the dampers must be tuned to the frequency of the mode of interest. Since the bladders are as yet unpredictable in terms of the way they influence the frequency of the damper, this tuning must be performed experimentally.

The dampers are tuned by first placing them on a rigid surface such as a slab on grade. An accelerometer is placed on the plate and the plate is given an initial displacement. The plate is then released and the free vibration response is recorded. An FFT of the acceleration determines the frequency. Tuning is performed by changing the mass, the span, or both.

Once tuned, the dampers are positioned on the floor such that one end is located over the point of maximum amplitude for the mode shape under consideration. It is also important that no modal node is located between the two ends of the damper. Otherwise, the damper will tend to rotate as a rigid body and will not achieve the desired result. Once in place, final adjustments are made to the mass in order to optimize floor performance.

RESULTS

A heel drop at the center of the test floor indicated that the floor had two strong modes of vibration, one at 7.3 hz and the other at 17 hz. A total of four dampers were used to control the floor, two for each mode.

Fig. 2 shows the Heel Drop acceleration histories for the test floor with and without the dampers. The figure shows that there was very little damping inherent in the original floor. Without the dampers, the vibration took over three seconds to decay. Conversely, the vibration of the floor with dampers took only one second to decay.

The frequency response due to the Heel Drop is shown in Fig. 3. The lack of damping in the original floor is again illustrated in Fig. 3a by the two sharp peaks at 7.3 hz and 17 hz. Fig. 3b shows that these peaks are almost eliminated by the dampers.

Fig. 4 shows the acceleration histories of the floor with and without dampers while a person walked along the midspan. Average peak accelerations were reduced from 0.06 g to 0.01 g after the dampers were installed. Not only was a significant reduction in peak accelerations observed, but the damped acceleration response consisted mainly of high frequency vibration which is generally found to be less annoying to occupants.

CONCLUSIONS

The multi-celled liquid damping device proved to be an effective solution to the problem of annoying floor vibrations in a laboratory test floor. Further research focused on optimizing the parameters of the damper is planned. In addition, permanent installations have been completed and are being monitored for effectiveness. Measurements and preliminary reports from occupants thus far are very favorable.

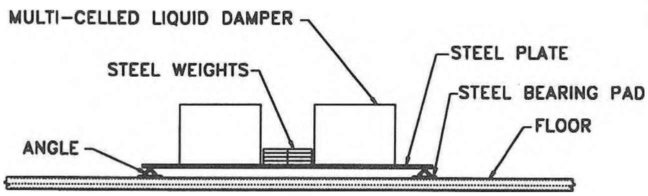


Figure 1. Prototype Damper

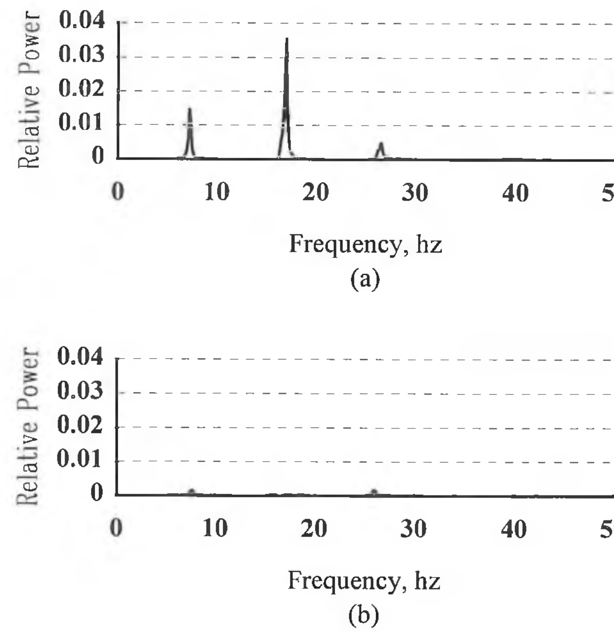


Figure 3. Heel Drop Frequency Response.
(a) Without Dampers, (b) With Dampers

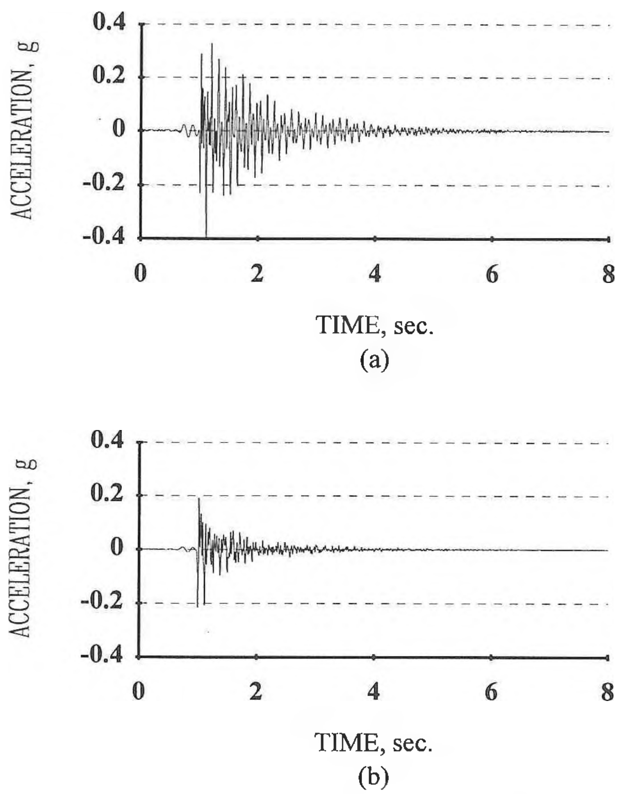


Figure 2. Heel Drop Acceleration Histories.
(a) Without Dampers, (b) With Dampers

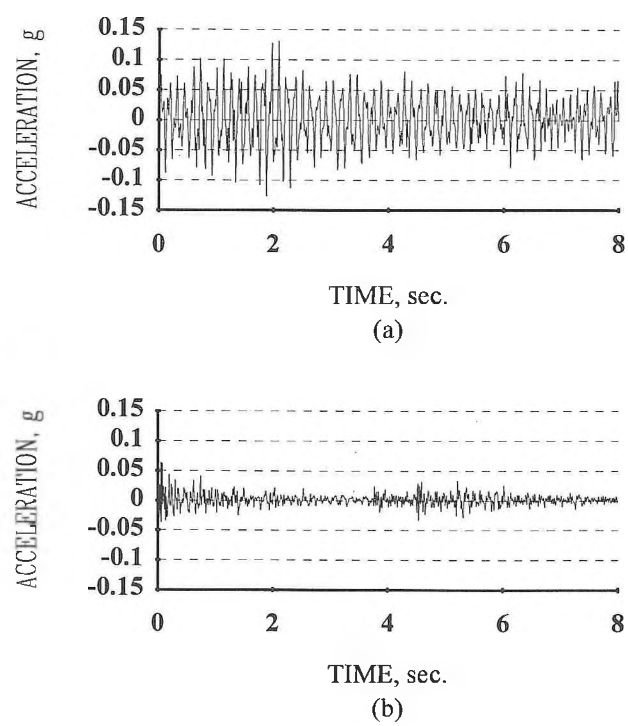


Figure 4. Walking Acceleration Histories.
(a) Without Dampers, (b) With Dampers

EXERCISE-INDUCED BUILDING VIBRATIONS: A MODERN-DAY HAPPENING

Gerry Pernica, Institute for Research in Construction, National Research Council of Canada, Ottawa, Ontario, K1A 0R6

INTRODUCTION

Rhythmic exercises in a small gymnasium on the top storey of a recently constructed office tower in downtown Ottawa were considered responsible for producing annoying vibrations in the upper floors of the 26-storey concrete building. The gymnasium was built and operated by a tenant which leased the top half of the office tower. The exercises, jumping, jumping jacks, side-to-side stepping, etc., were part of a corporate aerobics class which began each day at noon. The 1-hour class was generally well attended containing upwards of 50 participants. However, as in most office buildings, not all employees broke for lunch at the same time. Those who worked while the aerobics class was in progress found the induced floor vibrations in the upper storeys of the building highly annoying.

For operational reasons, management wanted the aerobics classes to continue but not at the expense of staff productivity. They, therefore, sought schemes to reduce the dynamic response of the floors to the rhythmic activities taking place in the gymnasium. This paper briefly describes the investigation that was undertaken i) to identify the structural elements within the building undergoing resonant or near-resonant behaviour during the classes and ii) to recommend appropriate remedial measures (tuned mass dampers, stiffening of floor elements) to rectify the situation.

BUILDING DESCRIPTION

The reinforced concrete building consisted of a central core of bidirectional shear walls which enclosed elevator shafts, strairwells and restrooms, and two rows of columns around the perimeter of the building (Fig. 1). Column spacings within the 40.8 m square, nearly symmetrical building ranged from 6.7 m to 9.1 m and column heights from 3.6 m (above 3rd storey) to 4.0 m. Column cross-sectional areas decreased with building height from 0.91 m^2 to 0.31 m^2 for the inner row of columns and from 0.68 m^2 to 0.22 m^2 for the outer row of columns. At each floor level, the columns supported 230 mm thick flat slabs which deepened to 340 mm over a 1.2 m wide strip along the perimeter of the building.

The gymnasium was situated at the south-west corner of the building between the two rows of columns (Fig. 1). Its floor was covered by a 6 mm thick rubber pad which was glued to the concrete slab. Aerobics classes were conducted in the western half of the gym which was separated from the neighbouring mechanical room by a full-storey blockwall partition. Other partitions in the upper storeys of the building were primarily of steel stud-drywall construction. However, most of these lightweight walls did not extend to the underside of the next floor slab but stopped just above the height of the suspended ceilings.

MEASUREMENT PROCEDURE

Vibration measurements were made to determine i) the natural frequencies of several of the concrete floor slabs in the upper storeys of the building and ii) the frequencies of the annoying vibrations induced in the floor slabs by rhythmic exercises during the aerobics classes.

Floor vibrations produced by instrumented hammer impacts, electrodynamic shaker inputs and ambient sources of excitations, were used to identify the natural frequencies of the concrete slabs. Hammer impacts were performed at several floor locations on the 25th storey directly below the gymnasium and harmonic shaker excitations (3 Hz to 13 Hz) at one location in the gymnasium.

Floor vibrations induced by staged rhythmic jumping at discrete frequencies from 2.0 Hz to 2.6 Hz were also measured to observe the sensitivity of the floors to exercise frequency. The exercises were performed in the gymnasium by two groups of people (two and eight participants) jumping in unison for at least 30 seconds at each frequency.

Vertical vibrations at the centre of several perimeter bays in the south-west corner of the building were measured on storeys 19, 22, 25 and 26. Measurements were made using servo-accelerometers having a sensitivity of 5 volts/gravity from 0-300 Hz. The accelerometers were taped either to aluminum plates which were epoxied to the underside of the floor slabs or to steel plates which were fitted with adjustable spike legs to penetrate the carpeting covering the floors in most office areas. Signals were amplified, low-passed filtered at 25 Hz and recorded on a multi-channel fm tape recorder for later analysis.

ANALYSIS PROCEDURE

Recorded signals were analysed on a 2-channel, narrowband frequency analyser. Natural frequencies of floor slabs on the four storeys were determined from Fourier spectra of hammer impacts and ambient excitations. Shaker excitations were ignored because the shaker (maximum force of 134 N) was unable to induce floor responses with good signal to noise ratios at most accelerometer locations. Dominant frequencies induced in the floor slabs during aerobics classes and staged rhythmic jumping were identified from Fourier and coherence functions.

RESULTS

Analyses indicated that i) the fundamental bending frequencies of the concrete slabs, about 10 Hz for each floor area, were substantially above the beat frequencies of the rhythmic aerobic exercises (1.8 Hz to 2.4 Hz) and the staged jumping exercises (2.0 Hz to 2.6 Hz); ii) the dominant frequency component of floor response during rhythmic activities occurred at twice the beat frequency of the activity and not at a higher harmonic in the vicinity of the fundamental bending frequencies; iii) the floors exhibited additional modes below 10 Hz with natural frequencies in the vicinity of the second harmonic of the rhythmic activities; iv) these modes were not simply other types of individual floor modes but vertical modes of vibration of the entire 26-storey building; and v) the lowest building modes (namely, 4.2 Hz, 4.4 Hz, 5.1 Hz and 5.5 Hz) were being strongly excited by the second harmonic of rhythmic activities during aerobics classes.

Calculations, assuming a vertical stress of 6.5 MPa in the two rows of columns throughout the height of the building, yielded a vertical building frequency of about 4 Hz. The floor vibration problem was, therefore, a resonant phenomenon associated with the dynamic properties of the entire 26-storey building and not simply those of floor slabs in the upper storeys of the building.

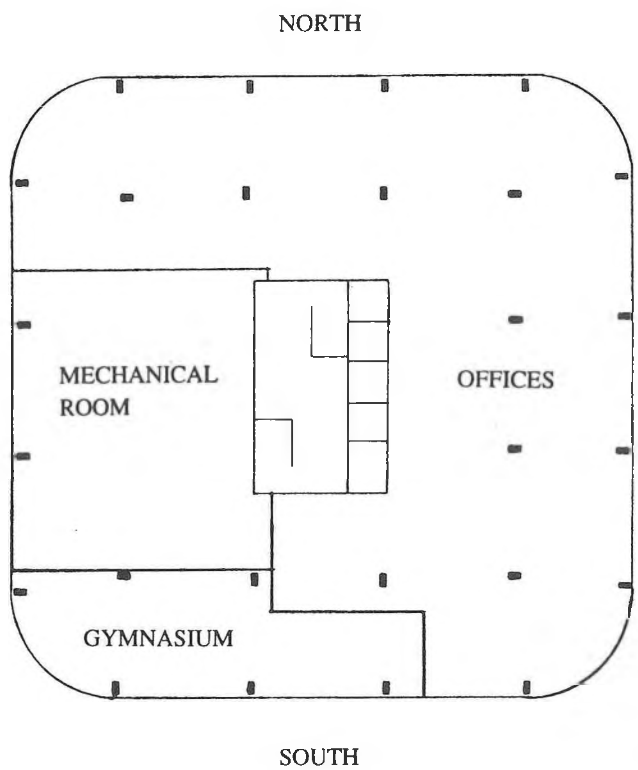


Figure 1 Layout of Structural Elements and Rooms on 26th Storey of Concrete Office Building

REMEDIAL MEASURES

The installation of tuned mass dampers to decrease the resonant floor responses of the building was not a practical solution because of the number of building modes that would need dampers and the large mass required for each of these dampers to be effective [1, 2]. Stiffening the building to raise its vertical natural frequencies was also impractical since significant changes to the building's vertical support system would be required.

Remedial measures included i) relocating the gymnasium to a much lower location within the building or nearer its central core to reduce the effects of the rhythmic forces in exciting the vertical modes of the building; ii) restricting the step frequency or beat of the rhythmic exercises to 1.5 Hz or less so that near-resonant/resonant effects are significantly reduced; and iii) changing the format of the aerobics class from a high to a low impact exercise program.

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New Design Criterion for Walking Vibrations

by **D.E. Allen**
Institute for Research in Construction, National Research Council Canada,
Ottawa, Ontario, Canada. K1A 0R6

The modern trend to lighter, more flexible floor systems with less damping has resulted in more complaints from walking vibrations. A new design criterion for walking vibration of wide applicability to all floor systems with natural frequencies less than 10 Hz has recently been developed (Allen and Murray, 1993). The new criterion is intended to replace the criterion contained in Appendix G of the CSA Standard CAN/CSA S16.1-M89. Recent experience has shown that the S16.1 criterion has limited application.

The new design criterion, plotted for use in practice in Figure 1, is given by

$$\beta W \geq K \exp(-0.35 f_o) \qquad (1)$$

where:

f_o = fundamental natural frequency of the floor structure (Hz)
W = weight of a floor panel representing its fundamental mode of vibration (kN)
β = damping ratio of the floor system (Table 1)
K = a constant (Table 1)

The basis for Equation (1), described in Allen and Murray (1993), is resonance. Resonance occurs when a harmonic component of a repeated force, such as footsteps, corresponds to the natural frequency of the floor structure. The peak resonance acceleration for any harmonic force is equal to the peak harmonic force divided by the equivalent mass (W/2g) times twice the damping ratio (2β). Walking produces significant harmonic forces at approximately 2, 4, 6 and 8 Hz, i.e. the first four harmonics. Resonance occurs if the floor frequency is equal to any of these harmonics, and this is often the case if the fundamental frequency of the floor is less than 10 Hz. The harmonic force, however, decreases with increasing harmonic. Also there are other factors reducing the real situation as compared to pure resonance, such as the number of footsteps near mid-span and the fact that the person annoyed is some distance from the walker. A more important factor is human reaction to vibration which depends very much on the use and occupancy of the floor. All these factors are combined together in the constant K given in Table 1 and used in Figure 1 (Allen and Murray, 1993).

To apply the design criterion (Figure 1 or Equation 1), the designer needs to estimate the parameters β, f_o and W. The damping ratio, β, can be estimated with the help of Table 1, which applies to typical concrete deck and steel floor systems. The other two parameters require more care:

Natural Frequency. Natural frequency, f_o, is a function of floor stiffness and mass. The stiffness of the floor structure, however, is determined by the flexibility of the floor joists or beams, plus the flexibility of the supporting girders. A useful formula for design of simply-supported joist-and-girder floor systems is

$$f_o = 18 / \sqrt{\Delta_j + \Delta_g} \qquad (2)$$

where Δ_j and Δ_g are the deflections (in mm) of the joist and girder under the weight that they support. Composite action can often be assumed for most concrete-deck-steel-floor systems but there is a reduction for certain systems, such as concrete decks separated from girders by joist shoes (Allen and Murray, 1993).

Weight of Equivalent Panel: In the case of a simply-supported footbridge the panel weight is simply equal to the weight of the suspended footbridge, i.e.,

$$W = w\,B\,L \qquad (3)$$

where w is the distributed mass weight (in kPa), L is the span, and B is the width of the panel. For simple one-way joist, beam or girder systems on rigid supports the equivalent panel is defined by the span L with width, B, determined from

$$B = C\,[D_y / D_x]^{1/4}\,L \qquad (4)$$

where C = constant,

= 2.0 for joists or beams in most areas,
= 1.0 for joists or beams beside interior openings,
= 1.6 for girders supporting joists on top,
= 1.8 for girders supporting beams connected to webs,

D_y = flexural rigidity (per unit width) transverse to the direction of the span

D_x = flexural rigidity (per unit width) in the direction of the span.

When flexible joists or beams rest on flexible girders, the equivalent weight is determined from the interaction formula:

$$W = \frac{\Delta_j}{\Delta_j + \Delta_g} W_j + \frac{\Delta_g}{\Delta_j + \Delta_g} W_g \qquad (5)$$

where the subscripts j and g refer to the joist and girder panels respectively.

More guidance on estimating these parameters is contained in Allen and Murray (1993).

REFERENCE

Allen, D.E. and Murray, T.M. 1993. *Design Criterion for Vibrations due to Walking.* AISC Journal, 30(4), December 1993, p. 117-129.

Table 1. Values of K and β for use in Eqn. (1)

	K kN	β
Offices, residences, churches	58	0.03*
Shopping Malls	20	0.02
Footbridges	8	0.01

* 0.05 for full height partitions, 0.02 for floors with few non-structural components (ceilings, ducts, partitions, etc.) as can occur in churches.

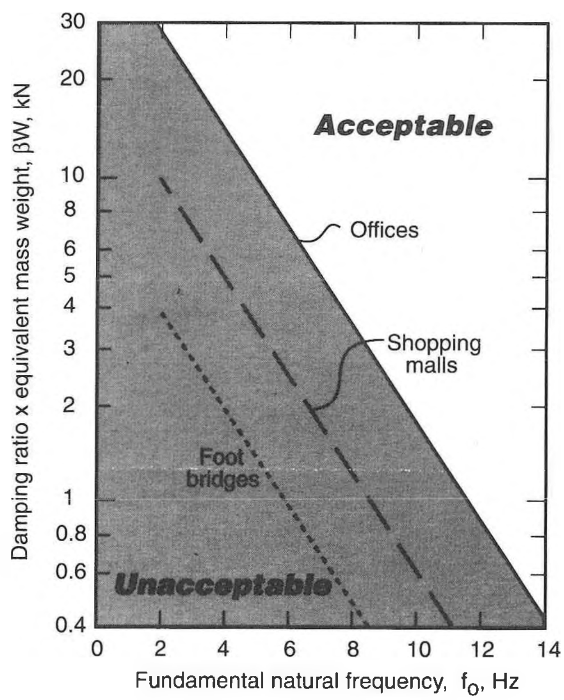


Figure 1. New Design Criterion for Walking Vibrations

FORCED VIBRATION OF A STEEL CANTILEVER BEAM WITH THICK VISCOELASTIC DAMPING LAYER

D.C. Stredulinsky and J. P. Szabo

Defence Research Establishment Atlantic, P.O. Box 1012, Dartmouth, Nova Scotia B2Y 3Z7

Introduction

The control of radiated noise is important for naval applications. DREA has been conducting research on the application of elastomeric materials to anechoic, decoupling and vibration damping tiles for ship hulls and machinery vibration isolation systems. As well, over the past twenty years DREA has developed, in-house and through contract, the general purpose finite element (FE) code VAST [1] for vibration and strength analysis of complicated structures. Recently a direct frequency response method was incorporated in VAST [2, 3] to allow modelling of frequency dependent dynamic mechanical properties. DREA has also developed methods for measuring the dynamic mechanical properties of elastomeric materials in the frequency domain [4].

This paper considers the vibration of a cantilever beam with a thick layer of viscoelastic damping material bonded to one surface. The measured forced response is compared to numerical results obtained using the VAST direct frequency response method in conjunction with measured dynamic mechanical properties for the damping material. Predictions of the composite system loss factor, made using VAST and independently using a code PREDC, are also compared to the experimental data. PREDC is a computer program obtained from University of Dayton, Ohio which employs analytical equations for free and constrained damping treatments of beams and rectangular plates.

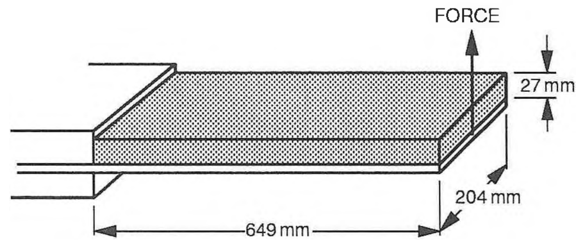


Figure 1: Steel cantilever with 27 mm thick damping layer

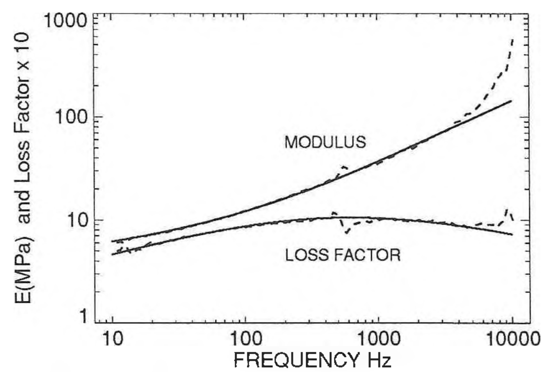


Figure 2: C-1002 properties (— Fitted , - - - Measured)

The VAST Direct Frequency Response Method

The VAST direct frequency response method assumes a steady state harmonic forcing function and results in the following system of equations for the displacement of nodes in the FE model

$$\left([K] + j\left[K'\right] - \omega^2[M]\right)\{\delta\} = \{F(\omega)\} \qquad (1)$$

where $[K]$ and $[K']$ are the real and imaginary parts of the global stiffness matrix (generally frequency dependent), $[M]$ is the mass matrix, $\{F(\omega)\}$ is the complex load vector, $\{\delta\}$ is the complex nodal displacement vector, ω is the forcing frequency and j is $\sqrt{-1}$. This system of equations was solved at each specified forcing frequency to obtain the amplitude and phase for each component of the nodal displacement vector $\{\delta\}$.

The dynamic mechanical properties for groups of elements in the VAST FE model can be represented using a complex Young's modulus $E^*(\omega) = E(\omega)\,(1 + j\eta(\omega))$ where η is the material damping or loss factor. The frequency dependence of $E(\omega)$ and $\eta(\omega)$ can be specified using tables of frequency weighting values with linear interpolation between values or by using quadratic polynomials over a specified frequency range.

The Cantilever Beam Description

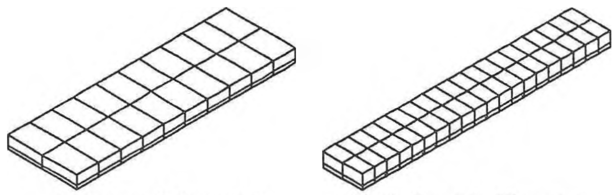
The cantilever beam considered is shown in Figure 1. The steel beam was 9.5 mm thick and clamped between steel blocks at one end. A 27 mm thick layer of EAR Isodamp C-1002 viscoelastic damping material was bonded to the upper surface of the beam. The assumed properties of the steel were Young's modulus $E = 2.07 \times 10^5$ MPa, Poisson's ratio $\nu = 0.3$ and density of 7870 kg/m^2. The Young's modulus E and loss factor η for the damping material, shown in Figure 2, were determined using a direct stiffness method [4]. The fitted curves, required for the PREDC program, were also used to generate tables of frequency weighting values for the FE analysis. The density of the damping material was 1280 kg/m^3.

Viscoelastic damping materials typically have Poisson's ratios near 0.5 (incompressible) in the 'rubbery' region, decreasing through a transition region to a value of 0.3 in the 'glassy region' [6]. The PREDC code assumed a Poisson's ratio of 0.5. The VAST FE code presently cannot consider a truly incompressible material so that a Poisson's ratio of 0.47 was used in the FE analysis.

The Beam Forced Vibration Response

The measured forced response was reported in reference [5] both for the bare beam and the beam with damping layer. A vibration exciter was used to apply a load at the centre-line, 9 mm from the tip, normal to the bottom surface of the steel beam. The applied force was measured with a force transducer and the acceleration of the top surface of the beam measured with an accelerometer placed at several locations along the centre-line.

The forced response of the damped beam was predicted using the direct frequency response method in VAST Version 7.1. Loss factors extracted from the bare beam experiment were used for steel in the FE analysis of the damped beam.



a) Coarse full model b) Refined half model

Figure 3: Finite element meshes of cantilever

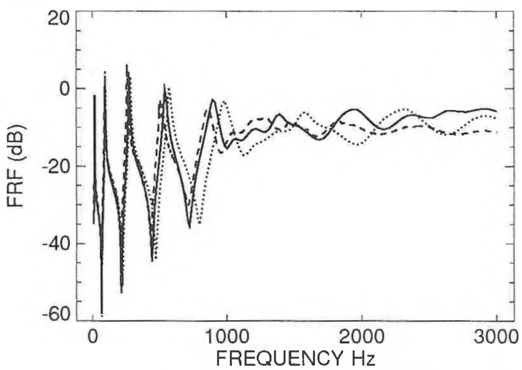


Figure 4: Frequency response at the centre of the tip; (— refined FE, · · · coarse FE, - - - measurement)

The FE models were constructed using 20-noded isoparametric solid elements. Two meshes were used; a coarse model of the entire beam and a refined 'half' model taking advantage of symmetry (see Figure 3).

The predicted nodal displacement amplitudes δ(mm) were converted to acceleration amplitudes and reduced to a frequency response function FRF (magnitude in dB) using $\mathrm{FRF} = 20\log\left(\omega^2\delta\right)$ at each forcing frequency ω(radians). The predicted FRF's for the coarse and refined meshes are compared to the measured FRF's in Figure 4. The phase information was also predicted but no experimental measurements were available for comparison. Below 1800 Hz there was reasonably good agreement between the refined FE model result and the experimental curve. The predicted frequency of the first resonant peak was ten percent higher than measured. The next four resonant peaks ranged from four to seven percent higher than measured, consistent with five percent differences obtained between the bare beam experiment and bare beam FE predictions. Over this frequency range the experimental and refined FE response levels differed by 3 dB or less. The predicted FRF levels over the frequency range 1800 to 3000 Hz were up to 6 dB higher than measured.

The composite loss factors predicted with the VAST direct frequency response module are compared to the measured loss factors and PREDC results in Figure 5. There was reasonable agreement between the VAST loss factor predictions and the measured points. Above 1000 Hz there was a significant frequency shift between the curves for the coarse and refined meshes, suggesting that a further refinement of the mesh should be used in this frequency range. The loss factors predicted by the PREDC code were significantly lower than both the measured points and the VAST curves at very low frequencies and at frequencies above 500 Hz. The PREDC code only considers the damping material loss factor. When the bare beam loss factors were neglected in the VAST damped beam analysis, the VAST results were close to the PREDC points in the lower frequency range.

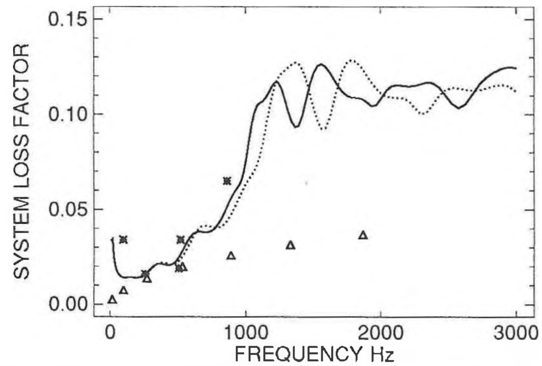


Figure 5: Composite loss factor; (— refined FE, · · · coarse FE, * measurement, △ PREDC)

Concluding Remarks

The VAST finite element predictions of the forced vibration response of the steel cantilever beam with a thick damping layer were in good agreement with experimental results up to 1800 Hz. Above this frequency the analysis indicated that a further refinement of the FE mesh should be considered. The FE analysis also showed that there was significant axial and transverse shear deformation in the damping layer at higher frequencies. Modification of the VAST code, to include frequency dependent values of Poisson's ratio, should improve modelling of the shear properties which may improve response predictions for higher frequencies.

The PREDC code considered only flexural bending. The VAST FE analysis predicted much more complicated wave motion in the damping layer at higher frequencies. This may explain why the measured loss factors and those predicted by VAST were significantly larger than obtained with the PREDC code at forcing frequencies above 500 Hz.

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DÉTECTION VIBROACOUSTIQUE DES FISSURES DE FATIGUE DANS LES POUTRES

K. El Bikri, A. Berry, R. Gauvin
Département de Génie Mécanique, Université de Sherbrooke
Sherbrooke J1K 2R1

1- INTRODUCTION:

Certaines structures immergées sont soumises à des chargements cycliques qui, avec le phénomène de corrosion, constituent les principales causes d'initiation et de propagation des fissures dites de fatigue. Pour cela les techniques de contrôle non destructif font l'objet d'un champ de recherche actif pour détecter et identifier ces fissures qui peuvent être, dans certaines situations, très nocives. Dans les dix dernières années, une stratégie de détection des fissures a été développée utilisant une technique d'analyse modale mais dont le potentiel est encore mal exploité; l'objet de cette présente étude sera d'examiner dans un premier volet, la faisabilité de la méthode en vue de localiser et de caractériser les fissures de fatigue dans une poutre, puis dans un deuxième volet, déterminer le rayonnement acoustique de la même poutre afin d'estimer la sensibilité de cette approche quant à la profondeur de la fissure. L'idée de base de cette méthode est qu'une fissure dans une structure introduit une flexibilité locale qui va affecter la réponse vibratoire de la structure et par conséquent son rayonnement acoustique.

2-FORMULATION THÉORIQUE:

Pour étudier l'effet d'une fissure sur la réponse dynamique d'une poutre élastique sollicitée par un moment de flexion pure P_5 tel que montré dans la figure 1., nous avons à établir l'expression du coefficient de flexibilité locale C_{55} en fonction de la profondeur a de la fissure. La théorie classique de la mécanique de la rupture [1] permet, en appliquant l'équation de Paris et le théorème de castigliano, permet d'aboutir au résultat suivant:

$$C_{55} = \frac{6\left(1-\nu^2\right)h}{EI} F\left(\frac{a}{h}\right) \tag{1}$$

$$\text{où } F\left(\frac{a}{h}\right) = \int_0^{\frac{a}{h}} \left(\frac{\pi a}{h}\right) f^2\left(\frac{a}{h}\right) d\left(\frac{a}{h}\right) \tag{2}$$

$f\left(\frac{a}{h}\right)$ étant une fonction de $\frac{a}{h}$ dont l'expression est déterminée pour la configuration géométrique considéré.

La poutre fissurée est modélisée comme deux éléments de poutre qui doivent satisfaire les conditions de continuité en plus des conditions aux limites imposées, de ce fait chaque élément de poutre vérifie l'équation de vibration de flexion d'une poutre uniforme soit:

$$\frac{d^4W_i}{dx^4} - k^4 W_i(x) = 0 \qquad i = 1,2 \tag{3}$$

où $k^4 = \frac{\omega^2 \rho S}{EI}$, S est la section de la poutre, ρ est la densité du matériel, $(W_i)_{i=1,2}$ est le déplacement vertical pour chaque élément de poutre.

la solution générale de l'équation (3) s'écrit

$$w_i(x) = A_i\cosh(kx) + B_i\sinh(kx) + C_i\cos(kx) + D_i\sin(kx) \tag{4}$$

Pour une fissure située à une position $x=x_0$, les conditions de continuité se traduisent par :

$$W_1(x_0) = W_2(x_0)\,,\; W_1''(x_0) = W_2''(x_0)\,,\; W_1'''(x_0) = W_2'''(x_0) \tag{5-7}$$

$$EIC_{55}\, W_1''(x_0) + W_1'(x_0) = W_2'(x_0) \tag{8}$$

les conditions aux limites par contre, elles sont arbitraires. En substituant la solution (4) dans les équations correspondant aux conditions aux limites et aux conditions de continuité, nous aboutissons à un système d'équations algébriques linéaires de huit coefficients inconnus A_i, B_i, C_i, D_i, i-1,2 . les fréquences naturelles sont alors calculées en posant le déterminant du système égal à zéro.

En ce qui concerne le rayonnement acoustique, la formule de Rayleigh [2] fournit l'expression de la pression acoustique en champ lointain pour une poutre plane, bafflée et de dimension finie .

$$p(R,\theta,\varphi) = -\rho_0\omega^2 \frac{e^{-jkR}}{2\pi R} \tilde{W}(\lambda,\mu) \tag{9}$$

où $\tilde{W}(\lambda,\mu)$ est la transformé de Fourrier spatiale bidimensionnelle du déplacement de la surface qui vibre.

Avec $\lambda = \frac{\omega}{c_0}\sin\theta\cos\varphi$, $\mu = \frac{\omega}{c_0}\sin\theta\sin\varphi$

En régime forcé, vue la linéarité des équations, nous pouvons décomposer le déplacement dans la base modale soit $W(x) = \sum_{i=1}^{\infty} a_i W_i$

la vitesse quadratique moyenne prend alors l'expression suivante:

$$\langle \dot{W}^2 \rangle = \frac{\omega^2}{2\ell} \sum_{n=1}^{\infty} |a_n|^2 N_n \tag{10}$$

$$\text{où } N_n = \int_0^{\ell} W_n^2(x)\,dx\,|\tilde{W}|^2 \tag{11}$$

3-RÉSULTATS ET DISCUSSION:

La figure 2. montre la courbe théorique et les valeurs expérimentales des fréquences de résonance du troisième mode de vibration d'une poutre, libre à ses deux extrémités, en fonction de la profondeur de la fissure, située au centre de la poutre. La chute des fréquences propres avec l'augmentation de la profondeur de la fissure est due à la diminution de la rigidité. La comparaison des résultats montre que les valeurs théoriques et expérimentales concordent presque parfaitement, nous remarquons également que les variations des fréquences propres avec la profondeur de la fissure ne sont significatives qu'àpartir de 30% de l'épaisseur de la poutre. Quant à l'effet de la fissure sur la déformée du mode 3, il est de plus en plus marqué au fur et à mesure que la fissure est profonde comme indiqué par la figure 3.. Cet effet très local, peut-être exploité pour localiser la position de la fissure. Sur le plan rayonnement acoustique, l'intensité acoustique modale donnée par le paramètre $|\tilde{W}|^2$, qui lui est proportionnel, est représentée à la figure 4. en terme de directivité et . en fonction de la variable kl à la figure 5., dans ce cas pour une fréquence donnée, i.e pour une valeur du vecteur d'onde acoustique, nous avons la gamme des vecteurs d'onde mécanique correspondant aux composantes qui rayonnent. En régime forcé, la vitesse quadratique moyenne dans le cas d'une poutre sur appuis avec une fissure au centre et une excitation au 1/4 de la portée de la poutre est représentée à la figure 6. , le résultats obtenu confirme bien le glissement des fréquences , en outre on voit que le pic correspondant au mode 2 n'est pas affecté par la présence de la fissure puisqu'elle est située au centre de la poutre.

4-CONCLUSION:

Le modèle présenté permet de déterminer la réponse vibroacoustique d'une poutre avec une fissure de fatigue; il constitue un outil très utile pour une étude paramétrique modale complète permettant de dégager les critères de détection des fissures dans les poutres. La sensibilité des fréquences naturelles à la présence de la fissure est médiocre, toutefois la méthode peut-être utilisée de façon satisfaisant pour la localisation des fissures àpartir des déformées propres. Le rayonnement acoustique représente un moyen de détection des fissures plus sensible que l'analyse modale.

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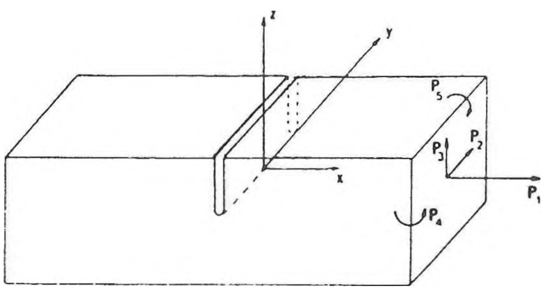

figure 1. élément de poutre chargé avec une fissure transversale

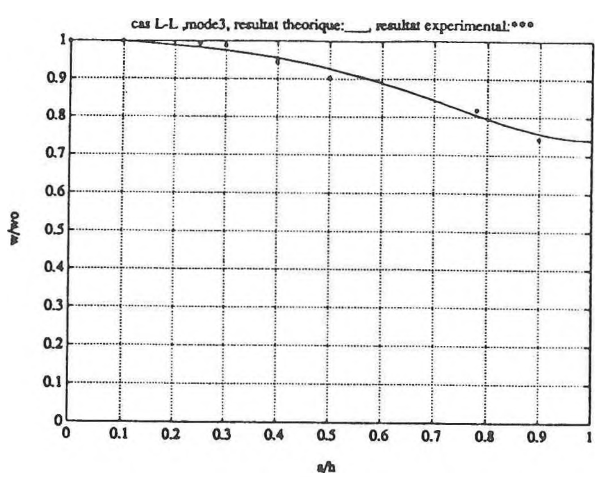

figure 2.

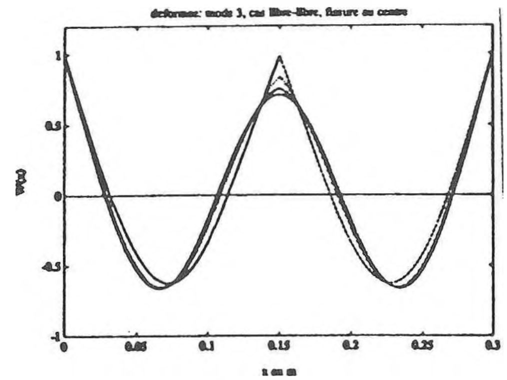

figure 3.

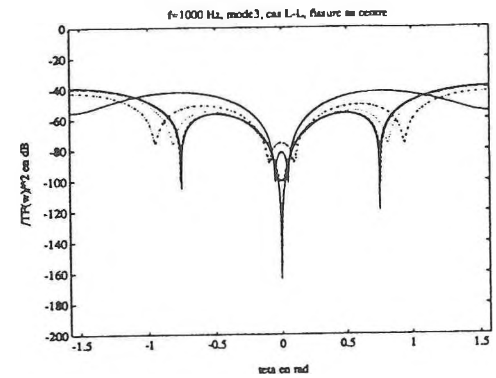

figure 4.

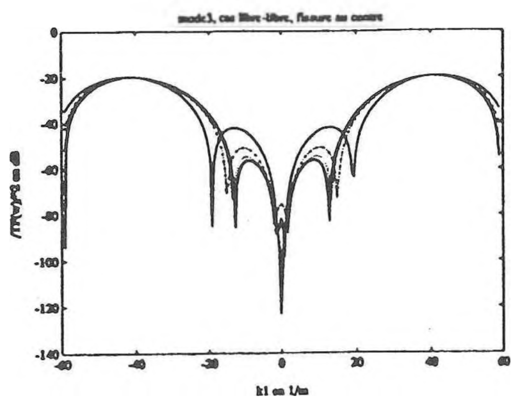

figure 5.

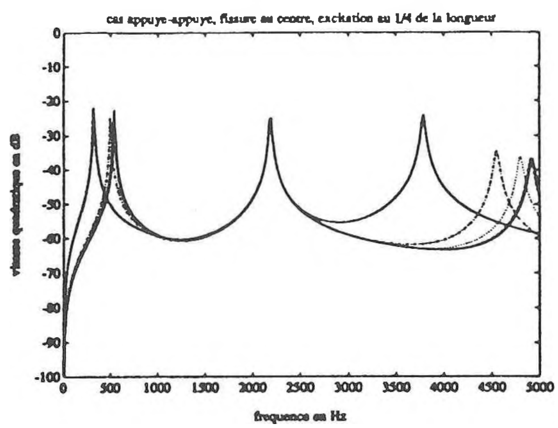

figure 6.

DETERMINATION OF DYNAMIC PROPERTIES OF NON-LINEAR STRUCTURES USING THE PULL-RELEASE TEST

J.Hans Rainer & Gerry Pernica
Institute for Research in Construction, National Research Council, Ottawa, ON, K1A 0R6

INTRODUCTION

For the determination of the dynamic properties - natural frequencies, mode shapes and damping - of an existing structure, a number of experimental methods are available. Among the most common ones are: the shake table test for small and medium size structures; excitation by a shaker mounted on the structure and locating the resonances by sweeping across a frequency band either slowly or in discrete steps; monitoring the ambient vibrations and determining the properties by signal analysis; and pulling (or pushing) the structure and suddenly releasing it and measuring the resulting decaying vibrations. This pull-release test is one of the simplest dynamic methods for determining a complete set of dynamic properties. While it is usually employed at low amplitudes of vibration, the method can also be employed at large amplitudes and consequently to obtain amplitude-dependent dynamic properties for structures with non-linear deformation and damping properties.

This presentation focuses on the theoretical and analytical issues that have to be considered in planning such a test and interpreting the results. The practical aspects in carrying out the pull-release test and instrumentation considerations are not addressed here.

THEORETICAL CONSIDERATIONS

While the pull-release method is simple to perform, certain limitations and requirements need to be observed in order to obtain reliable results. These requirements flow from the physical characteristics of the test method. A simple SDF system will be considered first.

As the name of the test implies, the structure is pulled (or in some cases pushed) a certain distance and then suddenly released. This gives rise to the time history of vibration decay for displacement. Differentiation gives the velocity and a further differentiation the acceleration. See Fig. 1. These three interrelated time histories contain all the information necessary to characterize the dynamic properties of the structure:

i. damping can be calculated from the amplitude variation as a function of time

ii. natural frequency, is given by the zero crossings of the decay curve

iii. for non-linear systems, variations of damping or natural frequency as a function of amplitude can be determined

iv. and for a multi-degree of freedom (MDF) system, the mode shapes can be obtained from the responses at various locations on the structure.

CALCULATION OF DAMPING AND NATURAL FREQUENCY

Damping. For a viscously damped system, damping is given as a fraction of critical , or in % thereof, by the following relationships: From the logarithmic decrement

$$\delta = \ln\left(A_m / A_{m+n}\right)$$

$$n = \frac{1}{2}, 1, 1\frac{1}{2}, 2 \ldots\ldots.$$

$$\beta = \delta\sqrt{4\pi^2 n^2 + \delta^2{}_n}$$

For small δ this simplifies to:

$$\beta = \frac{1}{2\pi n} \ln \frac{A_m}{A_{m+n}}$$

where A denotes the peak amplitude in a decay curve, ***n*** the nth peak in the decay curve, and ***m*** is the number of cycles after ***n***. ***m*** can be any positive integer including zero or any integer plus 1/2.

Natural Frequency. The natural frequency can be determined at any stage of the decay curve by measuring the time T between zero crossings for one cycle, or T/2 at 1/2 cycle, and then taking the inverse (Fig. 1a). This is feasible along the entire decay curve.

This measurement of zero crossings yields the damped natural frequency T, which then can be corrected to yield the undamped natural frequency $T_0 = T / \sqrt{1 - \beta^2}$.

MEASUREMENT IMPLICATIONS

The properties of the decay curve described above lead to important consequences for the measurement of the relevant qualities.

The initial acceleration upon release of the structure has an infinitely steep rise time. This can only be achieved with an accelerometer of infinite frequency response and with zero time delay in release mechanisms, neither of which is achievable in practice. Consequently, the first acceleration peak will always be distorted and will not provide a reliable initial amplitude. It follows then if that initial amplitude and its rate of decay characteristic is important, the direct displacement measurement is the only one that will yield the desired information. In that case, an independent reference frame is needed to serve as a mounting platform for the displacement transducer. If on the other hand, the first amplitude peak is not of interest, then accelerometers mounted on the structure can give very satisfactory acceleration vibration decay curves.

AMPLITUDE DEPENDENT DAMPING AND FREQUENCY

The damping ratios and natural frequencies for a structure can be computed over a number of cycles, thereby averaging out small computational or experimental variances. If, however, the structure's dynamic properties are amplitude dependent, these properties have to be computed over each cycle or even half cycles. The results then represent an average value over that cycle, or half cycle, and can be plotted as damping versus mean displacement. A mean displacement can be defined as being the average of two adjacent peak amplitudes.

MULTIMODE RESPONSE

When the structure exhibits more than one mode, ie. the structure is an MDF system, the time decay record will include the combined response of a number of modes. In order to extract the desired properties of damping, frequency and modal amplitudes for any particular mode, the pure signal for that mode alone needs to be extracted. This can be done by filtering, where, the following needs to be observed. For the acceleration record, the filter cannot deal with a sudden jump at the beginning of the signal shown in Fig. 1c but will result in distortions in the peaks near the beginning of the decay curve. This can be circumvented by filtering the signal in the reverse order, in which case the "ringing" occurs before the acceleration jump, which is of no interest in the investigation. The first peak also needs to be ignored in any case since its amplitude is not realistic right from the start of the measurement phase, as was explained before.

The displacement record, Fig. 1a, experiences problems with filtering because the mean of the record is not zero and this causes a distortion of the base line. Again, a reversing of the signal permits the production of reliable filtered results right up to the time of the pull release. Thereafter (or in the original signal before the release) severe distortions will occur but these can again be ignored.

USE OF TEST DATA IN SEISMIC QUALIFICATION

One important application for the data obtained from a pull-release test is the seismic qualification of equipment for seismic response. This can relate to facilities such as boilers or heat exchangers in nuclear power plants, or free-standing structures such as high voltage power switches or line-conditioning platforms for power lines. Seismic qualification can be carried out according to the standards CAN3-N289.4-86 (CSA 1986) or IEEE Std. 344-1987 (IEEE 1987). Both standards recognize the pull-release method as one component of a combined testing and analysis approach. They view it as a low-amplitude method and caution about amplitude dependent results. With the above techniques, however, amplitude dependent result can be fully accounted for and treated in a consistent manner.

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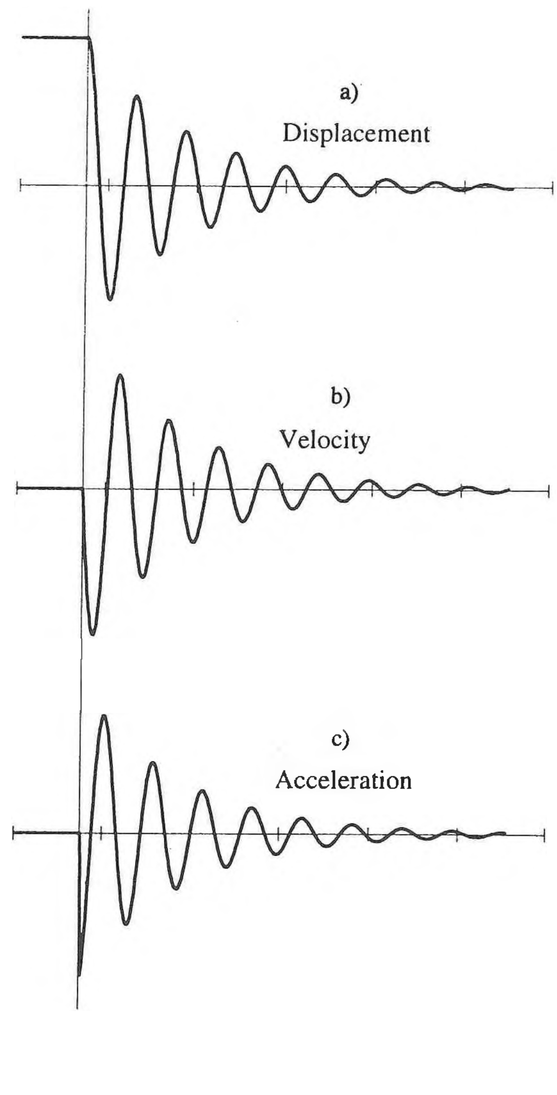

Fig. 1: Theoretical Vibration Decay Records for Pull-Release Test

PC-BASED MEASUREMENT AND ANALYSIS OF TRAFFIC VIBRATIONS

M.Osama Al-Hunaidi, J.Hans Rainer, and Gerry Pernica
Institute for Research in Construction, National Research Council of Canada, Ottawa, ON, K1A 0R6

INTRODUCTION

The purpose of this paper is to present the general requirements and procedures for measurement and analysis of traffic-induced vibrations and to discuss the methods that are currently in use at IRC/NRC. The following topics are addressed: (i) measurement systems with special emphasis on recording of vibration signals using portable multi-channel PC-based data acquisition systems (ii) 1/3 octave spectrum analysis and frequency weighting of vibration time-histories using digital filtering on PCs, and (iii) measurement practices such as transducers, mounting methods and sampling rates.

MEASUREMENT SYSTEM

Instrumentation for the measurement of vibration signals usually consists of the following components: (i) vibration transducers, (ii) signal conditioners, and (iii) recording equipment. The characteristics of the complete measurement system should be appropriate for measuring and recording vibration signals in the frequency and amplitude ranges of interest, e.g. a flat frequency response, flat amplitude response, linear phase shift, sufficient resolution and sensitivity, low noise level, sufficient dynamic range, and appropriate environmental operating conditions. For vibrations induced in buildings by road and rail traffic: (i) frequency range is generally from 4 Hz to 80 Hz; sometimes up to 125 Hz, and (ii) amplitudes are in the range 0.005 to 2 m/s^2 for acceleration and 0.05 to 25 mm/s for velocity. It is recommended that the noise level introduced by the complete measurement system be at least 10 dB below the minimum anticipated vibration levels and the dynamic range of the system be greater than 40 dB.

Vibration Transducers. Either velocity and acceleration transducers, also known as geophones and accelerometers, respectively, can be used for the measurement of traffic-induced ground and building vibration. The characteristics of geophones and accelerometers vary widely from one manufacturer to another. The appropriate transducer type should be selected according to the frequency and vibration amplitude ranges which are of interest or anticipated. At present, the use of accelerometers is becoming more widespread than geophones because of their wide variety of types and ranges.

Accelerometers are generally classified into two types: (i) deflection type, and (ii) servo null-balance type. Deflection type accelerometers utilize mechanical spring restoring forces and are usually identified by the type of transducer element they utilize, e.g. potentiometer type, strain-gauge type (includes piezoresistive transducers), and piezoelectric type. The higher the sensitivity of deflection-type accelerometers, the greater is their mass. Servo-type accelerometers, on the other hand, utilize servo electro-magnetic restoring forces, i.e. "electrical springs", to maintain a mass element in almost a zero-position. Servo accelerometers provide good accuracy, especially at low frequencies. They have very low acceleration thresholds and high sensitivity even though their mass is small. The particular type used by the authors has an acceleration threshold equal to 1 μg, selectable sensitivity (up to 10 V/g), a resonance frequency greater than 800 Hz, and a weight of about 175 grams which includes an aluminum mounting base.

Signal Conditioners. Signals from vibration transducers are passed through signal conditioners before they are displayed or recorded to perform the following: (i) to amplify the voltage output of vibration transducers so that it is sufficiently above the noise level of the display or recording system, and (ii) to eliminate noise or undesired vibrations outside the frequency range of interest. It is advantageous to have the following signal conditioners: (i) amplifiers with gain settings at several values, (ii) a high-pass filter with a cut-off frequency at about 1 Hz to remove any DC offset and eliminate low frequency drifting, and (iii) a low-pass filter with selectable cut-off frequencies to remove unwanted noise above the frequency range of interest and to avoid frequency aliasing during analog-to-digital conversion of vibration signals. Signal conditioning hardware can be of the stand-alone type or included in a self-contained data acquisition system as described below.

Data Recording. Vibration signals can be recorded directly in digital form using modern portable PC-based systems. Recording the data directly in digital form offers speed and convenience and the opportunity for on-site inspection and analysis of data. PC-based data acquisition systems also have the advantages of being self-contained, flexible, readily upgradable, and relatively inexpensive. On the other hand, recording the data in analog form is advantageous since the signals can always be re-digitized differently in the future if needed, or used as a back-up to the PC system.

Data recording can also be performed using hand-held "vibration meters". These devices usually provide a single vibration descriptor, e.g. rms or peak level of the frequency weighted vibration signals that can be readily compared with satisfactory vibration levels provided by relevant standards. It must be ensured, however, that the weighting and averaging procedures are appropriate for the evaluation of transient vibration.

PC-Based Data Acquisition Systems. Portable PC-based systems can be integrated using commercially available multi-channel analog-to-digital (A/D) conversion boards, and multi-channel signal conditioning boards. These boards, which are software controlled, fit into standard expansion slots inside PCs. Many makes of these plug-in A/D and signal conditioning boards do not require software programming by the user.

Essential features of a system for multi-channel measurements of traffic vibration might be as follows: 8 to 16 channels, sampling rate of 1000 Hz/channel, 12-bit resolution, A/D board with built-in amplifiers with gains at 1, 2, 4 and 8, signal conditioning boards with gain amplification at 1, 10, 100 and 1000, and anti-aliasing filters with a cutoff frequency between 100 and 200 Hz and attenuation rate of about 70 dB/octave or greater. The above combination of gain settings on the A/D and signal conditioning boards provides a versatile selection of gains which is beneficial for utilizing most of the voltage range of the A/D board to increase the signal-to-noise ratio. Preferably, it should be possible to select gain settings for each channel individually in order to accommodate any wide variation in vibration levels at different measurement locations when performing multi-channel measurements.

One of the PC-based systems that is used by the authors features 16 channels, 12-bit resolution, 400 KHz total sampling rate, simultaneous-sample-and-hold (SSH) capability to reduce inter-channel sampling/conversion delay, pre-trigger capability which is useful for unattended measurements, A/D board with gain amplification at 1, 2, 4 and 8, signal conditioning boards with gain amplification selectable in linear increments from 1 to 200, and anti-aliasing filters with an attenuation rate of 80 dB/octave and a cutoff frequency selectable from 1 Hz to 100 KHz. All components are driven with a single data acquisition software package and fit into a "lunch-box" type PC. This system was optimized for several applications including traffic-induced vibration.

ANALYSIS OF VIBRATION SIGNALS

The degree of detail required in the analysis of traffic-induced ground and building vibration signals depends on the nature and purpose of the investigation. For a preliminary evaluation of the effect of vibration with respect to human response and building damage it might be sufficient to just find the peak of the vibration signal and determine the predominant frequency of vibration by simply counting the number of negative and positive peaks in a given time interval. For in-depth evaluation, however, 1/3 octave band analysis, frequency weighting of acceleration signals, and spectral analysis have to be performed.

General Digital Signal Processing. Digital signal processing and analysis can be performed conveniently using several commercially available user-friendly PC software. The software packages typically include the following signal processing and analysis functions: low-pass/high-pass/band-pass filtering, signal extraction and concatenation, algebraic functions, FFT analysis and spectral estimates, e.g. auto-spectrum, cross-spectrum, transfer function, and coherence function. Some software packages support the creation of macro commands, i.e. command sequence, which are very beneficial for automating lengthy tasks and the processing and analysis of large amounts of vibration signals. For example, the authors obtain velocity from acceleration time records by integrating the latter indirectly in the frequency domain using a macro command that performs the following procedures: extract 50% overlapping segments from the time record, perform an FFT with a Hanning window on each segment, set FFT components below 2.5 Hz to zero, divide FFT components above 2.5 Hz by frequency, inverse modified Fourier spectra (IFFT), and finally, add time segments using 50% overlapping. The macro can be run on selected channels or all channels in a data file.

1/3 Octave Analysis. PC software was developed for performing 1/3 octave analysis (Al-Hunaidi et al., 1992) using recursive digital filters which conform to specifications by the standard ISO 2631/2 (1989) for evaluation of human exposure to vibration in buildings. This software can be run in both interactive and batch modes. The latter makes it possible to automate the analysis when a large number of data files have to be analyzed. For instance, both 1/3 octave analysis and frequency weighting (see below) of vibration data collected from 42 runs of test vehicles involving 11-channel measurements, 30 seconds each, were completed in less than 2.5 h on an IBM-compatible 486 DX PC (clock speed of 50 MHz). Similar analysis using conventional analyzers would have taken several days to several weeks. The software can accomodate up to 16 channels and can deal with several processing parameters specified by the user, such as format of data files, channels to be analyzed, and frequency range of the 1/3 octave spectra. Other information such as sampling frequency, sensitivities and gains of instrumentation are read directly from header files created by the data acquisition software. The user can also specify how the signals should be processed, including integration time, start time, and number of initial integration windows to be ignored. Currently, this program can only access data files created by two different commercial data acquisition software packages; however, it was written in modular form so that it can be easily modified to access files acquired with other software packages.

Frequency Weighting. Several options for frequency weighting of acceleration time histories is incorporated in the above software. These were realized using recursive digital band-limiting and band-pass filters which conform to specifications in ISO 8041 (1990) and BS 6841 (1987). The use of frequency-weighted vibration levels is the preferred analysis method by several standards for the evaluation of human exposure to vibration in buildings, e.g. ISO 2631/2 (1989) and BS 6472 (1984).

PRECAUTIONS DURING MEASUREMENTS

Vibration transducers should be mounted on the ground or on building components using mounting methods which are capable of faithfully transmitting to the transducer the actual motion of the ground or building components over the frequency range of interest. If the mounting method is suspected of distorting the motion, alternative methods should be tried and compared. For ground vibration measurements, the standard ISO 4866 (1990) suggests fixing vibration transducers to a stiff rod (having a diameter not less than 10 mm), driven into the ground. The use of a tapered stake having cruciform section (Al-Hunaidi & Rainer, 1990; Nolle, 1978) is believed to provide a more reliable method than that suggested by the ISO standard.

Precautions should be taken against inductive pickup, electrostatic pickup, and ground loops. These can cause large spurious vibration signals often concentrated at the power-line frequency. Instruments should be properly shielded to decrease electrostatic interference; transmission cables should be twisted conductors wrapped in foil to reduce both inductive and electrostatic pickup; and the complete measurement system should grounded at one point only.

When acquiring vibration signals directly in digital form, one should sample at an appropriate frequency in order to obtain: (i) valid results when data is analyzed in the frequency domain, and (ii) accurate amplitudes of vibration signals in the time domain. To obtain valid results in the frequency domain, vibration signals while still in analog form should be passed through a low-pass filter with its cutoff frequency F_c set at least at the maximum frequency of interest. The appropriate sampling frequency F_s should be greater than $(2 + 40 / M)\ F_c$ where M is the attenuation rate of the low-pass filter. For accurate peak values of the time signal, the sampling frequency usually depends on the form of the transient signal and its predominant frequency. A sampling frequency equal to about 10 times the predominant frequency is normally sufficient.

Finally, calibration of individual components of the complete measurement system should be performed periodically in accordance with relevant standards and (or) recommendations by instrument manufacturers. In addition, end-to-end calibration of each channel of the complete system is recommended at the beginning and end of every vibration measurement session.

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EVALUATION OF APPROPRIATE SAMPLE SIZE FOR MEASUREMENT OF VIBRATION LEVELS INDUCED BY RAIL TRANSIT VEHICLES

M. Osama Al-Hunaidi

Institute for Research in Construction, National Research Council of Canada, Ottawa, ON, K1A 0R6

and

Marc Bracken

Aercoustics Engineering Limited, 50 Ronson Road, Unit 127, Rexdale, ON, M9W 1B3

1 INTRODUCTION

Building vibration levels induced by rail transit vehicles under usual operational conditions can vary from one vehicle to another. In order to obtain meaningful vibration levels, measurements of building and (or) ground vibrations should be performed for an appropriate number of vehicle pass-bys. In a recent rail transit system vibration study by the authors, the number of vehicle pass-bys for a particular vehicle population was chosen so that the mean vibration level of the sampled vehicles is reasonably close to the true mean of vibration levels induced by all vehicles in the population. This paper presents statistical analysis procedures, based on Kreyszig (1979) and Ang & Tang (1975), that can be used to determine an appropriate sample size to achieve a certain confidence range of the true mean vibration level, e.g. ±3 dB of the sample mean, for a certain confidence level, e.g. 95%. The results obtained using these statistical procedures are reported for subway train and streetcar populations of the rail transit system studied by the authors. In addition, verification of the appropriateness of the calculated sample sizes is presented.

2 CONFIDENCE INTERVAL AND LEVEL

Supposing that a sample of (n) vehicle pass-bys is used, the mean vibration level is calculated as

$$\bar{x} = \frac{1}{n}\sum_{i=1}^{n} x_i \tag{1}$$

and the standard deviation is calculated as

$$s = \left(\frac{1}{n-1}\sum_{i=1}^{n}(x_i - \bar{x})^2 \right)^{1/2} \tag{2}$$

In this study, vibration signals induced by rail transit vehicles were processed individually to obtain 1/3 octave band and overall frequency-weighted vibration levels using the computer program TOAP-Version 4 (software for 1/3 octave analysis and frequency weighting using recursive digital filtering on PCs; see Al-Hunaidi et al. (1992) for details). Rms levels for 1/3 octave bands in the range from 10 to 125 Hz and for the frequency-weighted vibration signals were calculated using linear time integration with a 1 second integration time. For each vehicle pass-by, only maximum rms levels for each 1/3 octave band and the frequency-weighted signal were retained. The vibration level (x_i) in Eqs. 1 and 2 is the maximum rms vibration level of the i^{th} vehicle pass-by. Assuming that the vibration levels (x_i) induced by the population of each type of vehicle, e.g. subway trains or street cars, to have a normal density distribution, there is a certain chance (γ) that the true mean of vibration levels (μ) is in the range

$$\bar{x} - k \le \mu \le \bar{x} + k \tag{3}$$

where

$$k = s.c / \sqrt{n} \tag{4}$$

and c is the solution of the equation

$$F(c) = \frac{1}{2}(1+\gamma) \tag{5}$$

found from t-Distribution tables with (n-1) degrees of freedom. In this study, the confidence level was taken equal to 95%. If the true standard deviation of the population (σ) is known, Eq. (4) becomes (for a 95% confidence level):

$$k = 1.96\,\sigma / \sqrt{n} \tag{6}$$

When the sample size (n) is large, e.g. >20, the standard deviation of the sample (s_{large}) is a good estimation of the standard deviation of the total population (σ). Hence using s_{large} as an approximation for σ, the sample size required for a 95% confidence interval (k) can be calculated using Eq. (6).

In order to determine how accurate is the estimated standard deviation of the total population (taken as the standard deviation of a very large sample s_{large}), the following equation can be used to calculate the 95% confidence interval for the true standard deviation (σ):

$$\frac{s_{large}^2}{1+1.96\sqrt{2/(n_{large}-1)}} \le \sigma^2 \le \frac{s_{large}^2}{1-1.96\sqrt{2/(n_{large}-1)}} \tag{7}$$

3 GOODNESS OF FIT: CHI-SQUARE TEST

The hypothesis of normal distribution for the vibration levels induced by the population of a type of vehicles can be tested using the chi-square test. Only overall frequency-weighted levels were used in this test. The test is performed as follows:

- Subdivide the vibration levels x_j into K intervals $I_1, I_2, \ldots, I_K$ such that each interval contains at least 5 values. Determine the number b_j of values in each interval.
- Using the normal distribution function

$$F(x_j) = \phi\left(\frac{x_j - \bar{x}}{s}\right) \tag{8}$$

calculate the probability p_j that a vibration level assumes a value in the interval I_j (note: $\phi(z)$ is the distribution function of the normal distribution with mean equal to 0 and variance equal to 1. It is tabulated under normal distribution in most statistics textbooks). Then compute the number of vibration levels theoretically expected in each interval, i.e.

$$e_j = np_j \tag{9}$$

where

$$p_j = \phi\left(\frac{x_j - \bar{x}}{s}\right) - \phi\left(\frac{x_{j-1} - \bar{x}}{s}\right) \tag{10}$$

- Compute the deviation

$$\chi_o^2 = \sum_{j=1}^{K} \frac{(b_j - e_j)^2}{e_j} \tag{11}$$

- Choose a significance level α (e.g. 2.5% or 5%)
- Determine the solution for c in the following equation

$$P(\chi^2 \le c) = 1 - \alpha \tag{12}$$

from chi-square distribution tables with $K - r - 1$ degrees of freedom, where r is the number of estimated parameters (e.g. r=2 if the mean and standard deviation are estimated). In Eq. 12, P designates the probability of the event $\chi^2 \le c$.

- If $\chi_o^2 \le c$, do not reject the hypothesis.

4 CALCULATION PROCEDURE

In step form, the procedure used in this study to determine the appropriate sample size (n) was as follows.

Step 1

At selected locations, one for subway trains and another for streetcars, measurement of vertical ground vibration level for about 100 vehicle pass-bys was performed. It is assumed that operational conditions for transit vehicles at other locations along the rail transit system are similar to those at the selected locations. This is a reasonable assumption, except perhaps for a sloping rail track. For each 1/3 octave band and the overall frequency-weighted signal, the standard deviation of the vibration levels in these large samples is calculated using

$$s_{large} = \left(\frac{1}{n-1} \sum_{i=1}^{n} (x_i - \bar{x}_{large})^2 \right)^{1/2} \tag{13}$$

where

$$\bar{x}_{large} = \frac{1}{n} \sum_{i=1}^{n} x_i \tag{14}$$

The 95% confidence range of the true standard deviation is calculated using Eq. 7.

Step 2

The assumption of normal distribution for the measured vibration levels is verified for the overall weighted vibration level using the chi-square test with a 2.5% significance level as described above.

Step 3

The appropriate sample size (n) for each 1/3 octave band is determined as that which yields a confidence range defined in Eq. 3 with k = 3 dB and a 95% confidence level. The size (n) is calculated using Eq. 6, but using s_{large} instead of σ, i.e.

$$n = \left(1.96\, s_{large} / k\right)^2 \tag{15}$$

The appropriate sample size is taken as the maximum of the values obtained for the various 1/3 octave frequency bands or overall frequency weighted signal.

Step 4

The appropriateness of the sample size (n) calculated in Step 3 is verified using a few randomly selected samples from the large sample described in Step 1. This is done by checking if the confidence range obtained with these samples and calculated using Eq. 4 for a 95% confidence level is within the range used in Step 3.

Note: For measurements to be performed after deciding on the sample size, it is suggested that the confidence interval be calculated using Eq. 4 for the 95% confidence level and reported for each 1/3 octave band at each measurement station.

5 RESULTS

The statistical procedures described in the previous sections were applied to determine appropriate sample sizes for measurement of vibration levels induced by subway train and streetcar populations of a major rail transit system.

Subway Trains

The assumption of normal distribution for vibration levels was checked using overall frequency-weighted rms vibration levels induced by 99 subway train pass-bys. Vibration signals were measured in the vertical direction on the ground in front of a residential home. Calculations for the chi-square test, described in Section 3, yielded χ_o^2=5.349. Finding the solution for c in Eq. 12 as 11.14, using 4 degrees of freedom and 2.5% significance level, the hypothesis of normal distribution for vibration levels induced by the subway train population was not rejected. The required sample size for a 95% confidence interval with k equal to 3 dB was then calculated using Eq. 15 for each 1/3 octave band in the range from 10 to 125 Hz and for the overall frequency weighted signal. The maximum required sample size occurred at the 1/3 octave band with a 16 Hz centre frequency and was equal to 9. The corresponding sample size was equal to 2 for the 40 Hz 1/3 octave band, which was the predominant frequency at the location of measurements, and for the overall frequency-weighted vibration level. This is much lower than the sample size required for 16 Hz. An explanation for this is that ground vibration at the 16 Hz frequency was caused by several sources of vibration in addition to subway trains, e.g. traffic on adjacent roads. To be on the conservative side, the recommended sample size was taken to be equal to 10 subway trains.

Streetcars

The assumption of normal distribution for vibration levels was checked using overall frequency-weighted rms vibration levels induced by 101 streetcar pass-bys. Vibration signals were measured in the vertical direction on the street curb. Calculations for the chi-square test yielded χ_o^2=9.247. Finding the solution for c in Eq. 12 as 9.35, using 3 degrees of freedom and 2.5% significance level, the hypothesis of normal distribution for vibration levels induced by the streetcar population was not rejected. The maximum required sample size occurred at the 1/3 octave band with a 10 Hz centre frequency and was equal to 6. Hence, the recommended sample size was equal to 6 streetcars.

6 VERIFICATION OF SAMPLE SIZE

Verification of the sample size recommended in the previous section was performed using arbitrarily selected samples at different times during the period in which the large samples were acquired. For subway trains, the error was well below 3 dB, and for streetcars the error slightly exceeded 3 dB in some instances.

REFERENCES

Al-Hunaidi M.O., Hofmeister M., and Halliwell R.E. (1992). A program for 1/3 octave analysis of traffic vibrations using digital filtering on PCs. *Inter-Noise 92*, Toronto, Vol. 2, pp. 1153-1156.

Ang A. H-S., and Tang W.S. (1975). *Probability Concepts in Engineering Planning and Design (Volume 1)*. John Wiley and Sons, Inc., New York.

Kreyszig E. (1979). *Advanced Engineering Mathematics*, 4th edition. John Wiley and Sons, Inc., New York.

ACKNOWLEDGMENTS

Messrs. W. Guan and L. Wang of IRC assisted in performing the vibration measurements. Mr. L. Wang also programmed and performed the statistical calculations.

The Canadian Acoustical Association
L'Association Canadienne d'Acoustique

MINUTES OF THE BOARD OF DIRECTOR'S MEETING

10:00 am, June 12th, 1994
Château Champlain, Montreal

Present:	R. Hetu	E. Bolstad	D. Quirt
	T. Nightingale	M. Hodgson	C. Sherry
	J. Hemingway	R. Ramakrishnan	F. Laville
	D. Chapman	B. Gosselin	
Regrets:	D. Jamieson	S. Forshaw	E. Slawinski
	M. Roland-Mieszkowski		

The Meeting was called to order at 10:08 hours.

1. **President's Report**

The World Conference on Ultrasound (WCU) is proceeding with CAA as co-sponsor. Hugh Jones will act as liaison person for CAA.

CAA will also act as co-sponsor, with no financial assistance, for the upcoming International Conference on Structural Dynamics in Hong Kong.

2. **Executive Secretary's Report**

The paid membership for 1994 stands at 294, which is an increase of 10% over June, 1993. Final Notices will be sent to unpaid Members in June and all unpaid Members removed from the Mailing List on August 1st, which is the deadline for the September issue of the Journal.

The increase in membership reflects the work of D. Jamieson and W. Sydenborgh which was appreciated by the BoD.

J. Hemingway announced his intention to resign as Secretary at the October, 1994 meeting.

3. **Treasurer's Report**

The Association continues in good financial health. It was suggested that a 1 year GIC be taken out as part of the operating fund to maximise interest.

E. Bolstad announced his intention to retire at the October 1994 Meeting. J. Hemingway offered to take on the post of Treasurer.

4. **Editor's Report**

The Journal continues to be in good shape, in a break even situation, but with a lack of Papers. M. Hodgson continues to look into the feasibility of forming an Editorial Board.

There has been a request from a second advertiser for the centre spread. It was agreed that the centre spread should be made available to other advertisers.

5. **Membership/Recruitment**

A mailing regarding Membership and Awards has been made to University departments. The BoD expressed its appreciation of D. Jamieson's efforts in increasing the Membership.

Updating of the Brochure was discussed. D. Chapman agreed to contact M. Mieszkowski and D. Jamieson regarding progress. R. Ramakrishnan agreed to help if necessary.

6. **Awards Committee**

Directors' Awards

4 papers are eligible, 1 graduate and 3 professional over 30.

Bell Speech Prize

5 applications were received, recommendation of a winner made and accepted.

Fessenden Prize

1 application was received, recommended and accepted.

Eckel Award

2 applications were received, a winner recommended and accepted.

Student Oral Presentations

No report available. T. Nightingale to contact A. Behar.

Science Fair

No report, however the fee has been paid. R. Hétu to contact A. Cohen.

Postdoctoral Prize

1 application was received, recommended and accepted. It was recommended that the prize could be awarded twice, subject to the approval of the Treasurer.

7. **Acoustics Week Reports**

Toronto, 1993

105 registered resulting in a surplus of $4,500. Thanks were expressed for the Local Planning Committee, Exhibitors and Local Consultants.

Ottawa, 1994

To be held at the Citadel Hotel, which is downtown. CAA BoD Meeting to be 7:00 pm on October 18th.

Quebec City, 1995

The offer of B. Gosselin and R. Hetu to host the 1995 Meeting in Quebec City was gratefully accepted.

1996 ?

A meeting in the West was suggested, E. Bolstad to investigate.

1997 ?

Halifax was suggested.

1998 ?

Sherbrooke is a possibility, hosted by J. Nicolas.

8. **Other Business**

Student Travel Subsidy Policy

Motion: "That the CAA Student Travel Policy be as follows:

1. The Student Travel Subsidy will be administered by the CAA Treasurer.

2. Before each annual meeting, student members of CAA intending to present papers may apply, at the time of submission of an abstract, to the CAA Local Planning Committee for a student travel grant.

   The application must be accompanied by a statement from their supervisor stating that the student is a full time student and the CAA Travel Subsidy combined with other travel funds that the student may receive will not exceed student travel costs to attend the Meeting.

3. The Travel Subsidy will be awarded to students who have travelled more than 100 km. to attend the Meeting.

   The minimum amount of the subsidy will be $150.

   Past experience has shown that the total amount of the Travel Subsidy will not exceed $2,500 per year.

4. The Local Organizing Committee of the Annual CAA Meeting will provide to the Treasurer a list of the candidates names, addresses and Social Insurance Numbers.

5. The Treasurer will inform the Local Planning Committee of the amount of subsidy for each student.

6. Students will be given their travel grant at the CAA meeting after they have presented their paper."

Proposed: J. Hemingway
Seconded: R. Ramakrishnan

Carried

Thanks were expressed to R. Ramakrishnan for his work on developing the policy.

Seminars

It was agreed that the seminar portion of the annual meeting be administered by the local planning committee.

Nominations

The offer of J. Hemingway to stand as Treasurer was accepted. 2 Directors and a Secretary are required. D. Chapman to investigate.

9. **Visibility of CAA**

The theme of Hearing Accessibility was accepted as the theme of AWC 1994, Ottawa, supported by the UBC Institute of Hearing Accessibility. M. Hodgson to co-ordinate with T. Nightingale and other Visibility Sub-committee members.

10. **By-Law Review**

It was agreed that the By-Law review be discontinued. D. Chapman to look into the possibility of translation into French.

The meeting was adjourned at 15:37.

NEWS / INFORMATIONS

CONFERENCES

129th Meeting of the Acoustical Society of America: May 31-June 4, 1995, Washington, DC, USA. Contact: Elaine Moran, Acoustical Society of America, 500 Sunnyside Blvd., Woodbury, NY 11797, USA. Telephone: +1 (516) 576-2360, Fax: +1 (516) 349-7669.

2nd International Conference on Acoustics and Musical Research: 3rd week, May 1995, Ferrara, ITALY. Contact: Conference Secretariat, CIARM95, National Research Council of Italy, Cemoter Acoustics Department, Via Canal Bianco, 28-44044 Ferrara. Tel. +39 532 731571-Fax +39 532 732250. E-mail CIARM95@CNRFE4.FE.CNR.IT

International Symposium in Music and Concert Hall Acoustics (MCHA95): May 15 to 18 ,1995, Kirishima, Kagoshima-Prefecture, JAPAN. Contact: The Kirishima International Concert Hall, Kagoshima, Japan for further details.

15th International Congress on Acoustics: 26-30 June, 1995, Trondheim, NORWAY. Contact: ICA'95, SEVU, Congress Department, N-7034 Trondheim, Norway, Telephone +47 7359 5251/7359 5254, Fax +47 7359 5150,Electronic Post ica95@sevu.unit.no

INTER-NOISE 95: July 10-12, 1995, Newport Beach, California, USA. Contact: Institute of Noise Control Engineering, P.O. Box 3206, Arlington Branch, Poughkeepsie, NY 12603, USA. Tel. (914) 462-4006, Fax. (914) 473-9325.

17th Boundary Element International Conference: 17-19 July, 1995, Wisconsin, USA. Contact: Lis Johnstone, Conference Secretariat, BEM 17, Wessex Institute of Technology, Ashurst Lodge, Ashurst Southampton, SO40 7AA. Tel 44 (0) 703 293223, Fax 44 (0) 703 292853, EMail CMI@uk.ac.rl.ib, Intl EMail CMI@ib.rl.ac.uk

1995 World Congress on Ultrasonics: September 3 to 7, 1995, BERLIN. Contact: WCU'95 Secretariat, Prof. Dr. J. Herbertz, Gerhard-Mercator-Universitat, D-47048 Duisburg, Germany. Tel +49 (203) 379-3243,Fax +49 (203) 37 35 34

BETECH 95: September 13-15 1995, Liege, BELGIUM. Contact: Liz Johnstone, Conference Secretariat - BETECH 95, Ashurst Lodge, Ashurst, SO40 7AA UK. Tel +44 (0) 703 293223, Fax +44 (0) 703 292853, EMail CMI@uk.ac.rl.ib., Intl EMail CMI@ib.rl.ac.uk

Second International Conference on Theoretical & Computational Acoustics: August 21-25, 1995, Hawaii, USA. Contact: Dr. Ding Lee (Code 3122), Naval Undersea Warfare Center, Detachment New London, New London CT 06320 USA. Tel 203-440-4438 Fax 203-4406228

130th Meeting of the Acoustical Society of America: November 27-December 1, 1995, St. Louis, Missouri, USA. Contact: Elaine Moran, Acoustical Society of America, 500 Sunnyside Blvd., Woodbury, NY 11797, USA. Telephone: +1 (516) 576-2360, Fax: +1 (516) 349-7669.

CONFERENCES

129e rencontre de l'Acoustical Society of America: Washington, DC, du 31 mai au 4 juin 1995. Renseignements: Elaine Moran, Acoustical Society of America, 500 Sunnyside Blvd., Woodbury, NY 11797, USA. Téléphone (516) 576-2360; télécopieur (516) 349- 7669.

2e conférence internationale sur la recherche en acoustique et en musique: Ferrara, Italie, 3e semaine de mai 1995. Renseignements: Conference Secretariat, CIARM95, National Research Council of Italy, Cemoter Acoustics Department, Via Canal Bianco, 28-44044 Ferrara, Italie. Téléphone 39 532 731571; télécopieur 39 532 732250; courrier électronique CIARM95@CNRFE4.FE.CNR.IT.

Symposium international d'acoustique musicale et de salles de concert (MCHA95): Kirishima, Kagoshima-Prefecture, Japon, du 15 au 18 mai 1995. Renseignements: The Kirishima International Conceert Hall, Kagoshima, Japon.

15e congrès international d'acoustique: Trondheim, Norvège, du 26 au 30 juin 1995. Renseignements: ICA'95, SEVU, Congress Department, N-7034 Trondheim, Norvège. Téléphone 47 7359 5251/5254; télécopieur 47 7359 5150; courrier électronique ica95@sevu.unit.no.

INTER-NOISE 95: Newport Beach, Californie, du 10 au 12 juillet 1995. Renseignements: Institute of Noise Control Engineering, P.O. Box 3206, Arlington Branch, Poughkeepsie, NY 12603, USA. Téléphone: (914) 462-4006, télécopieur (516) 473-9325.

17e conférence internationale sur les éléments de contour: Winconsin, États-Unis, du 17 au 19 juillet 1995. Renseignements: Lis Johnstone, Conference Secretariat, BEM 17, Wessex Institute of Technology, Ashurst Lodge, Ashurst Southampton, SO40 7AA. Téléphone 44 (0) 703 293223; télécopieur 44 (0) 703 292853; courrier électronique CMI@uk.ac.rl.ib; courrier électronique international CMI@ib.rl.ac.uk.

Congrès mondial de 1995 sur les ultrasons: Berlin, Allemagne, du 3 au 7 septembre 1995. Renseignements: WCU'95 Secretariat, Prof. Dr. J. Herbertz, Gerhard-Mercator-Universitat, D-47048 Duisburg, Allemagne. Téléphone 49 203 379 3243; télécopieur 49 203 37 3534.

BETECH 95: Liège, Belgique, du 13 au 15 septembre 1995. Renseignements: Liz Johnstone, Conference Secretariat, BETECH 95, Ashurst Lodge, Ashurst, SO40 7AA, Royaume-Uni. Téléphone 44 (0) 703 29 3223; télécopieur 44 (0) 703 29 2853; courrier électronique CMI@uk.ac.rl.ib, courrier électronique international CMI@ib.rl.ac.uk.

2e conférence internationale sur l'acoustique théorique de calcul: Hawaï, du 21 au 25 août. Renseignements: Dr. Ding Lee (code 3122), Naval Undersea Warfare Center, Detachment New London, New London CT 06320, États-Unis. Téléphone 203 440 4438; télécopieur 203 440 6228.

130e rencontre de l'Acoustical Society of America: St. Louis, Missouri, États-Unis, du 27 novembre au 1er décembre 1995. Renseignements: Elaine Moran, Acoustical Society of America, 500 Sunnyside Blvd., Woodbury, NY 11797, États-Unis. Téléphone (516) 576-2360; télécopieur (516) 349- 7669.

NEW PRODUCTS

Proposed Worker Protection Warning Act

Consistent, nationwide warnings for workplace safety equipment, including personal protective equipment, would be provided under the Worker Protection Warning Act now under consideration. Under the proposed legislation, OSHA would be charged with creating and requiring warnings and guidelines that would preempt inconsistent state warnings.

The legislation is designed to simplify instructions, limit training and retraining times, and reduce the problems created by multiple state guidelines. Ultimately, consistency in the warnings for safety equipment will contribute to worker protection in the workplace.

Acoustic Fire Detection `Sounds` Good at NIST

Few sounds are more appealing than the cracking and popping of wood blazing in the fireplace. But fire can also produce sounds the human ear cannot hear.

Researchers in the Building and Fire Research Laboratory at the National Institute of Standards and Technology are using acoustic emission transducers to detect a fire hazard in the making. Heat from a small fire inside a wall, ceiling or an overloaded electrical circuit can cause the surrounding materials to expand and emit sound at very high frequencies.

NIST researchers found that trapped moisture boiling or plastic bubbles bursting sent out a characteristic "acoustic signature" that can be detected before the material bursts into flames. In recent laboratory tests, NIST's fire experts were able to "hear" these sounds up to 3 meters from the heat source.

Additional real-life fire testing is needed as well as more research to screen out false alarms. For technical information, contact William Grosshandler, B356 Polymer Bldg., NIST, Gaithersburg, MD 20899-0001, (301)975-2310, EMail wgrosshan@enh.nist.gov(via internet).

PEOPLE IN THE NEWS

Below is a letter, for your interest, that I received from Mr. R. Wayne Gatehouse, Psychology Department, University of Guelph:

Note: Special Awards Judging
Canada Wide Science Fair
University of Guelph
May 18, 1994

On May 18, 1994 I, along with Mr. Greg Wheeler a teacher of Science and French from Cambridge Ontario, had the privilege of judging the 17 entries for the CAA special award of $400.00.

Students who reach the cross Canada finals for the Science Fair self-nominate their projects for the special awards and most put themselves into competition in several different categories within three broad categories of Life Sciences, Engineering, and Computers. During the general judging which is done on the preceding days, the students compete for medals (and other monetary prizes and trips) which are awarded in Junior (Primary school level); Intermediate (early High school); and Senior categories. The prizes are awarded in this stage of the competition on the bases of Scientific Thought (45%); Creativity (25%); Display (20%); and, project Summary (10%). Several of the students who competed for the CAA special award won such prizes.

NOUVEAUX PRODUITS

Projet de loi américain sur les mises en garde pour l'équipement de protection

Les États-Unis étudient un projet de loi destiné à uniformiser à la grandeur du pays les mises en garde de séurité accompagnant les équipements de protection utilisés en milieu de travail. En vertu de cette loi, la Worker Protection Warning Act, l'OSHA serait chargée de la création et de la mise en application des mises en garde.

Cette loi vise à simplifier les instructions, à minimiser le besoins de formation et de recyclage, et à réduire la confusion créée par l'existence d'un ensemble disparate de mises en garde dans les différents États, le tout dans le but d'améliorer la protection des travailleurs.

La détection acoustique des feux

Peu de sons sont aussi réconfortants que le craquement et le pétillement du bois qui brûle dans la cheminée. Mais le feu peut également produire des sons inaudibles.

Des chercheurs du Building and Fire Research Laboratory du National Institute of Standards and Technology (NIST) aux États-Unis étudient la possibilité d'utiliser des transducteurs pour détecter les feux naissants. En effet, sous l'action de la chaleur qui se dégage d'un feu naissant à l'intérieur d'un mur, d'un plafond ou d'un circuit électrique surchargé, les matériaux environnants, en prenant de l'expansion, peuvent émettre des sons de très hautes fréquences.

Les chercheurs du NIST ont remarqué que le bouillonnement de l'humidité emprisonnée dans les murs ou l'éclatement des bulles de plastique ont des «signatures acoustiques» qui peuvent être détectées avant que le matériel prenne feu. Lors d'essais en laboratoire, ils ont été capables de repérer ces sons jusqu'à trois métres de la source de chaleur.

Des recherches plus poussées sur le terrain sont nécessaires afin d'éliminer les fausses alarmes et de perfectionner le système. Renseignements: William Grosshandler, B356 Polymer Bldg., NIST, Gaithersburg, MD 20899-0001; téléphone 301 975 2310; courrier électronique (internet) wgrosshan@enh.nist.gov.

LES GENS QUI FONT PARLER D'EUX

Voici la traduction d'une lettre que j'ai reçue de monsieur R. Wayne Gatehouse, du departement de psychologie de l'université de Guelph, et qui, je crois, saura vous intéresser.

«Le 18 mai 1994, j'ai eu l'honneur, en compagnie de monsieur Greg Wheeler, professeur de sciences et de français de Cambridge (Ontario), de juger les mérites de 17 projets dans le cadre de la remise du prix spécial de 400 $ de l'ACA.

Les étudiants qui se rendent aux finales canadiennes de l'expo-sciences soumettent leurs projets pour le prix spécial et la plupart d'entre eux s'inscrivent également dans plusieurs catégories générales regroupées sous Sciences de la vie, Ingénierie et Informatique. Pendant le concours général qui se déroule en premier, les étudiants sont en compétition pour des médailles et autres prix remis au niveau junior (école primaire), intermédiaire (premier cycle de secondaire) et senior (deuxième cycle de secondaire). A cette étape de la compétition, les prix sont remis en fonction des critères suivants: contenu scientifique (45%), créativité

For the judging of the CAA special award your judges decided after consultation, to concentrate on the first two of these categories, and to stress the latter in so far as it pertained to Innovative aspects and possible applications within the broadly defined area of Acoustics.

Of the 19 self nominated projects two were withdrawn prior to judging and three looked at in the "first pass" were not considered to meet the above definitional criteria in that their relationship to acoustics was limited to the use of existing technologies within the scope of primarily investigating another area (e.g. an ultrasonic irradiation device being used to fraction out benzene etc. within compounds). Of the remaining 14, three were presented in French.

The judging was very difficult since there were several entries of diverse emphases which met the criteria. Nearly all of them were heavily involved with computer and programming technologies. The topic areas ranged from: a new method of training for persons using morse code; several in the general area of enhancing the qualities of sound outputs from computer sound cards; several others in the general area of computer generated music (e.g. recognition of tone colour, recognition of the differences between live and computer generated music, factors affecting the pitch); a study of effects of wood chips as parts of various compositions of roadside noise barriers.

The third and second place entrant's topics were a study of sound localization for the purpose of determining equations that would be able to produce an understanding of signal location in virtual reality helmets and a computerized hearing screener system.

The winning entry Les Orielles ça de Protège! (J. Marchand) devised an ingenious method of controlling sound volume in personal (i.e. Walkman-like) stereos. Basically, the innovation consisted of a series of LED's mounted into the casing of the stereo that successively lit up in response to increased volume above about 85 dB so that users could have some sort of visual feedback that the volume was quite high. Once the volume was > 95 dB a delay circuit (a small board that the winner developed and put inside the Walkman; he went through three prototypes before he was satisfied with it's performance) cut in such that after 15 seconds the volume was automatically reduced to 85 dB for another 15 seconds after which it was allowed to increase again to reflect the setting which the user had initially set it at. The process of "enforced volume reduction" was again instituted after another delay period. The student was asked why he let the volume go back up after it's initial reduction and his reply was to the

As implied there were many very good projects which were all worthy of some recognition and as judges we suggest that perhaps CAA might consider establishing three equal value prizes of say $200.00 to $300.00 each. Additionally, it is suggested that your judges be given some criteria that meet with the guidelines of the Canada Wide Science Fair classifications for Junior, Intermediate and Senior entrants. If something like this was done, the judges would find the task easier and a little more equitable.

(25%), présentation (20%) et résumé du projet (10%). Plusieurs des étudiants inscrits pour la bourse de l'ACA ont remporté des prix dans cette compétition.

Pour la remise du prix spécial de l'ACA, les juges ont décidé, après consultation, de n'évaluer que les projets entrant dans les catégories Sciences de la vie et Ingénierie, et de juger ceux de la catégorie Informatique que dans la mesure où ils sont innovateurs et qu'ils présentent des applications possibles dans le domaine de l'acoustique.

Des dix-neuf projets en lice, deux ont été éliminés au départ et trois ont été éliminés après une première évaluation parce que leur relation avec l'acoustique se limitait à l'utilisation de technologies existantes dans un nouveau contexte (p. ex., l'utilisation d'un appareil d'irradiation par ultrasons pour le fractionnement du benzène). Des quatorze projets restants, trois ont été présentés en français.

Les juges n'ont pas eu la tâche facile puisque plusieurs projets répondaient à des degrés divers aux différents critères. La presque totalité des projets empruntaient beaucoup à l'informatique et à la programmation. Plusieurs projets portaient en effet sur l'amélioration de la qualité du son produit par les cartes de son pour ordinateurs. D'autres relevaient du domaine plus général de la musique informatique (p. ex. la reconnaissance des différences entre la musique produite par instrument et la musique produite par ordinateur; les facteurs affectant la hauteur de son). Notons également une nouvelle méthode de formation pour les personnes utilisant les codes en morse ainsi qu'une étude sur l'utilisation des copeaux de bois dans la fabrication d'ouvrages antibruit placés le long des routes.

Les troisième et deuxième place ont été attribuées à une étude sur la localisation du bruit dans le but de déterminer les équations qui permettraient de comprendre l'emplacement des signaux dans les casques à réalité virtuelle, et à un système de dépistage auditif informatisé.

Le première place est allée au projet «Les oreilles ça se protège!» de J. Marchand pour l'invention d'un dispositif de réglage du volume du son dans les baladeurs (Walkman). Le dispositif est composé d'une série de diodes électroluminescentes qui s'allument une à une à mesure que le volume dépasse environ 85 dB, de façon à avertir visuellement l'auditeur que le son est trop fort. Lorsque le volume dépasse 95 dB, un circuit à retard (une carte de circuit mise au point par le jeune inventeur et installée à l'intérieur du baladeur; it a fabriqué trois prototypes avant d'être satisfait du résultat) abaisse automatiquement le volume à 85 dB après quinze secondes puis remet le réglage initial après un autre quinze secondes. Lorsqu'on lui a demandé pourquoi il laissait le volume remonter, le jeune inventeur a expliqué qu'on ne pouvait pas priver totalement l'utilisateur de sa liberté de choix et qu'il était à espérer qu'avec les signaux visuels et la réduction répétitive du volume, ce dernier apprenne à régler

Comme vous pouvez le constater, un grand nombre de projets auraient mérité une certaine reconnaissance et, en tant que juges, nous avons suggéré que l'ACA considère la création de trois prix de même valeur, de 200 ou de 300 $ chacun. Nous avons également suggéré que l'on nous donne certains critères tenant compte des classifications junior, intermédiaire et senior et sur lesquels nous pourrions baser vos évaluations. Ces critères rendraient notre tâche plus facile et nos évaluations plus équitables.

The Canadian Acoustical Association
l'Association Canadienne d'Acoustique

PRIZE ANNOUNCEMENT

A number of prizes, whose general objectives are described below, are offered by the Canadian Acoustical Association. As to the first four prizes, applicants must submit an application form and supporting documentation to the prize coordinator before the end of February of the year the award is to be made. Applications are reviewed by subcommittees named by the President and Board of Directors of the Association. Decisions are final and cannot be appealed. The Association reserves the right not to make the awards in any given year. Applicants must be members of the Canadian Acoustical Association. Preference will be given to citizens and permanent residents of Canada. Potential applicants can obtain full details, eligibility conditions and application forms from the appropriate prize coordinator.

Edgar and Millicent Shaw Postdoctoral Prize in Acoustics

This prize is made to a highly qualified candidate holding a Ph.D. degree or the equivalent, who has completed all formal academic and research training and who wishes to acquire up to two years supervised research training in an established setting. The proposed research must be related to some area of acoustics, psychoacoustics, speech communication or noise. The research must be carried out in a setting other than the one in which the Ph.D. degree was earned. The prize is for $3000 for full-time research for twelve months, and may be renewed for a second year. Coordinator: Sharon Abel, Mount Sinai Hospital, 600 University Avenue, Toronto, ON M5G 1X6. Past recipients are:

1990	*Li Cheng*	*Université de Sherbrooke*
1993	*Roland Woodcock*	*University of British Columbia*

Alexander Graham Bell Graduate Student Prize In Speech Communication and Behavioural Acoustics

The prize is made to a graduate student enrolled at a Canadian academic institution and conducting research in the field of speech communication or behavioural acoustics. It consists of an $800 cash prize to be awarded annually. Coordinator: Don Jamieson, Department of Communicative Disorders, University of Western Ontario, London, ON N6G 1H1. Past recipients are:

1990	*Bradley Frankland*	*Dalhousie University*
1991	*Steven D. Turnbull*	*University of New Brunswick*
	Fangxin Chen	*University of Alberta*
	Leonard E. Cornelisse	*University of Western Ontario*
1993	*Aloknath De*	*McGill University*

Fessenden Student Prize in Underwater Acoustics

The prize is made to a graduate student enrolled at a Canadian university and conducting research in underwater acoustics or in a branch of science closely connected to underwater acoustics. It consists of $500 cash prize to be awarded annually. Coordinator: David Chapman, DREA, PO Box 1012, Dartmouth, NS B2Y 3Z7.

1992	*Daniela Dilorio*	*University of Victoria*
1993	*Douglas J. Wilson*	*Memorial University*

Eckel Student Prize in Noise Control

The prize is made to a graduate student enrolled at a Canadian academic institution pursuing studies in any discipline of acoustics and conducting research related to the advancement of the practice of noise control. It consists of a $500 cash prize to be awarded annually. The prize was inaugurated in 1991. Coordinator: Murray Hodgson, Occupational Hygiene Programme, University of British Columbia, 2206 East Mall, Vancouver, BC V6T 1Z3.

Directors' Awards

Three awards are made annually to the authors of the best papers published in *Canadian Acoustics*. All papers reporting new results as well as review and tutorial papers are eligible; technical notes are not. The first award, for $500, is made to a graduate student author. The second and third awards, each for $250, are made to professional authors under 30 years of age and 30 years of age or older, respectively. Coordinator: position vacant.

Student Presentation Awards

Three awards of $500 each are made annually to the undergraduate or graduate students making the best presentations during the technical sessions of Acoustics Week in Canada. Application must be made at the time of submission of the abstract. Coordinator: Alberto Behar, 45 Meadowcliffe Drive, Scarborough, ON M1M 2X8.

The Canadian Acoustical Association
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ANNONCE DE PRIX

Plusieurs prix, dont les objectifs généraux sont décrits ci-dessous, sont décernés par l'Association Canadienne d'Acoustique. Quant aux quatre premiers prix, les candidats doivent soumettre un formulaire de demande ainsi que la documentation associée au coordonateur de prix avant le dernier jour de février de l'année durant laquelle le prix sera décerné. Toutes les demandes seront analysées par des sous-comités nommés par le président et la chambre des directeurs de l'Association. Les décisions seront finales et sans appel. L'Association se réserve le droit de ne pas décerner les prix une année donnée. Les candidats doivent être membres de l'Association. La préférence sera donnée aux citoyens et aux résidents permanents du Canada. Les candidats potentiels peuvent se procurer de plus amples détails sur les prix, leurs conditions d'éligibilité, ainsi que des formulaires de demande auprès du coordonateur de prix.

PRIX POST-DOCTORAL EDGAR ET MILLICENT SHAW EN ACOUSTIQUE

Ce prix est attribué à un(e) candidat(e) hautement qualifié(e) et détenteur(rice) d'un doctorat ou l'équivalent qui a complèté(e) ses études et sa formation de chercheur et qui désire acquérir jusqu'à deux années de formation supervisée de recherche dans un établissement reconnu. Le thème de recherche proposée doit être relié à un domaine de l'acoustique, de la psycho-acoustique, de la communication verbale ou du bruit. La recherche doit être menée dans un autre milieu que celui où le candidat a obtenu son doctorat. Le prix est de $3000 pour une recherche plein temps de 12 mois avec possibilité de renouvellement pour une deuxième année. Coordonnatrice: Sharon Abel, Mount Sinai Hospital, 600 University Avenue, Toronto, ON M5G 1X6. Les récipiendaires antérieur(e)s sont:

1990	*Li Cheng*	*Université de Sherbrooke*
1993	*Roland Woodcock*	*University of British Columbia*

PRIX ÉTUDIANT ALEXANDER GRAHAM BELL EN COMMUNICATION VERBALE ET ACOUSTIQUE COMPORTEMENTALE

Ce prix sera décerné à un(e) étudiant(e) inscrit(e) dans une institution académique canadienne et menant un projet de recherche en communication verbale ou acoustique comportementale. Il consiste en un montant en argent de $800 qui sera décerné annuellement. Coordonnateur: Don Jamieson, Department of Communicative Disorders, University of Western Ontario, London, ON N6G 1H1. Les récipiendaires antérieur(e)s sont:

1990	*Bradley Frankland*	*Dalhousie University*
1991	*Steven D. Turnbull*	*University of New Brunswick*
	Fangxin Chen	*University of Alberta*
	Leonard E. Cornelisse	*University of Western Ontario*
1993	*Aloknath De*	*McGill University*

PRIX ÉTUDIANT FESSENDEN EN ACOUSTIQUE SOUS-MARINE

Ce prix sera décerné à un(e) étudiant(e) inscrit(e) dans une institution académique canadienne et menant un projet de recherche en acoustique sous-marine ou dans une discipline scientifique reliée à l'acoustique sous-marine. Il consiste en un montant en argent de $500 qui sera décerné annuellement. Coordonnateur: David Chapman, DREA, PO Box 1012, Dartmouth, NS B2Y 3Z7.

1992	*Daniela DiIorio*	*University of Victoria*
1993	*Douglas J. Wilson*	*Memorial University*

PRIX ÉTUDIANT ECKEL EN CONTROLE DU BRUIT

Ce prix sera décerné à un(e) étudiant(e) inscrit(e) dans une institution académique canadienne dans n'importe quelle discipline de l'acoustique et menant un projet de recherche relié à l'avancement de la pratique en contrôle du bruit. Il consiste en un montant en argent de $500 qui sera décerné annuellement. Ce prix a été inauguré en 1991. Coordonnateur: Murray Hodgson, Occupational Hygiene Programme, University of British Columbia, 2206 East Mall, Vancouver, BC V6T 1Z3.

PRIX DES DIRECTEURS

Trois prix sont décernés, à tous les ans, aux auteurs des trois meilleurs articles publiés dans l'*Acoustique Canadienne*. Tout manuscrit rapportant des résultats originaux ou faisant le point sur l'état des connaissances dans un domaine particulier sont éligibles; les notes techniques ne le sont pas. Le premier prix, de $500, est décerné à un(e) étudiant(e) gradué(e). Le deuxième et le troisième prix, de $250 chacun, sont décernés à des auteurs professionnels âgés de moins de 30 ans et de 30 ans et plus, respectivement. Coordonnateur: poste à combler.

PRIX DE PRESENTATION ÉTUDIANT

Trois prix, de $500 chaqun, sont décernés annuellement aux étudiant(e)s sous-gradué(e)s ou gradué(e)s présentant les meilleures communications lors de la Semaine de l'Acoustique Canadienne. La demande doit se faire lors de la soumission du résumé. Coordonnateur: Alberto Behar, 45 Meadowcliffe Drive, Scarborough, ON M1M 2X8.

INSTRUCTIONS TO AUTHORS PREPARATION OF MANUSCRIPT

Submissions: The original manuscript and two copies should be sent to the Editor-in-Chief.

General Presentation: Papers should be submitted in camera-ready format. Paper size 8.5" x 11". If you have access to a word processor, copy as closely as possible the format of the articles in Canadian Acoustics 18(4) 1990. All text in Times-Roman 10 pt font, with single (12 pt) spacing. Main body of text in two columns separated by 0.25". One line space between paragraphs.

Margins: Top - title page: 1.25"; other pages, 0.75"; bottom, 1" minimum; sides, 0.75".

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Abstracts: English and French versions. Headings, 12 pt bold, upper case, centered. Indent text 0.5" on both sides.

Headings: Headings to be in 12 pt bold, Times-Roman font. Number at the left margin and indent text 0.5". Main headings, numbered as 1, 2, 3, ... to be in upper case. Sub-headings numbered as 1.1, 1.2, 1.3, ... in upper and lower case. Sub-sub-headings not numbered, in upper and lower case, underlined.

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Page numbers: In light pencil at the bottom of each page.

Reprints: Can be ordered at time of acceptance of paper.

DIRECTIVES A L'INTENTION DES AUTEURS PREPARATION DES MANUSCRITS

Soumissions: Le manuscrit original ainsi que deux copies doivent être soumis au rédacteur-en-chef.

Présentation générale: Le manuscript doit comprendre le collage. Dimensions des pages, 8.5" x 11". Si vous avez accès à un système de traitement de texte, dans la mesure du possible, suivre le format des articles dans l'Acoustique Canadienne 18(4) 1990. Tout le texte doit être en caractères Times-Roman, 10 pt et à simple (12 pt) interligne. Le texte principal doit être en deux colonnes séparées d'un espace de 0.25". Les paragraphes sont séparés d'un espace d'une ligne.

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L'abonnement pour la présente année est dû le 31 janvier. Les nouvels abonnements reçus avant le 1 juillet s'appliquent à l'année courante et incluent les ancient numéros (non-épuisés) de l'Acoustique Canadienne de cette année. Les abonnements reçus à partir du 1 juillet s'appliquent à l'année suivante.

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